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Polynomial based H_∞ robust governor for load frequency control in steam turbine power systems

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ABSTRACT

This work presents an approach to design a load frequency controller (LFC) for power systems with steam turbines. The goal is to damp the oscillations of the output frequency deviations as fast as possible. The design is based on the polynomial H_∞ robust control theory. The robust governor is synthesized by assuming parameter's variations, negligible dynamics, and a constant main steam pressure. The proposed controller will adequately ensure the internal stability and the robust performance of the closed-loop system. The closed-loop control system is tested by subjecting the system to different disturbance signals to show the robustness characteristics, and the well damping of the output frequency under parametric perturbations. The simulation results point out that the system performance is substantially improved.

1. Introduction

Power system stability is the property that enables it to remain in a state of operating equilibrium under normal operating conditions and to regain an acceptable state of equilibrium after being subjected to a disturbance [1,2]. The quality of the power supply must meet certain standard requirements with regard to specific factors. These factors are the constancy of the frequency, the constancy of the voltage, and the level of reliability. Our main concern is regarding the first factor mentioned above.

The most universal method of electric generation is accomplished by the thermal generation using the steam turbine-driven generator units. The steam is produced in steam generators or boilers using either fossil or nuclear fuels as primary energy sources [2]. The poor balancing between the generated power and demands can cause the system frequency to deviate away from the nominal value, and create inadvertent power exchanges between control areas. To avoid such a situation, load frequency controllers are designed and implemented to perform automatically this balancing in each control area [1,3,4].

In [5], the speed governors have been designed based on PID techniques. Fuzzy sliding mode controller for LFC has been designed in [6] to account for the system's parameters variations and the governor backlash. The researchers in [7] used genetic algorithm GA for tuning the control parameters of the Proportional–Integral (PI) control subject to the H_∞ constraints in terms of linear matrix inequality LMI. Modern control techniques have been reported in [8,9], in which a load frequency controller for LFC has been designed using linear quadratic regulation LQR techniques. The work in [10] investigates the design problem or robust load frequency controller using LMI methods for solving the H_∞ control problem. The optimization by the sequential quadratic programming technique is utilized to design a robust load frequency control [11]. In [12], the design of a self-tuning for a PID behavior controller is investigated. An adaptive fuzzy control PID a like controller is designed for an isolated turbine speed control system. Another very interesting technique is the active disturbance rejection control ADRC, which solves the FLC problem by estimating the disturbance on-line, and determining an efficient nonlinear feedback control [13]. In [14], the ADRC is used to design a robust frequency load controller for interconnected power system. The ADRC-based FLC solution is developed for the power systems with turbines of various types, such as non-reheat, reheat and hydraulic.

The FLC problem is not only taken place in isolated power generator systems but also in interconnected electric power systems. Recent works consider the LFC problem in of several-area interconnected reheat thermal power system, where different control schemes are used. In [15], the decentralized LFC problem is solved by using robust optimal PID controller for two-area power systems. For four-area power system, in the work [16], the fuzzy logic technique is employed to solve the problem.

This paper presents a procedure to design a robust H_∞ governor based on the polynomial approach. The work herein considers

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parameters and model uncertainties, which are the main reason of causing the inefficiency of usual frequency load control such as PI and PID or even adaptive fuzzy control PID a like controller. Moreover, the robustness features can be attained within the framework of the linear theory. Therefore, in practice, the implementation of the obtained controller (or a reduced order one) will be easier than implementing the nonlinear feedback control usually used with the ADRC.

2. Steam turbines and speed governing system

A steam turbine converts the stored energy of a high pressure and high-temperature steam into a mechanical energy, which is in turn converted into electrical energy by the generator. The heat source for the boiler may be a nuclear reactor or a furnace fired by fossil fuel (coal, oil, or gas) [1,2].

Steam turbines normally consist of two or more turbine sections or cylinders coupled in series. Most units placed in service in recent years have been of the tandem-compound design. In a tandem-compound, the sections are all on one shaft, with a single generator.

A typical mechanical-hydraulic speed governing system consists of a speed governor (SG), a speed relay (SR), a Hydraulic Servomotor (SM), and a Governor-Controller Valve (CVs). In a steam turbine-generator system, the governing is accomplished by a speed transducer, a comparator, and one or more force-stroke amplifiers. Fig. 1 depicts a conventional block diagram of a closed-loop control system of a steam turbine generator [2]. Appendix 1 gives the detailing of the used symbols and the transfer functions of the individual components [17].

3. Polynomial robust governor

In this work, a different configuration for the problem of LFC of steam turbine has been proposed as shown in Fig. 2. In the proposed configuration, the controller (governor) is placed in the feed forward path in contrast to the conventional governor in which the controller is positioned in the feed backward path. Since the main purpose of the droop feedback is to provide the steady-state speed regulation, in the process of governor control system design, we will temporarily assume that the droop feedback is of unity gain. The proposed configuration will be used to set up the problem within the framework of the H_∞ design methodology. The polynomial methods will be used for design the desired controller.

To start, let us assume a mixed sensitivity configuration for the steam turbine plant as shown in Fig. 3. It includes the performance shaping filters $V(s)$ and $W_1(s)$, and the uncertainty filter, $W_2(s)$. The additive uncertainty is used to compensate for neglected dynamics, which are represented as unstructured uncertainty through $W_2(s)$.

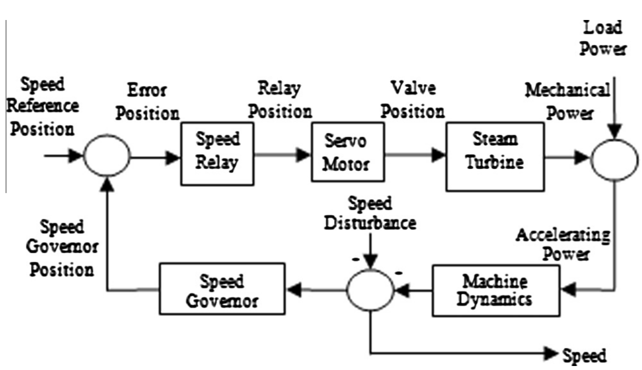

Fig. 1. Block diagram of steam turbine system.

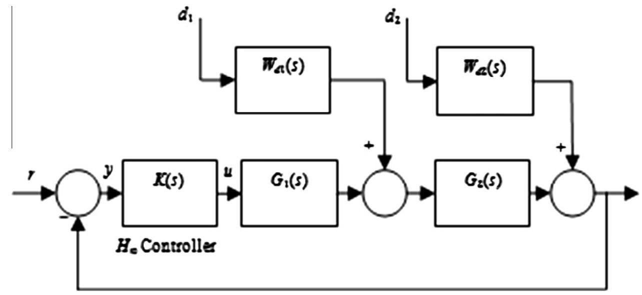

Fig. 2. Proposed configurations for LFC of a steam turbine power system.

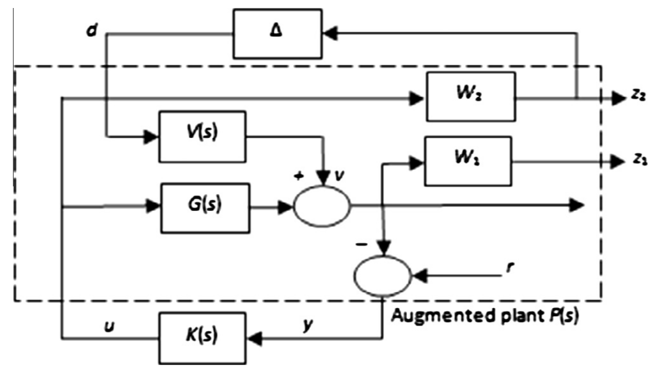

Fig. 3. A mixed sensitivity configuration.

The exogenous input d generates the disturbance v after passing through a shaping filter with transfer function $V(s)$. The control error z has two components z_1 and z_2, which are corresponding to the plant output and input respectively. The transfer functions of the different blocks are given by scalar polynomials as

$$G(s) = \frac{N(s)}{D(s)}, V(s) = \frac{M(s)}{D(s)}, W_1(s) = \frac{A_1(s)}{B_1(s)}, W_2(s) = \frac{A_2(s)}{B_2(s)} \tag{1}$$

where the transfer function $G(s)$ is given by

$$G(s) = G_1(s) * G_2(s) = \frac{N(s)}{D(s)} \tag{2}$$

The design of the shaping filters is highly depended on the model at hand, and certain considerations have to be taken during the design of these shaping filters like uncertainty, high frequency roll-off, and integral control. The system dynamic is described by

$$\begin{bmatrix} z_1 \\ z_2 \\ y \end{bmatrix} = P \begin{bmatrix} d \\ u \end{bmatrix} \tag{3}$$

where the transfer function matrix P of the generalized plant is described by [18,19]

$$P = \begin{bmatrix} p_{11} & p_{12} \\ p_{21} & p_{22} \end{bmatrix} = \begin{bmatrix} W_1 V & W_1 G \\ 0 & W_2 \\ -V & -G \end{bmatrix} = \begin{bmatrix} \frac{A_1 M}{B_1 D} & \frac{A_1 N}{B_1 D} \\ 0 & \frac{A_2}{B_2} \\ -\frac{M}{D} & -\frac{N}{D} \end{bmatrix} = D^{-1} N = [D_1 \;\; D_2]^{-1} [N_1 \;\; N_2] \tag{4}$$

The mixed sensitivity problem schematized in Fig. 3 is the problem of minimizing the H_∞-norm of the closed-loop transfer function matrix

$$T_{zw}(s) = \begin{bmatrix} W_1 SV \\ -W_2 RV \end{bmatrix}, \quad \text{where} \begin{bmatrix} z_1 \\ z_2 \end{bmatrix} = T_{zw}(s)[d] \tag{5}$$

where S is the sensitivity function, $1/(1+KG)$ and R is the control sensitivity function, $K/(1+KG)$.

The robust stability is the property that the closed-loop system remains stable under changes of the plant and the controller. For open-loop stable system, i.e. when both plant and controller are minimum phase and have only right-hand-side poles, then before checking for robust stability, we assume the existence of a nominal feedback loop gain. With respect to numerator–denominator perturbations, a robust stability is guaranteed if the following inequality holds [19].

$$|\delta_L = [-\delta_D \quad \delta_N]|_\infty \leqslant 1 \Rightarrow |\delta_D(jw)|^2 + |\delta_N(jw)|^2 \leqslant 1,\ w \in \Re \tag{6a}$$

The coefficients δ_D and δ_N are scaling coefficients of the additive perturbations in the denominator and numerator of the plant given by

$$\Delta_D = V\delta_D W_1, \Delta_N = V\delta_N W_2 \tag{6b}$$

where Δ_D and Δ_N represent proportional perturbation of the denominator and of the numerator. Fig. 4a and b illustrate these perturbations and the scaling modification.

For stable additive perturbation δ_L, the closed-loop system is robustly stable if the controller K stabilizes the nominal plant and satisfies the following inequality:

$$H_\infty = \left\| \begin{bmatrix} W_1 SV \\ -W_2 RV \end{bmatrix} \right\|_\infty = \left\| \begin{bmatrix} W_1(1+KG)^{-1}V \\ -W_2 K(1+KG)^{-1}V \end{bmatrix} \right\|_\infty < 1 \tag{7}$$

where H_∞ is the interconnected matrix of the mixed sensitivity structure shown in Fig. 4b.

The [.] cost function can also be interpreted as the design objectives of nominal performance, good tracking, disturbance rejection, and robust stabilization, with regard to additive perturbation.

Alternatively, when the inequality 7 cannot be satisfied with a specific controller, then one way to find the stabilizer is to minimize the H_∞ norm of the closed-loop transfer function, T_{zw}. If $T_{zw} > 1$, then there is no controller that stabilizes the system for all perturbations satisfying the inequality 6a. In that case, the stability robustness is only obtained for perturbations satisfying inequality 6a with the right-hand side replaced with $1/\lambda^2$. The constant value λ^2 is given by

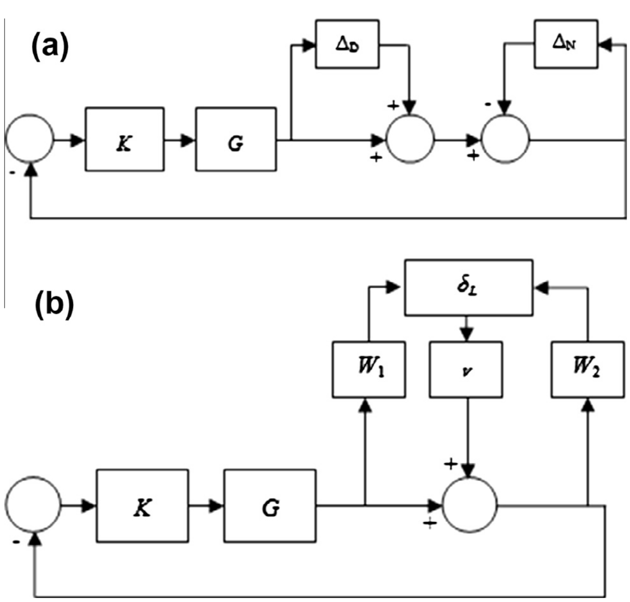

Fig. 4. (a) Fractional perturbation model, and (b) perturbation model with scaling.

$$\lambda^2 = |W_1(jw)S(jw)V(jw)|^2 + |W_2(jw)R(jw)V(jw)|^2 \tag{8}$$

The first term of Eq. (8) dominates at low frequencies whereas the second term dominates at high frequencies. Therefore, within the frame of the optimal solution, it can be easily conducted that the following two inequalities hold

$$|S(jw)| \leqslant \frac{\lambda}{|W_1(jw)V(jw)|}, \quad w \in \Re \tag{9a}$$

$$|R(jw)| \leqslant \frac{\lambda}{|W_2(jw)V(jw)|}, \quad w \in \Re \tag{9b}$$

Therefore, by appreciating choice of the weighting functions W_1, W_2, and V (in particular, with W_2V large at low frequencies and W_2 V large at high frequencies), the functions S and R may be made small in appropriate frequency regions. In such a case, the correct choice gives a scaled perturbation $[-\delta_D \delta_N]$, which satisfies $\|[-\delta_D\text{-}\delta_N]\|_\infty \leqslant 1$, or $\|[-\delta_D \delta_N]\|_\infty \leqslant 1/\lambda$ and hence the stability robustness is obtained in either of the two cases.

For SISO systems and for the mixed sensitivity problem, the H_∞ norm of closed-loop T_{zw} is given by [20].

$$\|T_{zw}\|_\infty = \sup_{-\infty<w<\infty} \begin{pmatrix} |W_1(jw)S(jw)V(jw)|^2 + \\ |W_2(jw)R(jw)V(jw)|^2 \end{pmatrix} \tag{10}$$

The design procedure is reduced to the choice of the weighting filters W_1, W_2, and V. Since the selection of these filters depends on the problem at hands, then the design procedure often involves a thumb of rules, an ad hoc, and a fine tuning such to make the functions S and R small in appropriate frequency regions. Therefore, the design based on the mixed sensitivity problem cannot only give a robust stable closed-loop system, but also to achieve a number of important objectives for the one-degree-of-freedom feedback configuration.

The first step to design a robust governor requires the choice of a nominal plant $G_{nom}(s)$, which is of a simpler transfer function than the original plant $G(s)$. One possible proposed choice of the nominal model for steam turbine system is

$$G_{nom}(s) = G_{1nom}(s)G_{2nom}(s) = \frac{(F_{HP}T_{RH}s+1)}{(T_{CH}s+1)(T_{RH}s+1)} \frac{1}{(T_M s + K_D)} \tag{11}$$

The dynamics of the speed relay $SR(s)$, servo motor $SM(s)$ and the crossover piping in the turbine transfer T_{CO} have been neglected as they have very small-time constants. These neglected high-frequency dynamics will be replaced by an additive uncertainty. The detailed model will be replaced by a nominal model $G_{nom}(s)$, and the uncertainty filter, $W_2(s)$ that accounts for these neglected dynamics and parameter variations.

4. Design procedure using polynomial approach

The design procedure can be carried out by the following five steps:

i. Obtain the nominal transfer function of the steam turbine system.
ii. Obtain the perturbed transfer function, which describes the parameter's uncertainties as well as preparing an additive model of perturbation.
iii. Determine appreciably the filters for the mixed sensitivity model.
iv. Apply the linear H_∞ control theory to obtain the robust controller.
v. Simulate for testing the robust stability and robust performance.

Concerning the data in Appendix 1, the nominal transfer function is given by

$$G_{nom}(s) = \frac{(2.1s+1)}{(0.25s+1)(7s+1)} \times \frac{1}{(8s+2)} \tag{12}$$

The design procedure starts by assuming two real perturbations that are added to the system with the characteristics shown in Table 1. Then, one can determine the perturbed model $G_p(s)$ as

$$G_P(s) = \frac{1}{0.1s+1}\cdot\frac{1}{0.2s+1} \times \frac{0.3(0.4s+1)(T_{RH}s+1)+0.4(0.4s+1)+0.3}{(T_{CH}s+1)(0.4s+1)(T_{RH}s+1)} \cdot \frac{1}{8s+2} \tag{13}$$

where the steam chest time T_{CH}, and the reheat time T_{RH}, are uncertain parameters and their ranges of values are (0.1–0.4) and (3–11) with nominal values of 0.25 and 7 respectively. The set of uncertain plants as defined by the variations of the two parameters T_{CH} and T_{RH}, is given as,

$$\Pi = \{G_p(T_{RH}, T_{CH}); \quad \forall\, T_{RH}, T_{CH}\} \tag{14}$$

To account for the neglected dynamic as well as the parametric uncertainties inherently exist within the system, uncertainty needs to be considered during the design stage of the H_∞ controller. Based on the forgoing analysis, the next step is to design the uncertainty filter $W_2(s)$ which represents the neglected dynamics as well as the variations in T_{CH}, and T_{RH}. At the same time, this filter still acts as a shaping filter to the control the sensitivity function, which in turn improves the performance and provides a high frequency roll-off. Accordingly, a proposed transfer function of the filter, $W_2(s)$ is

$$W_2(s) = W_2'(s) \times rolloff(s) = \frac{(272\ s+0.08)}{(1000\ s+80)} \cdot \frac{(s+80)}{160} \tag{15}$$

where $W_2'(s)$ is the filter that reflects the unstructured and parametric perturbations.

The additive errors $|(G_p(jw) - G_{nom}(jw)|$ and the $W_2(jw)$ are plotted in Fig. 5. As it can be seen from the figure, the filter exactly fits the additive perturbations and covers it entirely. Hence this filter is qualified to represent the uncertainty in the system.

Next step is to design the shaping filter $W_1(s)$, which has to limit the effect of the disturbances up to the bandwidth of the closed-loop system. In this design, the bandwidth of the closed-loop system will set to be at least equal to 0.85 rad/s. Hence, the transfer function of $W_1(s)$ will be

$$W_1(s) = \frac{s+0.85}{s} \tag{16}$$

Further, the disturbance shaping filter $V(s)$ will take the form

Table 1
Nominal value of system parameters.

Parameter	Description	Value	Units
K_D	Damping factor torque (pu)/speed (pu)	2	pu
T_M	Mechanical starting time	8	sec
F_{IP}	IP turbine power fraction	0.4	–
F_{LP}	LP turbine power fraction	0.3	–
F_{HP}	HP turbine power fraction	0.3	–
T_{CO}	Crossover time constant	0.4	sec
T_{SR}	Speed relay time constant	0.1	sec
T_{SM}	Servomotor time constant	0.2	sec
T_{CH}	Steam chest time constant	0.25	sec
T_{RH}	Reheater time constant	7	sec
P_{vmax}	Maximum valve position	1	pu
P_{vmin}	Minimum valve position	0	pu

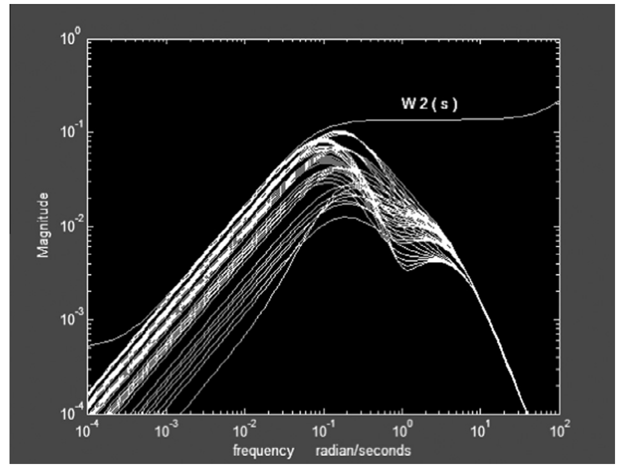

Fig. 5. Additive uncertainties for 36 plant combinations of T_{RH} and T_{CH}.

$$V(s) = \frac{M_{nom}(s)}{D_{nom}(s)} = N\frac{M_{nom}(s)}{(0.25s+1)(7s+1)(8s+2)} \tag{17}$$

where $D_{nom}(s)$ is the denominator of the nominal plant. The numerator polynomial $M_{nom}(s)$ will be selected to be a 3rd-order polynomial as

$$M_{nom}(s) = (s+2)(s^2+1.2s+0.72) \tag{18}$$

The number N is a scale factor, which is introduced to normalize the numerator such to satisfy the condition, $W_1(\infty)V(\infty) = 1$. Therefore, it can easily be found that $N = T_{CH}T_{RH}T_M = 14$. Finally, the transfer function of the filter $V(s)$ becomes

$$V(s) = \frac{14(s+2)(s^2+1.2s+0.72)}{(0.25s+1)(7s+1)(8s+2)} \tag{19}$$

According to the transfer functions $V(s)$, $W_1(s)$, and $W_2(s)$, the polynomial matrix fraction of the generalized plant can be calculated.

$$D_1 = \begin{bmatrix} 10^{-10}+s & 0 \\ 0 & 1+13s \\ 0 & 0 \end{bmatrix}, \qquad D_2 = \begin{bmatrix} s+0.85 \\ 0 \\ 2+23s+62s^2+14s^3 \end{bmatrix} \tag{20}$$

$$N_1 = \begin{bmatrix} 0 \\ 0 \\ -20-44s-45s^2-14s^3 \end{bmatrix}, \qquad N_2 = \begin{bmatrix} 0 \\ 5*10^{-4}+1.7s+0.021s^2 \\ -1-2.1s \end{bmatrix} \tag{21}$$

With these matrices of polynomials in hand, the mixed sensitivity problem is solved with aid of the robust Matlab toolbox to give the H_∞ robust controller.

$$K(s) = \frac{1594.2197(s+4.172)(s+0.08)}{s(s+79.96)(s+3.301)} \times \frac{(s^2+0.6475s+0.1349)}{(s^2+0.3868s+0.04608)} \tag{22}$$

The frequency response of the closed-loop transfer function of the controlled system (the complementary sensitivity transfer function $KG_p/(1 + KG_p)$) has H_∞-norm equal to 0.64308.

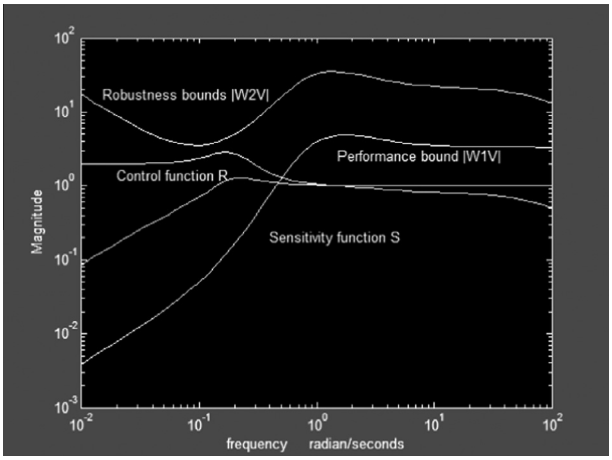

Fig. 6. Singular value plots of S and R with their bounds.

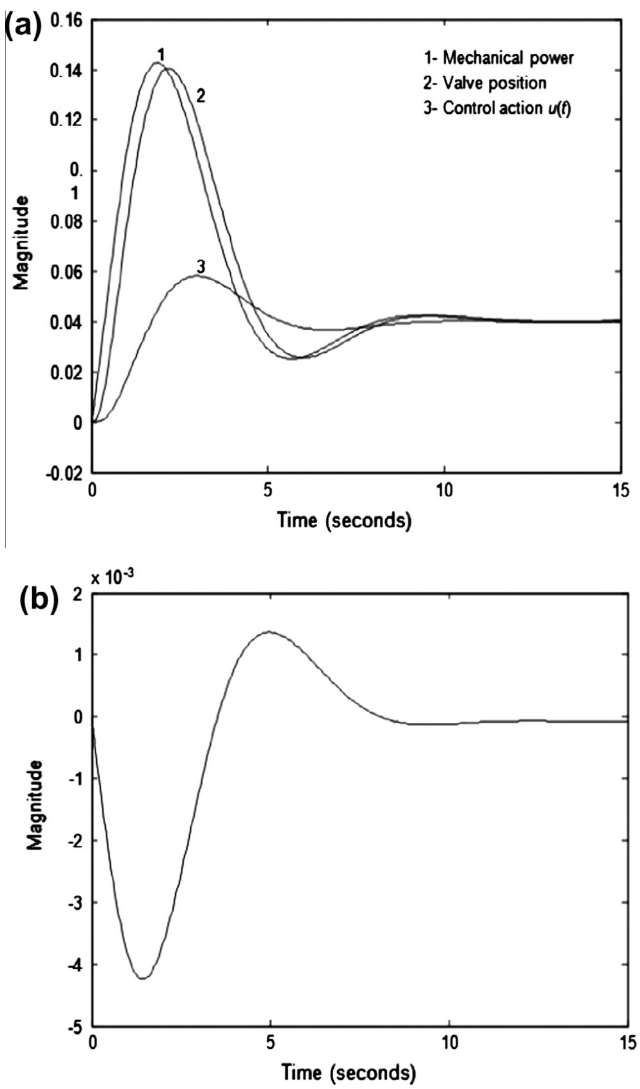

Fig. 7. (a) Transient responses of the mechanical power, valve position and control action. (b) Transient response of the frequency deviation.

5. Simulation and results

The frequency response of the sensitivity S, and control sensitivity R functions together with their bounds are depicted in Fig. 6.

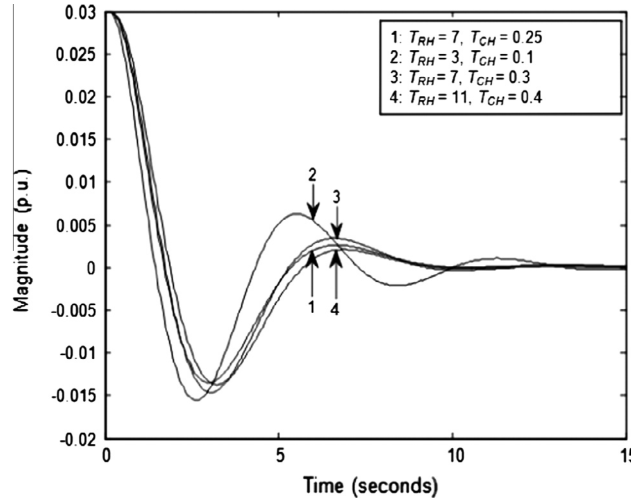

Fig. 8. Transient responses of Δw_r due to parameter variations.

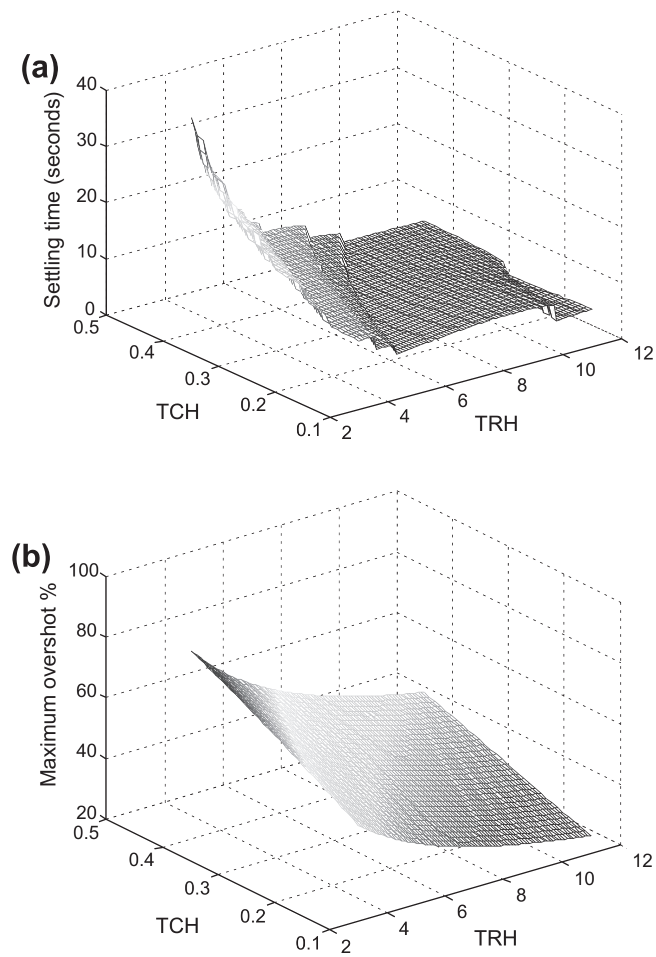

Fig. 9. (a) Settling time, and (b) maximum overshoot for different values of T_{RH} and T_{CH}.

As it is evident in the figure, the sensitivity S and the control sensitivity R functions lie below their bounds. This indicates that the design made quite effective control over the performance of the closed-loop system. This translated into adequate bandwidth of 1.66 rad/s, and very good margins; a gain margin of 37.5 dB, and a phase margin of 50.6°. The nominal stability (NS) of the

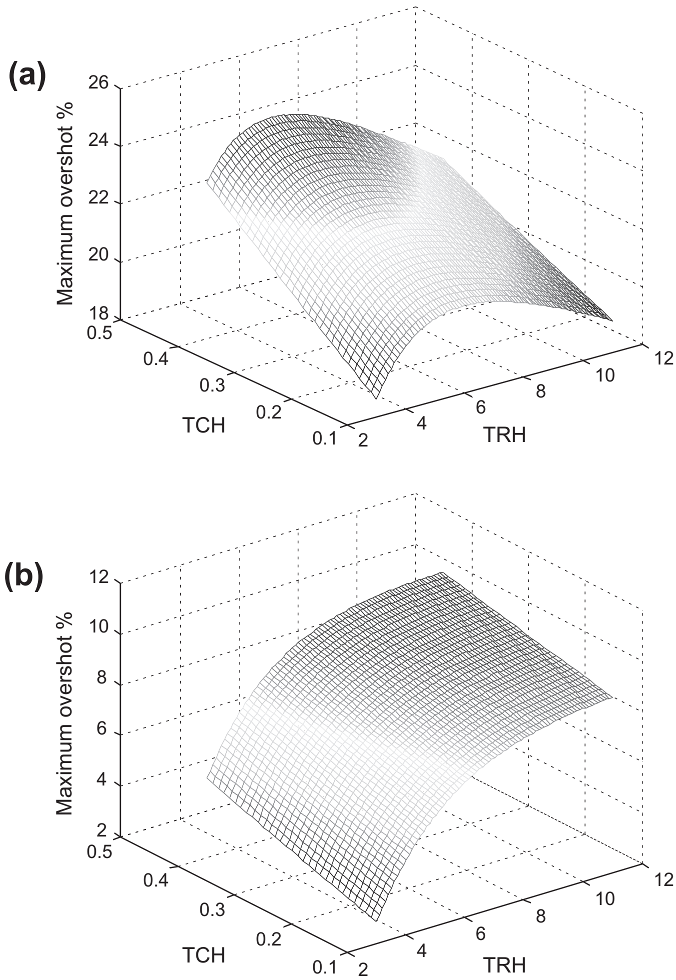

Fig. 10. Maximum overshoot (a) with 0.1 of the forward gain, (b) with 0.05.

closed-loop system is achieved. On the other hand, since the performance bound, $1/|W_1V|$ covers the frequency response of the sensitivity function S over the entire frequency range, then the nominal performance (NP) is guaranteed. Furthermore, since the frequency response of the R function lies below the bound $1/|W_2V|$, then the proposed governor provides the required robust stability (RS) of the closed-loop system. Therefore, consequently, the robust performance (RP) is satisfied.

For a step change in the load ($\Delta P_L = 0.04$ pu), the transient responses of the output frequency deviation Δw_r, and the mechanical power, the control action and the valve position, are shown in Fig. 7a and b respectively.

The settling time of the output frequency deviation Δw_r is about 10.4 s, with undershoot of about 0.004. It is worth mentioning that for larger load demands, the settling time will be almost the same, but with larger undershoot.

To test the system for uncertain parameters, the response of the output frequency deviation Δw_r is plotted for a 0.03 pu speed disturbance input. In Fig. 8, the plots show this deviation for the nominal and different non-nominal values of T_{RC} and T_{CH} parameters. Only with the second choice of T_{RC} and T_{CH} (3, 0.1 respectively), the output frequency deviation slightly oscillates before going to the zero steady-state value; the other two choices result in almost the same behavior as the nominal does.

Finally, it is important to test the stability for whatever the T_{RC} and T_{CH} values are in the assumed ranges. For this purpose, a unit step input is applied to obtain the time response for 1800 pair of values of the two parameters T_{RC} and T_{CH}. Fig. 9a shows the 2% criterion settling time surface, which indicates the achievement of the asymptotically stable response and hence a robust stability. The settling time is ranging from 6.65 to 38.29 s. Correspondingly, the overshoots are ranging from 26% to 82% as shown in Fig. 9b. Therefore, even when the system is strongly under damped, it tracks the input after a finite settling time. On the other hand, by reducing the forward controller gain, the maximum overshoot can be reduced efficiently. Fig. 10a and b show the overshoot behavior for one tenth and one twenties of the forward gain. Obviously, the settling time will be increased with forward gains.

6. Conclusions

A robust governor for load frequency control is achieved using the H_∞ control theory. The design is carried out by the polynomial approach, which improves the robustness behavior of the steam turbine power system to track demands and reject sudden disturbances. Besides the improvement of the system response in time and frequency domains, both nominal and robust stabilities are secured with parameter uncertainties. The robust performance is as well controlled to achieve an adequate system response. Furthermore, integrating action and high frequency roll-off are already met by a relatively law order H_∞ controller.

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Authors would like to thanks and appreciate the assistance and encouragement of the colleagues at the electric department -college of engineering-Baghdad University.

Appendix A.

The transfer functions of the steam turbine control system are as follows [10]:

1. Speed governor $S_G = 1/R$, where R is value of the steady-state regulated speed. The value of R determines the steady-state speed load characteristic of the generating unit.
2. Speed relay $S_R(s) = 1/(T_{SR}\,s + 1)$, where T_{SR} is the time constant of the speed relay.
3. Servomotor $S_M(s) = 1/\,(T_{SM}\,s + 1)$, where T_{SM} is the time constant of the servomotor. The servomotor delivered power with respect to valve position from a minimum value of 0 pu to a maximum value of 1 pu.
4. The steam turbine has the transfer function.

$$S_T(s) = \frac{\Delta P_m}{\Delta P_V}$$

$$S_T(s) = \frac{(F_{HP}(T_{CO}s+1)(T_{RH}s+1) + F_{IP}(T_{CO}s+1) + F_{LP})}{(T_{CH}s+1)(T_{CO}s+1)(T_{RH}s+1)}$$

where ΔP_m is the incremental change of the turbine mechanical power, ΔP_v is the deviation in valve position; F_{HP} is the HP turbine power fraction; T_{CO}, T_{RH}, T_{CH} are the time constants for the cross over, reheater, and steam chest respectively; F_{LP} is the LP turbine power fraction; F_{IP} is the IP turbine power fraction.

1. The machine dynamic has the transfer function.

$$S_M(s) = \frac{\Delta w_r}{\Delta P_a} = \frac{1}{T_M s + K_D}$$

where Δw_r is the deviation of the angular speed of the synchronous generator; ΔP_a is the incremental change of the accelerating power; T_M is the mechanical starting time; K_D is a damping factor.

Table 1 shows the nominal values of the steam turbine power system [17].

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