



Mitigating Oscillations in Hydraulic Pumping Systems by Using a Supplementary Damping Controller

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Abstract

A feedback control strategy for mitigating pressure oscillations in centrifugal pumping systems is proposed herein. Severe system damage may occur if such oscillations are not adequately damped. According to the experimental data observed, this dominant oscillating mode is operating point-dependent and may become poorly damped under low-speed regime. By using a theoretical control approach, the key parameters that affect the damping of the dominant oscillating mode are investigated. Linear model analysis show that the relative damping of the oscillations depends on the pumping system's average rotational speed and it is almost zero at low-speed operating condition. To cope with such a dangerous condition, a supplementary feedback damping controller is designed. The damping controller actuates via motor–pump variable-speed drive subsystem. The proposed controller modulates the rotational speed setpoint of the centrifugal pump rotor to produce an extra damping torque. Controller design is performed by using frequency domain techniques, and performance thereof is assessed through experimental tests in a laboratory rig pumping system. The results obtained show that by using the proposed control methodology, it is possible to safely operate the pumping at a low-speed regime, thus preventing damage to the equipment.

Keywords Damping control · Oscillation mode mitigation · Hydraulic pumping system

1 Introduction

Hydraulic pumping systems play an important role in many industrial applications requiring transport and distribution of fluids. Such applications include chemical processes (Smith and Corripio 1997), fresh water pumping (Ghosh et al. 2015), and auxiliary hydraulics systems for hydroelectric power plant operation (Cabrera et al. 1996), to name a few. Photovoltaic (Chandel et al. 2015), and mining (Deepak et al. 2007), pumping systems are examples of the pumping systems that need to be operated in variable-speed regime. In the former case, variable-speed operation is required due to variable sun irradiance on pumping systems supplied by photovoltaic panels, while in the latter case, variable-speed operation stems from the need to pump variable-density material in mining processes. In the former case, operating at low speed is necessary to ensure the continuity of the water supply service even when sun irradiance is low. In the latter case, low-speed pumping operation is necessary when pumping material is of varying density.

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However, due their physical characteristics, pumping systems may exhibit a poorly damped dominant oscillation mode when operating at low rotational speed. As a consequence, pumping systems may undergo severe vibration on the structure, leading to malfunctioning or damage in valves and/or pipes, with outages, as reported in Murphy et al. (2006), Song and Sepehri (2002), Manring (2005) and Isermann (2006), just to name a few. Therefore, to propose strategies to address the lack of sufficient damping in pumping system is a very important engineering problem and this is the main focus of the present work. Several control strategies to cope with such poorly damped oscillation modes in vibrating electromechanical structures have been reported (Csencsics et al. 2018). Mitigating such poorly damped pressure oscillations is advisable to prevent severe damage on valves and pipes. This contributes to preserving life expectancy of equipment and to avoid an increased energy consumption of the drive–pump system (due to turbulence).

The research reported in Kube et al. (2010) investigated pressure hydraulic oscillations occurring in a penstock of hydroelectric power plants, demonstrating that poorly damped oscillations may cause irreversible damage to the structure. Pressure variations in water distribution pipes (Mylapilli et al. 2015) can also lead to severe damage in pipes. Such undesirable situation could affect the regular water supply, leading to leakage or even to the total unavailability of water supply.

To the best of the authors' knowledge, it seems that most papers published so far focus on mitigating pressure oscillations by using expensive, open-loop passive mechanic–hydraulic apparatus, such as relief valves and surge chambers (Kube et al. 2010; Mylapilli et al. 2015).

Although effective, such purely passive solution may require expensive changes to the original hydraulic installations. Therefore, aiming to address this shortcoming of using expensive, purely passive solutions, this paper presents and discusses a low-cost supplementary control strategy for damping dominant oscillating mode in hydraulic systems. The main advantages of the proposed strategy, in contrast to those based on passive dampers, are: (1) it is easier to implement and (2) the proposed solution is less expensive than the use of passive dampers.

Performance tests are conducted in a laboratory-scale hydraulic system. The proposed damping controller has a standard lead–lag structure which, after discretization, has been embedded in a microcontroller subsystem. The controller modulates the speed setpoint of a built-in PID controller embedded into the voltage source inverter that drives the electrical motor of the pumping system.

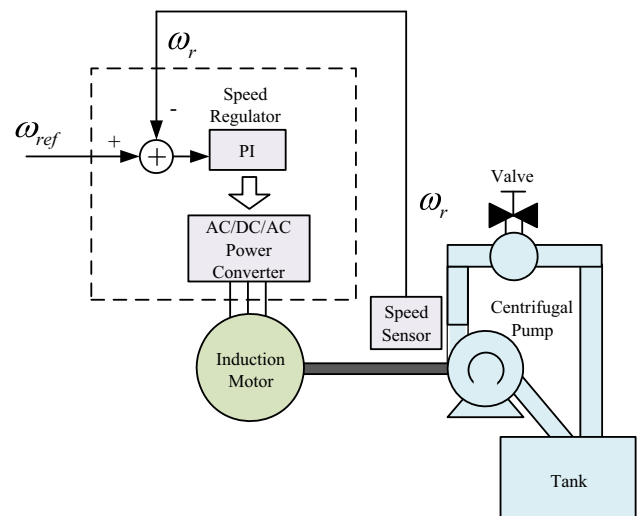


Fig. 1 Functional scheme for the laboratory pumping system

2 Phenomenological Model of a Pumping System

In order to gain an insight of the underlying hydraulic oscillating pressure phenomenon, a nonlinear model of the pumping system is firstly presented and then a linearized model is obtained. According to experimental data presented in Sect. 4, the damping value becomes almost zero under low-speed operation. As a consequence, very dangerous sustainable pressure oscillations may arise due to the lack of damping. Although such oscillations are rather common in hydraulic pumping systems, to the best of the herein authors' knowledge, it seems that this issue is scarcely covered from a control-based theoretical viewpoint. Therefore, an explanation of the oscillatory phenomenon is carried out in this section as based on linearization of the nonlinear model.

2.1 Nonlinear Model of the Motor–Pump System

Figure 1 shows a typical configuration of an industrial variable-speed pumping system. The three-phase electrical induction motor, which drives the centrifugal pump, is actuated by an AC/DC/AC power converter. In its standard configuration, the power inverter is equipped with an embedded speed-regulating control in the form of proportional–integral (PI) controller. The hydraulic load is a closed circuit consisting of a tank, pipes and a proportional valve.

The dynamic model of the rotating part in the motor–centrifugal pump system observes Newton's second Law. The resulting torque actuating on the rotational body (consisting of the combined rotational inertia of the electric motor and of the centrifugal pump) is given by

$$J \frac{d\omega_r}{dt} = T_e - T_{\text{pump}}, \quad (1)$$

where J is the combined motor–pump rotational inertia, ω_r is the rotor angular speed, $\frac{d\omega_r}{dt}$ is the angular rotor acceleration, T_e is the electrical torque supplied by the electromagnetic subsystem of the induction motor driving the centrifugal pump, and T_{pump} is the reaction (resistive) load torque due to the centrifugal pump.

By applying the moment conservation law, taking into account the mass of fluid contained in the impeller, along with some affinity laws, it is shown (Cherkasski 1984) that T_{pump} and ω_r must observe a parabolic law in the form:

$$T_{\text{pump}} = K_{\text{pump}} \cdot \omega_r^2, \quad (2)$$

where K_{pump} is a pump coefficient dependent upon the fluid density ρ of the pumped fluid, and upon the impeller geometry, (Cherkasski 1984; Mokeddem et al. 2011). By considering Eqs. (1) and (2), the following first-order nonlinear system can be obtained

$$J \frac{d\omega_r}{dt} + K_{\text{pump}} \cdot \omega_r^2 = T_e. \quad (3)$$

The nonlinear model (3) has the rotational speed, ω_r , as the output variable and the electrical torque, and T_e as the input variable.

Usually, T_e is supplied by means of an inverter (an AC/DC/AC converter) subsystem. Assuming ideal vector control, which is standard in industrial speed drive systems (Leonhard 2001), the instantaneous value of T_e is proportional to the control output of a speed-regulating proportional plus integral (PI) controller, namely

$$T_e(t) = K_T u(t), \quad (4)$$

$$u(t) = K_p e(t) + K_i \int e(t) dt, \quad (5)$$

$$e(t) = \omega_{\text{ref}} - \omega_r, \quad (6)$$

where the coefficient K_T represents a simplified model for a fast-actuating vector-controlled inverter, $u(t)$ is the instantaneous value of the control signal produced by the speed-regulating PI controller (embedded in the inverter converter), $e(t)$ is the speed error, and ω_{ref} is the angular speed setpoint (reference) value.

Equations (3)–(6) form an overall second-order nonlinear dynamic model for the inverter–motor–pump system. Figure 2 presents the block diagram for this nonlinear model. The load torque imposed by the centrifugal pump, T_{pump} , is the main source of nonlinearity on the system (cf. Fig. 2) because T_{pump} depends on the value of square of the rotational speed (a state variable of the mechanical–hydraulic system).

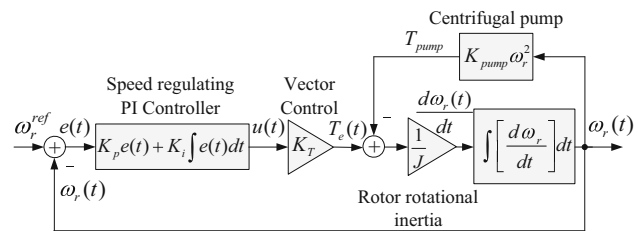


Fig. 2 Block diagram of the nonlinear model of the pumping system

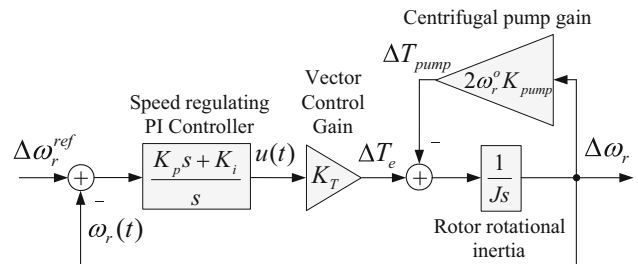


Fig. 3 Block diagram of the linearized (small signal) model for the inverter–motor–centrifugal pump system

2.2 Model Linearization

In order to derive a small signal model for the inverter–motor–pump system, it will be considered that the system is operating around a steady-state average speed ω_r^0 . The small signal model is then derived by linearizing the set of Eqs. (3)–(6), around operating condition characterized by, ω_r^0 .

Figure 3 presents the corresponding block diagram of the linearized model. In this linearized model, the net effect of centrifugal pump is to add a viscous damping, the gain $2\omega_r^0 K_{\text{pump}}$ of which being proportional to the value of the steady-state operating speed, ω_r^0 . Therefore, this viscous damping will be small for that operating condition where the system needs to be operated at reduced steady-state speed, ω_r^0 .

By examining the block diagram of the linearized model in Fig. 3, the closed-loop transfer function relating the speed deviation, $\Delta\omega_r$, to the PI controller speed reference deviation, $\Delta\omega_{\text{ref}}$, can be expressed as

$$G(s) = \frac{\Delta\omega_r}{\Delta\omega_{\text{ref}}} = \frac{b_1 s + b_0}{s^2 + 2\xi\omega_n s + \omega_n^2}, \quad (7)$$

where ξ and ω_n are, respectively, the relative damping and the natural angular frequency of the oscillatory dominant mode exhibited by the closed-loop system shown in Fig. 3. The numerator and denominator coefficients of $G(s)$ depend on the values of the PI controller's parameters (K_p and K_i), the system's physical parameters (K_T , J and K_{pump}), as well on the steady-state operating speed (ω_r^0), according to the following relationships (which can be easily computed from the closed-loop equation):

$$b_1 = \frac{K_p K_T}{J}, \quad (8)$$

$$b_0 = \frac{K_i K_T}{J}, \quad (9)$$

$$2\xi\omega_n = \frac{2K_{\text{pump}}\omega_r^0 + K_p K_T}{J}, \quad (10)$$

$$\omega_n^2 = \frac{K_i K_T}{J}. \quad (11)$$

By analyzing (10) and (11), the following remarks can be made: Eq. (10) shows that the decay rate value (product $\xi\omega_n$) depends linearly on the value of ω_r^0 . Therefore, the oscillations have a small decay rate when the system needs to be operated at a low speed. Equation (10) also shows that the oscillations decay rate, $\xi\omega_n$, is directly proportional to the PI controller's proportional gain, K_p . In principle, the decay rate could be improved by increasing the value of the PI gain K_p to obtain a desired time-decaying of the oscillation amplitude.

However, since the proportional action K_p exhibits a constant gain over all frequencies, there is a risk of the PI controller to react with high control efforts for high frequency noise or disturbance, with possible actuator and/or pump damage. Therefore, it is advisable to design a supplementary outer loop control which reacts only in the range of frequencies neighboring the frequency of the dominant oscillatory mode. Therefore, for safety reasons, it is advisable to keep K_p small in order not to induce undesirable hammer effects on the hydraulic system pipeline (Rocha et al. 2015). Equation (11) shows that the natural frequency of the oscillations, ω_n , is proportional to K_i .

It is advisable to design the regulating PI control in order to avoid inducing hammer effects on the hydraulic system, thus preventing damage. Therefore, from now on, only the cases having a K_p value sufficiently small will be considered, thus meeting the condition $2K_{\text{pump}}\omega_r^0 K_p K_T$ for all ranges of operating speeds ω_r^0 . This assumption allows to simplify (10), leading to the following approximated relationship between the decay rate of the oscillations ($\xi\omega_n$) and the average rotating speed at the current operating point (ω_r^0):

$$\xi\omega_n \approx \frac{K_{\text{pump}}}{J} \omega_r^0. \quad (12)$$

Note that for fixed values of J , and K_{pump} , the decay rate of the dominant oscillation is proportional to the average speed values, ω_r^0 , at a given operating condition (see Eq. 12). Therefore, low damping is expected at low-speed condition.

In order to avoid damage in the inverter–motor–pump system, it is advisable to design and implement a supplementary damping controller for mitigating the dominant oscillatory mode. This is the main objective of the control strategy presented herein.

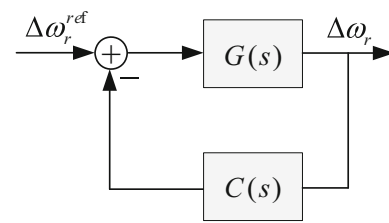


Fig. 4 Block diagram for the closed-loop system

In order not to induce harmful hammer effects on the hydraulic system structure, only the case of interest [corresponding to K_p small ($K_p \approx 0$)] will be considered. This assumption implies that the coefficients b_1 and b_0 , of the numerator of $G(s)$, in Eq. (7), take the values $b_1 \approx 0$ and $b_0 = \frac{K_i K_T}{J} = \omega_n^2$ [according to (8), (9) and (11)].

For the objectives of the damping control design, the overall hydraulic pumping (including the drive subsystem) will be assumed that can be adequately represented by a standard second-order plus time delay (SOPTD) in the form

$$G(s) = \frac{\Delta\omega_r}{\Delta\omega_r^{\text{ref}}} = \frac{\omega_n^2}{s^2 + 2\xi\omega_n s + \omega_n^2} e^{-\theta s}, \quad (13)$$

where θ is a time delay parameter included in the model in order to account for some uncertainties on the actual system. Such uncertainties include delays due to the internal processing of drive system along with non-minimum response due to the variable air content in the pipes.

3 Damping Controller Strategy

As explained in Sect. 2, the pumping system exhibits a dangerous poorly damped oscillation mode when operating at low-speed regime. Aiming to improve the damping of this dominant mode, this paper proposes a supplementary damping control strategy, described as follows.

3.1 Compensator Design

Let $C(s)$ be the transfer function of the damping controller to be designed. Consider the closed-loop feedback configuration shown in Fig. 4, where $G(s)$ is the transfer function of the plant model (the inverter–motor–centrifugal pump system, in our case). The corresponding closed-loop transfer function is given by

$$G_{cl}(s) = \frac{G(s)}{1 + G(s)C(s)}. \quad (14)$$

where $L(s) = G(s)C(s)$ is the open-loop transfer function.

According to classical control theory (Ogata 2010), a SOPTD transfer function $G(s)$, defined in (14), has a 3 dB

bandwidth frequency given by

$$\omega_{3\text{ dB}} = \omega_n \sqrt{1 - 2\xi^2 + \sqrt{(2\xi^2 - 1)^2 + 1}}. \quad (15)$$

According to Eq. (15), it is easy to verify that the ratio $\frac{\omega_{3\text{ dB}}}{\omega_n}$ becomes almost constant for small values for ξ , as follows

$$\frac{\omega_{3\text{ dB}}}{\omega_n} = \sqrt{1 + \sqrt{2}} \approx 1.553. \quad (16)$$

Therefore, Eq. (16) shows that there is a close relationship between $\omega_{3\text{ dB}}$ and ω_n . If the value of natural frequency ω_n is known, Eq. (16) can be used to obtain an estimate of $\omega_{3\text{ dB}}$.

The controller $C(s)$ must be designed so that the desired closed-loop behavior may be approximately represented by a SOPTD model having a desired damping ξ_d for the oscillation mode and a natural angular frequency ω_n . By imposing that the desired bandwidth of the closed-loop system bandwidth must be close to the plant model, the gain crossover frequency ω_{gc} must be approximately equal to $\omega_{3\text{ dB}}$, i.e.,

$$\omega_{gc} \approx \omega_{3\text{ dB}}. \quad (17)$$

By using the above assumptions, the angle and modules conditions of $L(s)$, at $s = j\omega_{gc}$, are, respectively, given by

$$\angle(L(j\omega_{gc})) = \angle(G(j\omega_{gc})) + \angle(C(j\omega_{gc})) = \pi + \varphi_m, \quad (18)$$

$$|L(j\omega_{gc})| = |G(j\omega_{gc})||C(j\omega_{gc})| = 1, \quad (19)$$

where φ_m is the desired phase margin.

The set of Eqs. (16)–(19) allow to design the controller $C(s)$. The designer must specify a desired phase margin φ_m (which is closely related to ξ_d , the desired damping for the closed-loop system).

Given the plant frequency response, $G(j\omega)$, the natural frequency ω_n , and the desired phase margin φ_m , the main steps of controller design algorithm are the following:

- (i) estimate $\omega_{3\text{ dB}}$ using Eq. (16) (or read it from the magnitude frequency response in the Bode diagram of $G(j\omega)$;
- (ii) set $\omega_{gc} \approx \omega_{3\text{ dB}}$;
- (iii) by using Eq. (18) (phase condition), calculate $\angle(C(j\omega_{gc}))$;
- (iv) by using Eq. (19) (modules condition), calculate $|C(j\omega_{gc})|$;
- (v) finally, by using $|C(j\omega_{gc})|$ and $\angle(C(j\omega_{gc}))$, choose a suitable transfer function structure to implement controller $C(s)$ (the details will be given in next subsection).

3.2 Selecting a Structure for the Compensator

After concluding the design steps (i) to (v), above, a suitable structure for the controller transfer function must be chosen.

In this paper, the controller transfer function is synthesized in the standard form

$$C(s) = K_c LL(s) F_{hp}(s), \quad (20)$$

where

K_c : a real scalar gain to be tuned according to the desired closed-loop damping of the dominant mode;

$LL(s)$: a lead–lag transfer function intended for compensating for the lag of the plant at the resonance frequency and having the following standard form:

$$LL(s) = \frac{T_1 s + 1}{T_2 s + 1}, \quad (21)$$

$F_{hp}(s)$: a high-pass filter designed to prevent any adverse interference of the damping controller $C(s)$ on the steady-state regime. $F_{hp}(s)$ has the following standard form

$$F_{hp}(s) = \frac{T_w s}{T_w s + 1}, \quad (22)$$

where the time constant T_w must be selected so that the corner frequency of the high-pass filter is lower than the resonance frequency of the dominant mode, namely

$$\frac{1}{T_w} < \omega_n. \quad (23)$$

4 Experimental Test System

The experimental tests described herein were conducted in a laboratory pumping system (Fig. 5). The system consists of a centrifugal pump [rated flow: 11.0 m³/h, rated pressure: 27.0 m water column (m.w.c.)], which is driven by a three-phase (220 V, 1.0 kW) electrical induction motor. The induction motor is actuated by an IGBT-based AC–DC–AC power converter.

Figure 6 exhibits the corresponding schematic diagram of the laboratory pumping system. Note that the rotational speed of the motor–pump set is regulated by using a PI controller, which is internal to the power converter subsystem. The speed reference, ω_{ref} , is set up by the operator, by means of a potentiometer. The PI speed regulator, along with the inverter power electronics, imposes an electrical torque on the electrical motor shaft to keep the rotational speed of the electric motor and centrifugal pump set regulated.

4.1 Dynamic Behavior of the Test Pumping System Under Different Operating Conditions

In order to gain an insight on the dynamic behavior of test pumping system, several step response tests have been car-

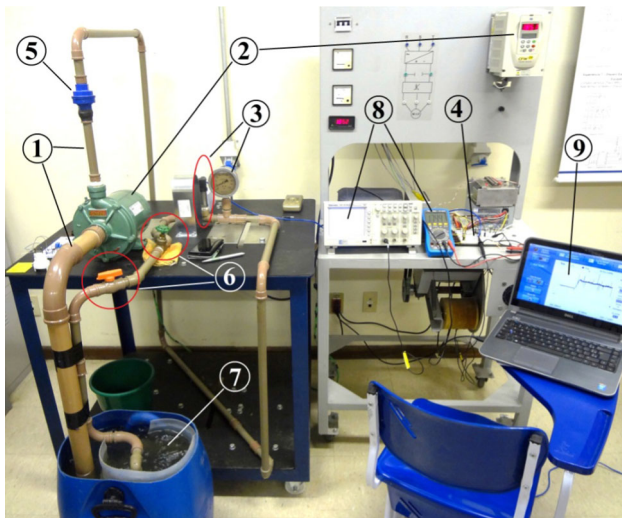


Fig. 5 Laboratory pumping system: (1) pipes, (2) inverter–motor–centrifugal pump system, (3) pressure measurement devices, (4) microcontroller board, (5) and (6) valves, (7) tank, (8) measurement devices

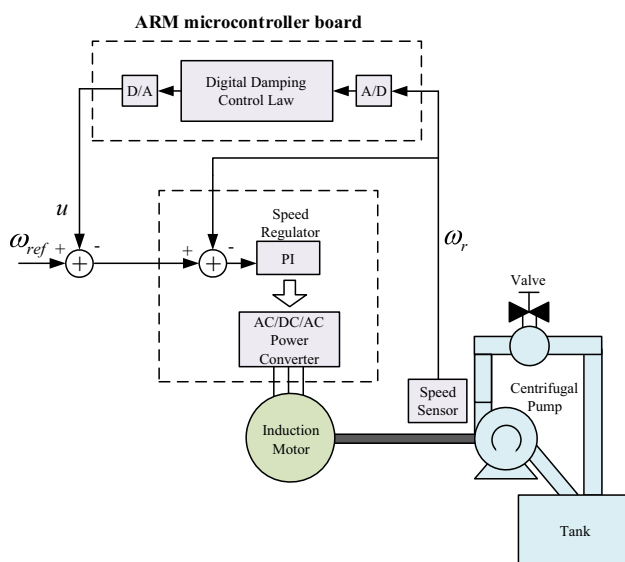


Fig. 6 Functional scheme for the laboratory pumping system, including the designed damping controller

ried out for a set of operating conditions. For the described tests, the schedule variable chosen to define the operating point was the steady-state average value of the pump's rotational speed. Three operating conditions are considered: high speed (speed values around 1500 RPM), medium speed (speed values around 1200 RPM) and low speed (speed values around 900 RPM). For all reported tests, the parameters of the internal PI controller were respectively set, by trial and error, to the following values: $K_p = 20$ and $T_i = 0.03$ s, where T_i is the integral time. Note that the value of the gain

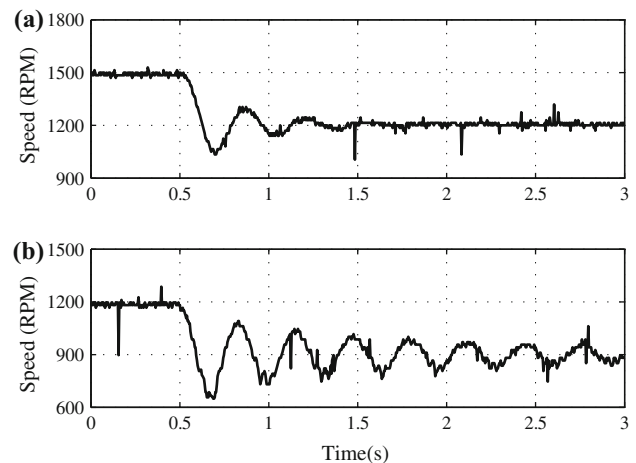


Fig. 7 System step response: **a** response for high-to-medium-speed transition; **b** response for medium-to-low speed transition

torque K_T of the test system is an uncertain unknown parameter in the reported experiments.

Figure 7 shows the results of two-step response tests carried out on the laboratory pumping test system. These tests correspond to the system's behavior after transitions from high-to-medium speed (cf. Fig. 7a) and from medium-to-low speed (cf. Fig. 7b). The dynamic system behavior is quite different for these different operating conditions. A severe reduction in the value of the system damping can be observed when the system is operating at low-speed regime (see Fig. 7b).

In Fig. 7a, the system is initially operating in a steady-state condition, around the operating point corresponding to high speed (1500 RPM). Around time instant $t = 0.5$ s, a negative step variation, of $\Delta\omega_r^{\text{ref}} = -300$ RPM is applied on the speed reference, leading the system to perform a transition from high (1500 RPM)-to-medium speed (1200 RPM). The system's response after disturbance is oscillatory but exhibits a satisfactory damping of the dominant oscillations (the oscillations are satisfactorily damped in less than 1.0 s after disturbance) as shown in Fig. 7a.

For Fig. 7b, the system is initially operating in a steady-state condition corresponding to medium speed (around 1200 RPM). Around time instant $t = 0.5$ s, a negative step variation, of $\Delta\omega_r^{\text{ref}} = -300$ RPM is applied on the speed reference, leading the system to perform a transition from medium (1200 RPM)-to- low speed (900 RPM). Note that the system's response is very oscillatory and dangerously undamped. In fact, under such operating condition, the pipeline structure itself is at risk of damage due to vibration on this non-damped oscillation mode.

Therefore, aiming to ensure a safe operation at low speed, it is advisable to design a supplementary damping controller to increase the damping of oscillations.

4.2 Identification of a Nominal Model

For the plant operating around a medium-speed condition, the step response tests are performed to identify a suitable linear plant model in order to design the damping controller. By using the collected input/output data, along with standard system identification procedures (Ljung 2001), the following SOPTD model is identified for the hydraulic plant described in Fig. 6.

$$G_{cl}(s) = \frac{\omega_n^2}{s^2 + 2\xi\omega_n s + \omega_n^2} e^{-\theta s}, \quad (24)$$

where the values of the estimated parameters ω_n , ξ (natural angular frequency and relative damping of the dominant oscillating mode) and time delay θ are the following: $\omega_n = 18.0 \text{ rad/s}$, $\xi = 0.15$ and $\theta = 0.08 \text{ s}$. The observed time delay is mainly due to the time taken by internal digital embedded system of the inverter to process the changes in reference signal of the PI speed-regulating controller. In addition, some non-minimum-phase behavior may be expected due to the air contained in the pipes. This effect is also accounted for by the time delay θ . For control purposes, the main effect of this small delay is to reduce the phase margin.

The non-minimum phase behavior is not easily observable in Fig. 7. Such behavior changes from an experiment to another, possibly, due to the uncertain of the air content on the pipe, at the beginning of each experiment. Therefore, this effect is taken into account by considering the time delay as an uncertain parameter.

Figure 8 shows the step response of the identified model along with the measured step response of the real hydraulic plant, for a step variation of +300RPM, from 1200 to 1500RPM, on the speed reference. A good relationship between the model and the real system responses can be observed. This identified SOPTD model is then used to design the damping controller.

4.3 Damping Controller Design

The damping controller design is based on the frequency domain approach described in Sect. 3. The designer must specify the desired phase margin. The obtainable closed-loop performance, in terms of damping of the dominant oscillation mode, is closely related to the specified phase margin.

The overall controller structure has the following form $C(s) = C_0(s)F_{hp}(s)$, where $C_0(s)$ is a lead-lag transfer function, which must be first designed to provide the desired closed-loop performance and phase margin, while $F_{hp}(s)$ is a high-pass filter chosen in order to block low-frequency components included in the measured signal. Let factor $C_0(s)$ be in the form

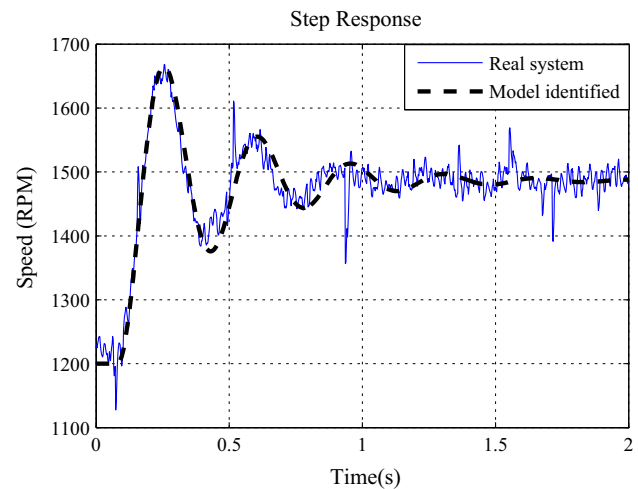


Fig. 8 Comparison between the step responses of the identified model (dashed line) and the actual plant (continuous line)

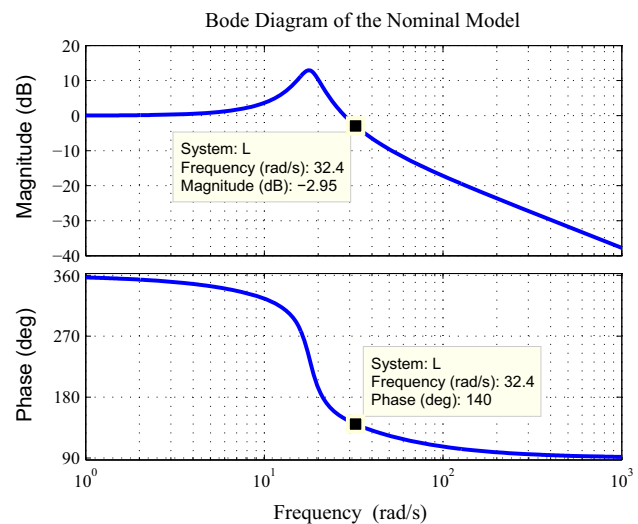


Fig. 9 Bode diagram of the loop transfer function $L(j\omega) = G(j\omega)$, for the uncompensated system

$$C_0(s) = K_c \frac{Ts + 1}{\alpha Ts + 1}, \quad (25)$$

where α and T are parameters to be calculated.

Assuming that the specified bandwidth of the closed-loop system is almost similar to the plant model, the value of gain crossover frequency, ω_{gc} , can be read from the Bode diagram of $G(s)$, as shown in Fig. 9. Note that the $\omega_{gc} \approx \omega_{3dB} \approx 32.4 \text{ rad/s}$.

Let $\phi_c = \text{angle}(C_0(j\omega_{gc}))$ be the maximal angle advance provided by $C_0(s)$ at gain crossover frequency ω_{gc} , and let φ_m be the specified phase margin. Then, from angle condition, the following relationship holds at ω_{gc} :

$$\phi_c = (\pi + \varphi_m) - \angle G(j\omega_{gc}). \quad (26)$$

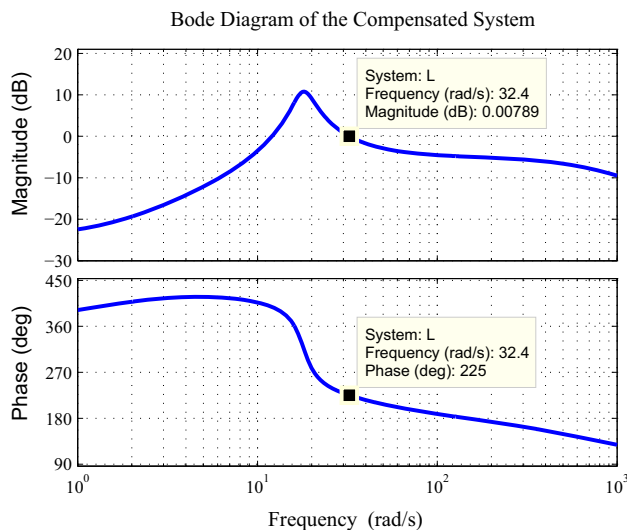


Fig. 10 Bode Diagram of the Loop Transfer Function $L_c(j\omega) = G(j\omega)C(j\omega)$, for the compensated system

By specifying a phase margin value of $\varphi_m = 45^\circ$ (degrees), which corresponds to a desired relative damping, the value of which being approximately $\xi_d = 0.4$ (Ogata 2010), and by using Eq. (26), the computed value for ϕ_c is the following $\phi_c = 180^\circ + 45^\circ - 140^\circ = 85^\circ$. After that by using $\phi_c = 85^\circ$, $\omega_{gc} = 32.4$ rad/s, $|G(j\omega_{gc})| = 1/\sqrt{2}$, the parameters of $C_0(s)$, namely α , T and K_c , are computed as follow

$$\alpha = \frac{1 - \sin(\phi_c)}{1 + \sin(\phi_c)} = 0.0019; \quad (27)$$

$$T = \frac{1}{\omega_{gc}\sqrt{\alpha}} = 0.7069, \quad (28)$$

$$K_c = \frac{1}{|G(j\omega_{gc})|} \frac{\sqrt{1 + (\omega_{gc}\alpha T)^2}}{\sqrt{1 + (\omega_{gc}T)^2}} = 0.06137. \quad (29)$$

Finally, the high-pass $F_{hp}(s) = T_w s / (1 + T_w s)$ is designed in order to reject unwanted low frequency disturbance components and also not to disturb the steady-state operating point. The filter time constant T_w is chosen to satisfy $T_w = 2/\omega_n = 0.111$ s [in order to satisfy the inequality (23)], which means that disturbances, which have frequency below $\omega_n/2 = 9.0$ rad/s, are rejected. In summary, the designed damping controller $C(s) = C_0(s)F_{hp}(s)$ is then

$$C(s) = 0.06137 \frac{(0.7069s + 1)}{(0.001347s + 1)} \frac{0.111s}{(0.111s + 1)}. \quad (30)$$

Figure 10 presents the frequency response of loop transfer function, for the nominal plant model together with the designed damping controller (Eq. 30). Note that the compen-

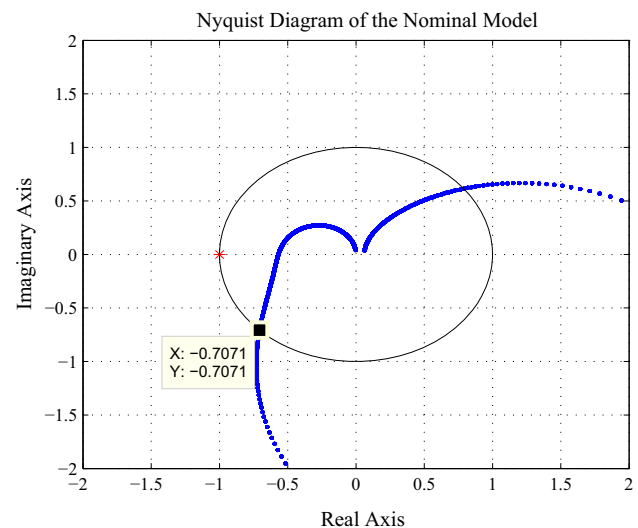


Fig. 11 Nyquist Diagram Loop Transfer Function $L(j\omega) = G(j\omega)C(j\omega)$, for the compensated system

sated system presents a phase margin, $\varphi_m \approx 45^\circ$, as specified ($\varphi_m \approx 225^\circ - 180^\circ \approx 45^\circ$, at $\omega = \omega_{gc} = 32.4$ rad/s).

Figure 11 shows the corresponding Nyquist diagram of L . Note that the compensated system presents, besides phase margin $\varphi_m \approx 45^\circ$, a gain margin around value 1.8.

In order to digitally implement the designed damping controller, the transfer function of $C(s)$, given by (30), is then transformed into a canonical discrete-time form (Landau and Zito 2006)

$$C(z^{-1}) = \frac{R(z^{-1})}{S(z^{-1})} = \frac{r_0 + r_1 z^{-1} + r_2 z^{-2}}{1 + s_1 z^{-1} + s_2 z^{-2}}. \quad (31)$$

By selecting a sample interval of $T_s = 0.02$ s (which satisfies the condition $\omega_s = \frac{2\pi}{T_s} \gg \omega_n$, where ω_s is the angular sample rate) and by using the Tustin method, the following pulsed transfer function is obtained for the digital controller

$$C(z^{-1}) = \frac{3.5575 - 7.0158z^{-1} + 3.4582z^{-2}}{1 - 0.0723z^{-1} - 0.6365z^{-2}}. \quad (32)$$

The control law given by Eq. (32) is now ready to be implemented in a digital system which, in this case, is a microcontroller-based environment. The performance of the designed damping controller is assessed by experimental tests, which will be detailed in the next section.

5 Experimental Performance Assessment

The damping control law designed in the previous section is implemented by using an electronic digital board based on a 32-bit, 84 MHz clock, microcontroller. The microcontroller implementation is performed to provide an outer damping

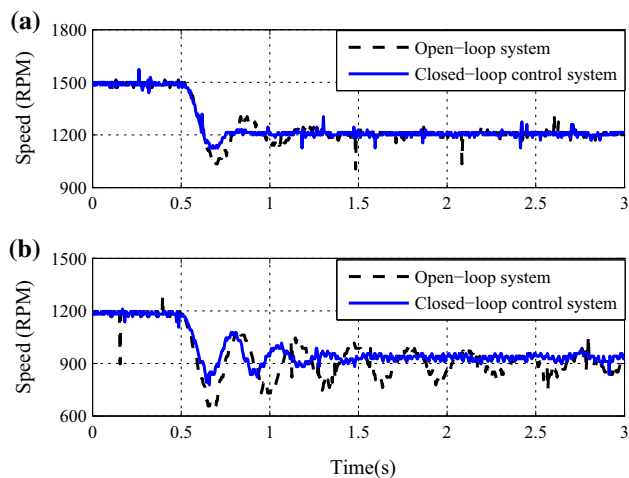


Fig. 12 Comparison between the system's response without (dashed line) and with damping controller (solid line), for two different initial operating conditions

control loop which is both low cost and has minimal interference on previous adjustments of inverters, a relevant fact for practical field applications. The reason is that sometimes the adjustment of inverter parameters' is not available or well documented.

After insertion of the designed damping controller, the system's final configuration of the motor–pumping system is represented in Fig. 6, where the damping controller actuates as a supplementary controller, by modulating the value of the reference signal for the speed-regulating PI according to detected oscillations on the rotor speed measurements.

5.1 Experimental Tests

In order to assess the performance of the designed damping controller, several tests are performed with the pumping system initially operating at steady-state condition at a high speed level (around 1500 rpm) and also at a medium speed (around 1200 rpm).

Figure 12a presents the measured system's response when the motor–pump system is initially operating around a high speed level (1500 rpm) and a negative step variation, of magnitude -300 rpm, is applied into the speed reference setpoint. Note that for this operating condition, the open-loop system response (i.e., without using the designed damping controller, dashed line) already exhibits a satisfactory oscillation damping. By including the damping controller action, the system damping is further improved, thus mitigating the oscillations in less than 1.5 s after the disturbance application.

Figure 12b shows the measured system's response when the motor–pump system is initially operating around a medium speed level (1200) RPM and a negative step variation, of magnitude -300 RPM, is applied into the speed

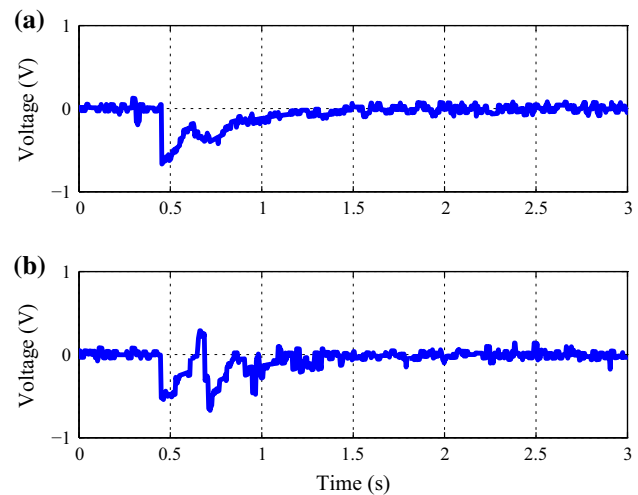


Fig. 13 Control signal response: **a** control signal for high-to-medium-speed transition; **b** control signal for medium-to-low-speed transition

reference setpoint. Note that the open-loop system response (dashed line) becomes unstable, presenting a sustainable dominant oscillation. This is a very dangerous condition, which leads to a system outage to prevent equipment damage. The corresponding closed-loop system response, when the motor–pump system is operating under damping controller action, is presented in Fig. 12b (see the signal represented by the solid line). The damping controller managed to safely mitigate the dominant oscillations damping in less than 1.5 s after the application of the disturbance (cf. Fig. 12b). Furthermore, by using the damping controller, it is possible to safely operate the plant in a low-speed regime, thus increasing the range of plant function.

Figure 13 presents the corresponding damping control signal. Note that the damping controller signal does not saturate and shows a smooth variation, which prevents induced shock waves on the pipe system. For low-speed operation, Fig. 13b shows that a controller detuning occurs, as expected. Therefore, robust or adaptive control should be considered if further performance improvements are desired. This particular issue is investigated by the authors and will be reported in future works.

5.2 Stability Margin Evaluation

In order to experimentally assess the phase margin when applying the designed damping controller (Eq. 32) to the real plant, several identification tests are performed on the pumping system to estimate linear parametric models representing the plant dynamics (without using controller $C(s)$) at operating conditions of high, medium and low speed. These models are named G_{high} , G_{medium} and G_{low} , respectively, and they are of the autoregressive with exogenous input (ARX) type.

Table 1 Identified ARX models for G_{high} , G_{medium} and G_{low} , sampling time $T_s = 0.020$ s

Op. condition	Plant identified model
High speed	$G_{\text{high}}(z^{-1}) = \frac{-0.1771z^{-1} + 0.2979z^{-2}}{1 - 1.769z^{-1} + 0.8843z^{-2}}$
Medium speed	$G_{\text{medium}}(z^{-1}) = \frac{-0.1797z^{-1} + 0.3012z^{-2}}{1 - 1.783z^{-1} + 0.9045z^{-2}}$
Low speed	$G_{\text{low}}(z^{-1}) = \frac{-0.1632z^{-1} + 0.2966z^{-2}}{1 - 1.8z^{-1} + 0.9329z^{-2}}$

Table 2 Loop transfer functions computed from identified models for G_{high} , G_{medium} , G_{low} and designed controller C

Loop transfer function $L(s)$ —high speed

$$L_{\text{high}}(s) = \frac{-417.1s^3 + 9849s^2 + 1.477e04s}{s^4 + 757.2s^3 + 1.161e04s^2 + 2.744e05s + 2.075e06}$$

Loop transfer function $L(s)$ —medium speed

$$L_{\text{medium}}(s) = \frac{-417.5s^3 + 9805s^2 + 1.471e04s}{s^4 + 756.1s^3 + 1.077e04s^2 + 2.764e05s + 2.16e06}$$

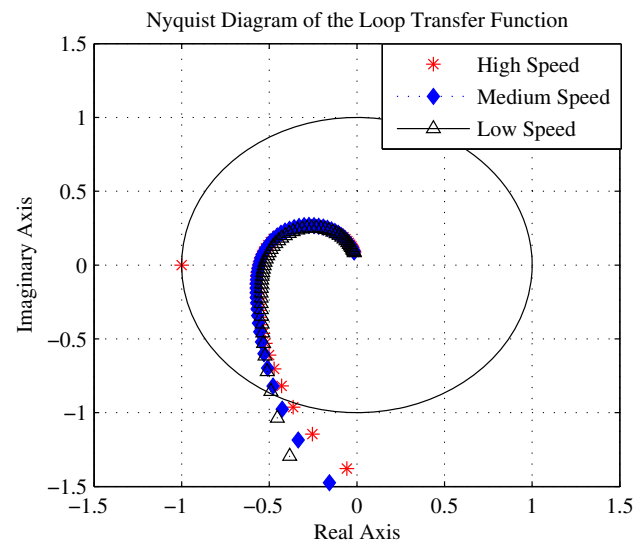
Loop transfer function $L(s)$ —low speed

$$L_{\text{low}}(s) = \frac{-393.4s^3 + 1.069e04s^2 + 1.591e04s}{s^4 + 754.6s^3 + 9636s^2 + 2.84e05s + 2.319e06}$$

These models are validated by using standard tests of autocorrelation of the residual (Ljung 2001). For all identification tests, a suitable excitation signal, of the pseudorandom binary sequence (PRBS) type, is designed to uniformly exciting a range of frequencies around the frequency of the dominant oscillation mode of the hydraulic pumping system. The obtained ARX models for G_{high} , G_{medium} and G_{low} are provided in Table 1.

The corresponding loop transfer functions is computed as follows $L_{\text{high}}(s) = G_{\text{high}}(s)C(s)$, $L_{\text{medium}}(s) = G_{\text{medium}}(s)C(s)$ and $L_{\text{low}}(s) = G_{\text{low}}(s)C(s)$, where the transfer functions $G_{\text{high}}(s)$, $G_{\text{medium}}(s)$ and $G_{\text{low}}(s)$ are obtained by converting the identified discrete-time plant models (given in Table 1) to the continuous-time domain (by using zero-order hold method, with sample time 0.020 s) and $C(s)$ is the designed damping controller given by Eq. (30). The values of L_{high} , L_{medium} and L_{low} are provided in Table 2.

Figure 14 exhibits the corresponding Nyquist diagram for the estimated L_{high} , L_{medium} and L_{low} . As expected, a trend for phase margin degradation as the speed decreases due to a reduction in the value of the phase margin angle. For medium-speed operating condition (blue diamond), the phase margin, verified experimentally, is approximately 63° , which corresponds to a safe phase margin, above that required in the design. This can be explained in part, by differences between the theoretical model used in the design and the real one obtained by identification tests. The controller robustness is demonstrated when the system is operating under the critical operating condition of low speed (black triangle), only a small phase margin degradation is observed. In this case, the phase margin is reduced to a value around 60° , which remains also a safe phase margin.

**Fig. 14** Nyquist diagram of experimentally estimated L . The designed controller C is kept fixed while the operating condition has been varied

6 Conclusion

The phenomenon of dominant oscillations in a motor-centrifugal pump was analyzed herein through a control-based theoretical approach. A feedback control strategy is proposed in order to mitigate dominant oscillations in such type of hydraulic systems. It is concluded that this class of motor-pump systems shows a lack of damping when the system operates at a low-speed regime, an undesirable limiting factor for the safe operation of pumping. To cope with this lack of damping, this paper proposes the design of a damping controller, which is embedded microcontroller-based system. The performance tests, performed in a motor-pump rig system, show that the designed damping controller managed to safely dampen the dominant oscillations. According to the results obtained, the system achieves more effectively compensates the oscillations and the safe operation of the pumping system at a low-speed regime.

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