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# Sliding Mode and PID Control for Pressure Regulation in Hydraulic Pipelines Under Leakage Disturbances

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# التحكم بالنمط الانزلاقي والتحكم التناسبي التكاملي التفاضلي (PID) لتنظيم الضغط في خطوط الأنابيب الهيدروليكية تحت اضطرابات التسرب

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**Abstract:**
A pipeline loses pressure to friction all the time, and to a leak some of the time. This study develops a nonlinear lumped-parameter model of a 1000 m hydraulic pipeline that accounts for momentum dynamics, Darcy–Weisbach friction, an orifice-type leakage disturbance, nonlinear valve pressure loss, and first-order actuator lag. The model is used to design and compare two pressure-regulation strategies: a PID controller with a causal derivative term and a sliding-mode controller based on an integral sliding surface formulated around a linearized equilibrium point. The controlled variable is the pressure immediately upstream of the regulating valve, with a desired value of 3 bar. Both controllers are evaluated against an uncontrolled fixed-valve baseline under an identical 350-second simulation scenario consisting of a startup period, a 100-second active-leak interval, and a post-leak recovery period. The results show that both controllers maintain the pressure close to its setpoint during the leakage event and reduce the mean absolute pressure error by more than 95% compared with the uncontrolled case. PID provides slightly faster recovery after the leak is removed, whereas the sliding-mode controller produces gentler valve motion and achieves comparable disturbance rejection. These findings demonstrate the potential of sliding-mode control for improving pressure regulation and robustness in hydraulic pipeline systems subject to leakage disturbances.

**Keywords**: pipeline pressure control, sliding mode control, PID control, leakage disturbance, hydraulic systems, nonlinear valve dynamics, actuator lag.

**الملخص**
يفقد خط الأنابيب جزءًا من ضغطه بسبب الاحتكاك بشكل دائم، وبسبب التسرب في بعض الأحيان. تعرض هذه الدراسة نموذجًا غير خطي ذا معاملات مُجمَّعة لخط أنابيب هيدروليكي بطول 1000 متر، يأخذ في الحسبان ديناميكية الزخم، والاحتكاك وفق معادلة دارسي-فايسباخ، واضطراب تسرب من نوع الفتحة (Orifice)، وفقد ضغط غير خطي عبر الصمام، وتأخرًا زمنيًا من الدرجة الأولى للمشغل (Actuator). يُستخدم النموذج لتصميم ومقارنة استراتيجيتين لتنظيم الضغط: متحكم PID بمشتقة سببية (Causal)، ومتحكم بالنمط الانزلاقي يستند إلى سطح انزلاق تكاملي مُصاغ حول نقطة توازن مُخطَّطة خطيًا. المتغير المتحكَّم فيه هو الضغط مباشرة قبل صمام التنظيم، بقيمة مرغوبة تبلغ 3 بار. جرى تقييم كلا المتحكمين مقابل حالة مرجعية غير متحكم فيها (صمام ثابت) ضمن سيناريو محاكاة مدته 350 ثانية يتضمن فترة بدء تشغيل، وفترة تسرب نشط مدتها 100 ثانية، وفترة تعافٍ بعد انتهاء التسرب. أظهرت النتائج أن كلا المتحكمين يحافظان على الضغط قريبًا من قيمته المستهدفة أثناء حدث التسرب، ويقللان متوسط خطأ الضغط المطلق بنسبة تزيد على 95% مقارنة بالحالة غير المتحكم فيها. ويوفر المتحكم PID تعافيًا أسرع نسبيًا بعد إزالة التسرب، في حين يُنتج المتحكم بالنمط الانزلاقي حركة أكثر سلاسة للصمام مع تحقيق رفض اضطراب مماثل. تُظهر هذه النتائج الإمكانات التي يوفرها التحكم بالنمط الانزلاقي في تحسين تنظيم الضغط ومتانة النظام في أنابيب الهيدروليك المعرضة لاضطرابات التسرب.

 

**الكلمات المفتاحية**: التحكم بضغط الأنابيب، التحكم بالنمط الانزلاقي، التحكم PID، اضطراب التسرب، الأنظمة الهيدروليكية، ديناميكية الصمام غير الخطية، تأخر المشغل.

**Abbreviations**

**Table 1. List of symbols and abbreviations used in the model.**

| Symbol | Description | Symbol | Description |
|---|---|---|---|
| Q | Volumetric flow rate [m³/s] | x | Fractional valve opening [0–1] |
| A | Pipe cross-sectional area [m²] | u | Controller command to actuator [0–1] |
| L, D | Pipe length, diameter [m] | τ | Actuator time constant [s] |
| ρ, SG | Fluid density [kg/m³], specific gravity | e | Pressure error [Pa] |
| f | Darcy friction factor | s | Sliding surface value |
| $C_v, C_{leak}$ | Valve/leak flow coefficients | λ, k, φ | SMC surface, reaching-gain, boundary-layer parameters |
| $K_f,\ K_{mom}$ | Friction and momentum lumped gains | $Kp, Ki, Kd$ | PID gains |
| $\Delta P_f, \Delta P_{leak}, \Delta P_v$ | Friction/leak / valve pressure drops [Pa] | $Q_0, x_0, u_0$ | Equilibrium flow, valve opening, command |

## Introduction

A pipeline carrying fluid from a fixed-pressure source to a downstream outlet has to keep its internal pressure near a target. Two things work against it. Friction along the pipe wall is always present. A leak, if one opens, acts like an extra orifice pulling pressure out of the line. That is the problem this paper looks at: hold the pressure just upstream of a regulating valve at 3 bar, starting from a 5 bar inlet and a 1 bar outlet, first from zero flow, then through a 100-second leak, then after the leak is sealed again.

A PID loop can hold a setpoint like this, but its valve command comes entirely from error that has already built up. Something sudden, a leak opening, only gets corrected after the pressure has already moved. Sliding mode control (SMC) works differently: it defines a surface in the state space of the plant and drives the system onto it through a switching action. Once on the surface, the closed loop follows the surface dynamics more or less regardless of how large the disturbance was that pushed it away. There is a price for that, and we come back to it later: the surface has to be derived from a model of the plant, and the switching term needs smoothing through a boundary layer, or it excites high-frequency chattering in the valve.

What follows is a four-term nonlinear model of the pipeline: momentum balance, friction, leak, and valve pressure drop, coupled to a first-order actuator lag. We linearize it about an equilibrium point to design an augmented-derivative PID controller and an integral-sliding-surface SMC. Then we run both, plus a fixed-valve baseline with no control at all, through the same three-phase leak scenario. That way, startup transient, disturbance rejection, and post-leak recovery can all be read off one simulation, under identical conditions.

## Related Work

A few things from prior work shape the comparison here. Industrial documentation on pipeline pressure regulators [6] points out a practical weakness of error-only PID valve control: the valve only reacts to error that has already built up, so a pressure surge can take several sampling cycles to correct. The same pattern shows up in pipeline flow-control studies directly. A PID controller regulating heavy-oil flow through motor-pump vibration stayed stable under 50% and 100% pipe leakage, though the paper reports a peak undershoot as large as 24% for the worse case [1]. A PLC-based fuzzy-PID scheme for oil-pipeline transport outperformed a cascade-PID baseline on settling time and overshoot [4]. Both results line up with what we see for the uncontrolled and PID cases here.

A separate line of work goes after the controller side directly, with sliding-mode designs. A PSO-tuned comparison of conventional PID, fractional-order PID, and SMC on an electrohydraulic actuator under leakage-related uncertainty [2], and a PSO-tuned hybrid sliding-mode-PID controller built and simulated in MATLAB/Simulink [3], both report better disturbance tolerance from the sliding-mode element than PID gives on its own. Closest to our application, an adaptive fuzzy sliding-mode scheme regulates airway pressure in a mechanical ventilator against an unknown leak, benchmarked against classical PID and SMC [5]. Sliding-mode observers, rather than controllers, have also been used on the leak side of the problem: a first-order and super-twisting sliding-mode observer pair locates and sizes leaks in a plastic pipeline from flow and pressure measurements alone [7], and a fuzzy-PID ARX-Laguerre observer does something similar for early leak estimation [8]. Closer to the controller side again, a fixed-time integral sliding-mode active-disturbance-rejection scheme regulating chamber pressure in an electro-hydraulic load simulator reports smoother, more accurate pressure tracking than plain PID under repeated disturbance cycles [9], the same qualitative gap between PID and sliding-mode behavior we find here. None of these works, though, derives its plant from first-principles pipe/valve/leak equations with an equilibrium-linearized sliding surface, and none tests the resulting controllers against a genuinely uncontrolled baseline over an explicitly staged, three-phase leak. That gap is what we are filling here.

## System Modeling

The dynamic behavior of the pipeline is governed by the conservation of momentum, which relates the rate of change of flow to the net pressure difference across the pipe segment. This net pressure is reduced by three distinct loss mechanisms: friction along the pipe wall, leakage through an orifice, and the pressure drop across the control valve. The following subsections present each component in detail, together with the physical assumptions underlying their mathematical formulation

### A. Momentum Conservation Equation

This equation expresses that the fluid acceleration is proportional to the net driving pressure (inlet minus outlet and losses) and inversely proportional to the fluid inertia (ρL/A). The term $\frac{dQ}{dt}$ represents the transient response of the flow, which is the key state variable in this lumped-parameter model

$$\frac{dQ}{dt} = \left(\frac{A}{(\rho.L)}\right)[\, P_{in} - P_{out} - \Delta P_f - \Delta P_{leak} - \Delta P_{\ v} \,] \tag{1}$$

Q is the volumetric flow rate, ρ the fluid density, and A the pipe cross-sectional area ($A = \pi D^2/4$).

### B. Friction Loss (Darcy–Weisbach)

The Darcy-Weisbach formulation is used here because it captures the quadratic dependence of friction loss on flow rate, which dominates in turbulent pipe flow. The lumped coefficient $K_f$ aggregates the geometric and fluid properties, allowing the friction term to be written compactly in the momentum equation.

$$\Delta P_f = K_f \cdot Q \cdot |Q|, \quad K_f = \frac{f \cdot L \cdot \rho}{(2 \cdot D \cdot A^2)} \tag{2}$$

The absolute-value form keeps the loss pointed the right way if the flow ever reverses.

### C. Leak Loss (Orifice Model)

The leak is modeled as an orifice with an effective flow coefficient $C_{leak}$, which is activated only during the disturbance interval (100–200 s). This formulation assumes the leak behaves like a fixed opening to atmosphere, with pressure drop scaling quadratically with flow through the leak path.

$$\Delta P_{leak} = SG \cdot \left(\frac{Q}{C_{leak}}\right)^2 \quad \text{(active only while the leak is present; zero otherwise)} \tag{3}$$

### D. Control Valve Pressure Drop

This equation models the valve as a variable orifice whose effective flow area scale with the opening fraction x. The quadratic dependence on $\frac{Q}{X}$ reflects the standard orifice relationship, where pressure drop increases with the square of flow velocity. The specific gravity SG accounts for the fluid density relative to water, ensuring the coefficient $C_v$ remains dimensionless. As x approaches zero, the pressure drop grows rapidly, which introduces strong nonlinearity near the closed-valve condition and motivates the careful gain selection in Section 3

$$\Delta P_v = SG \cdot \left(\frac{Q}{(C_v \cdot x)}\right)^2 \tag{4}$$

$x \in [0, 1]$ is the fractional valve opening and $C_v$ the valve flow coefficient.

 

### E. Actuator Dynamics

The first-order lag captures the finite response speed of the physical actuator, whether pneumatic, hydraulic, or electric. The time constant τ represents the combined effect of actuator inertia, friction, and any built-in filtering. This dynamic limitation means the controller cannot instantaneously reposition the valve, even if the computed command u changes abruptly. In practice, typical industrial control valves exhibit time constants in the range of 1–5 seconds, and the value τ = 2.0 s used here falls within that realistic band. The actuator model also implicitly enforces smooth valve motion, which helps reduce mechanical wear and avoids exciting high-frequency structural modes in the piping system.

$$\tau \cdot \frac{dx}{dt} = u - x \tag{5}$$

The valve actuator is modeled as a first-order lag with time constant τ between the controller command u and the actual valve position x.

### F. Model Parameters

**Table 2. Model parameters used in the simulation.**

| Parameter | Symbol | Value | Unit |
|---|---|---|---|
| Pipe length | L | 1000 | m |
| Pipe diameter | D | 0.3 | m |
| Fluid density | ρ | 850 | kg/m³ |
| Darcy friction factor | f | 0.02 | — |
| Specific gravity | SG | 0.85 | — |
| Valve flow coefficient | $C_v$ | 0.05 | — |
| Leak flow coefficient (active) | $C_{leak}$ | 0.0008 | — |
| Actuator time constant | τ | 2.0 | s |
| Inlet pressure | $P_{in}$ | 5.0 | bar |
| Outlet pressure | $P_{out}$ | 1.0 | bar |
| Pressure setpoint | $P_{desired}$ | 3.0 | bar |

### G. Modeling Assumptions

A few simplifications keep the model tractable. All are fairly standard for lumped-parameter hydraulic studies. The fluid is treated as incompressible, and the flow as one-dimensional along the pipe axis. Temperature effects on density, viscosity, and the friction factor are ignored, so f is held constant instead of being tied to Reynolds number. The leak is a single lumped orifice rather than a loss distributed along the pipe. Valve and actuator behavior come from a static orifice equation and a first-order lag; stiction and backlash are left out. Pipe wall elasticity, water-hammer effects, is not modeled at all. That is a reasonable simplification for the multi-second transients studied here, though it would need revisiting for fast surge analysis.

## Control Design

### A. Equilibrium Point and Linearization

Both controllers are built around the equilibrium point matching the 3 bar setpoint with no leak present. At equilibrium $\frac{dQ}{dt} = 0$ and $\frac{dx}{dt} = 0$, so the friction loss alone has to absorb the full pressure gap between the inlet and the setpoint:

$\Delta P_{f,0} = P_{in} - P_{desired} = 2\ bar$. That gives:

$$Q_0 = \sqrt{\frac{\Delta P_{f,0}}{K_f}} \approx 0.1878 \frac{m^3}{s} \tag{6}$$

The valve drop at that point is: $\Delta P_{v,0} = P_{in} - P_{out} - \Delta P_{f,0} = 2$ bar. The equilibrium valve opening follows from it:

 

$$x_0 = \frac{Q_0}{\left(C_v \cdot \sqrt{\frac{\Delta P_{v,0}}{SG}}\right)} \approx 0.00774, \quad u_0 = x_0 \tag{7}$$

Notice how small $x_0$ is. At the 3 bar setpoint the valve sits barely open, because the valve equation is very sensitive near that point: the pressure drops scales as $1/x^2$. That sensitivity is a big part of why the plant is so nonlinear close to equilibrium, and it is the reason we treat the SMC boundary layer carefully in Section C.

Linearizing the momentum and actuator equations about $(Q_0, x_0)$, and writing out the sensed-pressure sensitivity, gives the state matrices below (state vector $[\Delta Q, \Delta x]$, input $\Delta u$, all in deviation coordinates):

$$A = [-0.354 \;\; 4.296\,; \;\; 0 \;\; -0.500\,], \;\; B = [\,0\,; \;\; 0.500\,], \;\; \frac{\partial P_{sensed}}{\partial Q} = -2.130 \times 10^6 Pa \cdot \frac{s}{m^3} \tag{8}$$

The large off-diagonal term, $\partial\Delta Q/\partial\Delta x \approx 4.30$, says the flow rate is very sensitive to small changes in valve opening near equilibrium. That fits with how small $x_0$ turned out to be. The diagonal term, −0.354, points to mild self-stabilization in the flow dynamics on their own: more flow brings more friction, and that friction acts as natural damping. We use this linear model only to pick gains. Every result reported later comes from the full nonlinear model in Section 2.

**B. PID Controller**

A PID controller is the standard three-term feedback law: it drives an actuator based on the error between a measured signal and its setpoint, combining three actions that each do a different job. The proportional term reacts to how large the error is right now, and is what gives the loop its basic pull toward the setpoint. The integral term accumulates error over time, so even a small, persistent offset eventually forces the actuator to move enough to close it, which is what removes steady-state error a proportional-only loop would leave behind. The derivative term reacts to how fast the error is changing rather than its size, which lets the controller anticipate where the error is heading and damp out overshoot before it grows. Together, $\mathrm{u} = \mathrm{Kp} \cdot \mathrm{e} + \mathrm{Ki} \cdot \int \mathrm{e\,dt} + \mathrm{Kd} \cdot (de/dt)$, the three terms trade off responsiveness against stability, and tuning a PID loop is largely about finding gains that react quickly without ringing or saturating the actuator [10].

In this system, the PID loop regulates the sensed upstream pressure, $P_{sensed} = P_{in} - \Delta P_f - \Delta P_{leak}$, toward the setpoint, the same kind of loop used to regulate pipeline flow and pressure with plain PID in [1] and [4]. There is a subtlety here that is easy to get backwards, and an early version of this controller did: opening the valve raises Q, raising Q raises friction loss, and raising friction loss actually lowers $P_{sensed}$. The plant gain between valve opening and sensed pressure is negative. The first pass at this controller used the more intuitive error definition, $e = P_{desired} - P_{sensed}$, which drove the valve the wrong way: whenever the pressure ran high, the controller closed the valve further instead of opening it, which only made $P_{sensed}$ climb toward Pin and locked the loop at the wrong end of its range. Simulating the response made the sign error obvious almost immediately, and the fix is to flip the definition:

$$e = P_{sensed} - P_{desired} \tag{9}$$

with the control law

$$u = Kp \cdot e + Ki \cdot \int e\,dt + Kd \cdot (de/dt), \;\; u \in [0,1] \tag{10}$$

Numerical differentiation tends to amplify noise, so the derivative term is computed analytically instead, and causally, from the current state rather than the control action about to be taken: $\frac{de}{dt} = -2 \cdot K_f \cdot Q \cdot \left(\frac{dQ}{dt}\right)_{estimated}$.

This uses the current valve position, before the new command is applied. A conditional integral clamp, an anti-windup, freezes the integral term whenever the computed action is saturated in a direction that would push it further into saturation. The gains settled on after tuning:

$$Kp = 8 \times 10^{-6}, Ki = 3 \times 10^{-8}, Kd = 5 \times 10^{-6}. \tag{11}$$

**C. Sliding Mode Controller**

Sliding mode control takes a different approach from PID. Instead of shaping the actuator command directly from the error, it first defines a surface in the state space of the system, a combination of the state variables that equals zero along some desired reduced-order behavior, and then drives the system onto that surface through a switching control action, regardless of where it started or how large the disturbance pushing it away is. Once the state reaches the surface, it stays there (this is the "sliding" phase), and the closed loop then behaves according to the surface's own dynamics rather than the plant's original, often more complicated, dynamics. The appeal is robustness: as long as a disturbance enters the system through the same channel as the control input, matched uncertainty, the sliding motion rejects it almost completely, which is why SMC is a common choice when a plant is nonlinear or its parameters are not known precisely. The trade-off is that the switching action, if implemented as a hard sign function, causes chattering, rapid back-and-forth actuator motion, which is usually softened with a boundary layer at some cost to tracking precision [11].

The SMC is built on the same three-state deviation model, $z = [\Delta Q, \Delta x, \xi], where\ \Delta Q = Q - Q^0, \Delta x = x - x^0, and$ ξ is the integral of the bar-scaled pressure error. The sliding surface is defined as:

$$s = \Delta x - \lambda \cdot \xi \tag{12}$$

The minus sign on the integral term is not arbitrary, and it is worth explaining why, since the same sign mistake that affected the first PID attempt showed up again here in a different form. When the pressure error keeps accumulating positively, meaning pressure too high, the valve has to open further: $\Delta x > 0$. A surface defined as s = $\Delta x + \lambda \cdot \xi$ instead (plus rather than minus) looked equally plausible on paper, but drove the valve shut and kept it shut for the whole run once tested, because the reaching law then pushed $\Delta x$ in the wrong direction as ξ grew. The version above, with the minus sign, is what actually enforces the correct opening direction at s = 0. Applying the reaching law $\frac{ds}{dt} = -k \cdot sat\left(\frac{s}{\varphi}\right)$, a saturation function in place of the discontinuous sign function to cut chattering, and solving for u gives a closed-form control law:

$$u = x - k \cdot \tau \cdot sat\left(\frac{s}{\varphi}\right) + \lambda \cdot \tau \cdot \left(\frac{\frac{dP_{sensed}}{dQ}}{1e5}\right) \cdot \Delta Q \tag{13}$$

The first term is feedforward that cancels the actuator's own decay. The second is the reaching/switching term; it does the actual work of rejecting the leak. The third compensates for how flow rate feeds into the accumulated pressure error. After tuning, the values settled on were $\lambda = 0.08, k = 0.5, \varphi = 0.15$.

One numerical wrinkle came up during simulation. The saturation-based switching term is discontinuous enough that adaptive-step ODE solvers kept stalling on it, shrinking their step size near the switching surface instead of converging, a numerical form of chattering. Every simulation reported here uses a fixed-step 4th-order Runge-Kutta integrator instead, at $\Delta t$ = 0.02 s, and that sidesteps the problem.

### D. Stability Sketch

The stability argument here follows the usual reaching-condition logic. Take V = ½s² as a candidate Lyapunov function. Its derivative under the reaching law is:

$$V = s \cdot \dot{s} = s \cdot (-k \cdot sat(s/\varphi)) = -k \cdot s \cdot sat(s/\varphi) \leq 0 \tag{14}$$

Equality holds only at s = 0. So V never increases, and the surface pulls the state toward it from anywhere outside the boundary layer |s| < φ. Inside that layer, sat(s/φ) reduces to s/φ, and the dynamics become a locally exponentially stable first-order system: ṡ = −(k/φ) ·s. That is the trade a finite φ makes: a small, bounded chattering band at steady state, in exchange for no high-frequency switching. This argument only covers local attractivity for the nominal, disturbance-free linear plant. We check robustness against the leak itself empirically, in Section 5, rather than through a matched-disturbance bound, since the leak enters the full nonlinear plant multiplicatively through $\Delta P_{leak}$, not as a clean additive input.

### Simulation Scenario

Every controller runs on the full nonlinear model over a 350 s window, starting from zero flow (Q = 0) and an arbitrary initial valve opening (x = 0.3). A leak switches on at t = 100 s ($C_{leak}$ = 0.0008) and off again at t = 200 s. That splits the run into three phases: pre-leak (0 to100 s), leak (100 to200 s), and post-leak (200 to350 s). We also include an open-loop baseline, valve held fixed at its equilibrium opening x0 the whole time, for reference.

## Results and Discussion

### 1. Pressure Response

Figure 1 shows the sensed pressure for all three cases. All of them dip below the setpoint at the start, as flow ramps up from zero. That dip is not a controller weakness. It is a physical floor: friction loss, and therefore the sensed-pressure deviation, is exactly zero at zero flow and only grows as flow builds. No controller can avoid it without assuming an unrealistic nonzero starting flow. During the leak (red region), the open-loop baseline settles around 2.75 to 2.8 bar and stays there until the leak ends. PID and SMC, by contrast, actively push back and hold close to 3 bar the whole time.

 

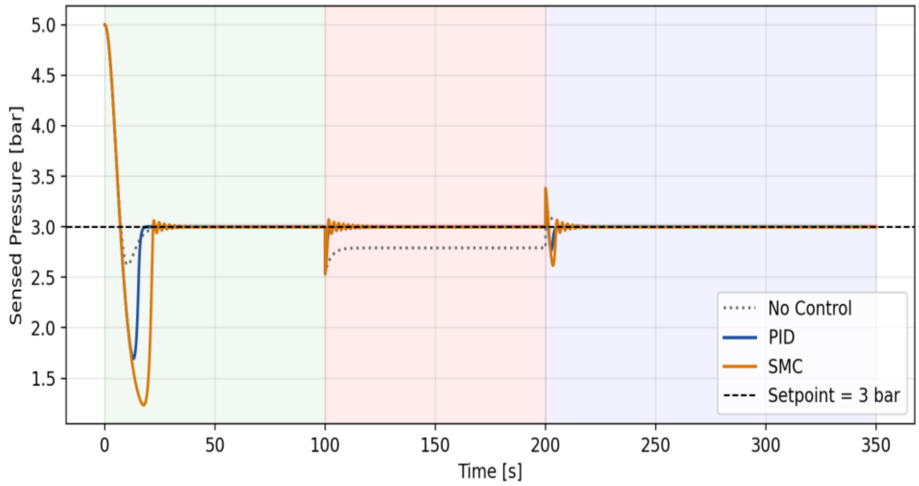


**Figure 1.** Sensed pressure response — No Control (dotted gray), PID (blue), SMC (orange). Shaded regions: green = pre-leak, red = leak active, blue = post-leak.

### 2. Response Characteristics

Table 3 lists the classical step-response numbers from the initial transient (0 → 3 bar).

**Table 3. Step-response characteristics for the three control strategies.**

| Characteristic | No Control | PID | SMC |
|---|---|---|---|
| Rise time (10%–90%) | 4.81 s | 4.67 s | 4.67 s |
| Peak value (undershoot) | 2.62 bar | 1.70 bar | 1.24 bar |
| Peak time | 10.4 s | 13.0 s | 17.6 s |
| Overshoot | 18.8% | 65.2% | 88.2% |
| Settling time (±2%)* | ~204 s | ~204 s | ~205 s |
| Steady-state error (pre-leak) | ~0 mbar | 0.05 mbar | ~0 mbar |

*Settling time in all three cases is really set by the ringing after leak removal (t = 200 s). The initial transient settles on its own within roughly 20 to 25 s.

### 3. Flow Rate Response

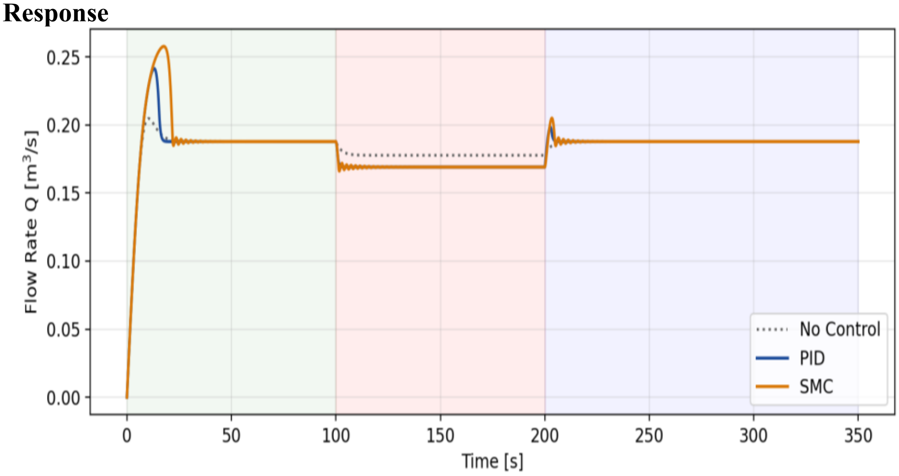


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**Figure 2**. Volumetric flow rate Q for all three cases.

Flow rate (Figure 2) looks fairly similar across all three cases. Q comes mostly from the momentum equation itself, not from the controller directly. The open-loop case settles at a noticeably lower flow during the leak, about 0.17 $m^3/s$, and never gets it back. Both closed-loop controllers push flow toward the nominal 0.188 $m^3/s$ despite the leak.

 

**4. Valve Opening (Controller Action)**

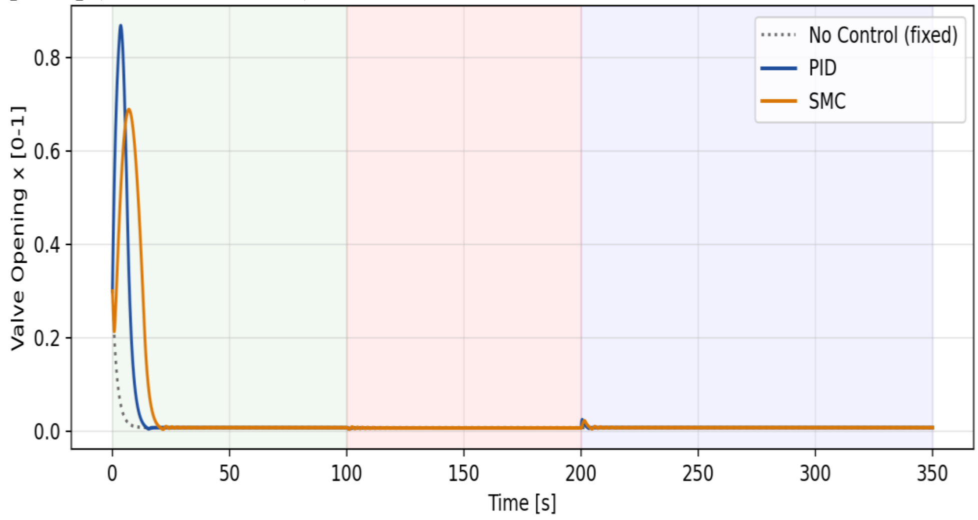


**Figure 3.** Valve opening x — the direct controller action.

Figure 3 shows the controller action directly. PID opens the valve harder and faster during the initial transient, peaking around 0.87, than SMC does, peaking around 0.69. That lines up with PID's higher overshoot. At leak onset and leak removal, both controllers show short, clearly defined valve pulses marking their intervention. The open-loop valve position barely moves at all.

**5. Pressure Loss Components**

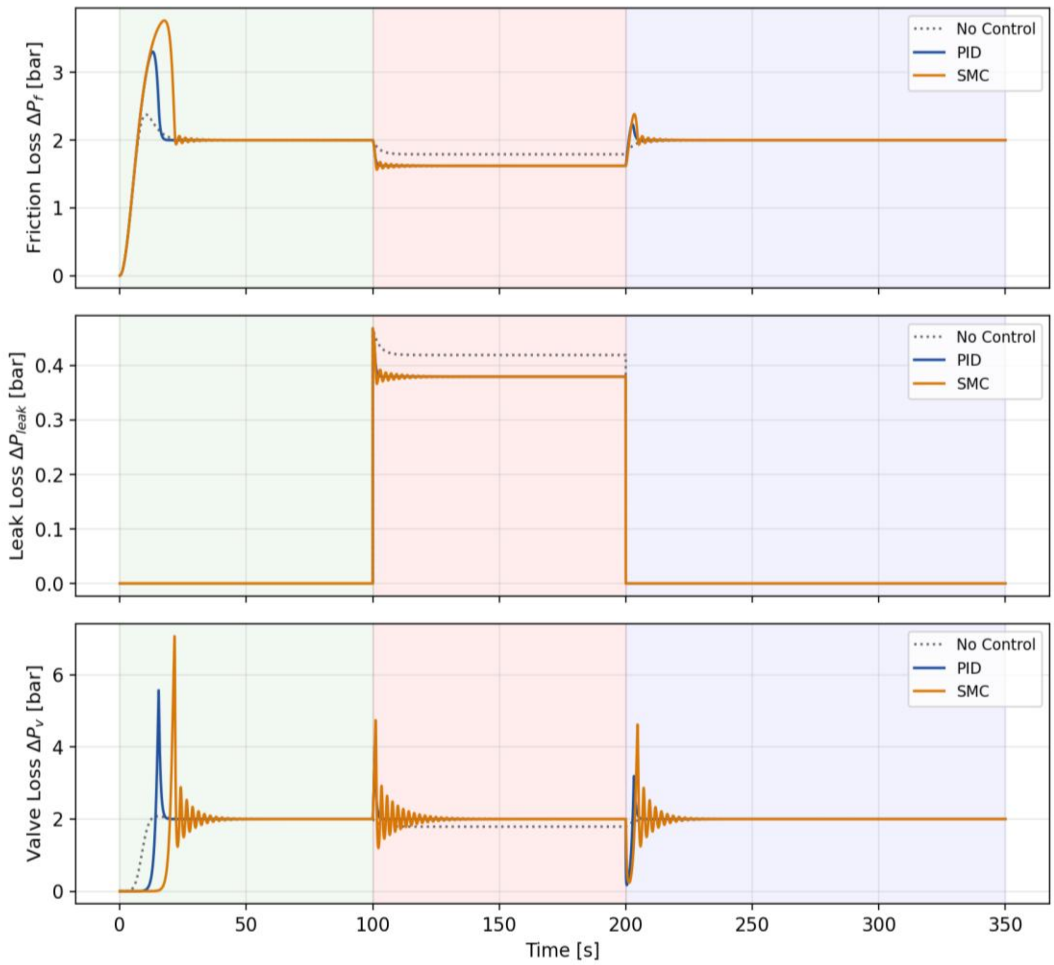


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**Figure 4.** Pressure loss components — friction (top), leak (middle), and valve (bottom) — for all three cases.

 

Figure 4 breaks the total pressure drop into its three physical pieces. The leak loss (middle panel) is nonzero only in the 100–200 s window by construction, and it is nearly identical across all three cases right at the moment the leak begins. None of the controllers can react instantly to a step disturbance. The valve loss (bottom panel) is where the controllers actually separate from each other: PID and SMC both show sharp transient spikes in $\Delta P_v$ lining up exactly with their corrective valve motion, while the open-loop $\Delta P_v$ barely changes.

**6. Disturbance Rejection and Recovery**

**Table 4. Disturbance-rejection and recovery metrics for the three control strategies.**

| Metric | No Control | PID | SMC |
|---|---|---|---|
| Mean \|error\| during leak (100 to 200 s) | 0.216 bar | 0.005 bar | 0.007 bar |
| Recovers to setpoint during leak? | No | Yes | Yes |
| Recovery time after leak ends (±2%) | 3.65 s | 1.00 s | 1.25 s |

This is where the three strategies really separate. The open-loop baseline runs a mean absolute pressure error more than 40 times larger than either closed-loop controller during the leak. PID recovers a little faster than SMC once the leak is gone, 1.00 s against 1.25 s, but SMC's valve motion is visibly gentler (Figure 3). That should mean less mechanical wear on a real actuator over time.

One thing worth flagging: the instantaneous peak deviation at leak onset, 0.468 bar, is the same for all three cases in Table 4. That is not a coincidence. At the exact instant the leak turns on, none of the controllers have had time to react, since valve position and flow rate are continuous states that cannot jump. The initial jump in $\Delta P_{leak}$ passes straight through into the sensed pressure no matter what controller is running. The number that actually distinguishes the controllers is the mean error over the whole disturbance, not that first instant.

**Limitations and Future Work**

A few limitations are worth being upfront about. Both controllers are tuned around one linearized operating point (Section 3.1), so performance could drop off for setpoints or leak sizes far from what was simulated here; gain scheduling or some form of adaptive retuning would probably be needed to cover a wider operating range. The SMC boundary-layer parameter φ trades chattering suppression against tracking precision, tuned here by hand rather than solved for analytically. The model also leaves out sensor noise, quantization, and communication delay. These tend to hurt sliding-mode performance more than PID in practice, because SMC leans on relatively high-gain switching near the surface. And the actuator is an idealized first-order lag: no rate limits, stiction, or hysteresis. A real control valve would probably smooth out the sharp position pulses seen in Figure 3.

The linearized model from Section 3.1 sets up a few natural next steps. An explicit Linear-Quadratic-Regulator with integral action could be benchmarked against PID and SMC on the same plant and leak scenario. A Model Predictive Control formulation is probably the more interesting direction, though: it would build the valve-opening constraint $x \in [0, 1]$ directly into the optimization instead of handling it after the fact through saturation and anti-windup, which should cut down the large overshoot both PID and SMC show in Table 3. Beyond that, running any of these controllers on a physical or high-fidelity test rig would show how they actually cope with the non-idealities listed above, rather than just how they cope in simulation.

**Conclusion**

We built a nonlinear pipeline pressure-control model, covering momentum conservation, Darcy-Weisbach friction, an orifice-type leak, a nonlinear control valve, and first-order actuator dynamics, and used it to design and compare PID and sliding-mode controllers under a leak disturbance. Both cut the mean pressure error during the leak by more than 95% compared to an uncontrolled baseline. PID, with a causally-derived derivative term added in, settled and recovered a bit faster. SMC matched it closely on disturbance rejection while moving the valve more smoothly, at the cost of more design complexity and some sensitivity to chattering during simulation. The overshoot both controllers show at startup turned out to be a physical floor set by the zero-flow initial condition, not a controller shortcoming. Next on the list: extending the comparison to LQR and Model Predictive Control, both able to handle actuator constraints directly inside the optimization rather than after the fact.

 

## Compliance with ethical standards

*Disclosure of conflict of interest*

The authors declare that they have no conflict of interest.

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