Proceedings of ASME Turbo Expo 2011
GT2011
June 6-10, 2011, Vancouver, British Columbia, Canada

GT2011-45605

ROBUST GAS TURBINE SPEED CONTROL USING QFT

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ABSTRACT

This paper provides a Quantitative Feedback Theory (QFT) robust control design of a gas turbine in the presence of uncertain parameters. Frequency domain analysis, disturbance rejection properties for SISO and MIMO plants, are among the distinctive features of QFT. In this paper, a QFT robust controller satisfying the required performance despite uncertainties and various constraints on the control effort and process is designed. The nonlinear gas turbine simulator employed in this paper is based on the gas turbine thermodynamic characteristics presented within MATLAB-SIMULINK. The accuracy of this simulator has been examined through several tests by real gas turbine responses.

INTRODUCTION

Gas turbines have various applications in different industrials such as generator driver for electrical power generation (heavy duty) and jet engine in Aeronautics. Having variety of nonlinear components, actuators, and sensors make the gas turbine a nonlinear complex system with many constraints to be satisfied.

Nonlinearity, different constraints on the actuators and control effort, system disturbances which arise from faults and load, directed the engineers towards advanced controller in order to satisfy the mentioned conditions with suitable performance [1], [2], and [3].

Gas turbines and other industrial systems are confronted with ageing, friction and erosion during their operation. Ageing and erosion influence the actuator geometric, valve's characteristic, value of the torque and power losses and other parameters. These changes in mechanical parameters of the gas turbine can enforce deteriorating effects on its dynamic. It has been shown that robust controller techniques can extend the operating life of these expensive systems. In [4] designed H_∞ robust controller improved the gas turbine performance. In comparison with the PI controller the H_∞ controller provides superior responses especially at high speeds.

In [5] and [6], multivariable control techniques for jet engine are presented. These techniques consist of a modified H_∞ method, an order reduction technique to reduce the order of controller and a simplified gain scheduling.

The goal of this work is to design a QFT robust controller to satisfy constraints and performance requirements for an uncertain nonlinear gas turbine. The QFT is a model based technique that requires an uncertain reliable model of the pant. Several models for different purposes have been developed during the recent researches. It has been shown that the mathematical models produced by thermodynamic characteristics and the geometric of the gas turbine components, are the most reliable models that describe gas turbine behavior over its different operating conditions. In [7], mathematical equations based on validated component models are derived and the off design performance of single shaft and twin shaft gas turbines are discussed.

From 1970, in order to examine the transient behavior of the gas turbine without endangering it, computer simulation techniques have been produced. Usually these simulators present the nonlinear complicated models of gas turbines which consist of the main components as the subsystems. In [8], a perfect simulator in MATLAB-SIMULINK is used to analysis on design and off design performances of the gas turbine engine. In on design performance analysis well-known thermodynamic relationships have been considered to calculate component's characteristic and for the off design analysis especial assumption and equation in steady state has been applied.

Since these nonlinear, high order and complex simulators are too complicated for control purposes, various methods have been used to simplify these models. Different identification techniques in time domain and frequency domain are included in these methods [9], [10], and [11].

In our application setup, a nonlinear simulator in MATLAB-SIMULINK describes a generator driver gas turbine engine dynamic behavior during it's under load normal operating. In this simulator each component of the gas turbine such as inlet, compressor, combustion, turbine, and load has been considered as a subsystem.

This study will be concerned with possible uncertainties in four parameters of the gas turbine system applied in the simulator. The uncertain gas turbine model is analyzed, then, a QFT robust control to satisfy constraints, ensure robust stability and performance of the gas turbine is designed.

PLANT DESCRIPTION

Our case study plant is a single shaft under load gas turbine operating normally in 1275 - 1660 rad/s in steady state. The model of this real industrial plant (a single shaft gas turbine) has been presented by a complex simulator in MATLAB-SIMULINK. This nonlinear SIMULINK model of the gas turbine was validated in steady state conditions against engine measurements when they are available, and against the prediction of a more rigorous steady state gas turbine model when measurements were not available. For the majority of variables the accuracy was within 1% [12]. The schematic of main blocks of the gas turbine system are depicted in Fig. 1. Figure 2 shows the gas turbine system in more details. By considering these two figures it is demonstrated that, the gas turbine block in Fig.1 includes the fundamental components of

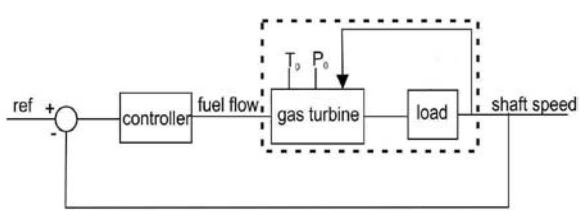

Figure1. Block diagram of the gas turbine system.

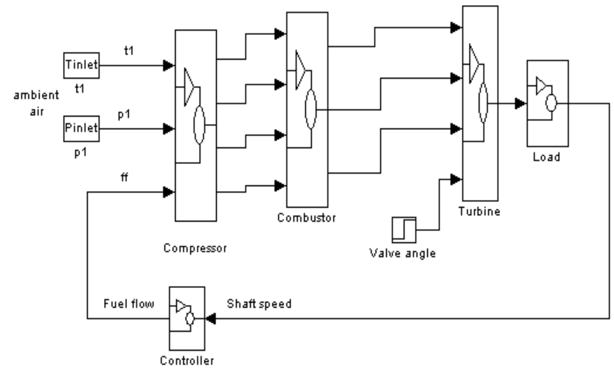

Figure2. The dashed region in more details.

the gas turbine system such as compressor, combustion chamber, turbine, pressuring valve, and other actuators. This block has two inputs T_0, P_0 as the environment conditions and a control input, fuel flow.

The load block calculates the turbine shaft speed based on the load torque (T_{Load}) and the torque losses coefficient ($T_{L\text{-}coef}$). The simplified equation Eq. (1) shows this block's function.

$$\begin{aligned}
\dot{N} &= \frac{1}{J}\left(F(t) - F_{ss}(N)\right)\frac{\Delta(T_{acc}(N))}{\Delta F} \\
T_{acc}(N) &= T_{Load}(N) - T_{loss}(N) \\
T_{Load}(N) &= T_{compressor}(N) - T_{turbine}(N) \qquad (1) \\
T_{loss}(N) &= N^2 T_{L-coef}
\end{aligned}$$

Where

N	Shaft speed, rad/s
J	Total inertia kg m^2/rad
F (t)	Fuel flow at time, kg s^{-1}
F_{ss}	Steady state fuel flow, kg s^{-1}
T_{acc}	Accelerating torque, Nm
$\frac{\Delta T_{acc}}{\Delta F}$	Accelerating torque per unit excess fuel, Nm s kg^{-1}
T_{Load}	Load torque, Nm
$T_{Turbine}$	Turbine torque, Nm
$T_{compressor}$	Compressor torque, Nm
T_{L-coef}	Torque loss coefficient, Nm/ [rad/s]^2

Finally, the controller block consists of two parts. First part is the start up controller which increases the fuel flow in a simple linear ramp to accelerate the turbine shaft speed until it reaches 1204 rad/s. At this specific point, the start up controller part is switched to a classic PI, the second part of controller. The PI controller determines the value of fuel flow based on the obtained error signal. Since this simple PI controller, regulates the turbine shaft speed to its reference set point, percentage of the maximum shaft speed ($\%N_H$) is considered as the system operating point. N_H is the maximum available turbine shaft speed that is 1660 rad/s.

In addition, the controller block contains a speed demand rate limit that does not allow to the gas turbine shaft speed change extremely fast. As a result, the variation of the turbine shaft speed is normally smooth.

The uncertain parameters mentioned in the previous section consist of total inertia (J), torque loss coefficient ($T_{L\text{-}coef}$), pressure valve final position (VF) and pressure valve's mechanical coefficient (CdV).

Total inertia contains the gas turbine inertia and the load inertia. In the mechanical equation, having been discussed in Eq.(1), turbine shaft speed depends on this factor inversely.

Since there are always some kind of torque losses because of mechanical shafts and bearings, the turbine generated torque is never delivered to the load exactly. With considering Eq. (1) it is clear that, the torque loss coefficient effects on the shaft speed indirectly.

VF is the final position of butterfly valve angle and the CdV is the butterfly valve mechanical characteristic coefficient. The pressurising valve is of the butterfly type and it is considered as a nozzle. The flow of this valve is a function of various parameters like A_e (nozzle effective area), CdV, outlet pressure, and outlet temperature. The nozzle effective area is considered to be a function of both valve angle and pressure ratio, as defined by graphical information supplied by the valve manufacturer [12]. The relation between the effective area of the nozzle and the pressure valve angle characteristics presented by Eq. (2) simply

$$F_{Pv} = f_1\left(\frac{P_{outlet} \times CdV \times A_e}{\sqrt{t_{outlet}}}\right)$$
$$A_e = f_2\left(\frac{P_{outlet}}{P_0}, VF\right) \qquad (2)$$

Where

F_{Pv} Pressure valve flow, kg s^{-1}

P_{outlet} Outlet pressure, Nm^{-2}

P_0 Inlet pressure assumed to be ambient, Nm^{-2}

A_e Nozzle effective area, m^2

VF Final pressure valve angle, deg

t_{outlet} Outlet temperature, K

f_1 A complicated mathematical function.

f_2 A graphical function based on valve manufacturer.

The pressure valve's flow determines the value of the turbine mass flow which is instrumental in turbine shaft speed. Empirical gas turbine studies reveal that the four mentioned parameters have approximately 15 or 20 percents of uncertainties.

The dynamic of the gas turbine changes during the whole range of operation, in case other conditions remain the same. As a result, it is an acceptable assumption to consider variation of the shaft speed as an additional system uncertainty.

The dashed section in Fig.1 has been identified as an uncertain single input single output plant which the fuel flow is the input signal and the shaft speed is its output.

MODEL IDENTIFICATION

In this paper, linear autoregressive identification with exogenous input has been used to achieve simple linear models in different operating points with uncertain parameters. For this investigation it is critical to feed the system input with a suitable signal and measure the output carefully. It is considerable that the environment conditions T_0 (area temperature) and P_0 (area pressure) are considered constant in this study.

As a complicated nonlinear system, gas turbine has a high order dynamic which requires an input with sufficient persistency excitation for identification purposes. This input can excite all modes of the plant and accurate system identification results will be obtained. After many experiments a PRBS signal with the amplitude of 0.05% of the reference set point has been selected and added to the turbine shaft speed reference signal. This signal makes a perturbation on the shaft speed set point during the simulator is running. Figure 3 shows this PRBS signal with its frequency spectral.

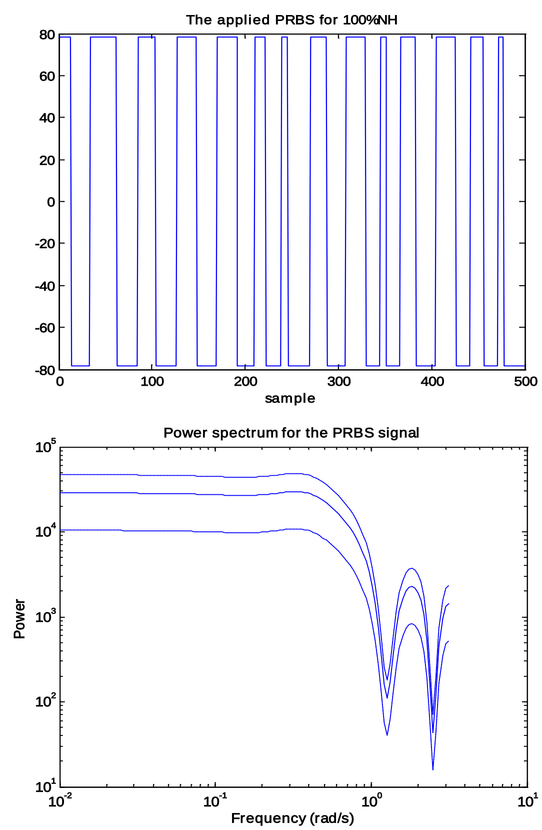

Figure3. The applied PRBS signal and its frequency spectral.

The measured input and output signals contain almost 500 samples. The first 400 samples are used for identification purposes and the rest are applied for the model validation process. In order to accept the identified model, the accordance between the measured output and the models generated response should be more than 85%. These linear polynomial models describe gas turbine's behavior in its full range operating point with parameter uncertainties.

Simulation studies reveal that variation of total inertia and torque losses affect the gas turbine response (turbine shaft speed) in inverse direction while, any changes in CdV and VF effect on the shaft speed directly. Therefore, in the following stage, 4 different values for the total inertia and the torque losses and also 5 different values for the CdV and VF have been considered as below

$$\begin{cases} J = (1+j)J_{no\min al} \\ T_{L-coef} = (1+j)T_{L-coef-no\min al} \\ CdV = i.CdV_{no\min al} \\ VF = i.VF_{no\min al} \end{cases} \qquad (2)$$

$$\begin{cases} i = \{0.9, 0.95, 1, 1.05\} \\ j = \{-0.2, -0.1, 0, 0.1, 0.2\} \end{cases}$$

So 20 samples for uncertain system will be achieved, which cover the most possible uncertainties of the gas turbine system in each operating point.

By varying the mentioned parameters and the operating set point, several uncertain linear models have been identified. The obtained models are in discrete time and the sampling time is considered equal to the simulator sampling time $T_s = 0.08_{(\mathrm{sec})}$. The structure of the obtained models remains the same as illustrated by Eq. (3).

$$G(z) = \frac{b_2 z^2 + b_1 z}{z^3 + a_2 z^2 + a_1 z + a_0} \qquad (3)$$

The produced discrete models have been converted to continous time transfer functions. During this change the order of the models increased to 4 such as Eq. (4).

$$T(s) = \frac{b_3 s^3 + b_2 s^2 + b_1 s}{s^4 + a_3 s^3 + a_2 s^2 + a_1 s + a_0} \qquad (4)$$

The set of these uncertain identified models presented by P. $P(s) = \{T_1(s), T_2(s), \ldots\}$. Upper and lower uncertainty bounds of the coefficients of the transfer function model are derived from the identified models. As a results, the system's uncertainties definition changes to parametric uncertainties in each operating point.

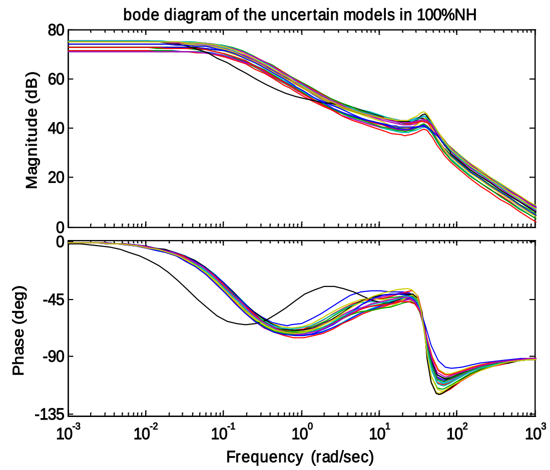

Figure4. Bode diagram of 20 sample models of uncertain gas turbine in 100%NH.

The bode diagram of the 20 samples of the identified models in 100%NH has been plotted in Fig.4, which obviously describes the uncertainty in the plant.

The aim of this paper is to build up a QFT robust control which is able to satisfy the desired performance and constraints and also has the ability to be applied for the whole operating range (in different gas turbine speed).

QFT CONTROLLER DESIGN

General structure of the QFT controller design is shown in Fig.5. The steps involved in the controller design are the computation of the G(s) and F(s) transfer functions, where G(s) expresses the compensator and the F(s) presents the pre-filter of the system.

After identifying, modeling and then, illustrating the model's parameters bounds, the first step to design a QFT robust control is to set upper and lower bound transfer functions. Based on the performance requirements these two LTI transfer functions should be found that the responses of the controlled identified models by QFT, lie between the upper and lower responses respectively.

Nowadays, reducing the energy consumptions is a priority in the national economy, so controlling gas turbines in order to decrease fuel consumptions and increase the energy efficiency is of great importance. This study tries to make the plant response faster with lower control effort as much as possible.

For this investigation the upper and lower bound transfer functions are considered as:

$$T_l(s) \le T(s) \le T_u(s) \qquad (5)$$

Both bound transfer functions are in second orders format and producing critical step responses with different settling times and undamped natural frequencies.

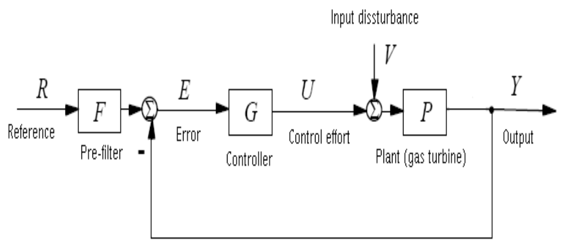

Figure5. General structure of QFT controller with uncertain plant.

Using the QFT toolbox in MATLAB the templates of uncertain model has been sketched on the Nichols chart for different system frequencies such as $w = \{.1, 1, 10, 100, 1000\}$.

These templates illustrate that if a robust design is achievable for the uncertain plant. In order to have a complete template, various conditions have been examined by changing the parameter value in their bounds. The obtained templates in the middle frequencies such as $w = 100Hz$ are expanded but in low and high frequencies are contracted.

One of the uncertain models is arbitrary chosen as the nominal plant.

For the next stage in control design, all necessary constraints are converted as a bound on the Nichols chart [13]. In our application setup, the main constraints include acceptable performance with suitable gain margin and phase margin in addition to control effort limitation.

It is desired that the closed-loop system be robust stable and has at least 50° phase margin for all $P(s) \in \mathcal{P}$ with at least 1.66 lower gain margin. The following formulae have been used to compute these margins in general [14].

Lower gain margin = $1 + W_s^{-1}$

Lower phase margin = $180^\circ - \theta, \theta = \cos\left(0.5W_s^{-1} - 1\right) > 0$

So it is considered that

$$\left|\frac{PG}{1+PG}(j\omega)\right| \le W_s = 1.2 \quad \omega \in [0, \infty) \quad (6)$$

On the other side, the amplitude of the controller effort relative to the shaft speed reference should be less than 0.0005 in order to prevent saturation in the system's actuators.

$$\left|\frac{U}{R}(j\omega)\right| = \left|\frac{G}{1+PG}(j\omega)\right| \le 0.0005 \quad \omega \in [0, \infty) \quad (7)$$

By using these bounds during the loop shaping, designer is able to judge whether all boundaries are critical or some of them are less important. For the first step of loop shaping the compensator and pre-filter are considered as unit transfer functions. During the loop shaping of the nominal loop gain in the Nichols chart an integrator is added to the controller to reduce the steady state error. Loop shaping is performed to satisfy all the mentioned constraints in all frequencies. At the end of this stage the final controller as G(s) is derived. Now the loop gains of the uncertain models satisfy the bounds. For the next step the pre-filter should be designed to enforce the close loop responses are between the upper and lower bounds.

The achieved G(s) and F(s) are as follows:

$$G(s) = \frac{1727s^3 + 3.084\times10^6 s^2 + 1.513\times10^8 s + 1.673\times10^7}{s^4 + 2.142\times10^4 s^3 + 4.691\times10^7 s^2 + 1.009\times10^{10} s} \quad (8)$$

$$F(s) = \frac{3.617}{s^2 + 4.801s + 3.617} \quad (9)$$

RESULTS

To have a correct comparison between the simulator PI governor and the designed QFT robust controller, both of them have been tested in two cases. At first, these controllers have been applied to the identified uncertain models of the gas turbine under the same condition, and secondly, the QFT controller has been implemented as a new block in MATLAB-SIMULINK replaced instead of PI governor.

Case 1: In this case, the PI governor of the nonlinear simulator and the QFT designed controller has been applied as the controller blocks for the identified models separately. This test is carried out in 80%NH and after 40 seconds when the system reaches to its steady state a step with amplitude as 0.1 the final value of shaft speed has been added to the reference signal. Applying this step to the reference signal makes an uncertainty in the speed form and examines the ability of tracking the reference signal in both controllers. Results are presented in Fig.6. The blue curves in this figure are the responses of the uncertain identified models with PI governor of simulator and two dashed violent responses belong to the upper and lower bound transfer functions which determine the desired region for close loop response of the controlled models by QFT method, and eventually, the red curves present the responses related to the same models controlled by QFT. As it is obvious, the uncertain identified models' responses are within the mentioned bounds because the designed QFT controller in the pervious section has been utilized. Figures 7 and 8 show the control effort generated by QFT and PI for each of these uncertain models. Since the upper and lower bounds have fast performances the closed loop performances of the QFT controller are faster than the PI controller. The peaks of the controller effort in QFT controller are almost twice of the PI.

Case 2: In this case, the QFT designed controller is presented as a Simulink block and substituted for the PI part in the controller block of origin nonlinear simulator. In order to complete this change, the new block should be matched with other parts of simulator. Therefore, such as the PI controller, QFT controller should start it application after that the turbine's shaft speed reaches to its suitable speed (1204 rad/s). Also, a speed demand rate limit should be considered. The results of the step responses of this new simulator for all range of mentioned uncertainties in 80%NH have been sketched in Fig.9. It is worth to mention that a step with amplitudes of 0.1 of the final value of the shaft speed has been applied to reference signal as the case 1. Like before, the blue curves belong to original nonlinear simulator and the red responses related to the new simulator which contains the designed QFT controller. Figures 10 and 11 show the control effort generated by new and original simulators for the uncertain gas turbine respectively.

The results clear that, the QFT controller satisfies the performance requirements completely. During the uncertainties in different parameters in the gas turbines system, all responses lie in the suitable region. The fuel flow signals in Fig.10 have lower peak in comparison with the Fig 11.

It is demonstrated from these two cases that, the QFT controller ensure the stability of the gas turbine system and satisfies its performance in spite of different uncertainties. In addition, because of the integrator factor in this designed controller the steady state error and tracking error is zero.

CONCLUSION

In this paper, a QFT robust controller that is able to control the gas turbine in the face of parameters uncertainties and different constraints has been designed. It is shown that the designed QFT controller controls the gas turbine in its whole operating range and also in the process of changing the operating point properly. Moreover, this kind of robust controller satisfies all constraints and performance requirements during the operation.

ACKNOWLEDGMENT

This research was supported by K. N. Toosi University of Technology. The authors would like to thank all people who help us with their technical comments during the preparation of this manuscript.

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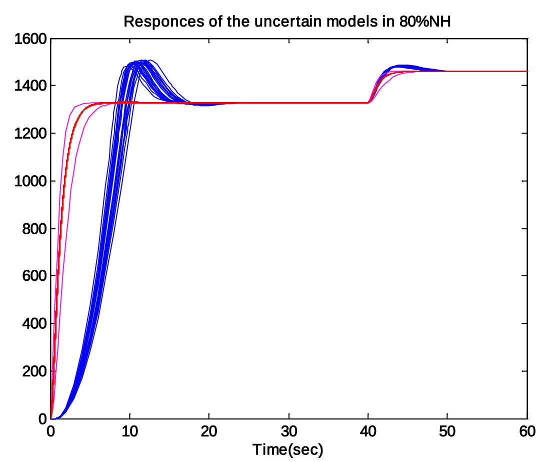

Figure6. Step responses of the uncertain models with PI and QFT controllers

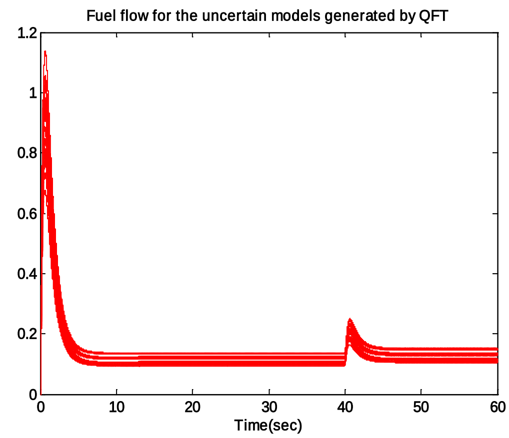

Figure7. Control effort signals for uncertain models generated by QFT controller in 80%NH.

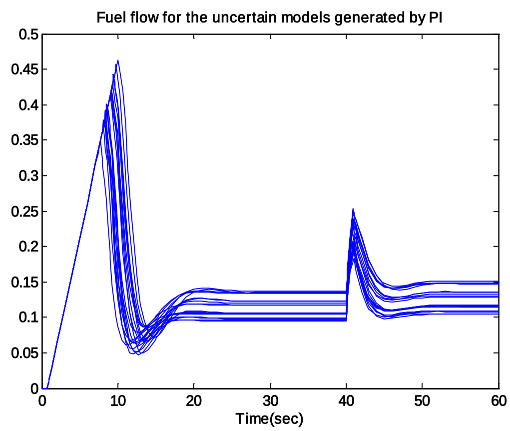

Figure8. Control effort of uncertain models generated by PI governor in 80%NH.

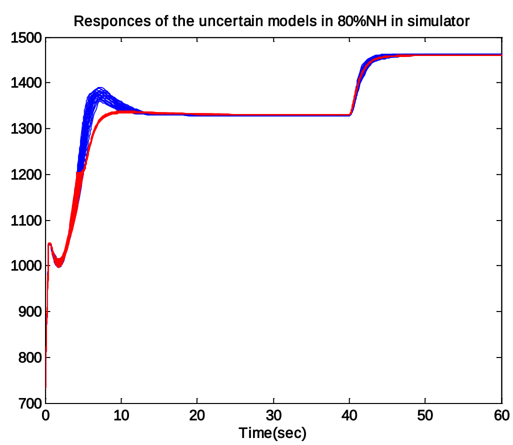

Figure9. Step responses of the uncertain models in steady state with original and new simulators.

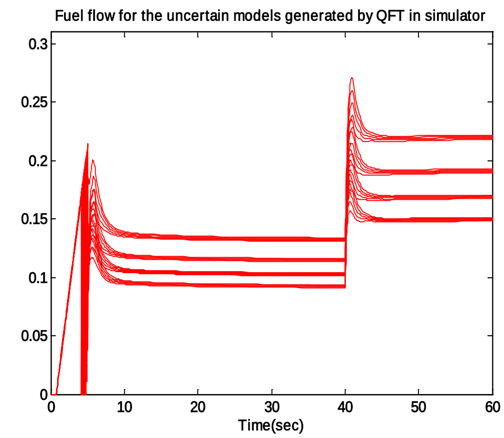

Figure10. Control effort of uncertain models generated by QFT controller in 80%NH in new simulator.

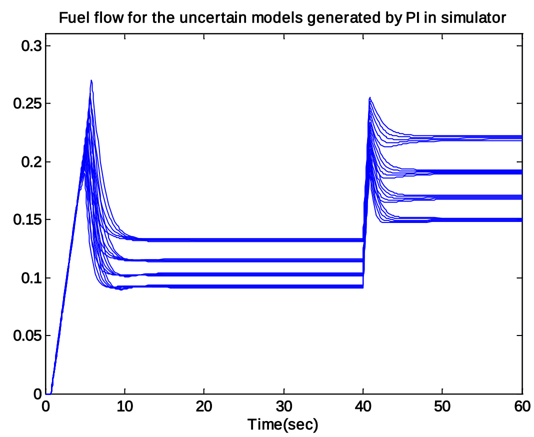

Figure11. Control effort of uncertain models generated by PI governor in 80%NH in original simulator.