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# **Design and Field Tests of a Digital Control System to Damping Electromechanical Oscillations Between Large Diesel Generators**

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Additional information is available at the end of the chapter

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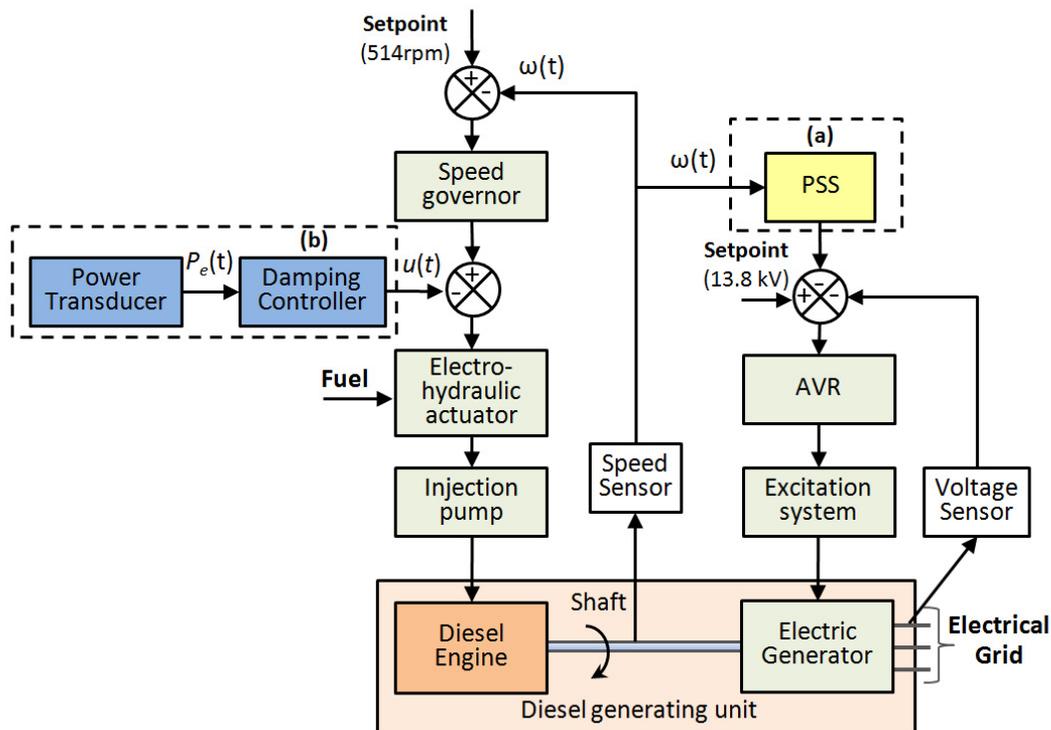
## **1. Introduction**

Electromechanical oscillations are natural phenomena in power systems having two or more synchronous generating units operating interconnected. These oscillations are undesirable because they can severely limit the power transfer between interconnected generating areas, due to reduced stability margins, as well as may decrease lifetime expectancy of system machines. If these electromechanical oscillations are not satisfactorily damped, they may, under some operating conditions, even increase (in amplitude), causing shutdown (tripping) of one or more interconnected generating units. As the oscillations are related to the physical nature of the electrical power system component's interactions, they cannot be avoided. However, by using efficient automatic control techniques, the electromechanical oscillations can be sufficiently attenuated in order assure a safe system operation, for all allowed operating conditions [1, 2].

Among the devices utilized to deal with electromechanical oscillations, the most common are the power systems stabilizers (PSS). As can be seen in Figure 1, dashed box (a), PSS devices usually actuates through the automatic voltage regulator (AVR) in order to increase the damping of poorly damped oscillations modes. The improvement of the damping is obtained through a torque component proportional to the machine speed deviation [1, 2].

An alternative technique, which has been investigated by several authors [3-5], is the application of a damping controller through the speed governor system (see dashed box (b), in Figure 1). This is the approach followed by this work. It is important to remark that this

technique is recommended only for generators systems having fast response actuators, such as diesel engines. The installation of a damping controller via the speed regulation system may be advantageous because theoretical studies show that there is a weak coupling between this control loop and excitation system controllers in other machines of the interconnected power system [3, 4].



**Figure 1.** Comparison between a conventional PSS (a) and a damping controller actuating through the speed governor system (b).

A complete real world example of a control development for safe operation of a power plant has been presented in this chapter. The chapter details the design, implementation and field tests of a digital damping controller applied to a diesel generating unit, at Santana thermoelectric power plant (located in north of Brazil), addressing the system identification and the implementation of the damping controller using digital control techniques. The experimental tests results are presented and discussed.

## 2. System description

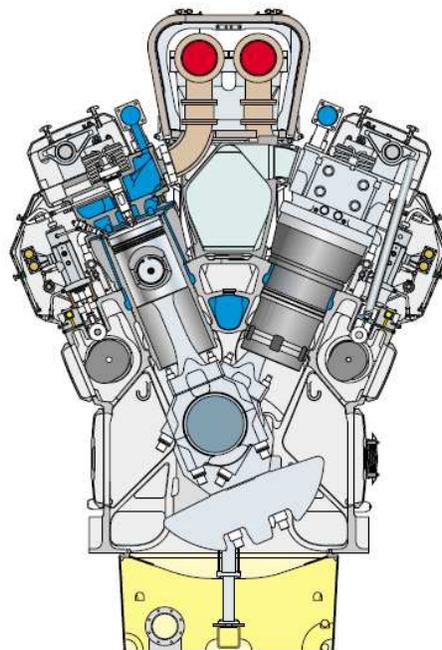
### 2.1. Diesel generating units at Santana Power Plant

Santana Power Plant has a set of seven generating units, including four identical 18-MVA Wartsillä diesel generating units (Figure 2). Due to economic operation constraints, it is advisable to preferentially operating the diesel engines because these machines have the lowest specific fuel consumption among the power plant generating units.



**Figure 2.** Diesel generating units, at Santana Thermoelectric Power Plant.

The diesel generating units are driven by 18V46 Wärtsillä four-stroke diesel engines, which have 18 cylinders in V-form (V-angle of  $45^\circ$ ), as shown in Figure 3. The engine presents a turbocharged and intercooled design, along with direct injection subsystem.

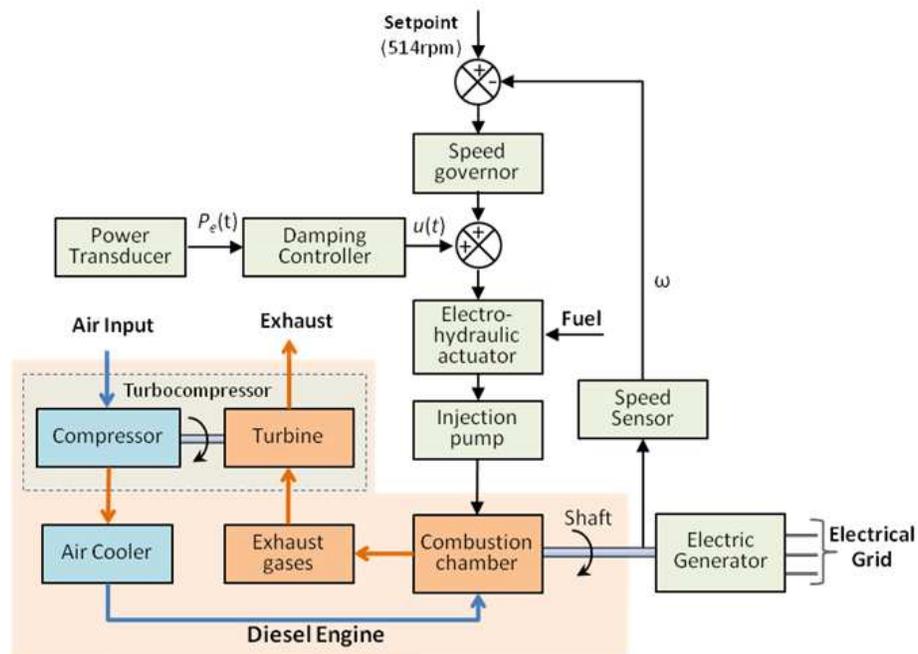


**Figure 3.** Cross section of the V-engine [6].

A simplified block diagram of the diesel generating units is shown in Figure 4. The turbocharging is performed by turbocompressors, which are driven by the exhaust gases from the combustion chamber. The turbine drives the compressor subsystem which in turn draws air from the environment increasing the air pressure. The compressed air is then cooled and, subsequently feedback into the combustion chamber, as illustrated in Figure 4. In order to keep the rotor speed at nominal value, during the system operation, a speed governor controls the loading of the diesel engine by actuating on the electro-hydraulic

actuator position (see Figure 4). Therefore, more or less fuel is injected in order to increase or decrease the mechanical power demanded by the electrical generator and its load.

The damping controller proposed in this chapter actuates in the output of the speed governor, modulating the electro-hydraulic actuator position according to the observed electromechanical oscillation on the measured electric power. Table 1 shows the main technical characteristics of the 18V46 diesel engine:



**Figure 4.** Simplified schematic of the diesel generating unit equipped with the damping controller.

Main technical data of engine		
Parameter	Unit	Value
Engine output	kW	17,550
Generator output	kW	17,025
Specific consumption (Diesel oil)	Liter/kWh	0.22
Cylinder bore	mm	460
Piston stroke	mm	580
Cylinder output	kW/cyl.	975
Engine speed	rpm	514
Piston speed	m/s	9.9
Mean effective pressure	bar	23.6
Firing pressure	bar	180
Charge air pressure	bar	3.1
Weight (Engine + generator)	Ton	290

**Table 1.** Main technical characteristics of the 18V46 diesel engine.

Depending on fuel availability, the 18V46 Wärtsillä diesel engines can be feed by one of the following fuel types: light fuel oil (light oil-diesel), heavy fuel oil, and natural gas. In Santana thermoelectric power plant, the light oil-diesel is the preferential choice. The light oil-diesel characteristics are presented in Table 2.

Light oil-diesel characteristics		
Parameter	Unit	Value
Viscosity, max	cSt a 50°C	11
Density, max	g/ml	0.92
Sulphur, max	% mass	2
Vanadium, max	mg/kg	100
Ash, max	% mass	0.05
Water, max	% vol	0.3
Pour point	°C	6

**Table 2.** Light oil-diesel characteristics.

The fuel injection system is composed of injection pumps, high pressure pipes, and injection valves. The nozzle is located at the top center of the cylinder head. The pressurized fuel in the low-pressure oil line (8.0 to 11.0 bar) has its flow controlled by the electro-hydraulic actuator, which is driven by the output signal of the speed governor (Figure 5). The fuel injected into the high-pressure combustion chamber (180 bar) is atomized and the combustion occurs after the compression cycle.

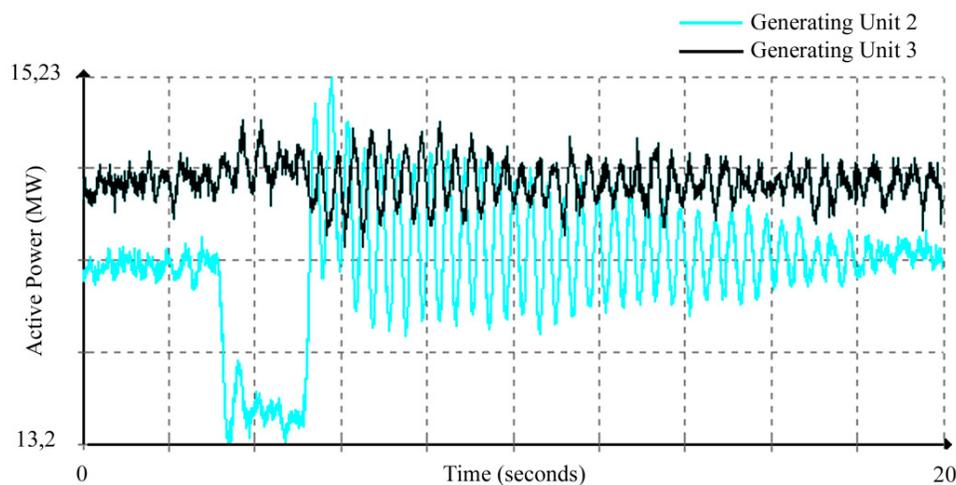


**Figure 5.** Electro-hydraulic actuator subsystem.

## 2.2. Low-damped electromechanical oscillations

Based on a priori information, provided by utility technical reports [7], a dominant 2.5 Hz intra-plant electromechanical oscillation mode was identified having a much reduced

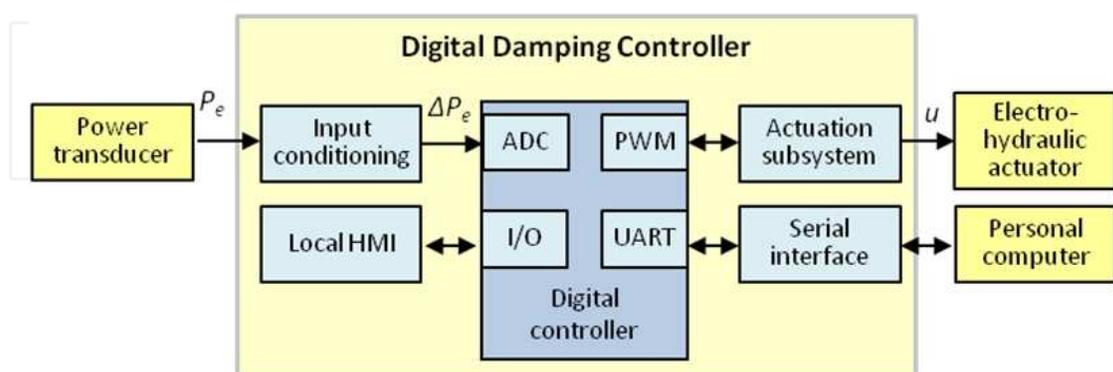
damping, which was observed from the measured machine terminal electrical power signal (Figure 6), when was applied a step in the electro-hydraulic actuator of the generating unit 2. As can be seen, diesel generating units 2 and 3 (G2 and G3) oscillated on phase opposition, which is interpreted as an indicative of intra-plant oscillation mode [2]. These tests were performed in order to investigate the reasons for the frequent machine shutdown (trip) due to actuating of torsional mode protection. Based on this a priori information, the 2.5 Hz intra-plant oscillation mode was then chosen as the target oscillation mode for which the digital controller presented in this chapter was designed.



**Figure 6.** G2 and G3 response to a step variation applied to the G2 electro-hydraulic actuator [7, 8].

### 3. Digital damping controller

In order to perform the field tests, the damping controller was implemented on an embedded system, composed by the main blocks: input conditioning system, digital controller, actuation system, local HMI, and communication interface (Figure 7).



**Figure 7.** Damping controller blocks.

In order to obtain a signal having enough information about the dominant power oscillation, it was necessary to pre-processing the measurement data. To that end, the active power signal ( $P_e$ ) has been chosen as the feedback signal to the damping controller. This

choice was based on field tests, which indicated that the signal/noise ratio can be improved. Therefore, the  $P_e$  signal was applied in the conditioning module, where it was filtered by a first-order low-pass filter in order to attenuate noise and, after that, the signal was processed by a first-order high-pass filter (washout) in order to eliminate the DC component.

The digital processing module of the damping controller is based on a digital signal controller (DSC). This device incorporates features of microcontrollers (variety of internal peripherals) and the DSP's (specific support for digital signal processing). The analog signal  $\Delta P_e$  is converted to digital through an internal 12-bits analog-to-digital converter (ADC). The firmware of the proposed damping controller control law was embedded in the digital signal controller and its control law was programmed by using C language.

The output of the digital controller is generated through an internal pulse width modulation (PWM) module of the DSC. The action of the damping controller in the electro-hydraulic actuator of the diesel engine is achieved by means of an actuator subsystem, which modulates the output current of the speed regulator. This electronic sub-system is implemented as an array of power transistors, which composes a current mirror scheme, which is commanded by the PWM signal.

The communication between the embedded system and other devices is performed through a RS-232 serial interface. Through this communication link, the embedded system is able to transmit the collected data to a personal computer (PC), to perform analyzing and control.

The developed embedded system was designed to operate in four different operational modes, namely: (i) "step response", (ii) "identification", (iii) "control", and (iv) "configuration mode". When in the first mode, the equipment applies a step variation to the electro-hydraulic actuator, collect the plant response and send the collected data to an auxiliary microcomputer PC, for a more complex data analysis. When operating in identification mode, the damping controller may generate a pseudo-random binary sequence (PRBS) and uses this signal test to modulate the electro-hydraulic actuator position of the diesel engine. Using a non-recursive least mean squares algorithm, pairs of input and output data are used to obtain plant estimated parametric models, which are used on the controller design. When in "control mode", the damping controller acquires the signal of active power deviation and processes the damping control law, generating a control signal that is applied in the actuator of the fuel valve of the diesel motor, in order to damp the electromechanical oscillations. Finally, in "configuration mode", the user can set up configuration parameters of the controller through a keyboard and a menu on a LCD display (local HMI).

## 4. Identification of linear models

### 4.1. Model structure

In order to design the digital damping controller, it is necessary to estimate a mathematical model that represents the system dynamics at a specific operation condition. This modeling step can be performed using identification techniques, establishing a system dynamical

model from measured input and output data, which are respectively, the current of command of the fuel valve and the signal proportional to the active power of the generator. The identified model captures the relevant information about the plant dynamic, for feedback control objectives.

A dynamic system model can be represented in several different ways, particularly in time and frequency domains. When the objective is to obtain a dynamic linear model around an operation point using sampled data, a discrete linear parametric model can be used such as an autoregressive with exogenous inputs (ARX) model. The ARX model can be represented in the discrete time domain by [9, 10]:

$$A(q^{-1})y(k) = B(q^{-1})u(k) + v(k) \quad (1)$$

Or by the transfer function:

$$G_p(q^{-1}) = B(q^{-1})/A(q^{-1}) \quad (2)$$

where  $q^{-1}$  is the discrete-time delay operator,  $y(k)$  and  $u(k)$  are the sequences of input and output data, respectively, and  $v(k)$  is assumed to be a white noise, a signal having a flat power spectral density, which is a very useful property for analysis and system identification [9, 10].  $B(q^{-1})$  and  $A(q^{-1})$  are discrete-time polynomials in the form:

$$B(q^{-1}) = b_1q^{-1} + b_2q^{-2} + \dots + b_{nb}q^{-nb} \quad (3)$$

$$A(q^{-1}) = 1 + a_1q^{-1} + a_2q^{-2} + \dots + a_{na}q^{-na} \quad (4)$$

Coefficients  $b_1, \dots, b_{nb}$  and  $a_1, \dots, a_{na}$  are the model parameters, and  $n_b$  and  $n_a$  are integer numbers used to define the model order.

## 4.2. Estimation algorithm

The input-output (I/O) representation (1) can be put into a linear regression form as follows. Let us define a  $n = n_a + n_b$  vector which contains all the coefficients to be identified:

$$\theta^T = [a_1 \ a_2 \ \dots \ a_{na} \ b_1 \ b_2 \ \dots \ b_{nb}] \quad (5)$$

We also define the extended regressor, which will be made up of past I/O data:

$$\phi^T(k) = [-y(k-1) \ -y(k-2) \ \dots \ -y(k-n_a) \ u(k-1) \ u(k-2) \ \dots \ u(k-n_b)] \quad (6)$$

The model output  $\hat{y}(k)$  can be calculated through the product  $\hat{\theta}^T(k)\phi(k)$ . Therefore, the estimating error  $\varepsilon(k)$  is the difference between the measured system output  $y(k)$  and the estimated output  $\hat{y}(k)$ , i.e.,  $\varepsilon(k) = y(k) - \hat{\theta}^T(k)\phi(k)$ .

The actualization of the model parameters  $\hat{\theta}$  can be performed through an iterative algorithm, such as the least mean square method (LMS):

$$\varepsilon(k) = y(k) - \hat{\theta}^T(k)\phi(k) \quad (7)$$

$$\hat{\theta}(k+1) = \hat{\theta}(k) + \gamma \varepsilon(k) \phi(k) \quad (8)$$

where the parameter  $\gamma$  is the step size.

### 4.3. Persistency of excitation and data acquisition

Before data acquisition, a previous system study is necessary to find the best way to obtain dynamic information of the plant. Analyzing system modes frequencies, an appropriate excitation test signal can be designed, which will excite the plant in a range of desired frequencies. This operation results in a better capture of dynamic information of the system, thus improving the dynamical model estimation.

The system to be identified has its dominant modes in a characteristic range of frequencies. Therefore, to excite the plant in an appropriate manner, the input signal must be designed to have an approximate uniform power spectrum in the dominant range. An exciting signal satisfying this property is the pseudo-random binary sequence (PRBS), which is a binary signal that can be generated by using digital techniques [10]. This can be done by using a feedback shift register having a length of  $N$  cells and a sample generation interval  $T_b$ . From knowledge of the minimum and maximum values  $f_{\min}$  and  $f_{\max}$  for the desired range of frequency excitation, the values of  $N$  and  $T_b$  can be calculated by using equations (9) and (10) [9, 10, 11]:

$$f_{\max} = \frac{0.44}{T_b} \quad (9)$$

$$f_{\min} = \frac{1}{(2^N - 1)T_b} \quad (10)$$

## 5. Design of the controller to damp electromechanical oscillations

### 5.1. Pole shifting technique

The damping controller goal is to increase the dominant mode damping, without changing significantly the natural frequency ( $\omega_n$ ) of this mode. In order to perform this task, the pole shifting technique was utilized to obtain a controller able to provide a stable closed loop system and performance characteristics as specified in accordance with the designer requirements. This method is a particular case of the general pole placement method.

In this technique, the open-loop dominant poles must be radially shifted to a new position toward the origin of the unitary circle in the  $z$ -plane. The amount of the radial displacement is specified by a contraction factor  $\alpha$ , according to the desired degree of damping for the closed loop system [12, 13]. Therefore, the designer first specifies a desired value,  $\xi_d$ , for the damping of the electromechanical mode and then calculates the value of the shifting factor  $\alpha$ , by using:

$$\alpha = e^{-(\xi_d - \xi)\omega_n T_s} \quad (11)$$

where  $0 \leq \alpha \leq 1$ ,  $T_s$  is the controller sampling period,  $\xi$  is the natural damping (system without damping controller).

The pole shifting method is based on the search of polynomials  $R(q^{-1})$  and  $S(q^{-1})$ , which satisfy the polynomial equation (12), known as Diophantine equation:

$$A(q^{-1})S(q^{-1}) + B(q^{-1})R(q^{-1}) = A_{cl}(q^{-1}) \quad (12)$$

Where the polynomials  $A(q^{-1})$  and  $B(q^{-1})$  are both known for the designer (model estimated of the plant),  $A_{cl}(q^{-1})$  is a polynomial with the desired closed-loop poles, and  $R(q^{-1})$  and  $S(q^{-1})$  are the polynomials with the parameters of the controller to be calculated:

$$R(q^{-1}) = r_0 + r_1q^{-1} + r_2q^{-2} + \dots + r_{nr}q^{-nr} \quad (13)$$

$$S(q^{-1}) = 1 + s_1q^{-1} + s_2q^{-2} + \dots + s_{ns}q^{-ns} \quad (14)$$

Assuming that  $n_a = n_b = n$  and  $n_r = n_s = n - 1$  (minimum order controller), and matching the coefficients of the same power in  $q^{-1}$ , the coefficients of the control law can be obtained directly by the solution of the following linear equation system:

$$\begin{bmatrix} 1 & 0 & \cdot & 0 & b_1 & 0 & \cdot & 0 \\ a_1 & 1 & \cdot & 0 & b_2 & b_1 & \cdot & 0 \\ \cdot & a_1 & \cdot & \cdot & \cdot & b_2 & \cdot & \cdot \\ a_{n_a} & \cdot & \cdot & 1 & b_{n_b} & \cdot & \cdot & b_1 \\ 0 & a_{n_a} & \cdot & a_1 & 0 & b_{n_b} & \cdot & b_2 \\ \cdot & 0 & \cdot & 0 & \cdot & 0 & \cdot & \cdot \\ \cdot & \cdot \\ 0 & 0 & \cdot & a_{n_a} & 0 & 0 & \cdot & b_{n_b} \end{bmatrix} \begin{bmatrix} s_1 \\ \cdot \\ \cdot \\ s_{n_s} \\ r_0 \\ \cdot \\ \cdot \\ r_{n_r} \end{bmatrix} = \begin{bmatrix} (\alpha - 1)a_1 \\ (\alpha^2 - 1)a_2 \\ \cdot \\ \cdot \\ (\alpha^{n_a} - 1)a_{n_a} \\ 0 \\ \cdot \\ 0 \end{bmatrix} \quad (15)$$

## 5.2. Smith predictor

Field tests performed at the diesel power plant revealed that there is a considerable dead time. For the tests described in this work, dead time is the time delay taken for the electric power signal starts to react after the application of a variation on the electro-hydraulic actuator. Smith Predictor is an adequate method to design controllers taking into account the observed delay [13]. The resulting controller with Smith Predictor consists of the digital controller  $C(q^{-1}) = R(q^{-1})/S(q^{-1})$  with two additional internal feedback loops (see Figure 8), where one is a linearized estimated model of the plant without considering the time delay  $\hat{G}_p(q^{-1})$ , while the other takes into account the model with the time delay  $q^{-d}\hat{G}_p(q^{-1})$ .

## 6. Experimental field tests

The damping controller was successfully installed and validated by experimental field tests carried out at Santana Power Plant. The controller equipment has been installed in a cabinet of the control system of an 18 MVA diesel generating unit (Figure 9). Tests for model

identification and estimation, with subsequent design and test of the proposed damping controller operating in closed-loop have been performed, with results showing an increase of the damping of the dominant electromechanical mode.

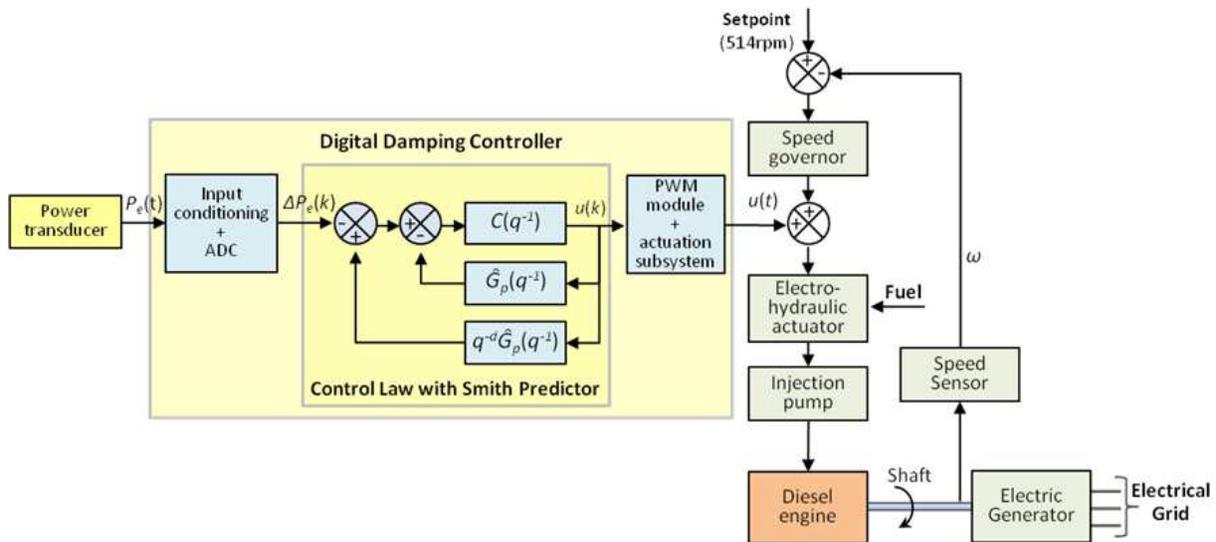


Figure 8. Digital controller with Smith Predictor.

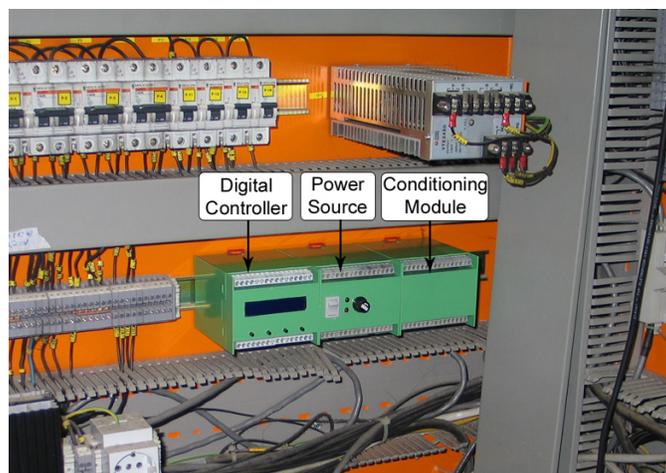


Figure 9. Field tests of the damping controller prototype, at Santana power plant [8].

### 6.1. Analysis of the system response for the application of a step variation

Step response is a useful way for an initial understanding of the system dynamic behavior, revealing some important characteristics that can be used for system modeling. Thus, with the damping controller disabled and the developed equipment programmed to operate only as a step generator, step variations were applied in the command of the fuel valve of the diesel engine. In consequence, the dominant oscillation mode with a frequency around 2.5 Hz, was observed in the electrical power signal. The amplitude of the test signal was configured to 5 mA, which is equivalent to an increase of 5% in the steady state opening of the fuel valve at the operation point considered, while the steady-state valve opening is set at 50 % (9 MW of active power).

These initial tests were also useful to identify possible nonlinearities of the system, such as deadband and time delay. These phenomena are usually found in hydraulic and thermal systems, such as the electro-hydraulic actuator and the diesel engine, and if they are ignored, may be difficult to tune the control system. In order to assess the nonlinearities, a series of tests was performed, which showed that the electro-hydraulic actuator has an excellent sensitivity and a dead zone that does not compromise the system control. So it was not necessary to implement any deadband compensation strategy in the controller. The field tests have shown that the diesel engine actuation system presents a dead time around 400 ms. In order to deal with this observed dead time, a Smith Predictor strategy was applied, as described in Section 5.2 of this chapter.

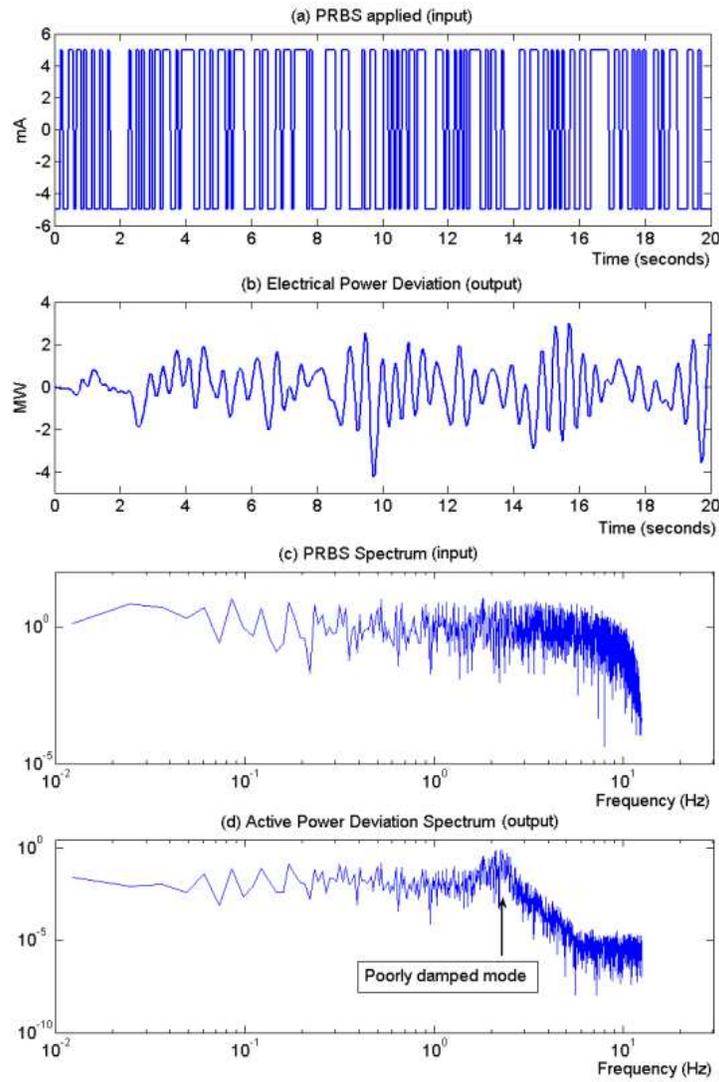
## 6.2. System identification tests

With the goal of obtaining a parametric model for the damping controller design, identification tests were performed in the 18-MVA diesel generating unit without using a damping controller. In order to excite the system electromechanical modes of interest, a PRBS sequence was used, with the parameter  $T_b$  equal to 80 ms and  $N$  equal to 9, resulting in a minimum frequency of 0.02 Hz and maximum frequency of 5.5 Hz, which excite uniformly the range of possible frequencies of the electromechanical oscillations modes (between 0.2 to 3 Hz) [1, 2]. The point of application of the exciting PRBS signal is the same point used for the application of the step response test, as already described in Section 6.1.

The input and output data sets were collected with a 40 ms sampling period and automatically transmitted to a PC, in which data sets were processed for purposes of identifying models representing the system dynamics in the current operating point. Figures 10a and 10b illustrate, respectively, the data obtained from the plant input variable (current variation in the diesel engine valve admission control) as well as from the plant output variable (power generator active power deviation).

From the acquired data, it is possible to make an estimate of the system response frequency spectrum on the operating point considered. The PRBS spectrum is characterized by being approximately uniform in the range specified by the project (Figure 10c), meaning that all modes between 0.1 and 5 Hz (approximately) were equally excited by the designed test signal. As can be seen in Figure 10d, a 2.5 Hz dominant intra-plant mode, having a small damping, can be observed from the electrical power deviation signal. Therefore, it is advisable to design a damping controller in order to improve the damping of this dominant oscillation mode.

The acquired data was divided into two data sets. The first one was used for the parametric model identification process, while the other set was used for model validating purposes. The identification process was carried out using a non-recursive least squares algorithm [10]. A fourth order ARX model structure was chosen, having 4 parameters in the numerator ( $B$ ), 4 parameters in the denominator ( $A$ ) and a discrete-time delay of 10 sampling intervals (400 ms).



**Figure 10.** (a) PRBS applied in generator (b) Active power deviation of the generator (c) PRBS spectrum (d) Active power deviation spectrum [8].

In this way, the resulting identified plant input-output ARX model, which was used for control design, has the following structure:

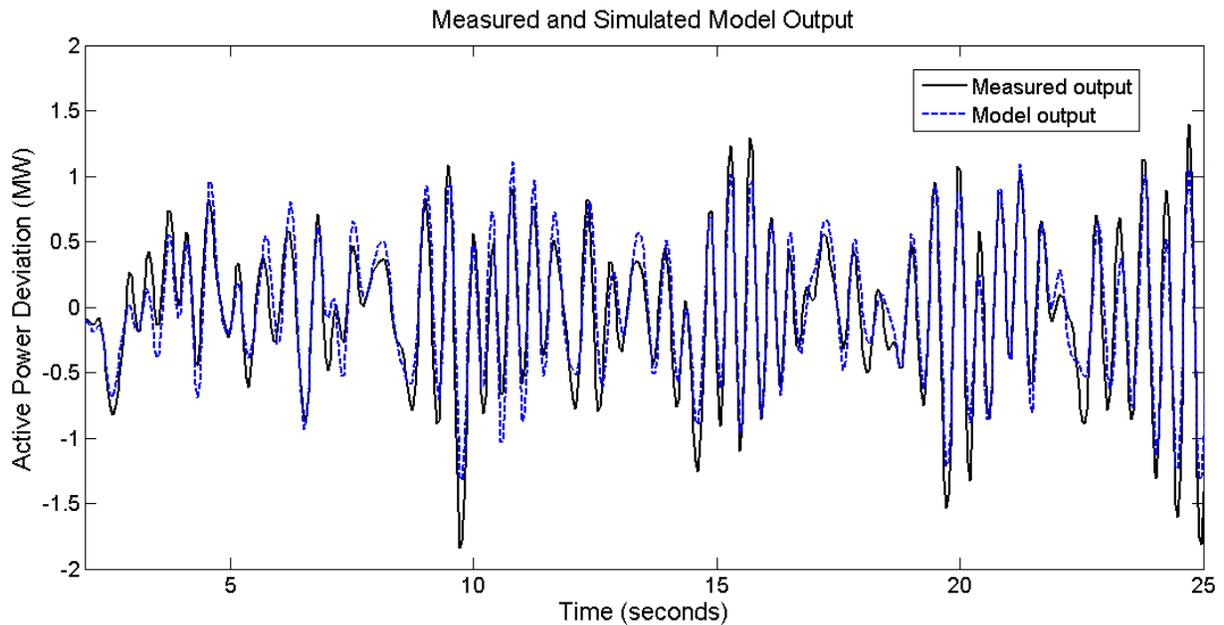
$$q^{-d} \hat{G}_p(q^{-1}) = q^{-9} \frac{B(q^{-1})}{A(q^{-1})} = q^{-9} \frac{b_1 q^{-1} + b_2 q^{-2} + b_3 q^{-3} + b_4 q^{-4}}{1 + a_1 q^{-1} + a_2 q^{-2} + a_3 q^{-3} + a_4 q^{-4}} \quad (16)$$

where the parameters values are presented in the Table 3.

Parameter	a1	a2	a3	a4
Value	-1.9980	1.8254	-0.8676	0.2626
Parameter	b1	b2	b3	b4
Value	0.0033	0.0034	0.0030	0.0028

**Table 3.** Parameters of the model estimated for the plant ( $T_s = 0.04$  s,  $d = 10$  sampling intervals).

Figure 11 illustrates the comparison between the real output of the system and the output estimated using the fourth order model identified in the tests. The result shows the good estimation of the model parameters. It is also verified that the model successfully captured the oscillatory dynamic of the intra-plant electromechanical mode, observed in field tests performed in the diesel generating unit.



**Figure 11.** Comparison between the measured output (solid black) and the output of the model (dashed blue), for the diesel generating unit of 18 MW [8].

### 6.3. Design of the damping controller control law

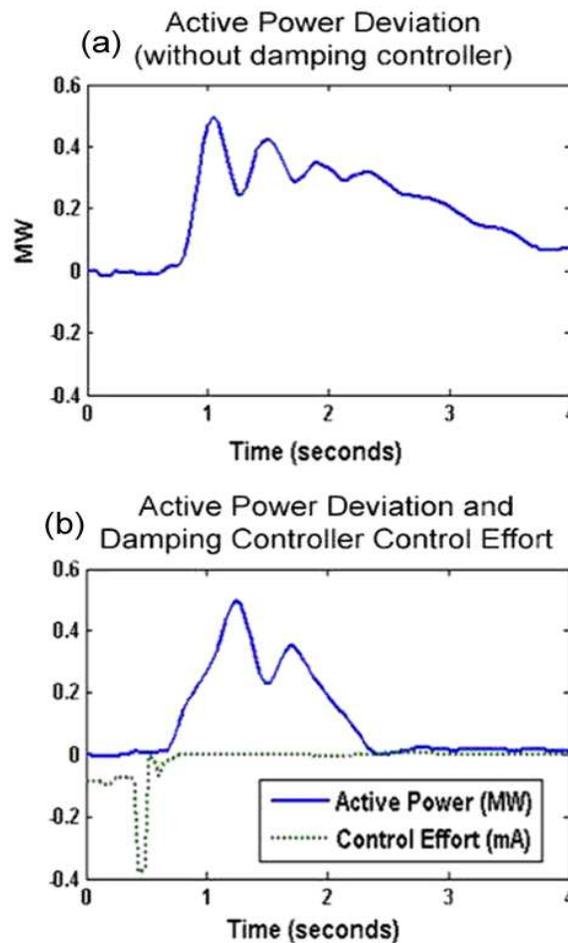
Using the identified model, the damping part of the control law was obtained using the pole shifting method and solving the matrix equation system (15). The controller design main goal is to provide an acceptable damping (in this case,  $\zeta_d = 0.2$ ), without affecting substantially the electromechanical dominant mode natural frequency and not exciting too much any unmodelled dynamics. In the plant fourth order ARX model (B / A), it is considered only one delay of sampling interval. The additional delay of 9 sampling intervals ( $d=1+9$ ) is accommodated through the control law scheme implemented using the Smith Predictor, as explained in Section 5. Table 4 shows the parameters of the damping controller used in field tests.

Parameter	r0	r1	r2	r3
Value	-1.6484	0.5728	-3.7522	-0.3732
Parameter	s1	s2	s3	-
Value	0.1018	0.0287	0.0020	-

**Table 4.** Parameters of the designed digital damping controller ( $T_s = 0.04$  s).

### 6.4. Closed-loop control tests

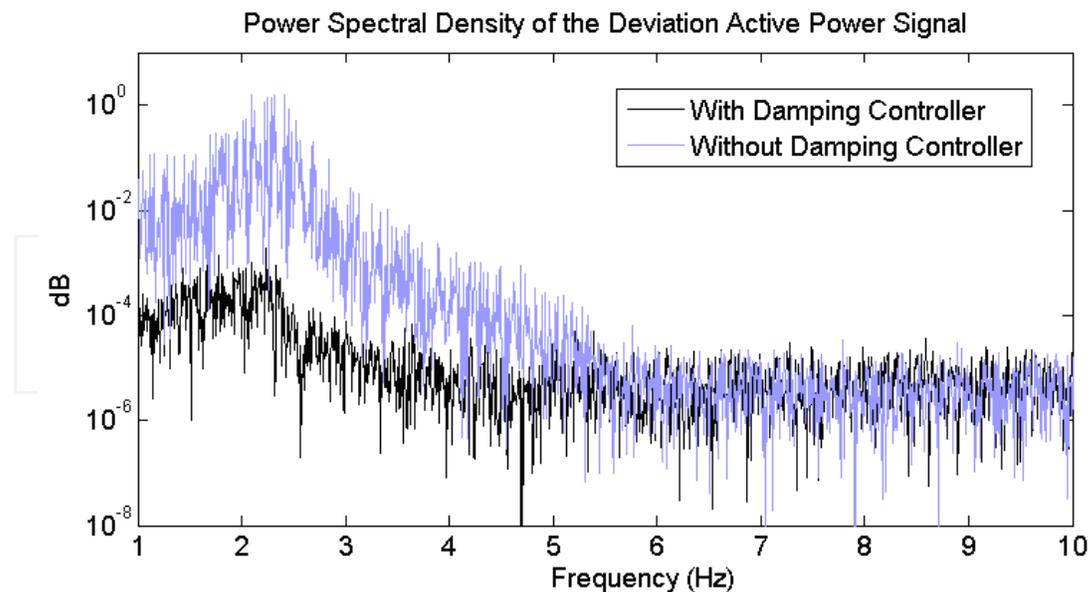
After the controller design, its parameters were inserted into the embedded controller. Then the device was programmed to act in the closed loop control mode (damping controller mode), to provide an additional damping signal to the system, through the speed control loop of the diesel generating unit. To evaluate the performance of the damping controller, step type disturbances were applied, as can be observed during the tests illustrated in Figure 12.



**Figure 12.** System step responses without damping controller (a) and with damping controller (b) [8].

Analyzing the graphics on Figure 13, it can be observed that the inclusion of the damping controller in the speed loop of Wärtsilä considerably improves the stability of the system, making it less oscillatory without affecting the speed governor performance.

In Figure 13 a comparison between the power spectral density of the active power deviation signal collected with the system with and without damping controller is shown. When the data were collected, the generator was excited by a PRBS signal. It is clear from this measurement that the observed electromechanical mode, around 2.5 Hz, is much more pronounced for the case in which the system operates without damping controller, than when the damping controller is acting on the system.



**Figure 13.** Comparison between the power spectral density of the deviation active power signals collected with damping controller and without damping controller [8].

## 7. Conclusion

This chapter presented the design and experimental tests of a digital damping controller which actuates through the speed governor system in order to damp an intra-plant electromechanical oscillation mode. The controller field tests were performed in a real generating unit of 18 MW in the Santana Thermolectric Power Station.

The actuation of the damping controller through the speed governor was only possible due to the fast response of the diesel engine. Thus, this technology is not recommended for use in systems with machines that have low speed of actuation, as in generating units driven by hydraulic turbines.

Results demonstrated that the system performance was improved after the inclusion of the damping controller, thus increasing the system dynamic stability margins. No adverse interactions have been observed in all performed field tests. This task and others, such as, studies of an adaptive controller design, considering a variable time delay in the loop, will be object of future investigations.

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## List of abbreviations

ARX	autoregressive with exogenous input
PWM	pulse width modulation
PSS	power system stabilizer
AVR	automatic voltage regulator
$G_i$	$i$ -th generating unit
HMI	human machine interface
ADC	analog digital converter
UART	universal asynchronous receiver and transmitter
I/O	input/output
DC	direct current
DSC	digital signal controller
DSP	digital signal processor
PC	personal computer
PRBS	pseudo random binary sequence
LMS	least mean squares

## List of symbols

$T_s$	data acquisition and control sample time
$q^{-1}$	discrete-time delay operator ( $q^{-1}y(t) = y(t - 1)$ )
$k$	normalized discrete time
$B(q^{-1}), A(q^{-1})$	numerator and denominator polynomials of the plant ARX model
$\theta$	vector of parameters
$\phi$	data regressor vector
$\varepsilon$	estimating error
$\gamma$	step size of LMS algorithm
$R(q^{-1}), S(q^{-1})$	numerator and denominator polynomials of the digital controller
$N, T_b$	number of cells and sample generation interval for the shift register
$f_{\max}, f_{\min}$	maximum and minimal values for the desired excited frequency range
$\omega_n$	dominant oscillation mode natural frequency
$A_{cl}(q^{-1})$	specified closed-loop polynomial
$\xi_d$	specified value of the damping for the dominant electromechanical mode
$\xi$	estimated value of the damping for the dominant electromechanical mode
$\alpha$	pole-shifting factor
$\hat{G}_p(q^{-1})$	estimated model of the plant without considering the time delay
$\hat{d}$	estimated process discrete-time delay

$q^{-d}\hat{G}_p(q^{-1})$	estimated model of the plant considering the time delay
$P_e$	active power signal
$u$	control signal
$\omega$	rotor speed

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