DAMPING INTER-AREA OSCILLATIONS IN POWER SYSTEMS USING A CDM-BASED PID CONTROLLER

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Abstract: *In this paper, controls of electro-mechanical inter-area oscillations between interconnected power regions are studied. If small amplitude oscillations with low-frequency in power systems do not damp as needed, these oscillations can cause instability of power systems by limiting the system's power transmission capacity. Many control methods have been applied to interconnected systems so far. In this study, a PID controller with derivative filter for damping the inter-area oscillations is designed by Coefficient Diagram Method (CDM) parameters. Time response performances of several previous control methods, such as the derivative filter PID controller, an optimization PID controller model that was developed using MATLAB/Simulink, and the controller designed using CDM are compared. Simulation results show that the CDM-based controller is more successful than the other controllers in damping the inter-area oscillations in power systems.*

Key words: *Inter-area oscillations, Coefficient Diagram Method, PID controller with derivative filter, interconnected power systems, damping oscillations*

1. Introduction

The interconnected energy transport system is the mechanism that provides the stability of energy, or the balance, between the centers of production and consumption throughout the country. The interconnected system adapts energy production to changes in consumption. For many years, because of long transmission lines or high-value power transmission in weak-connected power systems, one of the problems encountered is the existence of low-frequency oscillations between generator groups [1, 2]. These inter-area oscillations are a significant issue in power system stability and control areas. Modern power systems possess three main properties: (1) the measure of interconnected system is getting larger; (2) there is a closer operating point to the power system stability limit because of environmental and economic conditions; (3) being composed of new elements and keeping pace with technological developments unfortunately cause new uncertainties in the stability of power systems. These properties increase the risk of formation of low-frequency oscillations [3].

Electro-mechanical oscillations that emerge in group generators or group plants are called inter-area mode [1]. It is necessary to balance these electromechanical oscillations to ensure the security of the system. In interconnected systems, there are many factors that affect the inter-area oscillations. When inter-area power flow is increased, frequency and damping ratio of the inter-area mode and tie-line impedance are decreased. The effect of excitation systems on the frequency and damping ratio of inter-area mode changes according to the excitation type used. It has been reported in the literature [4] that the best damping in inter-area mode is seen with manually controlled exciters, and the worst damping ratio is seen in fast exciters with transient gain reduction. In addition, characteristics of the load affect the frequency and damping ratio of inter-area mode. Nonlinear loads have a diminishing effect on damping ratio.

Inter-area modes usually have frequencies in the range of 0.1 to 1.0 Hz. However, the properties of these oscillation modes are not known exactly [4]. An exact examination and control of these modes are quite complex. A detailed description of the whole interconnected system is needed to investigate inter-area modes. Recently, investigation of unstable oscillations of inter-area modes in large power systems and controlling them have become challenging tasks.

To date, several methods have been proposed in the literature, such as conventional PID controller, fractional order PID controller (FOPID) [5], adaptive controller based on unified power flow controller (UPFC) [6], and the effect of interline power flow controller (IPFC) [7]. In [8], a particle swarm optimization technique-based UPFC controller for damping inter-area oscillations was investigated. The authors of [9] developed a method for the design of an output feedback controller to damp electromechanical oscillations. In both of these studies, the utility of the proposed controller was tested, and the results obtained by means of nonlinear time domain simulations were shown. Shakarami et al. proposed a controller using static synchronous series compensator (SSSC) for damping inter-area oscillations [10]. In that study, a quadratic mathematic programming method was presented in the design of the stabilizer.

In this paper, a CDM-based PID controller for damping inter-area oscillations is designed, and the time response performances of previous control methods are compared with those of the presented control method. The simulation results show that the CDM-based controller is more successful in damping inter-area oscillations in power systems than the other controllers are. This paper is organized as follows. In Section 2, basic theory of linear systems is explained. In Section 3, the PID controller with derivative filter is explained in a few words. In Section 4, the steps used to design the controller with the proposed CDM method are explained. In Section 5, the CDM-based PID controller is designed. In Section 6, the CDM-based PID controller is applied to the power systems and compared with the PID controller with derivative filter and the optimization PID controller with derivative filter. In Section 7, the obtained results are evaluated.

2. Basic Theory of Linear Systems, Eigenvalues, and Eigenvectors

A power system typically comprises a large number of components. In addition, the behavior of most of these components is described through differential algebraic equations. Hence, in general, the dynamic behavior of a power system can be described by a set of n first-order nonlinear ordinary differential equations, denominated as state equations, together with a set of algebraic equations, developed on the basis of the system model. Using vector-matrix notation, the mathematical model of a dynamic system denoted in terms of a system of non-linear differential equations is as follows [4]:

$$\dot{x} = F(x, u) \tag{1}$$

$$y = G(x, u) \tag{2}$$

In the classification of power system stability, small signal stability is focused on small disturbances. Thus, to analyze the small signal stability of the system mathematically, the disturbances can be considered to be small in magnitude in order to be linear equations that describe the dynamics of the system.

For small perturbations of the system from its initial operating point, (1) and (2) can be expressed in linear form, as follows [4]:

$$\Delta\dot{x}(t) = A\Delta x(t) + B\Delta u(t) \tag{3}$$

$$\Delta y(t) = C\Delta x(t) + D\Delta u(t) \tag{4}$$

The eigenvalues of the state matrix A determine the time domain response of the system to small perturbations, and therefore, provide valuable information regarding the stability characteristics of the system. The stability of the system is determined by the eigenvalues, as follows:

a) A real eigenvalue corresponds to a non-oscillatory mode. A negative real eigenvalue represents a decaying mode. A positive real eigenvalue represents aperiodic instability.
b) Complex eigenvalues occur in conjugate pairs, and each pair corresponds to an oscillatory mode. If all eigenvalues have a negative real part, then all oscillatory modes decay with time, and the system is said to be stable [4].

For a complex pair of eigenvalues $\lambda = \sigma \pm j\omega$, the real component of the eigenvalues, σ, gives the damping, and the imaginary component, ω, gives the frequency of oscillation. A negative σ value represents a damped oscillation, whereas a positive σ value represents oscillation of increasing amplitude. The frequency of oscillation in Hz is given by $f = \omega/2\pi$. The damping ratio *(ζ)* is given by [4]:

$$\zeta = \frac{-\sigma}{\sqrt{\sigma^2 + \omega^2}} \tag{5}$$

The eigenvalue analysis of inter-area oscillations denotes that, in the range of inter-area oscillations, the number of eigenvalues becomes different. The new type of transfer function is given by

$$G(s) = \frac{1}{k}\sum_{i=1}^{k} \frac{1 - \zeta_i}{(s - \lambda_{1i})(s - \lambda_{2i})} \tag{6}$$

Herein, λ_{1i} and λ_{2i} are a conjugate pair of eigenvalues; ξ_i are the damping ratios corresponding to the conjugate eigenvalues; and k is the number of inter-area oscillations. The new damping ratios and oscillation frequency can be obtained by calculating the second-order transfer function, which has only one damping ratio [5].

3. PID Controller with Derivative Filters

Let a dynamic system *G(s)* be controlled by a PID controller *C(s)*, as shown in Fig. 1.

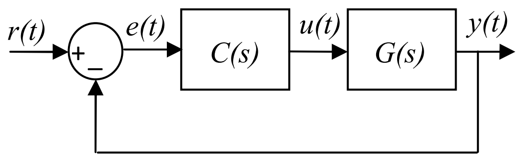

Figure 1. Block diagram of PID.

The PID controller has the form

$$C(s) = k_p + \frac{k_i}{s} + k_d s \tag{7}$$

where k_p, k_i, and k_d are the proportional, integral, and derivative constants of the controller, respectively [11]. The PID controller, *C(s)*, can be expressed in a time constant form, as follows:

$$C(s) = k_p\left(1 + \frac{1}{\tau_i} + \tau_d s\right) \tag{8}$$

where $\tau_i = k_p/k_i$ is the integration time constant and $\tau_d = k_d/k_p$ is the derivative time constant.

In practical control applications, the structure of the PID controller, shown in Fig. 1, is commonly modified through the addition of a derivative filter in order to satisfy the high gain effect induced by high-frequency measurement noise. The modified PID controller can be described as

$$C(s) = k_p + \frac{k_i}{s} + \frac{k_d s}{(1 + \alpha s)} \quad (9)$$

or

$$C(s) = k_p \left(1 + \frac{1}{\tau_i s} + \frac{\tau_d s}{1 + \left(\frac{\tau_d}{N}\right) s} \right) \quad (10)$$

where $\alpha = \tau_d/N$ is a design parameter [12].

4. Coefficient Diagram Method

The controller design with CDM is based on determination of the coefficients of the characteristic polynomial of the closed-loop system with respect to the suitable behavior criteria, such as equivalent time constant, stability index, and stability limit index [13]. The most important features of the method are: use of polynomial representation for the system and controller; use of control system structure with two-degree of freedom; usually, a non-overshoot unit step response of a closed-loop system; controller design with a predetermined required settling time; good robustness of the control system, according to the changes that may occur in system parameters; and having the controller with a sufficient gain and phase margin [13].

CDM's power comes from designing the most robustness and the simplest controller within practical limits for each system to be controlled. This method has been approved in a variety of systems [13].

The designed controller with proposed CDM has the lowest order and the most convenient band width and the time response of the closed-loop system has no overshoot. These properties guarantee robustness, sufficient damping of disturbance, and low cost [14]. A block diagram of CDM is shown in Fig. 2.

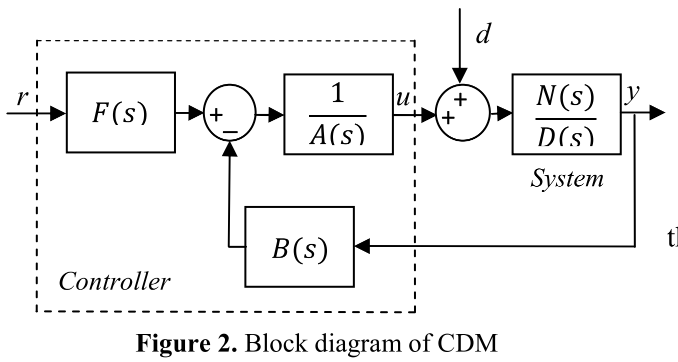

Figure 2. Block diagram of CDM

The terms in the diagram are: r is a reference input; y is an output; d is a disturbance signal; $N(s)$ and $D(s)$ are, respectively, numerator and denominator polynomials of transfer function of the system that it is desired to control. $A(s)$ is the denominator polynomial of the controller transfer function, $F(s)$ is the reference numerator polynomial, and $B(s)$ is the feedback numerator polynomial.

It is disadvantageous that there is a polynomial at the feedback system in practical application. To prevent this drawback, it can be transformed into a new block diagram of the control system, as shown in Fig. 3.

The output of the closed-loop system is

$$y = \frac{N(s)F(s)}{P(s)} . r + \frac{A(s)N(s)}{P(s)} . d \quad (11)$$

P(s) is a characteristic polynomial, and it can be shown as follows:

$$P(s) = D(s)A(s) + N(s)B(s) = \sum_{i=0}^{n} a_i s^i \quad (12)$$

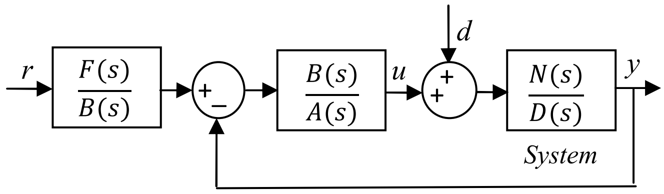

Figure 3. Block diagram of CDM used in practice.

The equivalent time constant τ, the stability index γ_i, and the stability limit index γ_i^*, which are design parameters of CDM, are calculated with the characteristic polynomial coefficients, as follows:

$$\gamma_i = \frac{a_i^2}{a_{i+1} a_{i+2}} \quad i = 1 \sim (n-1), \gamma_0 = \gamma_n = \infty \quad (13)$$

$$\gamma_i^* = \frac{1}{\gamma_{i-1}} + \frac{1}{\gamma_{i+1}} \quad (14)$$

$$\tau = \frac{a_1}{a_2} \quad (15)$$

In addition, coefficients of a_i are calculated as

$$a_i = \frac{a_0 \tau^i}{\gamma_{i-1} \gamma_{i-2}^2 \cdots \gamma_1^{i-1}} \quad (16)$$

The characteristic polynomial is given in terms of the design parameters by

$$P(s) = a_0 \left[\left\{ \sum_{i=2}^{n} \left(\prod_{j=1}^{i-1} \frac{1}{\gamma_{i-j}^j} \right) (\tau s)^i \right\} + \tau s + 1 \right] \quad (17)$$

This equation will be used as a target transfer function. $F(s)$ is a constant, and it is calculated as follows:

$$F(s) = \left(\frac{P(s)}{N(s)} \right) \Bigg|_{s=0} \quad (18)$$

Thus, the steady-state error which may occur in the steady-state response of a closed-loop system is corrected by F(s).

The equivalent time constant is determined as $\tau=t_s/2.5$. When determining the stability index and the stability limit index, according to desired attitude feature, $\gamma_i>1.5\gamma_i^*$ condition is considered.

During the design, the coefficients of controller are calculated as follows:

$$[C]_{sxs}\begin{bmatrix} l_i \\ k_i \end{bmatrix}_{sxs} = [a_i]_{sx1} \tag{19}$$

Herein, l_i and k_i are the unknown control parameters, matrix C is the coefficients of control parameters, and vector a_i is the set of coefficients of the desired target polynomial. Consequently, the parameters of the controller are obtained by solving an equation system with s unknown variables [13].

Because CDM is a method with a polynomial representation, the transfer function of the system is considered as two independent polynomials. These polynomials are the m order numerator polynomial, *N(s)*, and the n order denominator polynomial, *D(s)*. According to this, the polynomials of controller *A(s)* and *B(s)*, whose orders are *p* and *q*, respectively, are obtained as follows [13]:

$$A(s) = \sum_{i=0}^{p} l_i s^i \quad B(s) = \sum_{i=0}^{q} k_i s^i \tag{20}$$

The orders of polynomials show differences according to existence as well as type of disturbance. These orders of polynomials are determined according to Table 1 [13].

Table 1. The choosing order of A(s) and B(s) controller with different disturbance types.

	Without Disturbance	Step Type	Ramp Type	Impulse/ Sinus Type
order{A}	n-1	n	n+1	n-1
order{B}	n-1	n	n+1	n-1
Condition	-	l_0=0	l_0=l_1=0	-
order{P}	2n-1	2n	2n+1	2n-1

5. PID Controller Tuning by CDM

PID control is the most commonly used control method at the present time. It is estimated that more than 90% of control loops in the industry use PID control.

The designed controller using CDM, which is for a second-order system with step type disturbance, from Table 1, is as follows:

$$\frac{B(s)}{A(s)} = \frac{k_2 s^2 + k_1 s + k_0}{l_2 s^2 + l_1 s} \tag{21}$$

The polynomial of the designed controller using CDM for a second-degree system with step type disturbance is similar to the polynomial of the designed PID controller with derivative filter. Therefore, it is proposed in this study that the parameters of the PID controller with derivative filter are rewritten in terms of CDM parameters. (9) was restructured and then equaled with (21); it can be described as

$$\frac{(\alpha k_p + k_d)s^2 + (\alpha k_i + k_p)s + k_i}{\alpha s^2 + s} = \frac{k_2 s^2 + k_1 s + k_0}{l_2 s^2 + l_1 s} \tag{22}$$

From this equation, the parameters of the PID controller with derivative filter are calculated in terms of CDM parameters. They are found as follows:

$$k_p = k_1 - k_0 l_2 \tag{23}$$

$$k_i = k_0 \tag{24}$$

$$k_d = k_2 - k_1 l_2 + k_0 l_2^2 \tag{25}$$

$$\alpha = l_2 \tag{26}$$

6. The Applications of CDM Based PID Controller

6.1. Application I

In power systems, a typical inter-area oscillation mode occurs in the New England 39 bus system. There is only one inter-area oscillation mode in this system. This typical mode, shown in Fig. 4, was considered as an example.

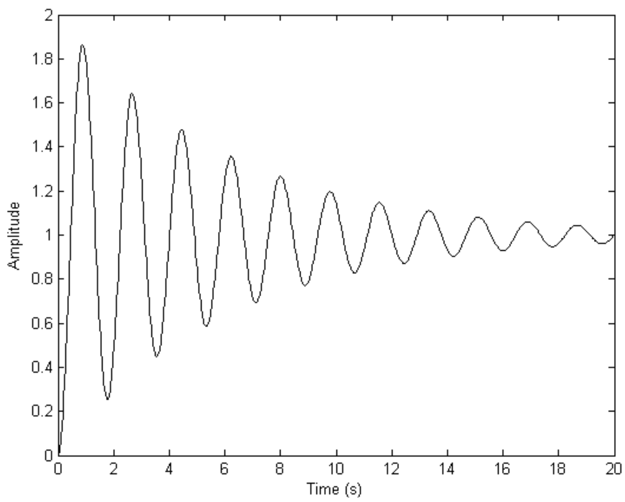

Figure 4. Inter-area mode oscillation in New England 39 bus system.

As shown in Fig. 4, closed-loop response of the plant without a controller had a maximum overshoot of 80%, and the frequency of inter-area oscillation was 0.5376 Hz. The transfer function given by (6) is obtained by using suitable parameters and becomes [5]

$$G(s) = \frac{0.9532}{s^2 + 0.3311s + 12.52} \tag{27}$$

The controller polynomials of the system given by (27) are obtained as $C_1(s)$ and $C_2(s)$ using the PID controller with derivative filter, designed by Zhong et al. [15], and the optimization PID controller with derivative filter designed by using Simulink MATLAB, respectively:

$$C_1(s) = -3.196\left(1 - \frac{1}{1.027s} + \frac{1.082s}{\left(\frac{1.082}{1.082}\right)s - 1}\right) \tag{28}$$

$$C_2(s) = 173.654\left(1 + \frac{1}{3.092s} + \frac{0.257s}{1 + \left(\frac{0.257}{86.392}\right)s}\right) \tag{29}$$

The step responses obtained from the application of Zhong et al., the optimization PID controller, and the designed CDM-based PID controller to the system are shown in Fig. 5.

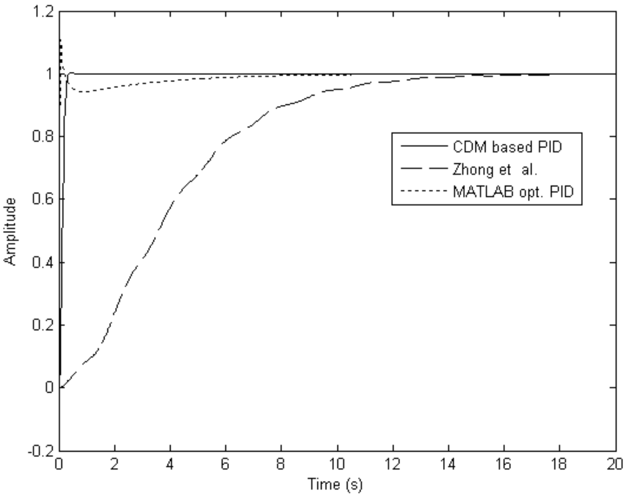

Figure 5. Step responses of controllers.

Looking at Fig. 5, the step response for the PID control with derivative filter [15] had no overshoot, but settling time was 10 s. As also seen in the same figure, the step response for the optimization PID control had a 10% overshoot and settling time of 4 s; however, the step response for the CDM-based PID control had no overshoot and a settling time of 0.6 s. These settling time values were computed using a tolerance band of 2% for the steady-state value. The results of these methods demonstrate that the CDM-based control provided a dramatically improved performance step for input tracking and for disturbance rejection. The shortest settling time is one of the favorable behaviors of the system with the CDM-based PID controller. In addition, the time response of the closed-loop system of the CDM-based PID control was without overshoot.

The disturbance damping performances of the controllers are shown in Fig. 6. In this application, disturbance was applied entirely at zero seconds to the system's input, and time responses given by the controllers were observed.

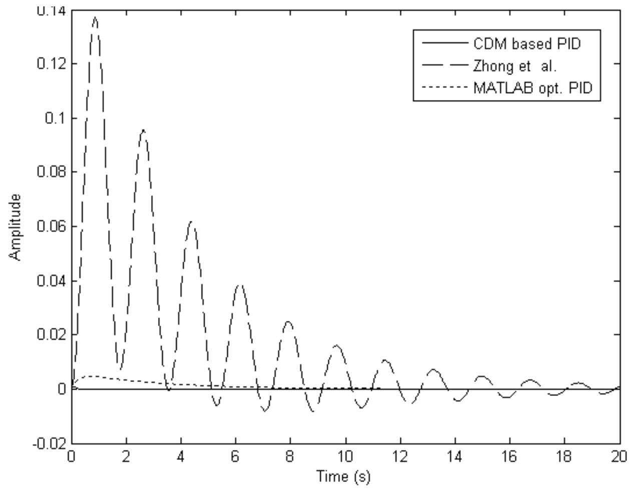

Figure 6. Disturbance damping performance of controllers.

In Fig. 6, it is clearly seen that the best disturbance rejection performance was shown by the CDM-based PID control when comparing these controllers.

In the other application, step type input, which is reference input, and disturbance were applied to the system at the same time. The step responses of the controllers are shown in Fig. 7. As can be seen, there was no change in the step responses of the CDM-based PID control and the optimization PID control when compared with step responses without disturbance of the controller. However, the PID control with derivative filter [15] had a settling time of 12 s.

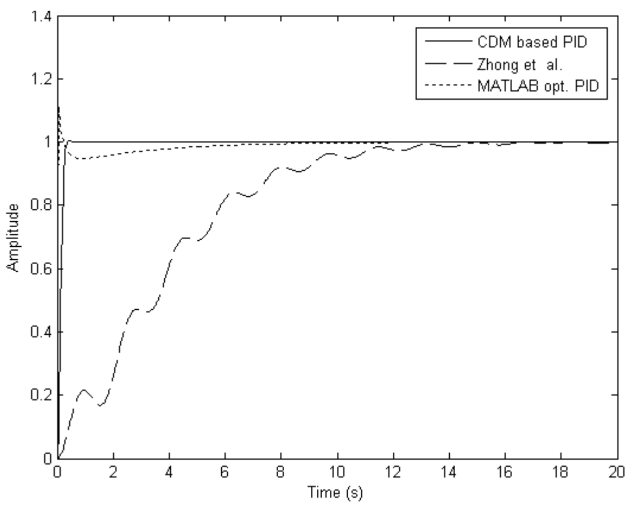

Figure 7. Step responses of controllers for system with disturbance.

The robustness of the proposed controller and the previous control methods were also tested. The ideal robustness analysis method for the power system control design problem would be capable of analyzing the stability and performance robustness of a nonlinear system to parametric uncertainty and unmodeled dynamics. However, such an approach does not exist at the present time, nor will it in the near future.

The step responses of the CDM-based PID controller, the optimization PID controller, and the PID controller with derivative filter [15] for ± 10% changes in the operational point's parameters of the system are shown in Figs. 8, 9, and 10, respectively.

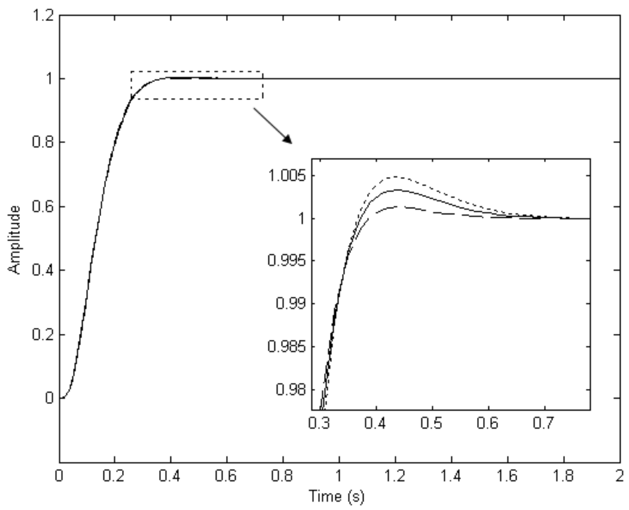

Figure 8. Step responses of CDM based PID controller for ± 10% changes in the operating point's parameters of the system.

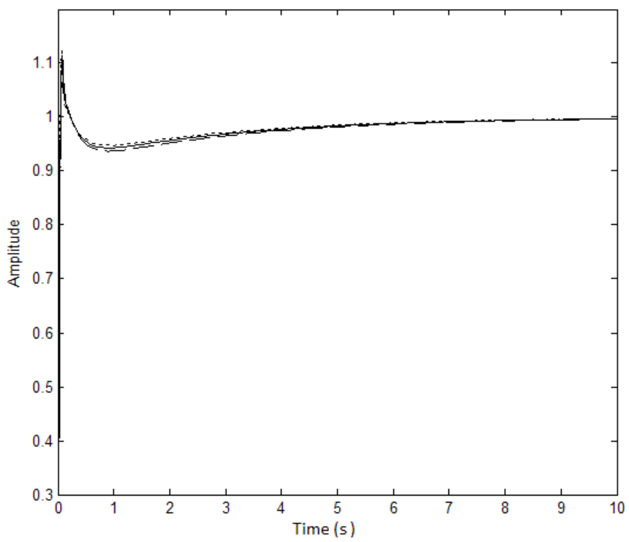

Figure 9. Step responses of optimization PID controller with derivative filter for ± 10% changes in the operating point's parameters of the system.

The CDM-based PID controller was found to be robust enough to change in the operating point of

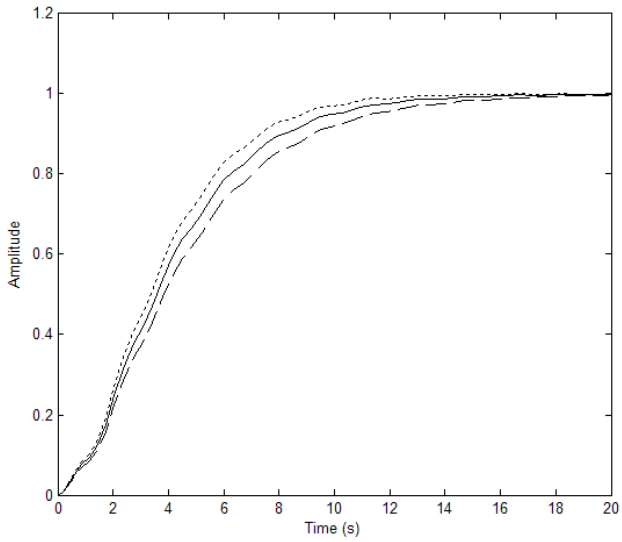

Figure 10. Step responses of PID controller with derivative filter [24] for ± 10% changes in the operating point's parameters of the system.

system parameters, as it adapts itself to generate a suitable variation of the control signals, depending on the operating conditions of the system.

The proposed control structure exhibited a substantial robustness to plant parameter changes and a low susceptibility to inaccuracy of the model adopted to tune the controller.

6.2. Application II

The second example concerns a model with a 2-area 4-machine power system [16] which is depicted in Fig. 11.

In this power system, there is a pair of typical inter-area oscillation modes, $\lambda_{1,2}=-0.501\pm j3.77$ [16]. When using (5) and (6), the transfer function of oscillation modes was obtained, as follows:

$$G(s) = \frac{0.8682}{s^2 + 1.002s + 14.4639} \quad (30)$$

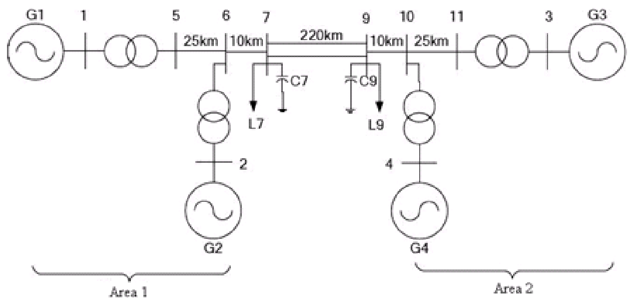

Figure 11. 2-area 4-machine power system.

The inter-area mode oscillation of this system is shown in Fig. 12. It can be seen that closed-loop response of the plant without a controller has a maximum overshoot of 65%, and the frequency of inter-area oscillation is 0.5747 Hz.

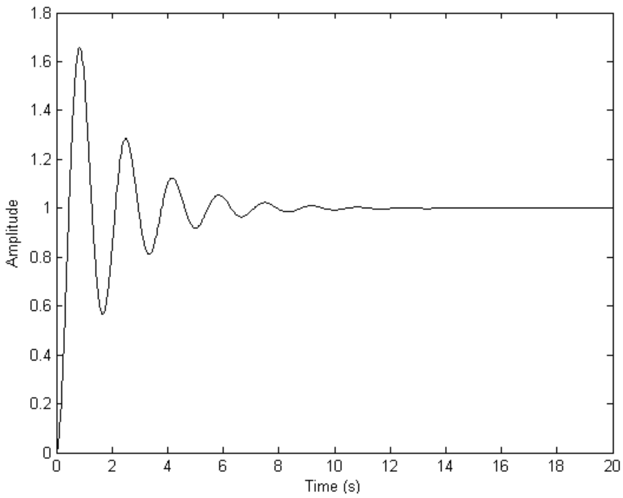

Figure 12. Inter-area mode oscillation in 2-area 4-machine power system.

The controller polynomials of the system given by (30) are obtained as $C_1(s)$ and $C_2(s)$ using the PID controller with derivative filter designed by Zhong et al. [15] and the optimization PID controller with derivative filter designed by using MATLAB/ Simulink, respectively:

$$C_1(s) = -4.0512\left(1 - \frac{1}{0.4114s} + \frac{2.5284s}{\left(\frac{2.5284}{1.0114}\right)s - 1}\right) \quad (31)$$

$$C_2(s) = 281.939\left(1 + \frac{1}{2.689s} + \frac{0.195s}{1 + \left(\frac{0.195}{97.984}\right)s}\right) \quad (32)$$

The step responses, obtained when using the designed CDM-based PID control, the PID control with derivative filter [15], and the optimization PID control for controlling this system, are given in Fig.13.

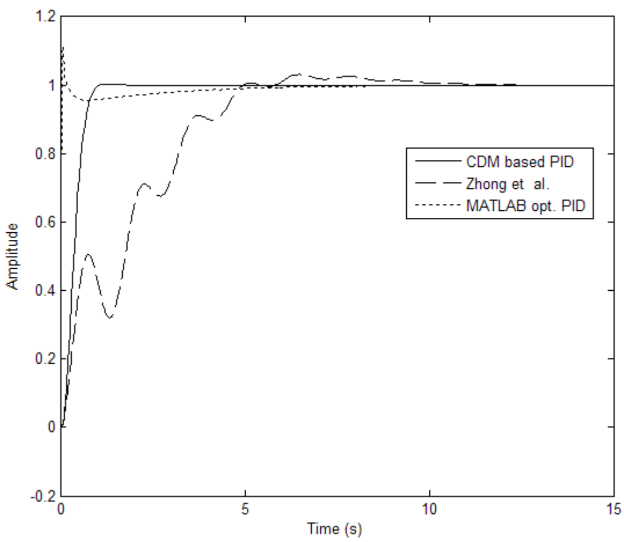

Figure 13. Step responses of controllers.

It can be seen in Fig. 13 that the step response for the PID control [15] and the optimization PID control had a 5% and 10% overshoot, and settling times of 8 s and 4 s, respectively. On the other hand, the step response for the CDM-based PID control had no overshoot and a settling time of 0.9 s. In addition, these settling time values were computed using a tolerance band of 2% for the steady-state value. In this case, the percentage overshoot of the system with optimization PID control and the PID control with derivative filter [15] were greater than that of the CDM-based PID control, and their settling times were longer than that of the CDM-based PID control in the system. That is to say, the time response of the closed- loop system was better with the CDM-based PID control when compared with both of the other controllers. In addition, the CDM-based PID controller had the best rejection of disturbance signal.

The disturbance damping performance of the controllers is shown in Fig. 14. In this application, disturbance was applied entirely at zero seconds to the system's input, and the time responses given by the controllers were observed.

In Fig. 14, it is clearly seen that the best disturbance rejection performance was shown by the CDM-based PID control when compared with the other controllers.

In another application, step type input which is reference input and disturbance were applied to the system at a parallel time. The step responses of the controllers are shown in Fig. 15.

As can be seen, there was no change in settling time of the CDM-based PID control, the PID control with derivative filter, and the optimization PID control when compared with the step responses without disturbance of the controller. However, the PID control with derivative filter [15] had an initial ripple was higher than the previous one, without the disturbance.

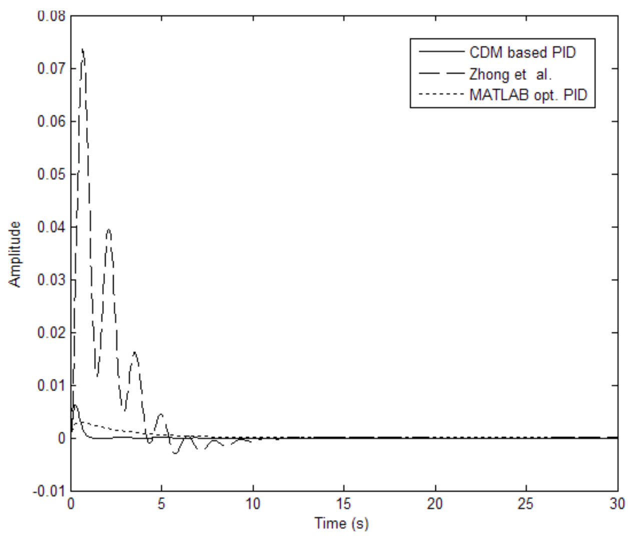

Figure 14. Disturbance damping performance of controllers.

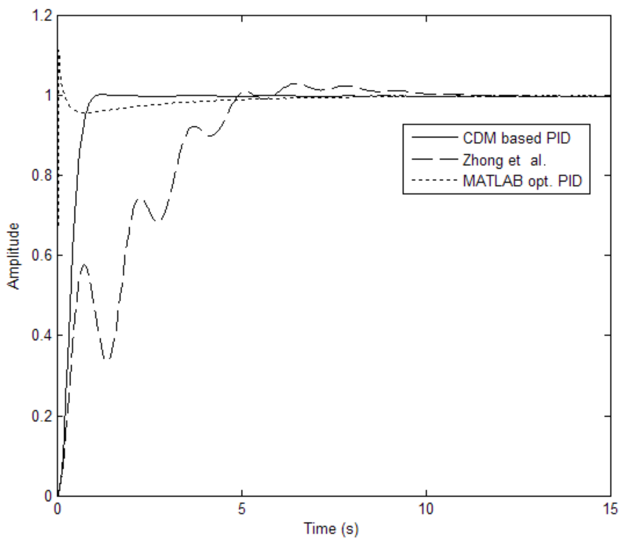

Figure 15. Step responses of controllers for system with disturbance.

7. Conclusions

In this paper, the performance of the designed control method, which is named the CDM-based PID controller, was compared with the performances of the optimization PID controller with derivative filter designed by using MATLAB/Simulink and the PID controller with derivative filter for damping of inter-area oscillations formed between interconnected power regions. In the time domain simulations, it was found that the CDM-based PID controller has many advantages, such as rapidity, simplicity, and robustness.

The PID controller is the most commonly used control method today. In reality, CDM is not widely applied in regard to PID controllers in the industry. However, the results were in accordance with our expectations— short settling time, without overshoot, and satisfactory rejection of disturbance in the response of the system. These results prove that the designed control method using CDM provides the best response, accuracy, and reliability. These, in turn, show that the CDM-based PID controller is feasible in controlling modes that emerge as inter-area oscillations in interconnected power systems.

For the controller design, transfer function of the system, which is a linear form around the operating point, is used, and CDM is employed to optimally tune the parameters of the controller. The effectiveness of the proposed CDM-based damping controller in power-system stability was demonstrated in a New England 39 bus system and 2-area 4-machine power systems. It was observed that the proposed CDM-based controller provides efficient damping to power system oscillations. Further, the proposed controller was found to be robust enough to change in operating points of system parameters, as it adapts itself to generate a suitable variation of the control signals, depending on the operating conditions of the system.

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