# Improved Control of Pressure Reducing Valves in Water Distribution Networks

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Bogumil Ulanicki
De Montfort University

# Improved Control of Pressure Reducing Valves in Water Distribution Networks

Simon L. Prescott[1] and Bogumil Ulanicki[2]

**Abstract:** The behavior of transients in water pipe networks is well understood but the influence of modulating control valves on this behavior is less well known. Experimental work on networks supplied through pressure reducing valves (PRVs) has demonstrated that, in certain conditions, undesirable phenomena such as sustained or slowly decaying oscillation and large pressure overshoot can occur. This paper presents results from modeling studies to investigate interaction between PRVs and water network transients. Transient pipe network models incorporating random demand are combined with a behavioral PRV model to demonstrate how the response of the system to changes in demand can produce large or persistent pressure variations, similar to those seen in practical experiments. A proportional-integral-derivative (PID) control mechanism, to replace the existing PRV hydraulic controller, is proposed and this alternative controller is shown to significantly improve the network response. PID controllers are commonly used in industrial settings and the methods described are easy to implement in practice.

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## Introduction

United Kingdom water companies commonly split their networks into district metered areas (DMAs) to facilitate monitoring and control activities. DMAs are small areas and are closed apart from designated inlets and outlets through which flow is monitored. Hydraulically controlled pressure reducing valves (PRVs) maintain a specified outlet pressure, irrespective of a higher fluctuating inlet pressure, and are often used to supply and control pressure in DMAs. Under the 15 min sampling adopted by water companies, PRVs are observed to perform well in normal operating conditions and excessive pressure transients only appear following extreme events such as large or sudden demand changes. However, when data are sampled more quickly, the PRV outlet pressure can be seen to fluctuate significantly with minor demand changes in the network. Examples of these two phenomena are given in Fig. 1, which show large pressure transients caused by a sudden demand change and persistent smaller pressure fluctuations caused by the poor response of a PRV to smaller, more routine demand changes. Both of these examples show data collected during field experiments using a 30 s sampling rate.

Pressure transients caused by the combined behavior of a network and PRV propagate through a DMA and result in water quality problems, a higher number of pipe bursts, and premature wear of the pipe infrastructure. It is impossible to eliminate demand changes from a network so it is important to control PRVs appropriately to minimize their impact on the system. The main aim of this paper is to demonstrate the performance of a control system that can be implemented in place of the standard control loop used on PRVs. This controller consists of a gain that amplifies the PRV output error, an integral term that takes into account the amount of time errors are present and a derivative term that is dependent on the rate of change of the errors. These controllers are, therefore, known as proportional, integral, derivative (PID) controllers and are used in many industrial applications.

The interaction between automatic control valves and transients and the resulting behavior have been investigated in several publications. Bergant et al. (2001) investigated the effect of valve closing time on the transient response in a pipeline and compared measured data with a simulation model. The effect of automatic control valves in a real pipe network was shown by Brunone and Morelli (1999), and used to estimate the friction in a transient model, and the effects of having multiple PRVs in a water distribution system were investigated by Prescott and Ulanicki (2004). Control of network transient dynamics using electrohydraulic servovalves was described in Driels (1975) and the application of PID control to a theoretical single input DMA was demonstrated by Prescott et al. (2005). The present paper extends this work to multi-input DMAs and also considers a real water network configuration.

It is well known that steady-state friction, used in the traditional method of characteristics transient model, underestimates attenuation of fast transient pressure waves. Several modifications to the standard model have been proposed to address this discrepancy, most notably by Brunone et al. (1991), and are reviewed and compared by Bergant et al. (1999). In this paper, unsteady

[1]Research Fellow, Water Software Systems, De Montfort Univ., Queens Building, The Gateway, Leicester, LE1 9BH, U.K. E-mail: slp@dmu.ac.uk

[2]Director of Research, Water Software Systems, De Montfort Univ., Queens Building, The Gateway, Leicester, LE1 9BH, U.K. E-mail: bul@dmu.ac.uk

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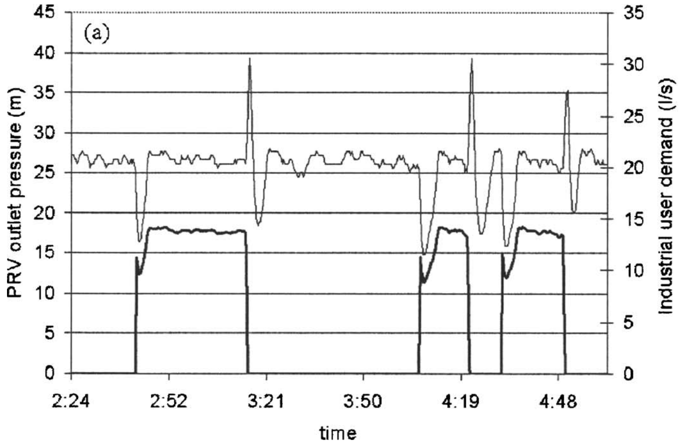

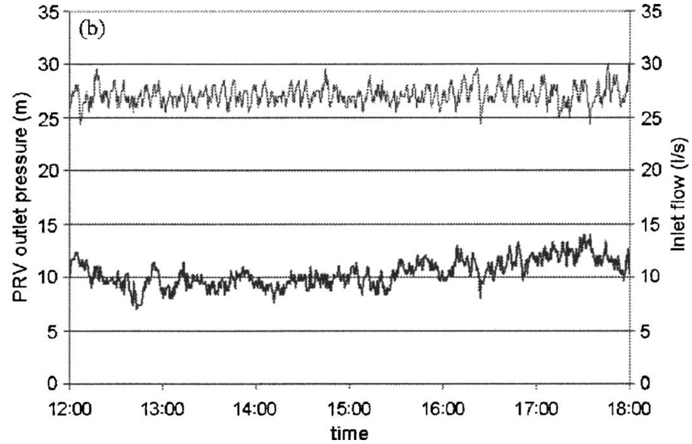


**Fig. 1.** Examples of undesirable transient behavior: (a) response to sudden demand change at large user; (b) persistent oscillations caused by small demand changes. PRV outlet pressure (thin line) and flow (thick line).

friction effects are not considered so the worst case scenario (in terms of sustained transients) is considered.

Rather than developing models of transient pipe networks, the main focus of this paper is to show how the dynamic problems shown in Fig. 1 occur, and investigate the suitability of using a PID controller to reduce the effects of these problems. The simplicity and flexibility of PID controllers make them the most obvious choice of controller for almost any industrial process and the most common tuning techniques will be discussed. The performance of a suitably tuned controller will be demonstrated through modeling studies and a method for physical implementation of the controller will also be briefly described.

The hydraulic networks simulated throughout the paper include random demands. The demand model is based on those of Buchberger and Wells (1996) and Garcia et al. (2003) and generates a demand profile for a single residence from a series of demand events. The interval between events is randomly generated, as is the event duration and intensity, and the sum of these individual residence demands gives the total network demand. One industrial user, with substantial demand changes, is also added. These two demand profiles combined enable both scenarios of Fig. 1 to be simulated and the PRV response investigated.

Throughout the paper, all pipes are represented by the standard method of characteristics transient model and a PRV is described by a behavioral model. Each is described more thoroughly in the following section. The random demand model and the PID controller are then briefly introduced and case studies shown. The paper finishes with some conclusions from the work.

## Models

In this section, the models used throughout the paper are presented. The usual one-dimensional transient pipe model, given in a range of texts (e.g., Chaudhry 1987; Streeter et al. 1998), is used to represent each pipe and the behavioral PRV model of Prescott and Ulanicki (2003) is used. This model was shown to represent the main PRV dynamics well in comparison to a full phenomenological model and is considered sufficient for this work. A demand model, comprised of random household events is used to replicate the real world situation of changing demand.

### *Sample Network*

Transient flow in a closed conduit is represented by the basic waterhammer momentum and continuity equations in space, $x$ and time, $t$

$$g\frac{\partial h}{\partial x} + \frac{1}{A}\frac{\partial q}{\partial t} + \frac{f}{2DA^2}q|q| = 0$$

$$\frac{\partial h}{\partial t} + \frac{a^2}{gA}\frac{\partial q}{\partial x} = 0 \tag{1}$$

where $h$=piezometric head; $q$=fluid flow; $D$=inside pipe diameter; $A$= pipe area; $f$=Darcy–Weisbach friction factor; $a$=speed of the shock wave; and $g$=acceleration due to gravity. In this work, the roughness of a pipe is specified and the friction factor determined using the Colebrook–White equation.

The waterhammer equations are traditionally solved by the method of characteristics (Streeter et al. 1998), which transforms the two parts of Eq. (1) to produce the compatability equations, valid along $C^+$ and $C^-$ characteristic lines defined by $dx/dt = \pm a$

$$dh \pm \frac{a}{gA}dq \pm \frac{f}{2gDA^2}q|q|dx = 0 \tag{2}$$

This is solved using a finite-difference approach where the space–time grid is selected to ensure that the characteristic lines are satisfied. Solving Eq. (2) for a single pipe gives a response similar to a decaying square wave but real networks consist of many interconnected pipes and the aggregated response is typically slower and smoother. For this reason, a theoretical DMA is represented by a mesh of pipes, shown with one PRV supply, in Fig. 2. Each node of the mesh has a demand and pressure-dependent leakage and the head and flow along each pipe of the DMA is governed by the transient model [Eq. (2)]. Leakage is assumed to occur mostly at pipe connections and is, therefore, distributed at every demand node across the DMA. The head at the nodes is determined by combining the upstream and downstream compatability equations for all pipes connected to the node, as described in Streeter et al. (1998) and Karney and McInnis (1992). All pipes are identical and their parameters are length=200 m, diameter=150 mm, roughness=0.5 mm, and wavespeed=1,200 m/s. Node 1 is a fixed head source of 50 m

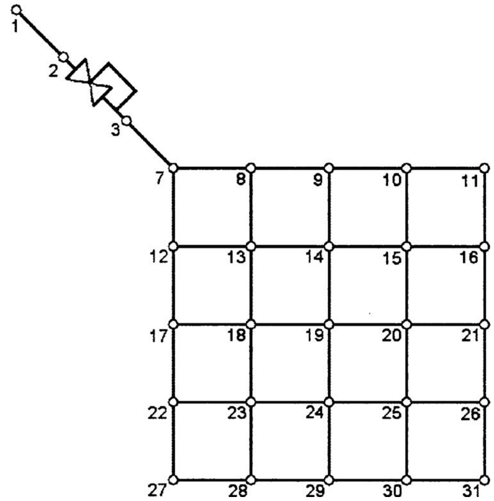


**Fig. 2.** Theoretical network supplied with PRV

and the leakage at Nodes 7–31 is given by $q_{\text{leak}}=0.01p^{0.5}$ L/s. The demands at Nodes 7–31 are determined by the demand model, described later. It is generally accepted that both demand and leakage can be pressure dependent and the nature of pressure dependency varies for different types of demand (or leakage). The model in this paper uses fixed demand and pressure-dependent leakage so there is an element of both types of outflow. Increasing (decreasing) the amount of pressure dependency will result in increased (decreased) damping in the system and therefore reduce (increase) oscillations and settling time.

### PRV Equations

The purpose of a PRV is to maintain a specified steady outlet pressure irrespective of fluctuations in a higher inlet pressure. This task is performed with a hydraulic control loop which senses the outlet pressure and injects or removes water, via a needle valve, from the control space (a chamber above the main valve element) which opens or closes the main valve as required. Several PRV models of varying complexity exist but, for the purposes of this paper, a PRV is represented with a behavioral model similar to that of Prescott and Ulanicki (2003)

$$q_m = C_v(x_m)\sqrt{h_{\text{in}} - h_{\text{out}}} \tag{3}$$

$$\dot{x}_m = \frac{q_3}{A_{\text{cs}}(x_m)} \tag{4}$$

$$q_3 = \begin{cases} \alpha_{\text{open}}(h_{\text{set}} - h_{\text{out}}) & \text{if } \dot{x}_m \geq 0 \\ \alpha_{\text{close}}(h_{\text{set}} - h_{\text{out}}) & \text{if } \dot{x}_m < 0 \end{cases} \tag{5}$$

where $q_m$=PRV flow; $h_{\text{in}}$=PRV inlet head; $h_{\text{out}}$=PRV outlet head; $q_3$=flow into or out of the control space (through a needle valve); $x_m$=opening of the valve; $h_{\text{set}}$=PRV set point; $C_v$=valve capacity (as a function of valve opening); and $A_{\text{cs}}$=cross-sectional area of the control space (as a function of valve opening). Eq. (3) relates the flow through the PRV to the head loss across it and opening, Eq. (4) represents the opening/closing rate of the PRV as a function of the flow into/out of the control space. This flow is determined by Eq. (5) and is dependent on the outlet head error and the needle valve setting. This setting is often different depending on the direction of needle valve flow so two values, $\alpha_{\text{open}}$ and $\alpha_{\text{close}}$, are given depending on whether the PRV is opening or closing.

The model, Eqs. (3)–(5), can be used for any PRV. For a particular PRV, it is necessary to specify relationships for valve capacity and cross-sectional area of the control space, as a function of valve opening, $x_m$. These are either available from PRV manufacturers or can be measured. In this paper, the experimentally obtained equations of Prescott and Ulanicki (2003) are used

$$C_v = 0.021 - 0.0296e^{-51.1x_m} + 0.0109e^{-261x_m} - 0.0032e^{-683.2x_m} + 0.0009e^{-399.5x_m}$$

$$A_{\text{cs}} = \frac{1}{3700(0.02732 - x_m)} \tag{6}$$

### Demand Model

This paper considers the response of a water network and PRV to two types of behavior—a large sudden demand in a network and a smaller, but constantly fluctuating, demand profile. The first of these, typically, occurs at a single large industrial user and is often easy to identify while the second is the aggregation of many users, which are much more difficult to individually distinguish. In all the simulations, the large industrial user demand is simply specified over time and the accumulated small demands are represented by a random demand model.

Demand events in a residence can be represented by a series of rectangular pulses each with an arrival time, an intensity, and a duration (Buchberger and Wells 1996). The arrival times can be represented by a Poisson process and the duration of events is characterized by many short events and a few longer ones (Garcia et al. 2003). This suggests the use of the exponential distribution for both the time between events and the duration of the events. Garcia et al. (2003) used a Weibull distribution for event intensity although in the present work, a normal distribution is used since the flow rate can realistically be assumed to have this type of characteristic.

The theoretical DMA of Fig. 2 is assumed to have 80 houses at each of Nodes 7–31, giving a total of 2,000 houses. A series of demand events is generated for each house, over a simulation time of 500 s. These are added to get the total demand for a given node then the process repeated for all 25 nodes. A single industrial user is assumed to be at Node 23 with a demand of 0 L/s for the first 100 s of simulation, then increasing to 8 L/s over a 5 s period, remaining at this value until 300 s and then decreasing back to 0 L/s, again over a 5 s ramp.

In the theoretical models, the parameters of the probability distributions used to generate the demand were chosen to give a total demand with mean and variance approximately 10 L/s and 1.65—the sample mean and variance of the recorded flow data in Fig. 1. To obtain this, the mean interevent time was 5 min at each house, average event duration was set to 15 s and the average event intensity was 6 L/min. The resulting profile was added to the industrial use to give the total demand shown in Fig. 3. This has a mean of 10.37 L/s and variance of 1.69.

### Combined PRV and Network

To determine the combined behavior of a DMA with random demand and a PRV, they have been modeled as blocks in Simulink and these models combined. A screenshot of the combined system, showing the interconnection between the blocks, is given in Fig. 4.

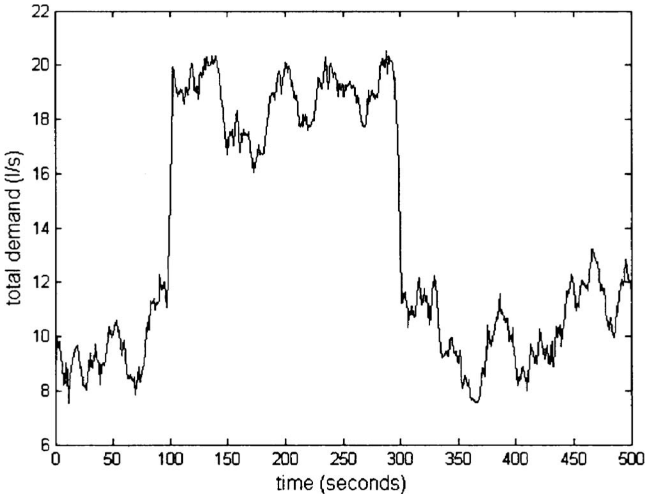


**Fig. 3.** Demand profile across the DMA

The PRV block incorporates the pipe upstream of the PRV so the inputs to this block are PRV set point, head at Node 1 and the PRV flow determined from the network. The output from the PRV is outlet pressure that forms an input to the DMA block. For clarity, the needle valve flow $q_3$, the outlet pressure error ($h_{set}-h_{out}$), and the PRV opening $x_m$, are also shown but these could be incorporated into the PRV block. The other input to the DMA is demand, generated by the model already described. The head and flow at each node of the DMA form inputs to the DMA for the next timestep and the average inlet flow at the beginning and end of a timestep is fed back to the PRV block as the PRV flow.

## Control of Pressure Reducing Valves

A PRV has a hydraulic control loop that senses outlet pressure and adjusts the valve opening as required. Apart from the desired outlet pressure, the only adjustable mechanism in this control loop is the needle valve setting, represented by $\alpha_{open}$ and $\alpha_{close}$ in Eq. (5). This determines the flow into or out of the control space and, therefore, the rate of PRV opening and closing and, in the field, is manually set during PRV commissioning.

In control terminology, the hydraulic control loop can be thought of as simple feedback system where the difference between set point and output is amplified by a gain $\alpha_{open}$ or $\alpha_{close}$ to provide a control signal, $q_3$. In reality, the PRV control loop contains a pilot valve with a spring but the stiffness of the spring is large and its dynamics can be neglected. Making the controller gain (needle valve setting) too large or too small can result in either of the problems shown in Fig. 1—for large gain, the PRV responds quickly to demand changes but the response will be oscillatory and take a long time to settle; if the gain is small then the response will be slow and the overshoot will be large (Prescott and Ulaniciki 2004). PRVs are set to obtain a balance between overshoot and settling time but it is difficult to obtain a good performance in all circumstances. Introducing dynamics into the control loop allows more sophisticated control and enables the response to different types of behavior to be improved.

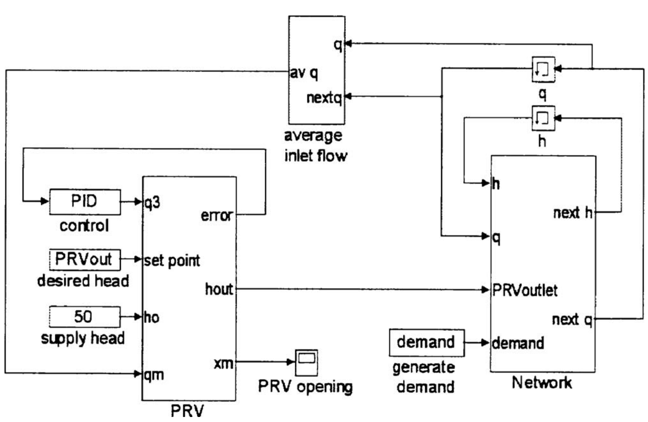


**Fig. 4.** Structure of Simulink model

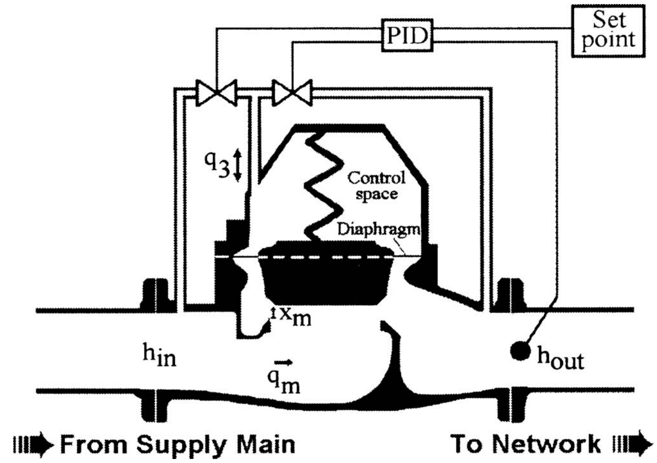


**Fig. 5.** Physical implementation of PID control system

To demonstrate this, a PID control mechanism is proposed to introduce the required dynamics into the controller. PID control adds two terms to the existing controller equation—an integral term that takes into account the amount of time errors are present and a derivative term that is dependent on the rate of change of the error. The control signal is still the flow into or out of the control space so Eq. (5) becomes

$$\varepsilon(t) = h_{set} - h_{out}$$

$$q_3 = K_p\left[\varepsilon(t) + \frac{1}{\tau_i}\int_{t=0}^{T}\varepsilon(t)dt + \tau_d\frac{d\varepsilon(t)}{dt}\right]$$

$$= K_p\varepsilon(t) + K_i\int_{t=0}^{T}\varepsilon(t)dt + K_d\frac{d\varepsilon(t)}{dt} \qquad (7)$$

where $K_p$, $K_i$, and $K_d$=proportional, integral, and derivative gains, respectively; and $\tau_i$ and $\tau_d$=integral and derivative time constants. Physical implementation of this system is shown in Fig. 5 and requires a pair of actuators to transfer water into and out of the control space and a programmable controller to implement the control law and operate the actuators. Since the water pressure in the control space is between the inlet and outlet pressures, water can be taken from the valve inlet when filling the control space and returned to the outlet when emptying it.

There are many methods to determine PID controller parameters, each of which have been successfully applied in certain conditions but also have limitations (Unar et al. 1996). The most well known are the Zeigler–Nichols design methods (Zeigler and Nichols 1942, 1943). The first of these involves estimating the ultimate gain and period (the parameters of closed loop oscillations at the limit of stability) by increasing the controller gain until sustained oscillations exist and then measuring them. The other estimates the system reaction rate from its open loop step response. However, the first of these is undesirable in practice since engineers are reluctant to drive systems to the verge of instability and the second cannot be implemented in this system—applying a nonzero $q_3$ will simply fully open or fully close the PRV.

A method better suited to this problem is Åström's relay method (Åström and Hägglund 1994), which also involves estimating the ultimate gain and period but uses relay controlled oscillations to do this. The method is also attractive since these parameters are evaluated by experiment so the technique can be easily implemented in the field. The procedure involves replacing the PID controller with a relay that following a transient period generates a stable oscillatory system from which the ultimate gain and period, $K_u$ and $P_u$, are measured. The PID parameters are calculated to move the critical point of the open-loop Nyquist curve to a desired position using

$$K_p = K_u \cos \phi$$

$$\tau_d = \frac{\tan \phi + \sqrt{\dfrac{4}{\mu} + \tan^2 \phi}}{4\pi} P_u$$

$$\tau_i = \mu \tau_d \tag{8}$$

where $\phi$=required phase margin; and $\mu$=constant. In the present case, these are taken as $\phi=60°$ and $\mu=4.3$, giving $K_p=0.5K_u$, $\tau_d=0.3P_u$ and $\tau_i=1.29P_u$.

Åström's relay method has been extended to form a diagonal controller for two-input–two-output (TITO) systems by Zhuang and Atherton (1994). The method is similar—replace PID controllers with relays, apply oscillations, and measure the ultimate gain of each loop $K_{u1}$, $K_{u2}$ and the ultimate period $P_u$, which will be the same for each loop. The critical points of the characteristic loci (a generalization of the Nyquist plot for multiple-input–multiple-output systems) are moved to desired locations by choosing suitable PID controller parameters. The first part of Eq. (8) becomes

$$K_{pi} = mK_{ui} \cos \phi \tag{9}$$

where the outer characteristic locus will have gain $m$ and phase $-180° + \phi$ at the critical frequency. The equations for $\tau_i$ and $\tau_d$ are the same as for the single input case.

Methods for selecting PID parameters are not exact and some adjustment is often necessary once the parameters are selected. In the next section, three case studies are presented that demonstrate the use of the techniques described. A good starting point is generated using the relay autotuning methods and then fine tuned to improve the performance, where necessary.

## Case Studies

To illustrate the interaction between transients in networks and PRVs, three case studies are presented. These are a theoretical single-input DMA, a single-input DMA with a real-life pipeline configuration (existing DMA) and a theoretical dual-input DMA. In each case, the response of the system to a demand profile is shown using both a standard PRV and a PID controlled PRV.

For digital control a suitable sampling rate has to be selected. If the sampling rate is low, the delay between samples will reduce accuracy and if it is high, integrator overflow may occur. In the field, a rule of thumb is to use a factor of between 0.1 and 0.01 of the system settling time. In these simulations, the sampling rate also determines the transient model discretization so this also affects the choice. In the case studies, the sampling rate was chosen using all these factors as a guide.

The simulations were carried out in Matlab/Simulink using a fixed time step of 0.01 s for the theoretical models and 0.005 s for the existing model—this timestep was required to give at least three nodes along each pipe since the shortest was 15.6 m with a wavespeed of 1,000 m/s. The simulation times, on a standard 3 GHz, 512 Mb PC, were approximately 14 min for the single inlet theoretical model, 33 min for the dual inlet model, and 94 min for the existing model.

### *Theoretical Single-Input DMA*

The response of the mesh of Fig. 2 has been determined by simulating Eqs. (2)–(5) with a random demand profile at each node, which results in the total demand of Fig. 3. Throughout this simulation, the PRV set point was 27 m and the needle valve settings, which govern opening and closing speeds of the PRV, were $\alpha_{open}=1.1\times10^{-6}$ m$^2$/s and $\alpha_{close}=10\times10^{-6}$ m$^2$/s. These values were selected to give a PRV outlet pressure profile over the first 100 s similar to that of the recorded data in Fig. 1—the resulting mean and variance of the PRV outlet head from simulating the models with these values were 26.4 m and 0.78, respectively (compared with 27.2 m and 0.76, for the recorded data). The PRV outlet pressure and PRV flow from the whole simulation is shown in Fig. 6.

The response of the system to demand changes is clear—for an increase in flow, the outlet pressure drops sharply because of the additional flow. The PRV opens at a rate governed by $\alpha_{open}$ to increase the outlet pressure back to the set point. When the flow decreases, the pressure increases quickly due to the flow reduction and then quickly drops as the PRV closes. Since the closing speed of the PRV is fast, the initial pressure increase due to the flow reduction is minor but the PRV overshoots its required steady-state position—it closes too much then has to open again to correct this. Most deviations from the set point are below the desired value, which is also confirmed by the mean value of 26.4 m—this is a result of the opening speed being slower than the closing speed. In some PRVs, the needle valve is reversed and, in these cases, the response of Fig. 6 would also be reversed. It is also evident that the oscillations are smaller in amplitude for the period when the flow is higher, between 100 and 300 s (standard deviation 0.62 compared with 0.81 for lower flow). This is consistent with observations by field technicians—PRVs tend to be less stable during low flow conditions when they are operating in an almost closed position.

The simulation demonstrates the balance between settling time and overshoot that needs to be addressed when commissioning a PRV. A fast response leads to oscillatory behavior and the PRV rarely settles since it is sensitive to small demand changes. A slow response results in large overshoot and severe pressure surges if there is a large demand change within the network. Replacing the standard pilot loop with PID controlled actuators offers scope to address both of these issues.

Åström's relay tuning method was applied to the theoretical system by inserting a relay in place of the PID controller in Fig. 4. Once transients had decayed, the amplitude and period of the

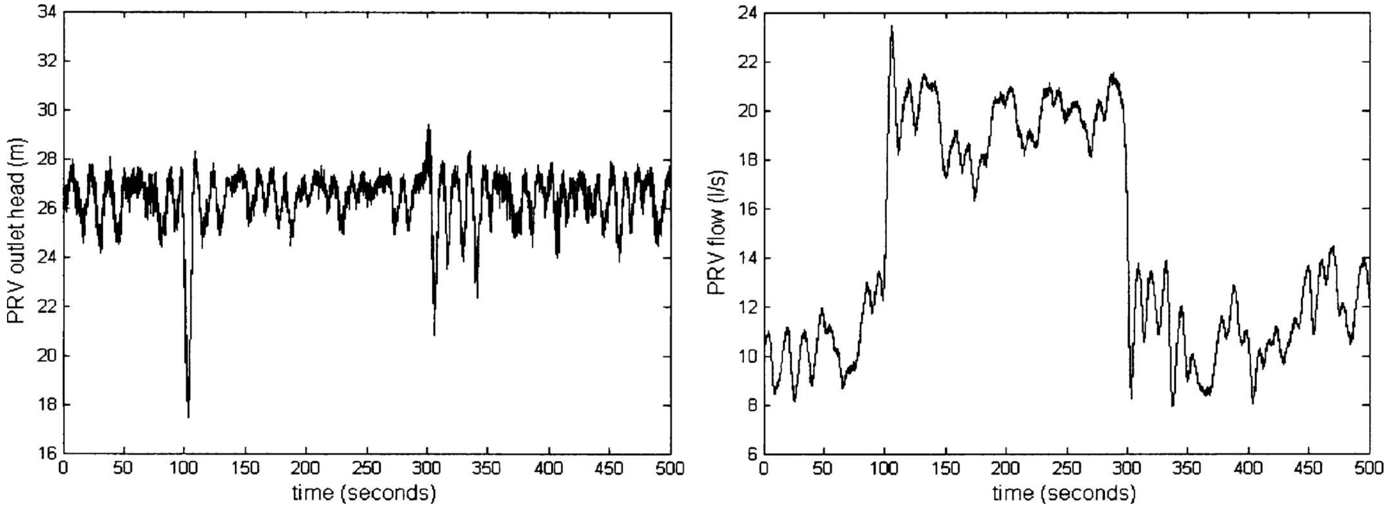


**Fig. 6.** Response of theoretical DMA to demand changes: PRV outlet and flow

steady oscillations were measured and used to calculate the PID parameters: $K_p=0.29\times10^{-3}$, $K_i=22.4\times10^{-3}$, and $K_d=0.87\times10^{-6}$. Replacing the PID controller and simulating the model with the same demand profile gave the response of Fig. 7(a). With PID control applied, the fluctuation in PRV outlet head is significantly reduced—the mean and variance of the outlet pressure is 27.006 m and 0.021—less than 3% of the variance from the standard control loop.

To determine the effect of measurement errors, the pressure outlet has been rounded to represent the resolution of an analogue to digital converter used in pressure loggers. Fig. 7(b) shows the outlet pressure of the PID controlled PRV when the outlet pressure measurement is rounded to the nearest 0.5 m—the outlet variance increases to 0.125, almost six times the value when this error was not considered. However, rounding the pressure outlet measurement to the nearest 0.1 m (easily within the capabilities of most pressure loggers) gives no discernable difference, and an almost identical variance of 0.0216, from the case in Fig. 7(a). The response of the PRV to faster changes in demand is shown in Fig. 7(c). In this case, the large demand changes at 100 and 300 s occur in a step rather than over a period of 5 s. The increased overshoot resulting from this is clear to see from Fig. 7 and is caused by the more sudden disturbance. However, the PID controller corrects the overshoot quickly.

## Existing DMA

During routine DMA investigation, the system shown in Fig. 8 was found to exhibit many pressure spikes throughout the day and night (recorded pressure and flow data are also shown). Further investigation revealed the causes of these spikes to be a single large user (at the node identified) who was making sudden demand changes, similar to those of Fig. 1. In these data, all pressure spikes are upwards and occur when the flow decreases suddenly. The PRV response to an increase in flow is very good so, from the experience of the previous case study, this indicates that the valve is able to open quickly but the closing rate is slow (i.e., $\alpha_{open}$ is large, $\alpha_{close}$ is small).

The DMA is mostly industrial with 130 users. Since a demand allocation model was available, this was used to allocate the correct number of users to each node and the demand model was used to generate a demand profile, shown in Fig. 9. Industrial use (per property) is higher than residential use and is assumed to consist of longer, more regular demand events. The demand model parameters were therefore changed to give a mean inter-event time of 200 s at each user, an average event duration of 90 s and an average event intensity of 6 L/min. This gives a random demand with an average of approximately 6 L/s and this is added to the large industrial user demand which is the same as in the theoretical case studies.

From the observed data, a good estimate can be made for the values of the needle valve settings which will reproduce the response—applying the generated demand to the model enabled suitable needle valve settings to be determined: $\alpha_{open}=50\times10^{-6}\ \mathrm{m^2/s}$, $\alpha_{close}=0.15\times10^{-6}\ \mathrm{m^2/s}$. Simulating the model, with these settings, gives the response shown in Fig. 10(a). The pressure remains almost entirely above the desired set point since the opening speed is fast and the only significant pressure spikes are following a decrease in flow. It is also clear that the peaks are more significant during the lower flow phase towards the end of the simulation.

The same PID tuning method was applied to this system and the ultimate gain and period estimated through this process were $K_u=1.59$ and $P_u=0.01$ resulting in the parameters $K_p=7.9\times10^{-5}$, $K_i=6\times10^{-3}$, and $K_d=2.4\times10^{-7}$. Running the PID controlled model using these parameters gave a response that was stable for small demand changes but went unstable as the large change was applied after 100 s—the controller overcorrected the error which resulted in a greater error in the opposite direction and the process continued, generating progressively larger errors as the controller output went back and forth. This was assumed to be caused by excessive integral action so the integral term was reduced by a factor of 10 (i.e., $K_i=0.6\times10^{-3}$). Simulating the model with this adjustment gave the more stable response shown in Fig. 10(b).

Again, there is a significant improvement in the system response through applying PID control—the mean and variance of the PRV outlet pressure under PID control is 165.999 m and 0.056 compared with 168.764 m and 10.846 for the standard PRV. Introducing PID control brings the mean outlet pressure to the 166 m set point and reduces its variance by almost 95%.

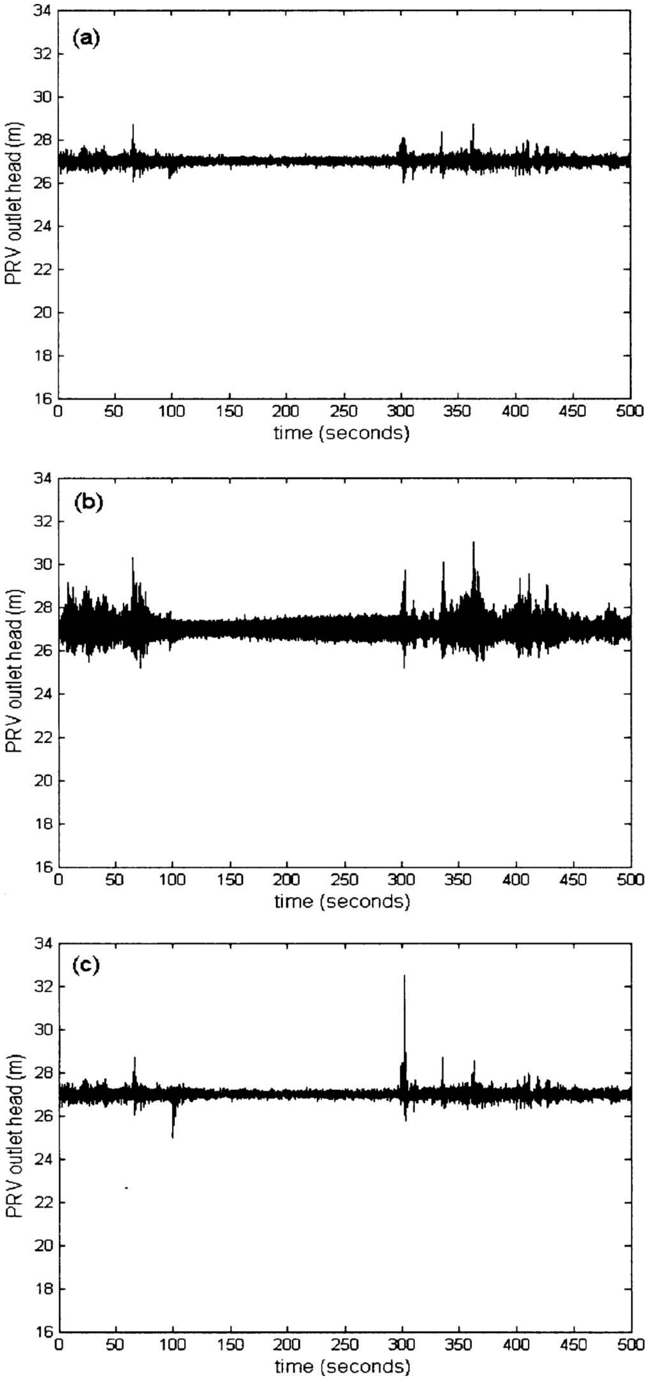


**Fig. 7.** PRV outlet head: (a) response with PID controller; (b) response with outlet pressure error; and (c) response with an instant large demand change

### Theoretical Dual-Input DMA

It is sometimes desirable to feed a DMA through more than one input because of DMA size, DMA head loss or security of supply. To demonstrate the potential for PID control in these situations, the theoretical network of Fig. 2 has been modified to include a second PRV supply which feeds the DMA at Node 31. In terms of modifying the model in Fig. 4, this involved adding an extra loop to the existing one—the relevant network flow variable was fed as an inlet to an identical PRV block and the PRV outlet from this block formed an extra input to the appropriate node of the network block. Both of these changes are shown in Fig. 11. Simulating this system with the same PRV parameters as before—$h_{\text{set}}=27$ m, $\alpha_{\text{open}}=1.1\times10^{-6}$ m$^2$/s, $\alpha_{\text{close}}=10\times10^{-6}$ m$^2$/s—and the same demand (Fig. 3), gives the outlet PRV pressure shown in Fig. 12(a). Only the first PRV outlet pressure is shown—the second PRV pressure is almost identical. Comparing these simulation results to those of Fig. 6 for the single input case, it is immediately apparent that the peak overshoot is less but the oscillations are faster. This can be attributed to the presence of a second modulating element in the system and also to the reduced flow through each PRV.

The relay experiment was applied to the system by replacing each PID controller with identical relays, determining the ultimate gain and period and calculating PID controller parameters using the charateristic locus method.

Different relay amplitudes were applied to the system but, despite this, the resulting gains were almost identical - $K_{u1}=3.55\times10^{-4}$, $K_{u2}=3.57\times10^{-4}$ - and the ultimate period was $P_u=0.01$ in both cases (as expected). Using a value of $m=1$ in Eq. (9) ensures that the phase margin of the PID controller is $\phi$, which was taken as 60°. These translate into PID parameters $K_{p1}=1.77\times10^{-4}$ and $K_{p2}=1.78\times10^{-4}$, $K_i\approx0.0138$, $K_d\approx0.53\times10^{-6}$. Running the model using these parameters led to the same instability problem as in the second case study for large demand changes. This was assumed to be caused by an accumulation of integral errors since the settling time of a dual PRV network is longer than for a single PRV system (Prescott et al. 2005). Applying the same solution - reducing the integral gain by factor of ten—resulted in a stable solution which is shown in Fig. 12(b). Once again, there is a significant improvement in the response when PID control is applied to each valve. The variances of the two outlet pressures are 0.0339, 0.0342 compared with 0.8044, 0.7969 for the case with standard PRV—a reduction of over 95% in each case.

### Case Studies—Summary

The three case studies described give an insight into how a PID control mechanism would perform in a practical setting. In all cases, PID parameters were estimated using a simple experiment which does not involve any knowledge of the network dynamics and in each case, suitable values were obtained for the proportional and derivative terms of the PID controllers. Using these, the controllers performed well with the estimated integral value for minor demand changes but, in two cases, they became unstable when a large change occurred. Fine-tuning the integral parameter—reducing it by a factor of 10—corrected this stability problem.

This can possibly be explained by the action of a PRV. If the outlet pressure is too high, the control loop injects water into the control space to close the valve and reduce outlet pressure. If outlet pressure is too low, water is removed from the control space to open the valve. Therefore the PRV acts as an integrator—the control action will be larger if errors are present for a longer amount of time. It is therefore necessary to exercise care when selecting the integral PID term, as demonstrated in the case studies.

In all cases, once suitable PID parameters were found, the PRV performance improved considerably from when the standard hydraulic control loop was used. These improvements can be quantified by measuring the sample variance of the outlet pressures—at worst the reduction in variance was almost 95%

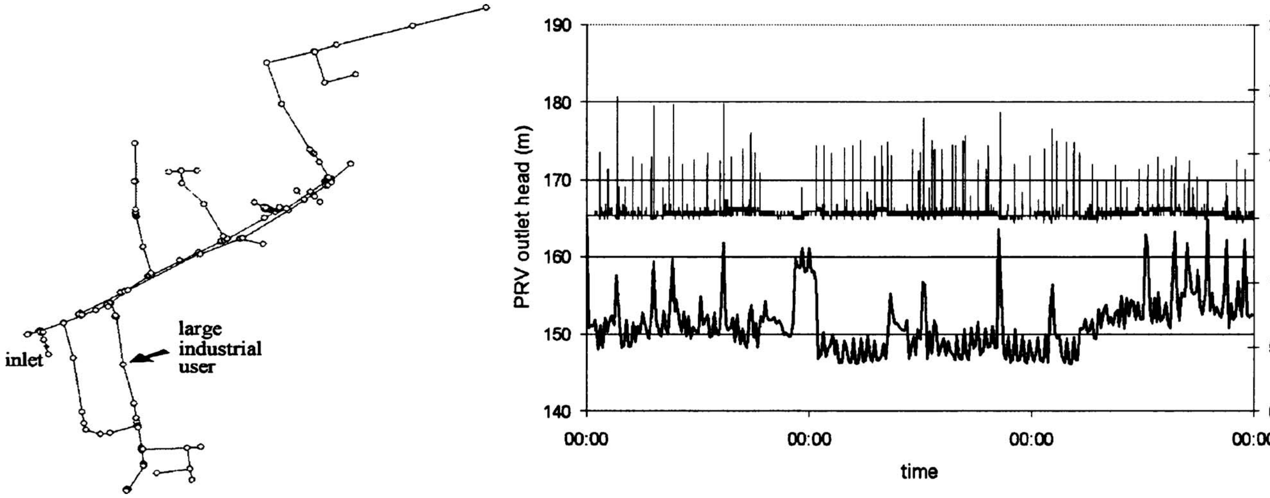


**Fig. 8.** Existing DMA and recorded data: PRV outlet pressure (thin line) and flow (thick line)

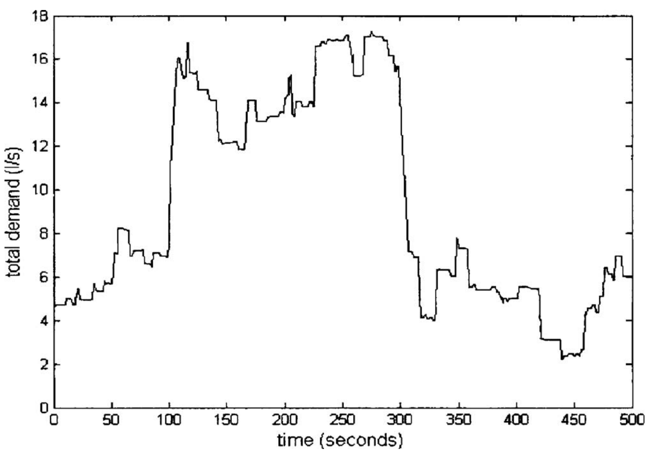


**Fig. 9.** Demand profile used in simulation

when the PID control was applied. The control system performs well in the presence of both persistent small demand changes and large sudden demand changes.

## Conclusions

This paper has shown examples of unwanted interaction between PRVs and transients in water networks and data collected during field experiments has illustrated the effects of this interaction. This includes large pressure peaks which propagate through a network and sustained oscillations which can lead to water quality problems, bursts and premature wear of the pipe network. PID controllers have been designed for three case study networks and their effectiveness at improving the interaction demonstrated.

The work in this paper focuses on problems caused by demand changes since these are caused by users and cannot be easily eliminated from the system. Demand profiles were obtained by adding a residential demand model to a large demand change. Both of these were chosen to mimic real life situations observed

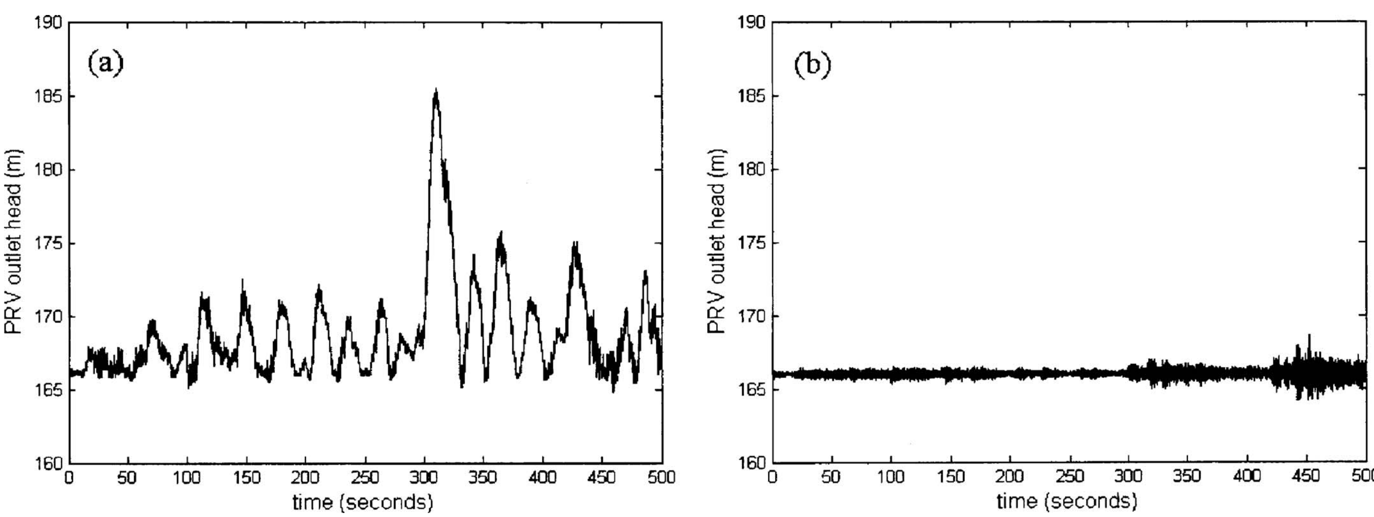


**Fig. 10.** PRV outlet for (a) standard PRV and (b) PID controlled PRV

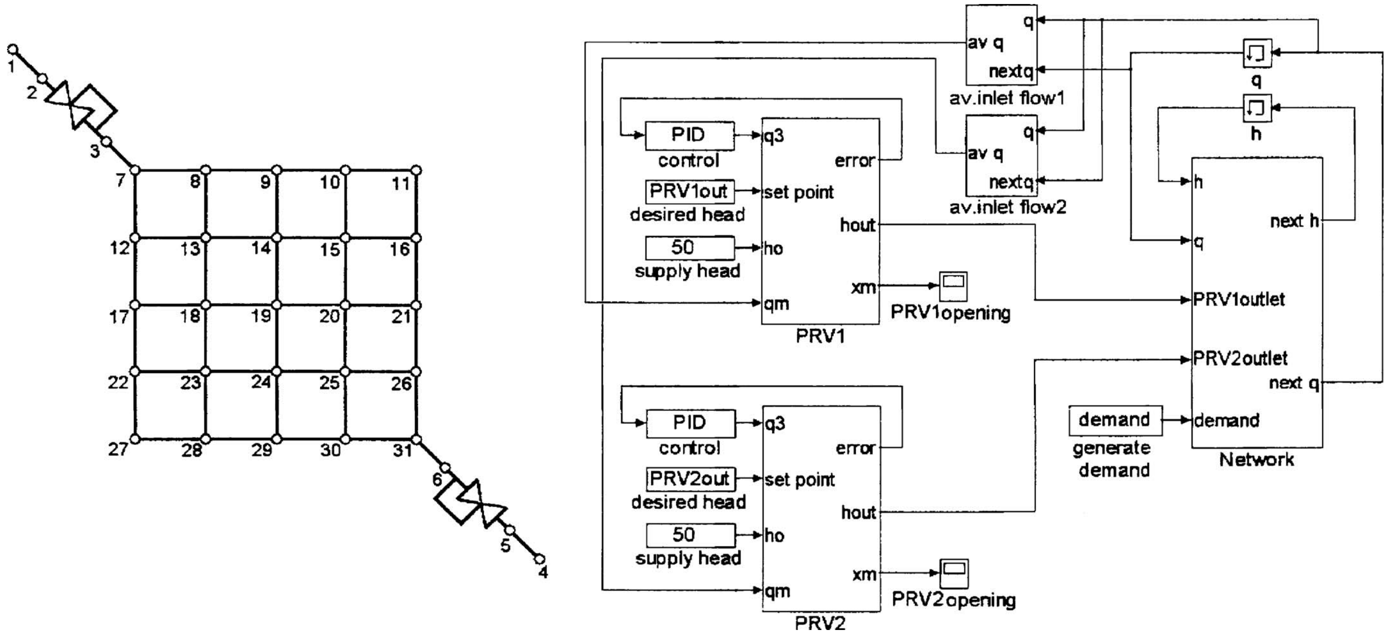


**Fig. 11.** Dual input mesh and corresponding Simulink model

during experimental work. The response of a standard PRV was described using experimental data and model simulations and the hydraulic control loop was then replaced with a PID controller to assess any changes in response.

There are many PID tuning techniques to estimate controller parameters but the one used was chosen because of its direct applicability to industrial practice. The tuning procedure does not require a system model or knowledge of system dynamics—the controller parameters can be directly estimated experimentally. In all cases, suitable proportional and derivative parameters were found although the integral term needed to be reduced to enable large demand changes to be controlled.

Physical implementation of the PID control system in practice requires pressure transducers to measure the outlet pressure of each PRV and a pair of actuators for each PRV (solenoid valves, for example) to transfer water from the PRV inlet to the control space and from the control space to the PRV outlet. These could be operated by a programmable controller using the PID algorithm described throughout this paper. From the modeling studies shown, significant improvements in PRV performance can be expected by using these methods.

## Acknowledgments

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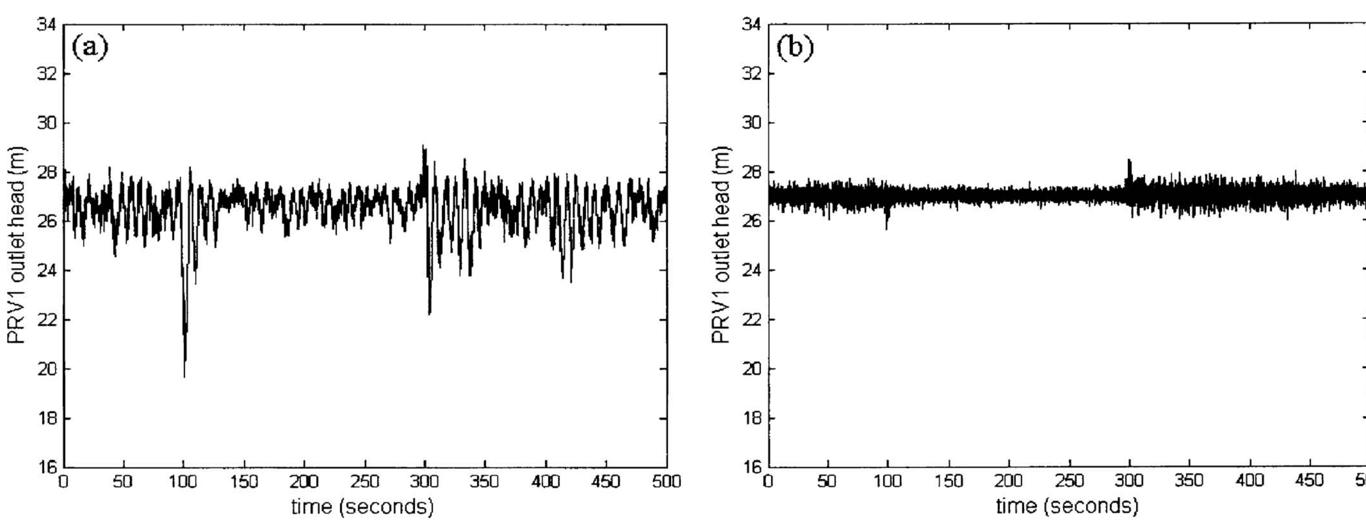


**Fig. 12.** First PRV outlet for dual feed network: (a) standard PRV; (b) PID controlled PRV

## Notation

*The following symbols are used in this paper:*

$A$ = inside pipe area;
$A_{cs}$ = cross sectional area of the PRV control space;
$a$ = transient wave speed;
$C_v$ = valve capacity;
$D$ = inside pipe diameter;
$f$ = Darcy–Weisbach friction factor;
$g$ = acceleration due to gravity;
$h$ = local piezometric head;
$h_{in}, h_{out}, h_{set}$ = PRV inlet head, outlet head, and set point;
$K_d, K_i, K_p$ = PID controller derivative, integral, and proportional gains;
$K_u$ = ultimate gain;
$m$ = gain of outermost characteristic locus at critical frequency;
$P_u$ = ultimate period;
$q$ = local flow rate;
$q_3$ = flow into or out of the control space;
$q_{leak}$ = node leakage flow;
$q_m$ = PRV flow;
$R$ = pipe resistance;
$t$ = time;
$x$ = axial distance;
$x_m$ = PRV opening;
$\alpha_{close}, \alpha_{open}$ = setting of needle valve for PRV closing and opening;
$\varepsilon$ = PRV outlet error;
$\mu$ = constant in PID parameter equations;
$\tau_d, \tau_i$ = PID derivative and integral time constants; and
$\phi$ = phase margin.

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