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Acoustic resonance in a reservoir-pipeline-orifice system

by

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ABSTRACT
The fundamentals of oscillating flow in a reservoir-pipe-orifice system are revisited in a theoretical study related to acoustic resonance experiments carried out in a large-scale pipeline. Four different types of system excitation are considered: forcing velocity, forcing pressure, linear oscillating resistance and nonlinear oscillating resistance. Analytical solutions are given for the periodic responses to the first three excitations. Analytical and numerical results for the large-scale pipeline are presented and some peculiarities are discussed.

Key words
Hydraulic transients; Acoustic resonance; Impedance; Orifice; Rotating valve; Analytical solution.

INTRODUCTION
Acoustic resonance in liquid-filled pipe systems is an undesirable phenomenon that cannot always be prevented. It causes noise, vibration, fatigue, instability, and it may lead to damage of hydraulic machinery and pipe supports. If possible, resonance should be anticipated in the design process and be part of the hydraulic transient analysis.

The prediction of resonance in liquid-filled pipelines is less straightforward than one might expect. First of all, the calculation of natural frequencies cannot always be based on simple formulas. Second, the excitation mechanism must be modelled correctly and care must be taken with excitation mechanisms that are influenced by the system response itself. Third, the influence and proper modelling of damping mechanisms is essential, in particular with regard to suppressing fluid transients and beat phenomena.

This study is a preliminary analysis of acoustic resonance tests carried out at Deltares, Delft, The Netherlands, within the framework of the European Hydralab III programme [1]. The (idealised) test system is a 50 m long pipeline of 200 mm diameter that is discharging water from a 25 m high reservoir through an 800 mm² orifice to the open atmosphere, as sketched in Fig.1. The outflow is partly interrupted by a rotating disc which generates flow disturbances at a fixed frequency in the range 3 Hz to 100 Hz. The system is simulated with four different models for the excitation.

Figure 1. Sketch of reservoir-pipeline-orifice system; pipe length \( L = 50 \text{ m} \) and inner diameter \( D = 0.2 \text{ m} \).
WATERHAMMER EQUATIONS

Classical waterhammer theory [2-4] adequately describes the low-frequency vibration of elastic liquid columns in fully-filled pipes. The two equations, governing velocity, $V$, and pressure, $P$, are

$$\frac{\partial V}{\partial t} + \frac{1}{\rho} \frac{\partial P}{\partial x} = - \frac{\lambda_f}{D} V|V|, \quad (1)$$
$$\frac{\partial V}{\partial t} + \frac{1}{K'} \frac{\partial P}{\partial t} = 0, \quad (2)$$

with

$$\frac{1}{K'} := \frac{1}{K} + \frac{D}{Ee}. \quad (3)$$

**Notation:** $D =$ inner pipe diameter, $E =$ Young modulus of pipe material, $e =$ wall thickness, $K =$ bulk modulus of liquid, $K' =$ effective bulk modulus including wall elasticity, $x =$ distance along pipe, $t =$ time, $\lambda_f =$ Darcy-Weisbach friction factor, and $\rho =$ mass density of liquid. The friction term is ignored herein, i.e. $\lambda_f = 0$, to concentrate on the orifice as the sole cause of damping. Equations (1) and (2) can be combined to the standard wave equations

$$c^2 \frac{\partial^2 V}{\partial t^2} - c^2 \frac{\partial^2 V}{\partial x^2} = 0 \quad \text{and} \quad \frac{\partial^2 P}{\partial t^2} - c^2 \frac{\partial^2 P}{\partial x^2} = 0, \quad (4a, 4b)$$

where the acoustic wave speed is

$$c := \sqrt{\frac{K}{\rho}}. \quad (5)$$

SINUSOIDAL EXCITATION

The pressure of the reservoir at $x = 0$ is taken constant, i.e.

$$P(0, t) = P_{res}. \quad (6)$$

Sinusoidal excitation at the downstream end (at $x = L$) is simply imposed by

$$V(L, t) = V(L, 0) + \dot{V} \sin \left(2\pi \frac{t}{T}\right) \quad \text{or}$$
$$P(L, t) = P(L, 0) + \dot{P} \sin \left(2\pi \frac{t}{T}\right), \quad (7a, 7b)$$

where $T$ is the period of oscillation, the circumflex ($\hat{\cdot}$) indicates the amplitude of oscillation, $V(L, 0) > \dot{V} > 0$ to prevent backflow at $x = L$ and $P(L, t) = P_{res}$ to guarantee equilibrium when $\dot{V} = \dot{P} = 0$ or $T = \infty$.

In practice it may be difficult to realise one of the boundary conditions (7). Sinusoidal excitation has been achieved by two typical forcing devices: 1) the oscillating piston and 2) the oscillating valve. The frequency-controlled oscillating piston can excite the system directly [5-6] or indirectly (oscillating liquid column) [7-9]. Typical frequency-controlled valve designs among others include a servo-valve unit [10], a siren-type valve [11] and a unit with variable periphery disc [12]. In the Hydralab III project [1] a Svingen-type rotating disc [13] has been used, which is described by the orifice equation below. The Svingen-type valve has been proved to be a cost-effective device of simple and robust design (with negligible FSI effects on the pipe test section).

**Nonlinear orifice equation**

In steady turbulent pipe flow the pressure loss, $\Delta P_0$, across an orifice is

$$\Delta P_0 = \xi_0 \frac{1}{2} \rho V_0^2 |V|, \quad \text{where} \quad \xi_0 := \left(\frac{A}{C_d A_{or,0}}\right)^2, \quad (8)$$

and $V_0$ is the steady flow velocity in the pipe, $A$ is the cross-sectional area of the pipe, $A_{or,0}$ is the steady outflow area of the orifice and $C_d$ is the coefficient of discharge [2, Section 3-3], [14, Section 9-5]. In a quasi-steady manner, the same relation is assumed to hold for an orifice with an area that varies in time,

$$\Delta P = \xi(t) \frac{1}{2} \rho V^2 |V|, \quad \text{where} \quad \xi(t) := \left(\frac{A}{C_d A_{or}(t)}\right)^2, \quad (9)$$

and $\xi(0) = \xi_0$ when starting the area variation from steady state at $t = 0$. Dividing (9) by (8) and introducing the dimensionless valve closure coefficient $\tau(t) := \sqrt{\xi(t)/\xi_0} = A_{or}(t)/A_{or,0}$ gives the nonlinear boundary condition

$$\Delta P_0 V|V| = \tau^2(t) V_0^2 |V_0| \Delta P, \quad V_0 \neq 0, \quad (10)$$

where $\tau(t) = 1$ corresponds to the steady state. The specific function $\tau(t)$ used to generate oscillating flow is

$$\tau(t) := \frac{2 - \alpha}{2} + \frac{\alpha}{2} \cos \left(2\pi \frac{t}{T}\right), \quad 0 \leq \alpha \leq 1, \quad (11)$$

in which $T$ is the period of the sinusoidal excitation and $\tau(0) = 1$, see Fig. 2. Equation (11) describes an orifice with
constant area $A_{or,0}$ that is partly covered by a varying area $A_{\text{disc}}(t)$ (rotating disc herein) according to the following relationships:

$$A_{or}(t) := A_{or}(0) - A_{\text{disc}}(t)$$

and

$$A_{\text{disc}}(t) := A_{\text{disc,max}} \left( \frac{1}{2} - \frac{1}{2} \cos \left( \frac{2\pi t}{T} \right) \right), \quad 0 \leq A_{\text{disc,max}} \leq A_{or}(0)$$

with

$$A_{or}(t) := A_{or}(0) \left( \frac{2 - \alpha}{2} + \frac{\alpha}{2} \cos \left( \frac{2\pi t}{T} \right) \right)$$

$$\alpha := \frac{A_{\text{disc,max}}}{A_{or}(0)}, \quad 0 \leq \alpha \leq 1 .$$

FUNDAMENTALS OF OSCILLATORY FLOW

The flow in the reservoir-pipe-orifice system (Fig. 1) is excited sinusoidally at $x = L$. If the frequency of excitation is high enough, say $f$ is of the order of $c/L$, the (elastic) liquid column will respond acoustically. In linear systems a steady oscillation builds up with a frequency equal to the constant frequency of excitation, something that is not necessarily the case when the nonlinear orifice equation (10) is applied. The amplitude of the oscillation strongly depends on the natural frequencies of the system, i.e. the frequencies of free vibration. For a system with a closed end (orifice area $A_{or,0} = 0$) the fundamental natural frequency is $c/(4L)$ and the higher harmonics are the odd multiples of $c/(4L)$. For a system with an entirely open end (orifice area $A_{or,0} = A$) the fundamental frequency is $c/(2L)$ and the higher harmonics are the even multiples of $c/(4L)$. Figure 3 shows the first three modes of oscillation. For a system with an orifice (with area $0 < A_{or,0} < A$) the fundamental frequency is expected to be in between $c/(4L)$ and $c/(2L)$. The question is: could it – similar to a reservoir-pipe-air-vessel system [15] – become $c/(3L)$ (Fig. 4)? In that case the anti-node of the first mode is not necessarily at a boundary or at the midpoint.

Reservoir reflection point

Wave reflection does not exactly take place at $x = 0$, but somewhat into the reservoir. For short pipes this effect can be significant and instead of the nominal pipe length $L$ an effective length $L_{\text{eff}}$ should be used. Alster [16, Eq. 40] derived

$$L_{\text{eff}} = \sqrt{1 + 0.48 \left( \frac{D}{L} \right)} \approx 1 + 0.24 \frac{D}{L}$$

(16)
Figure 3. The first three basic modes of pressure oscillation relative to the steady state: closed end ($c/(4L)$, top) and open end ($c/(2L)$, bottom).

Figure 4. The first four $c/(3L)$ modes of pressure oscillation relative to the steady state with constant impedance $\hat{P}/\hat{V} = \rho c \tan(\pi/3)$ at $x = L$.

to account for added-mass effects in the (infinitely large) reservoir. The semi-empirical formula (16) is valid for circular ducts of inner diameter $D$. It differs from the classical $L_{\text{eff}}/L = 1 + 4/(3\pi) D/L \approx 1 + 0.42 D/L$ given in standard textbooks on acoustics. In general $L_{\text{eff}}$ will be dependent on the frequency of oscillation.

**Orifice reflection criterion**

For very small and very large apertures the orifice resembles a closed end and an open end, respectively. Based on linear theory (Eq. 14 with $\tau' = 0$), Wylie, Streeter and Suo [2, Section 12-5] found for the free vibration:

- if $P_0 > \frac{1}{2} \rho c V_0$ the orifice behaves like a closed end (small pipe flow velocity means small orifice opening),
- if $P_0 < \frac{1}{2} \rho c V_0$ the orifice behaves like a fully open end (large pipe flow velocity means large orifice opening),
- if $P_0 = \frac{1}{2} \rho c V_0$ there is no reflection because the orifice impedance equals the system characteristic impedance.

(Criterion (17) has not (yet) been confirmed by time-domain solutions.)

**Resonance**

In the studied pipe-flow system (Fig. 1), the orifice – with its rotating disc – is the exciter and the reservoir is the energy source. Resonance may develop when the orifice is in its most open position ($\tau(t) = \tau_{\text{max}}$) when a low-pressure wave arrives and in its most closed position ($\tau(t) = \tau_{\text{min}}$) when a high-pressure wave arrives. At resonance a standing wave – in fact an excited natural mode – dominates the system. The resonance criterion used herein is

$$\hat{V}_{\text{max}} > 2\hat{V}_{\text{exc}} \quad \text{or} \quad \hat{P}_{\text{max}} > 2\hat{P}_{\text{exc}},$$

where $\hat{V}_{\text{exc}}$ or $\hat{P}_{\text{exc}}$ is the amplitude of the exciter and $\hat{V}_{\text{max}}$ and $\hat{P}_{\text{max}}$ are the calculated (or measured) maximal velocity and pressure responses. However, it is not always possible to clearly define $\hat{V}_{\text{exc}}$ or $\hat{P}_{\text{exc}}$.

**Beat**

Prior to the establishment of steady-oscillatory flow, a beat develops. The beat is the transient condition made up of the forcing function and the initial system response to the disturbance [2, p. 309]. Except for the start-up phase, beats may also develop when the forcing frequency is close to a natural
frequency of the system (e.g. in a spring-mass system). Because frequencies cannot coincide exactly, resonance often comes in the form of a beat. Two different forcing frequencies may also result in a beat. Beat is something unsteady (not steady-oscillatory), because the amplitude varies in time.

**Wave-speed versus phase-velocity**

Care must be taken in using the terms "wave speed" and "phase velocity". The wave speed is related to the front of a travelling disturbance and the phase velocity is related to a wave train (often of infinite length) [17]. The wave speed directly follows from a time-domain analysis and has values below the speed of sound in unconfined water (about 1480 m/s), whereas the phase velocity follows from a frequency-domain analysis and is a frequency-dependent complex number with absolute values possibly going up to infinity. The phase-velocity depends on the system's end conditions, whereas the wave speed does not. For dispersive waves it is difficult to identify the exact location of a wave front. Except for linear non-dispersive systems, wave speeds and phase velocities are not the same.

**TIME-DOMAIN SOLUTION**

The waterhammer equations (1) and (2), in combination with the boundary conditions (6), and (7) or (10), are solved exactly with the MOC-based method described in [18]. The quadratic equation (10) is solved simultaneously with one linear compatibility equation. This is done symbolically, but care should be taken of possible cancellation [19]. The initial condition is the undisturbed steady flow in the pipe. The reservoir pressure, $P_{\text{res}}$, and the resistance of the valve, $\zeta_0$, determine the initial flow velocity according to relation (8).

In principle it is not handy to calculate steady-oscillatory solutions by time-marching from an undisturbed system, because it may take a (too) long time for transients to damp out. Special techniques have been developed to deal with this problem [20-24]. In addition, in finding the spectrum, each single frequency of excitation requires its own simulation. On the other hand, time-domain analysis is the only option if one wishes to include nonlinearities.

The symbolic steady-oscillatory solution of Eqs (4a), (6) with $P_{\text{res}} = 0$ and (7a) for velocity excitation with $V(x, 0) = 0$, is [25]:

$$V_v(x, t) := \dot{V} \sin\left(2\pi \frac{t}{T}\right) \frac{\cos\left(2\pi \frac{x}{cT}\right)}{\cos\left(2\pi \frac{L}{cT}\right)} \quad \text{and}$$

$$P_v(x, t) := -\rho \, c \, \dot{V} \cos\left(2\pi \frac{t}{T}\right) \frac{\sin\left(2\pi \frac{x}{cT}\right)}{\cos\left(2\pi \frac{L}{cT}\right)}. \quad (19a)$$

Resonance occurs when $2\pi \frac{L}{cT} = \frac{\pi}{2} \mod \pi$ which is equivalent to $f = \frac{c}{4L} \mod \frac{c}{2L}$. The impedance at $x = L$ is

$$|\dot{P}_v / \dot{V}_v| = \rho \, c \tan\left(2\pi \frac{L}{cT}\right).$$

The symbolic steady-oscillatory solution of Eqs (4b), (6) with $P_{\text{res}} = 0$ and (7b) for pressure excitation with $P(x, 0) = 0$, is:

$$P_p(x, t) := \hat{P} \sin\left(2\pi \frac{t}{T}\right) \frac{\sin\left(2\pi \frac{x}{cT}\right)}{\sin\left(2\pi \frac{L}{cT}\right)} \quad \text{and}$$

$$V_p(x, t) := \frac{\hat{P}}{\rho \, c} \cos\left(2\pi \frac{t}{T}\right) \frac{\cos\left(2\pi \frac{x}{cT}\right)}{\sin\left(2\pi \frac{L}{cT}\right)}. \quad (19b)$$

Resonance occurs when $2\pi \frac{L}{cT} = 0 \mod \pi$ which is equivalent to $f = \frac{c}{2L} \mod \frac{c}{2L}$. The "resonance" at $f = 0$ represents rigid-column motion in one direction, which is oscillation with an infinitely large amplitude. The impedance at $x = L$ is $|\dot{P}_p / \dot{V}_p| = \rho \, c \tan\left(2\pi \frac{L}{cT}\right)$, too.

The analytical solutions (19a) and (19b) can be combined to satisfy any linear boundary condition at $x = L$. If one chooses

$$\dot{V} := \frac{\alpha}{2} \beta_0 V_0 \quad \text{and} \quad \dot{P} := \alpha \beta_0 P_0$$

with

$$\beta_0 := \left(\frac{2P_0}{\rho c V_0} - \tan\left(2\pi \frac{L}{cT}\right)\right)^2 \quad \text{and} \quad \left(\frac{2P_0}{\rho c V_0} + \tan\left(2\pi \frac{L}{cT}\right)\right)^2,$$ \quad (19c)

in the Eqs (19a) and (19b), respectively, then the symbolic steady-oscillatory solution

$$V(x, t) := V_v(x, t) + V_p(x, t) - \dot{V} \quad \text{and}$$

$$P(x, t) := P_v(x, t) + P_p(x, t) - \hat{P}.$$
satisfies the linearised orifice equation (14) with excitation (15). Equations (19) hold for zero initial conditions, but the steady states \(V_0(x)\) and \(P_0(x)\) may simply be added. Because \(\tau' = 0\) in Eq. (15) has a non-zero average, the velocity amplitude \(\tilde{V}\) has been subtracted in Eq. (19d). The ratio of pressure to velocity excitation at \(x = L\) is \(\frac{P}{V} = \frac{2P_0}{V_0}\), from Eq. (19c). For constant impedance at \(x = L\), that is \(\tau' = 0\) in boundary condition (14), \(\left|\frac{P}{V}\right| = \frac{2P_0}{V_0}\) too. If, looking at criterion (17), \(P_0 = \frac{1}{2}\rho cV_0\) the impedance equals the characteristic impedance \(\rho c\) and there are no wave reflections at \(x = L\). For \(2\frac{P_0}{V_0} = \rho c\tan(\pi/3)\) one might expect a system with \(c/(3L)\) fundamental frequency (Fig. 4).

**FREQUENCY-DOMAIN SOLUTION**

Frequency-domain solutions can only be found for linear systems. The waterhammer equations (1) and (2), or (4), in combination with the boundary conditions (6), and (7) or (14), are solved exactly with the TMM-based approach described in [26]. For non-dispersive wave problems, TMM transfer-matrices and MOC transformation-matrices are directly related [26]. The transfer matrix relating sinusoidal velocity and pressure fluctuations at two locations (a distance \(\Delta x\) apart) is [2, Section 12-3]:

\[
\begin{pmatrix}
 v_2 \\
p_2
\end{pmatrix} =
\begin{pmatrix}
\cos(\kappa \Delta x) & -i \sin(\kappa \Delta x) / (\rho c) \\
-i \sin(\kappa \Delta x) / (\rho c) & \cos(\kappa \Delta x)
\end{pmatrix}
\begin{pmatrix}
 v_1 \\
p_1
\end{pmatrix},
\]

(20)

where the subscripts 1 and 2 indicate the positions \(x_1\) and \(x_2 = x_1 + \Delta x\) along the pipe. This matrix is to be combined with two boundary conditions to find \(v\) and \(p\) at both the upstream and downstream location. An elegant introduction to the usual notation with complex numbers is given by Goyder [27].

Frequency-domain analysis discloses natural modes and it directly leads to all steady-oscillatory solutions as a function of the forcing frequency. It is not handy for the calculation of (sharp) transient solutions, because the superposition of many natural modes of oscillation is needed to match the initial and boundary conditions.

**TEST PROBLEM**

The reservoir-pipe-orifice system is simulated with the following input data: pipe length \(L = 50\) m, wave speed \(c = 1250\) m/s, pipe flow area \(A = 31416\) mm\(^2\), orifice area \(A_{or} = 800\) mm\(^2\), discharge coefficient \(C_d = 0.6\), orifice cover fraction \(\alpha = 0.2\), reservoir pressure \(P_{res} = 2.5\) bar, mass density \(\rho = 1000\) kg/m\(^3\) and friction factor \(\lambda_f = 0\). These values correspond to an idealised laboratory system [1] with pipe inner diameter \(D = 200\) mm, wall thickness \(e = 6\) mm and the reservoir water-level \(H_{res} \approx 25\) m above the elevation of the pipe’s central axis. The small end effect included in \(L_{eff} = 50.05\) m (Eq. 16) is neglected.

**Orifice**

The orifice is a horizontal slit of 100 mm width and 8 mm height. The atmospheric pressure downstream of the orifice has been set to zero, so that \(P_0 = \Delta P_0\) and \(P = \Delta P\) in Eqs (8-10) are gauge pressures. The resistance coefficient of the orifice is \(\xi_0 = 4284\) from Eq. (8). With a frictionless pipe at constant initial pressure \(P_0 = P_{res} = 250000\) Pa, this gives a constant initial flow velocity \(V_0 = 0.342\) m/s and a steady Reynolds number \(Re_0 = 68329\). The neglected steady pressure loss due to skin friction along the pipe is \(\lambda_f (L/D)(\rho V^2/2) = 292\) Pa assuming that \(\lambda_f = 0.02\). This is small compared to the steady pressure loss \(P_0\) at the orifice.

Initially the orifice is fully open. At \(t = 0\) the outflow is interrupted by a frequency-controlled rotating disc [12, 13] that has three 10 mm sinusoidal variations in its 263 mm radius as drawn in Fig. 5. The specific function \(\tau(t)\) in Eq. (10) used to describe the orifice with rotating disc is

\[
\tau(t) = 0.9 + 0.1 \cos \left( \frac{2\pi t}{T} \right), \quad \text{so that}
\]

\[
\tau'(t) = -0.1 + 0.1 \cos \left( \frac{2\pi t}{T} \right),
\]

(21)

where \(T\) is the period of the induced oscillation (see Fig. 2). The frequency range studied herein is from 1 Hz to 25 Hz. In its most closed position at \(t/T = 1/2\) (mod 1) (Fig. 2) the orifice is a horizontal slit of 80 mm width and 8 mm height (Fig. 5) with resistance coefficient \(\xi_0 = 6693\) from Eq. (8) and flow velocity \(V_0 = 0.273\) m/s. If \(\tau(t) = 0.9\) (average \(\tau\)-value) \(\xi_0 = 5288\) and \(V_0 = 0.307\) m/s. Thus, the velocity amplitude for very slow (\(f \ll 1\) Hz) variations is \(\tilde{V} = 0.0342\) m/s. Positive water displacements were needed for PIV measurements [1], so the induced flow is pulsating (not reversing direction).

**Fundamental frequencies**

The frequency range 1 Hz – 25 Hz covers the fundamental natural frequencies. If the forcing function is a velocity (Eq. (7a)) the natural frequencies are (Eq. 19a): \(c/(4L) = 6.25\) Hz plus odd multiples of \(c/(4L)\) (even harmonics are not excited). If the forcing function is a pressure (Eq. (7b)) the natural frequencies are (Eq. 19b): \(c/(2L) = 12.5\) Hz plus even multiples of \(c/(4L)\)
(odd harmonics are not excited). If the forcing function is the quasi-steady orifice equation (10) with oscillating coefficient (11), it is a combination of velocity and pressure, and the fundamental frequency is not clear in advance. The decisive velocity according to Eq. (17) (linear theory, free vibration, Eq. 14 with \( \tau' = 0 \), small fluctuations) is \( V_{0,\text{critical}} = 2P_0/(\rho c) = 0.4 \text{ m/s} \). Here \( V_0 \) is just below 0.4 m/s and the orifice is predicted to behave (as for the reflections) like a closed end (forcing velocity); for higher velocities it behaves like an open end (forcing pressure). The analytical solution (19cd) may shed further light on this matter.

\[ \mathbf{V} \]

**Figure 5.** Front view of downstream pipe end with orifice and Svingen-type rotating disc.

SIMULATIONS

**Velocity excitation**

It is understandable that the periodically interrupted outflow is modelled by the velocity excitation (7a). First, the low-frequency behaviour is quasi-steady flow with a constant pressure \( P_0 = 250000 \text{ Pa} \) and velocities varying in between \( V_0 = 0.342 \text{ m/s} \) and \( V_0 = 0.273 \text{ m/s} \), so that \( \hat{V} = 0.0342 \text{ m/s} \). Second, criterion (17) predicts closed-end \((V = 0 \text{ and } \hat{V} = 0)\) behaviour which corresponds to velocity-excitation with its imposed \( c/(4L) \) fundamental frequency.

The analytical solution (19a) reveals that the maximum velocity amplitude \( \hat{V}_{\text{max}} \) occurs at \( x = 0 \), so that in absolute value it is (see Fig. 6)

\[ \hat{V}_{\text{max}}(f) = \frac{\hat{V}}{\cos\left(2\pi \frac{L}{c} f\right)} \]  

(22)

The MOC time-domain solution [18] gives exactly the analytical solution (19a) only if the initial conditions are the same, that is: \( V_0(x) = V_0 \) and \( P_0(x) = P_{\text{res}} - \rho c \hat{V} \sin\left(2\pi x / (cT)\right)/\cos\left(2\pi L / (cT)\right) \).

The MOC solution differs from the analytical solution, if the time-marching does not start from the steady-oscillatory state. The difference is the transient generated – without the boundary excitation \((\hat{V} = 0)\) – by the counter-balancing initial non-equilibrium \( V_0(x) = V_0 \) and

\[ P_0(x) = P_{\text{res}} + \rho c \hat{V} \sin\left(2\pi x / (cT)\right)/\cos\left(2\pi L / (cT)\right). \]

Figure 7 shows system responses for excitation frequencies of 1, 6 and 12 Hz. The dashed blue line is the steady-oscillatory solution (Eq. 19a) and the continuous red line is the MOC transient solution starting from the constant steady state \( V_0(x) = V_0 \) and \( P_0(x) = P_{\text{res}} \). The velocities (at \( x = 0 \) and \( x = L \)) and pressures (at \( x = L \)) are displayed after subtraction of the steady state and the resulting oscillations are taken relative to the excitation amplitudes \( \hat{V} \) and \( \hat{P} = \rho c \hat{V} \). The steady-oscillatory velocities at \( x = 0 \) show an anti-node with maximal amplitude \( \hat{V}_{\text{max}} \), but the corresponding pressures at \( x = L \) do not because the (first) pressure anti-node is at \( x = cT/4 = \lambda/4 \) which corresponds to \( x = L \) only at resonance.

Equation (22) gives amplification factors \( \hat{V}_{\text{max}}/\hat{V} \) of 1.03, 15.9 and 1.01 for the excitation frequencies 1, 6 and 12 Hz, respectively. Resonance occurs at 6 Hz excitation, because this is close to the first natural frequency of 6.25 Hz \([\epsilon/(4L)]\). The transient response to 1 Hz excitation (Fig. 7a) includes waterhammer (free vibration) fluctuations with its characteristic frequency of 6.25 Hz. The effect is not so large [28], because the excitation time of 0.25 s (from zero to first peak) is larger than the wave-return time of 2L/c = 0.08 s, so that there is "no full Joukowsky" \((|P - P_0| < \hat{P} = \rho c \hat{V} \text{ at } x = L)\), but about "half Joukowsky" (Fig. 7a). Full Joukowsky occurs for \( f = 12 \text{ Hz} \) (Fig. 7c) where the waterhammer effect is maximal: \(|P - P_0| / \hat{P} \) is about 1 here and \( |V - V_0| / \hat{V} \) is about 2, because velocity waves double in magnitude upon reflection at the reservoir. This makes the waterhammer effect two times larger than the steady oscillation (Fig. 7c). The frequency-mismatch (12 Hz versus 6.25 Hz) leads to whimsical signals. Symmetry with respect to \( \nu T = 6 \) (in transient \( V \) at \( x = 0 \)) indicates a beat with a period of about \( T/12 \). The steady-oscillatory pressure at \( x = L \) (dashed line) has low amplitude, because it is close to a node. Conversely, it is close to an anti-node when resonance occurs for \( f = 6 \text{ Hz} \) (Fig. 7b). The steady-oscillatory flow has very large amplitude and the transient solution becomes a beat with its amplitude about twice so large and with a period of about \( T/24 \).

The problem here is – if one is interested in the steady-oscillatory situation only – that the initial transient does not die out at all, because one deals with a frictionless system. Optional line friction (lumped or distributed) will be small and gives rise to long simulation times, which is an annoyance especially in pipe networks [20-24]. The introduction of artificial damping that fades away in time would be an option. Frequency-domain solutions are fine and exact for this linearly modelled system.
Fig. 6. Frequency response to velocity excitation ($V_{\text{max}} = \hat{V}_{\text{max}}$ and $V_{\text{exc}} = \hat{V}$).

Figure 7. Velocity response (at $x = 0$), pressure response (at $x = L$) and velocity excitation (at $x = L$) at fixed frequency $f = 1/T$.  
Continuous line = transient solution; dashed line = steady-oscillatory solution (Eq. 19a); $V_{\text{exc}} = \hat{V}$ and $P_{\text{exc}} = \hat{P} = \rho c \hat{V}$;

a. $f = 1/T = 1$ Hz,  
b. $f = 1/T = 6$ Hz,  
c. $f = 1/T = 12$ Hz.
Figure 8. Velocity (at $x = 0$), pressure (at $x = L$) and velocity (at $x = L$) responses for nonlinear orifice excitation at fixed frequency $f = 1/T$. Continuous line = transient solution; dashed line = steady-oscillatory solution (forcing velocity, Eq. 19a);

- **a.** $f = 1/T = 1$ Hz,
- **b.** $f = 1/T = 6$ Hz,
- **c.** $f = 1/T = 12$ Hz.

Figure 9. Velocity (at $x = 0$), pressure (at $x = L$) and velocity (at $x = L$) responses for nonlinear orifice excitation at fixed frequency $f = 1/T = 6$ Hz. Continuous line = transient solution; dashed line = steady-oscillatory solution (forcing pressure, Eq. 19b).
On the other hand, it is not wise to ignore the transient phase preceding the steady-oscillatory state, because the system pressures can be up to two times higher than the steady-oscillatory ones.

**Nonlinear orifice excitation**

This (Eqs 10 and 21) is the real excitation (except for any unsteady effects [29], which might be of significance near resonance). Figure 8 shows system responses for excitation frequencies of again 1, 6 and 12 Hz. The dashed blue line is the steady-oscillatory solution for forcing velocity (Eq. 19a) and the continuous red line is the MOC transient solution starting from the constant steady state $V_0(x) = V_0$ and $P_0(x) = P_{res}$. The velocities (at $x = 0$ and $x = L$) and pressures (at $x = L$) are not displayed after subtraction of the steady state, because in nonlinear systems the unsteady solution may depend on the initial steady state. Here the boundary condition (10) depends on the initial condition in a nonlinear way. Also, there are no clear excitation amplitudes $\hat{V}$ or $\hat{P}$ anymore to normalise velocities and pressures. The resulting oscillations are shown as they are, except that the steady-oscillatory solutions are shifted in time to make them in phase with the transient solutions.

One remarkable result is that for 1 Hz and 12 Hz excitation the transient dies out within two forcing cycles and the remaining steady-oscillatory solution is that of velocity excitation with $\hat{V} = 0.0342$ m/s. The initiated oscillation at the orifice is steady almost immediately at 1 Hz (Fig. 8a) and at 12 Hz (which is close to anti-resonance) the transient pressure peak is strongly damped (Fig. 8c). At 6 Hz it is a transient velocity drop that is strongly damped (Fig. 8b) and, compared to Fig. 7b, there is no strong resonance anymore. The velocity excitation is more or less replaced by a pressure excitation with $\hat{P} = \rho c \hat{V} = 42.7$ kPa as displayed in Fig. 9, where the steady-oscillatory solution (dashed line) is according to Eq. (19b).

**Linear orifice excitation**

Because the amplitude $\hat{V}$ of the forcing velocity is (exactly) 10% of the steady velocity $V_0$ herein, it is justified to use the linear approximation (14) with excitation (21). This linearisation is used in frequency-domain analyses. The only concern is that at resonance the amplitudes may become too large for the approximation to be valid, in particular if these occur at $x = L$. Indeed, the results obtained with either relationships (10) or (14) are very close to each other (not shown), except for a subtle difference in the velocities at the orifice as depicted in Figs 10 and 11. The steady-oscillatory state computed with MOC [18] matches the analytical solution (19cd) (not shown).

**Discussion**

The linear solution in Fig. 10a shows that at resonance the downstream velocity directly drops to a constant value of 0.307 m/s, which corresponds to steady flow with the orifice 90% open ($\tau = 0.9$ and $\tau' = -0.1$) as expected. The velocity and pressure oscillations are $\lambda/4$ modes (not shown). From Eq. (14) – with $v = 0$ – it is evident that the pressure fluctuation $p$ directly follows the disc rotation represented by $\tau'$, such that $p = -2\tau'(t) P_0$.

![Figure 10](image1.png)

**Figure 10.** Velocity (at $x = L$) response for (a) linear and (b) nonlinear orifice excitation at resonance frequency $f = 1/T = 6.25$ Hz. Continuous line = transient solution; dashed line = steady-oscillatory solution (forcing pressure, Eq. 19b).

![Figure 11](image2.png)

**Figure 11.** Pressure (at $x = L$) response for (a) linear and (b) nonlinear orifice excitation at anti-resonance frequency $f = 1/T = 12.5$ Hz. Continuous line = transient solution; dashed line = steady-oscillatory solution (forcing velocity, Eq. 19a).

The nonlinear solution in Fig. 10b shows that at resonance the downstream velocity is not constant, but oscillating at twice the resonance frequency around an average value below the expected 0.307 m/s (dashed blue line). The latter nonlinear effect is known as acoustic streaming [30]. Both frequency-
doubling and DC velocity shift result from Eqs (7a) and (10) in combination with the equality

\[
\left[ V_0 + \dot{V} \sin \left( 2\pi \frac{t}{T} \right) \right]^2 = V_0^2 + \frac{1}{2} V^2 + 2V_0 \dot{V} \sin \left( 2\pi \frac{t}{T} \right) - \frac{1}{2} V^2 \cos \left( 4\pi \frac{t}{T} \right)
\]

(23)

The frequency-doubling effect caused by an orifice has already been observed by Wood et al. [31], however without any further explanation. Average orifice aperture and average steady flow rate do not correspond anymore in the (quasi-steady) nonlinear orifice.

Away from resonance, with 1 Hz and 12 Hz excitation in Figs 8a and 8c, the transient dies out remarkably fast because the orifice introduces much damping and, compared to pure velocity and pressure excitation, a more realistic model of the interaction between system dynamics and orifice is used. For the 1 Hz case there is no transient at all, and for both the 1 Hz and 12 Hz cases the analytical solution (19a) gives excellent predictions of the periodic state. Near resonance the analytical solution (19b) is better. The change from velocity excitation away from resonance to pressure excitation near resonance can be fully explained from the analytical solution (19cd).

An apparent paradox in the linear model: at resonance (\(f = 6.25\) Hz) the excitation is a forcing pressure and hence it is a \(c/(4L)\) system; here the forcing velocity maintains it as a \(c/(2L)\) system.

CONCLUSIONS

A frequency-controlled rotating valve that generates oscillating pipe flow in the acoustic range has been modelled in the time domain in four different ways: as a forcing velocity, as a forcing pressure, as a linear oscillating resistance and as a nonlinear oscillating resistance. Analytical steady-oscillatory solutions are presented for the first three cases. For the studied test case, away from resonance the forcing velocity is a simple and good model, except that fluid transients spoil the numerical solution. Near resonance the periodic behaviour transforms to that of a forcing pressure. The linear and nonlinear valve resistance models assure that fluid transients quickly damp out in the numerical simulations. The nonlinear model predicts near resonance a shift in the average outflow velocity and a doubling of its frequency of oscillation, where the linear model does not. The analytical solutions help in understanding and interpreting the results of the numerical simulations.

ACKNOWLEDGEMENTS

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REFERENCES


APPENDIX A: NONLINEAR SOLUTION
Meshless Water Hammer

Single pipe, no friction, nonlinear orifice excitation.

Input data:

Deltarees tests (idealised system)

rotating valve, no friction, damping due to valve resistance.

\[ \begin{align*}
&\text{Initial conditions} \\
&\begin{array}{l}
Q_0 = 0.341646 \text{ m} \\
V_0 = 0.292482 \text{ m} \\
L = 500 \text{ mm} \\
R = 200 \text{ mm} \\
\end{array}
\end{align*} \]

Longitudinal wave speed

\[ c = \frac{\sqrt{\left(1 - \frac{2c}{V_0} \right) c}}{\left(1 - \frac{2c}{V_0} \right)} \]

\[ c = 1254.99 \frac{\text{m}}{\text{s}} \]

\[ c = 125.5 \text{ Hz} \]

Matrices of coefficients (for waterhammer)

Note:

The matrices are made dimensionless through multiplication with the appropriate unit.

\[ A_{11} = 1 \\
A_{12} = \frac{1}{\rho V_0} \\
B_{11} = 1 \\
B_{12} = 1 \frac{\text{kg}}{\text{m}^2 \text{s}^2} \]

Transformation matrix \( T \)

\[ \begin{pmatrix} 1 & 0 \\ 0 & \frac{\text{kg}}{\text{m}^2 \text{s}^2} \end{pmatrix} \]

Transformation matrix \( S \)

\[ S = \begin{pmatrix} 0.5 & 0.5 \\ -0.25 \times 10^5 & -0.25 \times 10^5 \end{pmatrix} \]

\[ S = \begin{pmatrix} 1 & \frac{81 \times 10^{-7}}{1 - 4 \times 10^{-7}} \\ 1 - 4 \times 10^{-7} \end{pmatrix} \]

Initial conditions

Note:

Vectors are made dimensionless through multiplication with the appropriate unit.

\[ \text{Velocity:} \quad \begin{pmatrix} \phi_{1 \theta} \phi_{2 \theta} \end{pmatrix} = \begin{pmatrix} \text{P} \text{V} \end{pmatrix} \]

\[ \text{Scaling} \]

\[ T_{\theta} = \begin{pmatrix} 1 & 0 \rule{0pt}{2ex} \\ 0 & \frac{\text{kg}}{\text{m}^2 \text{s}^2} \end{pmatrix} \]

\[ T_{\theta}^{-1} = \begin{pmatrix} 1 & \frac{81 \times 10^{-7}}{1 - 4 \times 10^{-7}} \\ 1 - 4 \times 10^{-7} \end{pmatrix} \]

Boundary-condition matrices

Note:

Matrices are made dimensionless through multiplication with the appropriate unit.

\[ \text{Right-hand side vector} \]

Note:

Vectors are made dimensionless through multiplication with the appropriate unit.

\[ \text{Matrix } DS \]

\[ DS = \begin{pmatrix} 0.625 \times 10^5 & -6.25 \times 10^5 \\ 0.5 & 0.5 \end{pmatrix} \]
Coefficients α and β

\[ z = 0 \]
\[ z = L \]
\[ a_{12}(t) = \frac{\partial S_{012}}{\partial S_{\alpha \beta}}, \quad a_{12}(t) = 1 \]
\[ a_{21}(t) = \frac{\partial S_{021}}{\partial S_{\alpha \beta}}, \quad a_{21}(t) = -1 \]
\[ \beta_{1}(t) = \frac{1}{2}, \quad \beta_{2}(t) = 1.6 \times 10^{-6} \]
\[ \beta_{2}(t) = \frac{1}{2}, \quad \beta_{2}(t) = 2 \]

Constant coefficients α and β to speed up the calculation

\[ z = 0 \]
\[ z = L \]
\[ a_{12} = a_{12}(t) \]
\[ a_{21} = a_{21}(t) \]
\[ a_{21} = a_{21}(t) \]
\[ a_{21} = a_{21}(t) \]

Constant coefficients for oscillating valve at \( z = L \)

\[ a_{1} = \begin{pmatrix} 1 \end{pmatrix} \]
\[ a_{2} = \begin{pmatrix} 1 \end{pmatrix} \]
\[ a_{2} = 8 \times 10^{-6} \]

Time-dependent coefficients for oscillating valve at \( z = L \)

\[ (0) = 0.8 + 0.1 \cos \left(\frac{\pi}{2} \right) \]
\[ \tau(t) = \frac{-1 + 0.1 \cos(2 \pi T)}{2} \]
\[ b_{1} = 1 \]
\[ b_{2} = -0.2 \]
\[ b_{3} = 0 \]

Calculation intervals

\[ T_{1} > 0 \text{ s} \]
\[ T_{2} > 0 \text{ s} \]
\[ T_{1} = \frac{1}{10} \text{ s} \]
\[ T_{2} = 1 \text{ s} \]
\[ T_{1} = 0.003 \text{ s} \]
\[ T_{2} = 0.003 \text{ s} \]
\[ N = 400 \]
\[ i = 1, \ldots, 20 \]
\[ T = T_{1} + (i - 1) \times T_{2} \]

Analytical steady-oscillatory solution

\[ V_{V_{0}} = V_{V_{0}}(z, \lambda) \]
\[ V_{V_{L}} = V_{V_{L}}(\lambda, \lambda) \]
\[ V_{V_{V_{0}}} = V_{V_{V_{0}}}(z, \lambda) \]
\[ V_{V_{V_{L}}} = V_{V_{V_{L}}}(\lambda, \lambda) \]

\[ P_{P_{0}} = P_{P_{0}}(z, \lambda) \]
\[ P_{P_{L}} = P_{P_{L}}(\lambda, \lambda) \]
\[ P_{P_{V_{0}}} = P_{P_{V_{0}}}(z, \lambda) \]
\[ P_{P_{V_{L}}} = P_{P_{V_{L}}}(\lambda, \lambda) \]

Results (in nested arrays)

<table>
<thead>
<tr>
<th>( i )</th>
<th>( \eta_{i} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>( 0 )</td>
<td>( \eta_{0} = \eta_{\text{BOUNDARY}}(z, \lambda) )</td>
</tr>
<tr>
<td>( \lambda )</td>
<td>( \eta_{\lambda} = \eta_{\text{BOUNDARY}}(z, \lambda) )</td>
</tr>
</tbody>
</table>

Pressures

\[ P_{P_{0}} = (S_{\varphi_{0}}) \quad P_{P_{L}} = (S_{\varphi_{L}}) \]

Wave travel times

\[ \text{Recursion "coast to coast"} \]

\[ \eta_{1} = \text{BOUNDARY}(z, \lambda) \]
\[ \eta_{2} = \eta_{1} + \eta_{1} + \eta_{1} + \eta_{1} \]
\[ \eta_{3} = \eta_{2} + \eta_{2} \]
\[ \eta_{4} = \eta_{1} + \eta_{1} \]

Fluid velocities

\[ V_{B_{0}} = (S_{\varphi_{0}}) \quad V_{L} = (S_{\varphi_{L}}) \]

Results (write to file)

\[ \text{PRECISION} = 16 \quad \text{PRINCOLWIDTH} = 32 \]

\[ \text{RESV} = \text{augment} \left( \frac{1}{i \times V_{0}} \right) \]
\[ \text{RESP} = \text{augment} \left( \frac{1}{i \times P_{0}} \right) \]
\[ \text{RESVL = augment} \left( \frac{1}{i \times V_{L}} \right) \]
\[ \text{RESPL = augment} \left( \frac{1}{i \times P_{L}} \right) \]
Solution in interior points

\[ \eta_{\text{INTERIOR}}(x, t) = \begin{cases} \eta_{\text{BOUNDARY}}(x, t), & 0 < x < L \\ \eta, & x = L \\ \eta_{\text{BOUNDARY}}(x, t), & L < x < L + L \\ \eta, & x = L + L \end{cases} \]

Calculation intervals (repeated) for time history

\[ \Delta t = \frac{T}{N} \quad N = 480 \]

Analytical steady-oscillatory solution

\[ V_v(x, t) = V_v(x, t) \]

\[ P_{pz}(x, t) = P_{pz}(x, t) \]

Results (in nested array) \[ \eta_{\text{INTERIOR}}(x, t) \]

Calculation intervals for spatial distribution

\[ \Delta z = \frac{L}{N} \quad N = 100 \]

Analytical steady-oscillatory solution

\[ V_v(x, t) = V_v(x, t) \]

\[ P_{pz}(x, t) = P_{pz}(x, t) \]

Results (in nested array) \[ \eta_{\text{INTERIOR}}(x, t) \]
Results (write to file)

PENPRECISION = 16
PENCOLWIDTH = 32

RESVt = augment(Vt)

WRITEPRN("d:\winmcad\mcad11\results\CASA-10-38-AppA_Vt.prn") = RESVt

RESPt = augment(Pt)

WRITEPRN("d:\winmcad\mcad11\results\CASA-10-38-AppA_Pt.prn") = RESPt
APPENDIX B: LINEAR SOLUTION
Meshless Water Hammer

Single pipe, no friction, linearised orifice excitation.

Input data:

Deltaris MRI tests (idealised system)

Rotating valve, no friction, damping due to valve resistance.

Initial pressure and velocity:

- Deltares MRI tests
- Input data:
  - Single pipe, no friction, linearised orifice excitation.
  - Meshless Water Hammer

Steady-state pressure loss over valve:

Initial pressure and velocity:

Longitudinal wave speed:

Analytical steady-oscillatory solutions

Forcing velocity

$\rho = \frac{\rho}{\omega V_{\text{amp}}} \quad P_{\text{amp}} = 4.271 \times 10^3 \text{Pa}$

$V_{\text{osc}}(t) = V_0 + V_{\text{amp}} \sin \left( \frac{2 \pi t}{T} \right)$

$P_{\text{osc}}(t) = P_0 + \rho \omega V_{\text{amp}} \sin \left( \frac{2 \pi t}{T} \right)$

$V_{\text{amp}}(t) = \frac{V_{\text{osc}}(t)}{\omega} \quad P_{\text{amp}}(t) = \frac{P_{\text{osc}}(t)}{\omega}$

$V_{\text{amp}}(t) = \frac{V_{\text{osc}}(t)}{\omega} \quad P_{\text{amp}}(t) = \frac{P_{\text{osc}}(t)}{\omega}$

At $z = 0$:

$V_{\text{amp}}(t) = \frac{V_{\text{osc}}(t)}{\omega} \quad P_{\text{amp}}(t) = \frac{P_{\text{osc}}(t)}{\omega}$

Phase shift $\phi$ at resonance

$V_{\text{osc}}(t) = \frac{V_{\text{osc}}(t)}{\omega}$

$P_{\text{osc}}(t) = \frac{P_{\text{osc}}(t)}{\omega}$

$V_{\text{amp}}(t) = \frac{V_{\text{osc}}(t)}{\omega} \quad P_{\text{amp}}(t) = \frac{P_{\text{osc}}(t)}{\omega}$

$z < cT/4 = \lambda/4$

$z = cT/4 = \lambda/4$

$z > cT/4 = \lambda/4$

Forcing pressure

$P_{\text{osc}}(t) = P_0 + \rho \omega V_{\text{osc}} \sin \left( \frac{2 \pi t}{T} \right)$

$V_{\text{osc}}(t) = V_0 + V_{\text{amp}} \sin \left( \frac{2 \pi t}{T} \right)$

$P_{\text{amp}}(t) = \frac{P_{\text{osc}}(t)}{\omega}$

$V_{\text{amp}}(t) = \frac{V_{\text{osc}}(t)}{\omega}$

Phase shift $\phi$ at resonance

$z < cT/4 = \lambda/4$

$z = cT/4 = \lambda/4$

$z > cT/4 = \lambda/4$
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Initial conditions

Note:

Vectors are made dimensionless through multiplication with the appropriate unit.

\[ \begin{align*}
V_0 \times \mathbf{V} & = 0, \\
P_0 \times \mathbf{P} & = 0
\end{align*} \]

Boundary-condition matrices

Note:

Matrices are made dimensionless through multiplication with the appropriate unit.

\[ D \begin{pmatrix} 0 & 0 \\ 0 & 1 \end{pmatrix} \]

Right-hand side vector

Note:

Vectors are made dimensionless through multiplication with the appropriate unit.

\[ \mathbf{q} \]
Analytical steady-oscillatory solution

\[ Vv_0 = \text{Vo}(x, \eta) \]
\[ VvL = \text{Vo}(L, \eta) \]
\[ VvR = \text{Vo}(1, \eta) \]

Results in interior points

\[ η_0 = \text{BOUNDARY}(x, \eta) \]
\[ η_1 = \text{BOUNDARY}(x, \eta) \]

Fluid velocities

\[ VvL = \text{Vo}(x, \eta) \]
\[ Vv0 = \text{Vo}(x, \eta) \]

Pressures

\[ P0L = \text{Vo}(x, \eta) \]
\[ P0R = \text{Vo}(x, \eta) \]

Results (in nested arrays)

\[ Vv0 = \left[ \begin{array}{c}
0.28 \\
0.30 \\
0.32 \\
0.34 \\
0.36 \\
\end{array} \right] \]

Mean outflow and inflow

\[ \text{max}(VL) = 0.288026 \quad \text{min}(VL) = 0.271142 \quad \text{max}(V0) = 0.246986 \]

Analytical steady-oscillatory solution

\[ VvL = \text{Vo}(x, \eta) \]
\[ P0L = \text{Vo}(x, \eta) \]

Results (in nested array)

\[ VvL = \text{Vo}(x, \eta) \]

Calculation intervals (repeated) for time history

\[ x = \frac{L}{2} \]
\[ x = 37.5 \text{m} \]
\[ T1 = 0 \text{s} \]
\[ T2 = 1 \text{s} \]
\[ At = 0.00208333333333 \text{s} \]
\[ M = \text{sqrt}(\frac{T1}{At}) \]
\[ N = 800 \]
\[ J = 1 \quad N = 20 \]
\[ t = T1 + (t - 1) \cdot At \]
Mathcad 11 CASA-10-38B.mcd

Results (write to file)

\[ V_{zt} = \text{augment} \begin{pmatrix} \frac{1}{t} & V_z \end{pmatrix} \]

\[ \text{WRITEPRN}(\text{resultscasa-10-38-appb_Vz.prn}) = \text{RESVz} \]

\[ \frac{P_t}{\text{bar}} = \text{augment} \begin{pmatrix} \frac{1}{t} & P_z \end{pmatrix} \]

\[ \text{WRITEPRN}(\text{resultscasa-10-38-appb_Pz.prn}) = \text{RESPz} \]

Calculation intervals for spatial distribution

\[ n = 3 \cdot T \quad n = 0.25s \]

\[ N_z = 100, \quad \Delta z = \frac{L}{N_z} = 0.5m \]

\[ j = 1, \quad N_j = 1, \quad \Delta z = j \cdot \Delta z \]

Mathcad 11 CASA-10-38B.mcd

Results (in nested array)

\[ \eta_t = \text{augment} \begin{pmatrix} \text{INTERIOR} (j, z) \end{pmatrix} \]

Analytical steady-oscillatory solution

\[ V_{vt} = V_c(t, z) \]

\[ P_{pt} = P_c(t, z) \]

Pressure

\[ \eta_j = \text{augment} \begin{pmatrix} \text{INTERIOR} (j, z) \end{pmatrix} \]

Fluid velocity

\[ \eta_j = \text{augment} \begin{pmatrix} \text{INTERIOR} (j, z) \end{pmatrix} \]

Results (write to file)

\[ V_{vt} = \text{augment} \begin{pmatrix} \frac{1}{t} & V_{zt} \end{pmatrix} \]

\[ \text{WRITEPRN}(\text{resultscasa-10-38-appb_Vt.prn}) = \text{RESVt} \]

\[ \frac{P_t}{\text{bar}} = \text{augment} \begin{pmatrix} \frac{1}{t} & P_{zt} \end{pmatrix} \]

\[ \text{WRITEPRN}(\text{resultscasa-10-38-appb_Pt.prn}) = \text{RESPt} \]
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