



ADDIS ABABA UNIVERSITY
ADDIS ABABA INSTITUTE OF TECHNOLOGY (AAiT)
SCHOOL OF ELECTRICAL AND COMPUTER ENGINEERING
ROBUST CONTROL OF INDUSTRIAL BOILER USING
QUANTITATIVE FEEDBACK THEORY (QFT)

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Robust Control of Industrial Boiler Using Quantitative Feedback Theory

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DECLARATION

I declare that

- (i) The research reported in this dissertation, except where otherwise indicated is my original work.
- (ii) This dissertation has not been submitted for any degree or examination at any other university.
- (iii) This dissertation does not contain other persons' data, pictures, graphs or other information, unless specifically acknowledged as being sourced from other persons.

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ABSTRACT

Most practical Boiler systems are characterized by high uncertainty which makes them difficult to maintain good stability margins and performance properties for closed loop system. In case of conventional control, if plant parameter changes we cannot assure about the system performance. Hence it is necessary to design robust control for uncertain plant. Among the various strategies proposed to tackle this problem, Quantitative Feedback Theory (QFT) has proved its superiority especially in the face of significant parametric uncertainty. The feature of QFT is that it can take care of large parametric uncertainty along with phase (degree) and magnitude (dB) information at each working frequency. To achieve good performance of industrial boiler system which reduce fuel consumption rate and improves efficiency, dynamic variables such as fuel flow, air flow and pressure of boiler must be controlled. In the first step plant is identified by using experimental data by the mean of converting into a group of linear time invariant (LTI) uncertain plants. After representation of the uncertain dynamic system in general control configuration and modeling the parametric uncertainties, nominal performance, robust stability and robust performance against disturbances are analyzed by the concept of robust control.

Finally, Linear time invariant (LTI) has been carried out and two controllers are compared. As in the PID controller case, the initial PID design balances reference tracking and disturbance rejection. In this case as well, the controller yields some overshoot in the reference-tracking response, and suppresses the input disturbance with a longer settling time. The output in response to a unit step disturbance should remain within the range $[-1, 1]$ at all times, and it should return to 0 as quickly as possible $y(t)$ should at least be less than 0.1 after 3 sec).

From simulation we observed settling time 20.9 seconds and overshoots 30.7%. But both Controller and pre-filter controllers guarantee robust performance of the system against the uncertainties and result in desired time responses of the output variables. By applying QFT robust control, system tracks the desire reference inputs in a less time and with smoother time responses. It is observed from the simulation results that the overshoot is **0%**, rising time is **0.0199** seconds and the settling time is **0.0369** seconds and Peak time is **0.0939** seconds with QFT controller. It is further observed that the proposed controller has robust stable and Performance.

Keywords: Boiler, Pressure, Air flow, Fuel flow, QFT, Stability Margin, robust, performance

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LIST OF SYMBOLS

$G(s)$	controller transfer function (s-domain)
$P(s)$	plant transfer function (s-domain)
$P_o(s)$	nominal plant transfer function (s-domain)
$F(s)$	pre-filter (s-domain)
$L(s)$	open loop transfer function (s-domain)
$L_o(s)$	nominal open loop transfer function(s-domain)
$R(s)$	reference input (s-domain)
$D(s)$	disturbance (s-domain)
$Y(s)$	output signal (s-domain)
P	parameter space
T_R	Input-output tracking bounds
T_D	Disturbance bounds
$T_R(s)$	Tracking control ratio
$\delta_R(j\omega)_{HF}$	spread of tracking bounds at high frequencies
M_p	peak overshoots
t_s, t_x	settling time
t_p	peak time
B_u	Upper tracking bound
B_l	lower tracking bound
a_p	The specified peak magnitude of the disturbance response for the MISO
a.l.	Arbitrarily large
a.s.	Arbitrarily small
$a_y = L_m T_y$	The desired lower tracking bounds for the MIMO system
$b_y = L_m T_y$	The desired upper tracking bounds for the MIMO system
B_h	Ultra high frequency boundary (UHFB) for analog design
ρ_{eg}	Density of boiler furnace gas(Kg/m ³)
h_{eg}	Specific enthalpy of furnace gas(Kcal/Kg)
F_f	Fuel flow (coal)(Kg/hr.)
F_a	Air flow (Kg/hr.)

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Fr.....	Recirculation gas flow (Kg/hr.)
Feg.....	Mass flow of furnace gas through the boiler (Kg/hr.)
qs.....	Heat transferred to Secondary Super Heater (SSH) (Kcal/hr.)
Vbf.....	Furnace combustion chamber volume (m ³)
E.....	Stoichiometric air/fuel ratio
hr.....	Specific enthalpy of recirculation gas(Kcal/Kg)
ex.....	Percentage excess air level(%)
Cf.....	Calorific value of coal (Kcal/Kg)
qr.....	Heat transferred by radiation to riser(Kcal/hr.)
Tg.....	Temperature of furnace gas (°C)
Pg.....	Furnace gas pressure (kg/cm ²)
d V.....	Volume of drum(cm ³)
dw V.....	Drum water volume(cm ³)
d ρ.....	Density of drum steam (Kg/ cm ³)
dw ρ.....	Density of water in drum(Kg/ cm ³)
ew F.....	Feed water flow(Kg/ sec)
d F.....	Steam flow from drum(Kg/ sec)
d p.....	Drum steam pressure(Kg/ cm ²)
d T.....	Saturated steam temperature(°C)
dw h.....	Enthalpy of feed water(Kcal/Kg)

ABBREVIATIONS AND ACRONYMS

CL.....	Closed Loop
E(s).....	Error Signal
HF.....	High Frequency
LHP	Left Hand Plane
LHS.....	Left Hand Side
LTI	Linear Time Invariant
MATLAB	MAtrix LABoratory
MIMO.....	Multi-input, Multi-output
MISO.....	Multi-input, Single-output
NC	Nichols Chart
NP.....	Nyquist Plane
OL.....	Open Loop
PD	Proportional plus Derivative
PI	Proportional plus Integral
PID.....	Proportional plus Integral plus Derivative
QFT	Quantitative Feedback Theory
RHP.....	Right Hand Plane
RHS.....	Right Hand Side
R	Reference Input
SISO.....	Single-input, Single-output
SVD	Singular Value Decomposition
2DOF	Two Degrees of Freedom
TF	Transfer Function

CHAPTER ONE

INTRODUCTION

1.1. Background

Boilers are commonly found in industry where they are used to supply turbines with steam or for heating of other chemical compounds. Although the steam production is varied during plant operation, output variables such as steam pressure must be maintained at their respected values. Therefore, tracking the load variation commands of drum pressure is expected from a power plant boiler system. However, the physical constraints exerted on the actuators must be satisfied by the control signals. These constraints can be the magnitude and saturation rate for the control valves of the fuel, steam and feed-water flow [1, 2]. A boiler unit is a nonlinear complex system. Several dynamic models of the boiler system have been developed [3]. Various control methods have been applied to boiler or boiler–turbine controller design, e.g., gain scheduling and feedback linearization [1, 2], quantitative Feedback Theory [4], and intelligent control [5-7].

Most practical Boiler systems have high uncertainty levels in their open-loop transfer functions which makes it very difficult to create suitable stability margins and good performance in command following problems for a closed-loop system. Therefore, a single fixed controller in such systems is found among the 'robust control' family. Some aspects related to robust control and mixed sensitivity [8, 9, and 15] have been applied in different ways. A close loop system is robust if it is insensitive to differences between the actual system and the model of the system which was used to design the controller. These differences are referred to as model uncertainty.

The H_∞ robust control technique is used to check if the design specifications are satisfied even for the worst-case uncertainty. In this approach, a mathematical representation of the model uncertainty is found. Then it is determined if the stability and performance specifications are satisfied for all plants in the uncertainty set (robust stability and robust performance) [10]. In this Thesis, a linear time invariant (LTI) model of the boiler system is considered which is obtained from experimental data. After modeling the parametric uncertainties, a QFT robust controller is designed. Then the uncertain system is represented in the form of general control configuration and unstructured uncertainty is considered in the form of multiplicative input uncertainty.

On the other hand, to achieve the goals of disturbance rejection and command tracking, the sensitivity function must have a special shape which is obtained by considering a suitable performance weight function. Using an algorithm for μ -analysis and using genetic algorithm, an optimal robust controller is designed. Both optimal robust controller and QFT robust controller guarantee the robust performance of the uncertain system. A particular area where this is applicable is the optimization of the overall energy generation cycle in large multi-fuel boilers. The benefits that can be achieved by improving control in these applications are often significant. Currently many control loops in the industry run in manual operation (more than 30% in some cases), controllers are sub-optimally tuned and equipment is often problematic. The shift to more efficient configurations (e.g. multi-fuel biomass and gas based boilers) is not therefore a trivial task.

This project focuses on a very important component in boiler operation, which is the control of combustion. As explained in the following sections combustion control consists of a series of interacting loops responsible for defining the correct combustible ratios to deliver the right amount of air and fuel whilst responding to the varying heat demand requirements of the boiler. The use of multi-fuel boilers adds extra complexity. For fuel/air supply control a straightforward approach would be to breakdown the process and treat each subsystem separately. This would require addressing interactions between them separately in terms of control loop disturbance rejection and also in relation to safety. The individual controllers in this case are mostly simple Proportional-Integral-Derivative (PID), often in combination with feed-forward control. The optimization of fuel consumption and combustion efficiency (with respect to load and economic factors) traditionally relies on manual intervention, particularly if there are changing set-points and fuel types.

Advanced control methods can cope with interactions and the changes in set-points, system parameters and disturbances. It monitors energy release and heat demand requirements online and generates the most efficient boiler operation for safety, fuel savings and emissions reduction. It is therefore an intermediate step bridging the supervisory level, in which optimal air/fuel quantities are computed online, with the final control elements that deliver air and fuel supply.

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Uncertainty of system comes from modeling error of plant, or diversity of exogenous disturbance. We often hope designed controller make system satisfying anticipant performance objectives in large range in control system design. That is to say, in certain extent disturbance range, when parameters of controlled model very in some way, the designed system still achieves specified performance.

The steam boilers are energy conversion units which transform the combustion energy of fuel into steam heat and mechanical power. High pressure steam boilers are applied in cogeneration plants for steam turbine operation and district central heating. Low pressure steam is utilized for technological needs and for autonomous heating of administrative and manufacturing buildings, like food production enterprises.

The main tasks of the steam boiler design, operation and automatic control are as follows:

- 1) To minimize heat losses from the furnace and boiler room using economizers, increasing the heat resistance and fuel combustion efficiency, as well as optimizing the heat load control;
- 2) To reduce flue gas emissions using emission reduction facilities and applying burner's optimal control according to flue gas analysis;
- 3) Continuous measurement, analysis and control of inner technological parameters – steam pressure and temperature, and drum water level and temperature, as well as outer disturbances – feed water flow and steam expenditure what directly affect the boiler parameters.

1.2. Statement of the Problem

This thesis focuses on the design of the robust control of Industrial boiler to provide efficient combustion in combustion chamber using initial PID controller and QFT with combining PID to overcome the following drawback which gained from convention and other modern robust controls:-

- ❖ Tuning convectional controller like PID is time consuming.
- ❖ Finding fuzzy rules and membership functions too much iteration
- ❖ It is not cost effective
- ❖ It is difficult to model uncertainty
- ❖ It is not robust and doesn't have good dynamic response,
- ❖ It has long rise time and large overshoot
- ❖ It has complexity of the controller structure

For this reason, it is desirable to introduce other type of controller such as Quantitative Feedback Theory based robust controllers because of these controllers have better performance major contribution of the thesis.

1.3. Motivation

Performance robustness has been an important issue in control and it has been increasingly recognized as an area of significance in many design applications. This issue has particular importance among the engineering community because it offers a solution to control system design problems for plant and processes that cannot be described by a single linear time-invariant model. The aim of this thesis is to describe one of the methods available for robust control system design called Quantitative Feedback Theory (QFT) and discuss its application to engineering problems. QFT is a powerful design technique based on multi degree-of-freedom use of feedback. The basic building block of a multi-degree-of-freedom system is a two-degree-of-freedom system in which two controllers are present, one inside the feedback loop, used to reduce closed-loop uncertainty, and the other prior to the loop, used to shape the input in order to achieve the required output.

1.4. Objectives of the Thesis

- The general objective of the thesis is to investigate and analyze how the Quantitative Feedback Theory is integrated to the Robust and Performance control of Industrial boiler systems.
- ❖ The Specific Objectives of my thesis are given as follows:-
 - ✚ To develop a model of QFT method in order to design a controller of robust control for the boiler plant pressure.
 - ✚ To analyze the Robust and performance control of Industrial Boiler.
 - ✚ To design and simulate the QFT controller for obtaining good performance of an industrial boiler in order to reduce fuel consumption rate and reaching higher efficiency.
 - ✚ To design controller $G(s)$ & Pre-filter $F(s)$ to meets various stability and performance specification for all $P(s) \in P$

1.5. Significance of the Thesis

The Quantitative Feedback Theory (QFT) has been successfully applied to many control systems: linear and non-linear, stable and unstable, SISO and MIMO, minimum and non-minimum phase, with time-delay, with lumped and distributed parameters, multi-loop, etc.

In this thesis, I propose to design an optimizing controller using a multivariable feedback technique using QFT robust control and compare pressure control performance of the boiler using conventional PID controller. The comparison of both the controller performance will be analyzed with simulation. In parametric uncertain systems, we must first generate plant templates prior to the QFT design (at a fixed frequency, the plant's frequency response set is called a template). Given the plant templates, QFT converts closed loop magnitude specifications into magnitude constraints on a nominal open-loop function (these are called QFT bounds). A nominal open loop function is then designed to simultaneously satisfy its constraints, as well as to achieve nominal closed loop stability. In a two-degree-of-freedom design, a pre-filter will be designed after the loop is closed (i.e., after the controller has been designed) [12].

1.6. Thesis outline

This master thesis is organized in to five chapters: The first chapter presents the overview of Industrial Boiler, statement of the problem, relevance and objectives of the study.

In chapter two we reviews Boiler, QFT background, the Related done paper and its limitations, , by defining the used methods and referencing the previous works developed in other relevant cases that allows fulfilling the current objectives. And the third chapter is mainly expressed the modeling and control of boilers. Specifically it discusses the boiler combustion control in boiler plant which is to validate the input fuel flow and air flow control to desire efficient combustion. After that QFT design and its procedure will be discussed deeply chapter three of the thesis. Then in chapter four we present and analyze the obtained results of the modeled systems and the controller simulations. Finally, an overall analysis is performed. It is analyzed if the principal objective of the thesis has been reached is discussed in chapter five in addition to suggesting and recommending the work

CHAPTER TWO

QUANTITATIVE FEEDBACK THEORY AND LITERATURE REVIEW

2.1. Feedback Control Methods

Feedback and control are almost everywhere. Feedback is an intuitive means for control. For example, when you feel cold (sensing), you add one more layer of cloth (decision and then control action) to keep yourself warm and comfortable (objective). This is biological feedback due to a change in the environment. In technological systems, the loop “sensing-feedback-decision-control” is implemented to change the system behavior into a desirable one. More specifically, we will mainly concentrate on analytical and simulation methods for linear feedback control systems and a few aspects of simulation for nonlinear systems. For multiple input–multiple output (MIMO) linear systems, good references are [4].

In practical control system design, the more general feedback control structure is sometimes used with the feedback block simplified. In such feedback control systems the pre-filter and controller, can be adjusted independently in control system design. This is often referred to as two-degrees-of-freedom control. In this book, we will focus on one-degree-of-freedom control problems.

Most, but not all, of the existing design procedures for a control system make use of mathematical models. It is therefore important to try to obtain accurate mathematical models for the system components. The system can then be analyzed and designed in a systematic way and its properties assessed using the mathematical models as approximations of its true behavior.

If the system model is not known, two methods can be used to build a model of the system for the analysis and design tasks. The first method is to derive the system model using existing physical laws or principles. The second method, more often used, is to find an approximate mathematical model based on the observed response data of the system. The former method is referred to as the physical modeling and the latter the system identification.

How to obtain a model of the system to be controlled is a large subject area and will not be fully pursued in this book. Instead, we will focus on how to manipulate the models.

The design of a control system according to classical methods assumes full knowledge of the plant and the controller. In practical systems, however, the plant model will always be an inaccurate representation of the actual physical system due to:

- Parameter changes
- Un-modeled dynamics
- Un-modeled time delays
- Changes in equilibrium point (operating point)

Robust Control methods address the problem of uncertainty systematically. They aim to maintain adequate performance and stability margins despite the presence of uncertainty in the dynamics of the plant. For the purposes of this project QFT is used in the design and analysis of control systems characterized by significant uncertainty and undesired external inputs [1, 2, 3 4].

A robust control system has the following characteristics:

- ✚ Low sensitivities to parameter changes
- ✚ Closed loop stability is maintained within the range of parameter change
- ✚ Its performance does not deteriorate rapidly with parameter change

The meaning of the above terms is illustrated in the example of figure below:

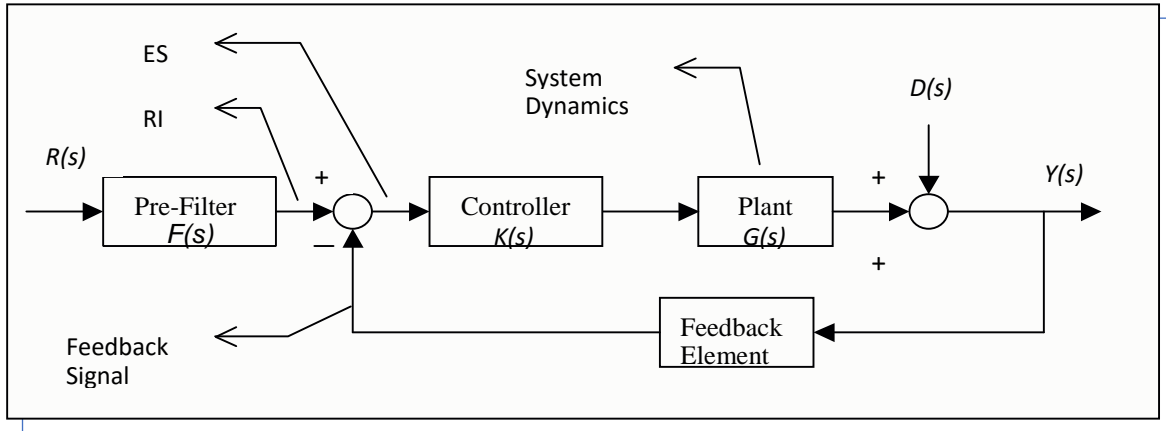


Fig 2.1: - 2DOF control system with external disturbance at the output of the plant [4, 10].

2.2. Quantitative Feedback Theory (QFT)

QFT Control of Single Input Single Output Systems

The task of a control engineer is to design a system such that the client specifications around the plant are met. The plant represents the dynamics of the system to be controlled. For all practical cases, the plant is uncertain but can be quantified. Neglecting the implementation of the controller elements, the control system usually contains sensors, actuators and the plant itself. The symbols used to represent the physical units and the signals in the control system are given in Table 2.1.

Table 2.1 Symbols that represent physical units and signals

Physical Units	Symbol in complex domain
Plant	$P(s)$
Sensor	$H(s)$
Pre-filter	$F(s)$
Feedback controller	$G(s)$

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Signals	Symbol
Reference	$R(s)$
Output	$Y(s)$
Noise	$N(s)$
Output disturbance	$d_o(s)$
Input disturbance	$d_i(s)$
Input	$U(s)$

Table 2.2:- Symbols that represent signals

Then the 2 degree of freedom structure Quantitative feedback theory is given as follows

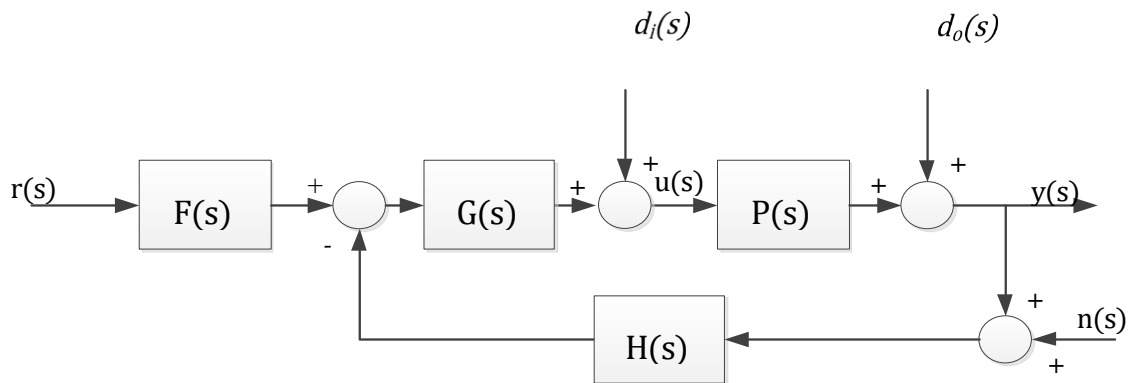


Fig 2.2:- QFT block diagram

The three common reasons for using feedback is to,

- Reduce uncertainty,
- Stabilize an open-loop unstable system,
- Reject disturbances, even when there is no plant uncertainty.

The progenitor of QFT, Prof. Isaac Horowitz made an important observation. He realized that fixed, stable systems (no uncertainty) do not require feedback to satisfy specifications. The specifications can be achieved by shaping the pre-filter appropriately.

His subsequent thought was what formed the fundamental philosophy of QFT. Horowitz' idea was that the amount of feedback should be directly related to the amount of uncertainty in the system; this idea explains the etymology of the title of the design philosophy.

The following describe the specifications that may be desired by the client:

- Reference tracking – The requirement for the output of the plant to follow (track) the reference signal
- Output/input disturbance rejection – The requirement for the attenuation of disturbances seen at the plant output due to disturbances at the plant output/input
- Robust stability margins – The stability requirement that takes unstructured uncertainties into account (uncertainties that are not a result of parameter uncertainty - structured uncertainty)
- Sensor noise rejection – The requirement for the transfer of sensor noise to the plant input to be below some specified level.

One of the attractions of QFT is that all the above stated requirements can be mapped to constraints on the controller during design time, whereas other methods may allow only certain requirements to be mapped and force an iterative approach to solving the whole problem.

To expound the idea of SISO QFT, the reference tracking design problem is developed further. The reference tracking problem is to design a feedback controller $G(s)$, and a pre-filter $F(s)$ such that the closed-loop reference to output magnitude response for all plants within the uncertain set lie between the upper bound, $A(j\omega)$ and lower bound $(j\omega)$. A typical set of upper and lower magnitude bounds are illustrated in Fig. 2.3. The frequency domain bounds are usually generated from reasonably desired time-domain responses.

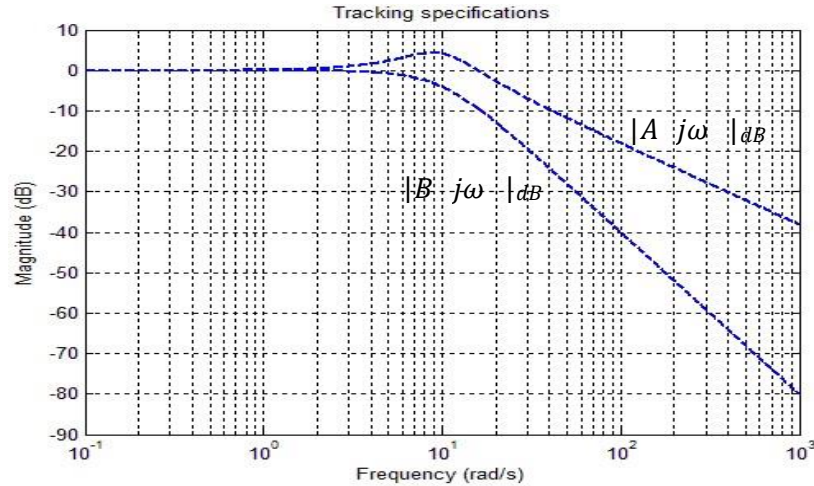


Fig 2.3:- Magnitude response of the tracking specifications [10, 12]

2.2.1. SISO QFT Design Approach

The Numerical Approach

Taking the sensor to be ideal (for simplicity), defining the loop gain $L=PG$ and neglecting the dependence on the complex variables for simplicity, the reference to output transfer function for the system shown in Fig. 2.1.3 is given by

$$T\left(\frac{y}{r}\right) = \left(\frac{L}{1+L}\right)F \quad (2.1)$$

The tracking specifications will be met if the maximum change of the logarithm of Eq. (2.1) is less than the change of the logarithm of the tracking specifications. Assuming loop i results in the maximum and loop j results in the minimum, the following must be satisfied.

$$\max \Delta \log \left| T \frac{y}{r} \right| = \log \left| \frac{L_i}{1+L_i} * F \right| - \log \left| \frac{L_j}{1+L_j} * F \right| \leq \log |A(j\omega)| - \log |B(j\omega)| \quad (2.2)$$

The pre-filter is fixed hence it gets cancelled in Eq. (2.1) when extracted from the logarithm. By simplifying Eq. (2.2) one obtains

$$\left| \frac{P_i 1 + G P_j}{P_j 1 + G P_i} \right| \leq \frac{|A(j\omega)|}{|B(j\omega)|} \quad (2.3)$$

Equation (2.3) is a linear fractional mapping on the controller G and after further manipulation one finally obtains a quadratic function which must be solved to obtain $|G|$ for some controller phase angle.

This is the approach one would take to design a controller using a computing machine. The quadratic function must be solved for each ordered pair of plants in the plant set and the intersection of the constraints are the final gain to be satisfied. This must be done for a set of design frequencies and phase angles.

The result of this computation is controller gain requirements for the set of phase angles chosen. The design task is to shape the controller such that it satisfies the constraints. Generally, a nominal loop $L_o = P_o G$ is chosen which is shaped by the designer; this is more relevant to the graphical approach to the QFT design. A tacit constraint is that $(1 + PG)$ does not have any right-hand plane (RHP) zero for all plants.

The Graphical Approach

QFT is well known for the use of templates in the graphical design procedure. A QFT template encapsulates the phase and gain of an uncertain plant at a particular frequency i.e. it is a physical representation of the uncertainty. Assuming the uncertain plant is given by a set of discrete functions, the template can be generated by determining the plant value for each plant in the set.

Generally, the discrete set of plant values is plotted on a graph that displays phase versus magnitude. Templates are generated for a finite set of frequencies (“working frequencies”). These frequencies are chosen such that the resulting templates appear significantly different from each other.

The graphical approach is an elegant way of designing because once the uncertainty is captured in the templates; the designer requires only a single loop to work with: the nominal loop. The nominal loop is the loop obtained when the controller is multiplied by one of the plants from the uncertain set; the plant chosen is referred to as the nominal plant. The point on the template that is an evaluation of the nominal plant is called the handle. The handle is an important concept that unites the nominal loop and the template.

Controller Preferences

An attractive feature unique to QFT is the transparency in design. At every stage of the design the designer can clearly “see” the effect of making changes to the system. The controller is actively constructed by the designer using the QFT philosophy as a guide.

This allows the designer to make trade-offs while designing between meeting boundary constraints (lower and upper boundary) and priority in the right to demand controller in our robust control design. These priorities are by choosing whether we increase gain or add poles and zeros. The three important preferences in controller design are

- Minimum controller gain,
- Minimum controller order and,
- Limiting minimum damping factor in controller elements.

Assuming ideal sensor, the sensor noise to plant input transfer function is given by

$$T \frac{u}{n} = -\frac{G}{1+L} \text{ where } L = PG \quad (2.4)$$

For a strictly proper loops (all real systems), the signal transfer from sensor to plant input goes with the controller gain at high frequency. $\lim_{\omega \rightarrow \infty} |L(j\omega)| \rightarrow 0$ This means that the noise at the sensor will be amplified via the controller if the gain of the controller is high at high frequency. All signals going to the plant input must pass the controller. Since the power transfer to the plant input is related to the integral of the magnitude of the controller frequency response and the error signal, the controller magnitude must be minimized in order to prevent actuator saturation. Unnecessarily high controller gain can be thought of as excessive actuation which will lead to quicker deterioration of the actuators.

Whether the controller is implemented digitally or using analog components an element of uncertainty is contributed by each element (pole/zero) of the controller. Hence, to reduce the unaccounted uncertainty in the controller implementation the designer must aim for minimum order controllers. It is also not desired to have very small damping factors in the controller elements. This is because the dynamics is very sensitive to the accuracy of the implementation. Small changes in the damping factor or corner frequency may result in large changes to the loop which in most cases no provision is made during the design.

2.2.2. Benefits of QFT

The benefits of QFT may be summarized as follows:

- a. The result is a robust design which is insensitive to plant variation.
- b. There is one design for the full envelope (no need to verify plants inside templates).
- c. Any design limitations are apparent up front.
- d. There is less development time for a full envelope design.
- e. One can determine what specifications are achievable up early in the design process. j,
- f. One can redesign for changes in the specifications quickly.
- g. The structure of compensator (controller) is determined upfront.

MISO Analog Control System

The overview of the MISO QFT design technique is presented in terms of the minimum-phase (m.p.) linear time-invariant (LTI) MISO system. The control ratios for tracking ($D = 0$) and for disturbance rejection ($R = 0$) are, respectively. The design objective is to design the pre-filter $F(s)$ and the compensator $G(s)$ so the specified robust design is achieved for the given region of plant parameter uncertainty. The design procedure to accomplish this objective is as follows:

Step 1: Synthesize the desired tracking model.

Step 2: Synthesize the desired disturbance model.

Step 3: Specify the J LTI plant models that define the boundary of the region of plant structured parameter uncertainty.

Step 4: Obtain plant templates, at specified frequencies, that pictorial described the region of plant parameter uncertainty on the NC.

Step 5: Select the nominal plant transfer function $P(s)*G(s)$.

Step 6: Determine the stability contour (U-contour) on the NC.

Steps 7-9: Determine the disturbance, tracking, and optimal bounds on the NC.

Step 10: Synthesize the nominal loop transmission function $L_0(s) = G(s)*P_0(s)$ that satisfies all the bounds and the stability contour.

Step 11: Based upon Steps 1 through 10 synthesize the pre-filter $F(s)$.

Step 12: Simulate the system in order to obtain the time response data for each of the J plants.

The QFT philosophy for feedback design fits a wide range of applications

- ❖ Plant uncertainty:-The controller should meet specifications in spite of variations in the parameters of the plant model.
- ❖ Plant Models from Experiments: - Many systems have complex dynamics and are very difficult to model analytically.
- ❖ Linear Plants from Nonlinear Dynamics: - Unlike the conventional small signal linearization about an operating point, Horowitz's idea is to replace the nonlinear plant with a set of linear, time-invariant (LTI) plants using assumed input and output responses.
- ❖ Several Performance Specifications:-The design problem consists of several closed-loop performance specifications and the objective is to synthesize a controller to meet simultaneously all specifications (a robust performance problem). QFT reveals via QFT bounds the “toughness” of each specification relative to others.

2.3. Review of Literatures

Over the last twenty years, many researchers have reported that fuzzy-logic controllers are very suitable for controlling objects with nonlinearity and even with unknown structure. However, one of the design methods widely used for fuzzy controllers is the definition of membership functions for linguistic variables, and then the formulation of fuzzy rules by control engineers (Braae and Rutherford, 1979; Tzafestas and Papanikolopoulos, 1990; Li, 1993). Unfortunately, such a fuzzy-logic controller cannot provide the desired control performance for a boiler combustion system, due to its uncertainties and large disturbances.

Besides, it is very time-consuming to find the necessary fuzzy rules and membership functions by adjusting a real boiler system on-line, because stoker-fired boilers that use solid fuel have large time lags. Another approach is to adapt a rule base or/and membership functions by means of self-organizing algorithms or a neural network, based on previous responses, until the desired control performance is achieved (Procyk and Mamdani, 1979; Scharf and Mandic,; Berenji[4, 5].

In China, industrial stoker-fired boilers are very widely used in many enterprises, under very poor conditions, and the fuel involved is solid fuel, mostly coal. Furthermore, most of these boilers are controlled by tedious manual operation.

Only few of them are controlled by PID-type controllers. However, such a control strategy cannot ensure a good control performance and good Robust.

A simple two-by-two model for a boiler-turbine unit is demonstrated by Tan w, Liu JZ, Fang F Chen YQ in 2007[2]. This model can capture the essential dynamics of a unit. The design of a coordinated controller is discussed based on the model. A PID control structure is derived, and a tuning procedure is proposed. The examples show that the method is easy to apply and can achieve acceptable performance.

A distance measure is proposed via the gap metric proposed by Tan W, Marquez HJ, Chen T, Liu JZ(2004)[3]. And the concept is applied to a boiler-turbine unit to analyze its dynamics. It is shown that the unit shows severe nonlinearity, but the nonlinearity can be avoided by careful choice of the operating range. A single linear controller can be designed to work in such an operating range. It is also shown that the controller constraint is another source of the nonlinearity, which can be compensated using anti-windup techniques. Simulation results are given to verify the conclusions.

The robust-two-degree of freedom multivariable control system using H_∞ , optimization method by Hwang CS, Kim DW (1995) [4]. This can achieve the robust stability and the robust performance property simultaneously. The feedback controller is designed using H_∞ optimization method. For mixed sensitivity function the feed forward controller is design using H_2 optimization method to minimize the error of both the transfer function of the optimal model and overall transfer function. The feedback controller can obtain the robust stability property; the feed-forward controller can obtain the robust performance property. Under modeling error the robust-two-degree of freedom multivariable control system is applied to the nonlinear multivariable boiler turbine system. From this thesis we generate the drawback of previously used methods. The control of boiler- turbine unit is widely studied using various control techniques, e.g., robust control, nonlinear control predictive control and intelligent control, the following problems make these control methods to be seldom applied in practice.

- a) The design methods are not generic in that for each unit a model should be identified and a controller should be designed. This process is called 'controller design'.

However, for control engineers, 'controller tuning' is probably more preferred. For example, many industrial processes can be modeled as first-order plus dead-time models and PID controller can be tuned for such processes for achieve satisfactory performance. Though a boiler-turbine units are different.

b) The control algorithms are not easy to implement and maintain. It is well known that the controllers designed by advanced control methods are generally complex.

c) The performance of a single controller cannot be guaranteed for wide-range load variations. Now a day, the units at thermal power plants need to follow the demand by the dispatch. As the capacity of boilers increases, the nonlinearity of the boiler- turbine units increases, which makes the linearized models at the operating points vary, and a single (possibly robust) linear controller may not meet the control objectives for the desired operating range.

This thesis presents a multi-objective QFT robust controller implemented with a signal-based approach [2]. The purpose of this work is to show the applicability of the Quantitative Feedback Theory control approach and the structured singular value μ tool in the context of Industrial Boiler systems, in particular on the Boiler plants. The QFT strategy can be applied to Multiple-Input Multiple-Output (MIMO) systems and allows designing a controller that stabilizes the plant and minimizes a fixed cost function.

However, this dynamic model may associate with uncertainties. Uncertainties in mentioned model are caused by lack of knowledge about the dynamics of the system, pay load changes, and air flow. Thus, application of robust control methods for high precise control of pressure is inevitable. In the first step plant is identified by using experimental data by the mean of converting into a group of linear time invariant (LTI) uncertain plants. After representation of the uncertain dynamic system in general control configuration and modeling the parametric uncertainties, nominal performance, robust stability and robust performance against disturbances are analyzed by the concept of structured singular value μ . This procedure gives a satisfactory controller at the presence of model perturbations. For more comparisons, quantitative feedback theory (QFT) as a well-known robust control approach is also applied to the plant.

CHAPTER THREE

MODELING & CONTROL OF BOILERS

3.1. Operating principles of Boiler

Boilers are, among other characteristics, distinguished by the layout of the evaporator. Due to material limits, evaporator tubes before were, operated at subcritical pressure, ($p < 221.2$ bar) utilizing the natural circulation principle. In order to obtain as high a heat transfer coefficient as possible, it is in this case important to keep the boiler tubes wetted throughout the whole tube length. Therefore, boilers are designed, so the steam quality nowhere along the evaporator exceeds about 0.3. Hereby the boiling crisis of first kind, ‘Departure from nucleate boiling (DNB)’ is avoided.

To improve the efficiency of steam plants, it is necessary to operate at supercritical pressures, in which case circulation mode is impossible. Instead, once-through boilers have been more frequently installed. For supercritical operation, the flow is to some extent similar to single-phase flow, and can be regarded as that. However, as one of the advantages of once-through boilers is their controllability by sliding pressure, it is necessary also to operate in the subcritical regime. By doing so, it is not possible to avoid DNB and dry-out.

To estimate heat transfer in this regime, one has to determine the positions along the evaporator tubes where the water/steam mixture changes its convective properties and to apply different correlations in each region. In subcritical operation mode, it is necessary to separate the two phases before the steam enters the super-heater section. Together, the radiative, conductive and convective heat transfers form the basis for the design and operation of the evaporator. They should all be correlated to build a model of a boiler.

In the flow of gasses in the furnace and water/steam in the evaporator, pressure losses are unavoidable. These are determined by looking at frictional, gravitational and losses due to acceleration of the fluid separately.

Frictional loss is correlated by a friction factor, f , determined as a function of the Reynolds number, which basically yields.

$$\Delta P = \frac{f(L\rho\omega^2)}{2di} \text{ where } f = f(Re) \quad (3.1)$$

Also pressure loss will be significantly different in single-phase and two-phase flow, and different correlations need to be applied in the two modes.

3.1.1. Boiler Combustion

There are two distinct aspects to the combustion-side model - heat generation and combustion gas composition.

✚ Heat Generation

The heat generation model is required to calculate how much energy is available to the boiler for steam generation. The motivation for modeling heat generation is obvious - the amount of energy available for steam generation is dependent on the amount of energy released by combustion.

✚ Combustion Gas Composition

The combustion gas composition model calculates the contents of the combustion gases as they leave the furnace. It is subdivided into a description of the composition of the products of gaseous fuel combustion and a description of the composition of the products of solid and liquid fuel combustion. Each of these sections is further subdivided into a description of stoichiometric and sub-stoichiometric combustion modeling.

There are two principal incentives for modeling the composition of the combustion gases.

Firstly, the model can be used to investigate combustion efficiency and means of improving it.

Combustion efficiency is decreased if an excess of air is available for combustion as this excess air must be heated up to combustion chamber temperatures. In this case the percentage of oxygen in the combustion gases will be greater than zero. If the amount of air available for combustion is less than or equal to the amount of air which is required for complete (stoichiometric) combustion of the fuel, the percentage of oxygen in the combustion gases will equal zero.

Secondly, the model can be used to investigate the amount of toxic and environmentally damaging gases which are generated by a boiler. Carbon monoxide, which is poisonous, is produced if the amount of oxygen available for combustion is insufficient to burn all the fuel (sub-stoichiometric combustion).

For this reason, it is important to keep the percentage of oxygen in the combustion gases above zero. Carbon dioxide, oxides of sulphur and oxides of nitrogen which are damaging to the environment are also produced by combustion. The heat which is generated in the combustion chamber has three sources:

1. Combustion of fuel
2. Heat contained in the incoming air
3. Heat contained in the incoming fuel

Assuming stoichiometric combustion, heat generation can be modeled;

$$Q_{cc} = C_{fu}M_{fu} + M_{fu}C_{fu}T_{fu} + M_{air}C_{air}T_{air} \quad (3.2)$$

Where

C_{air} = specific heat of air	O_{cc} = heat generated by combustion
C_{fu} = specific heat of fuel	T_{air} = temperature of air
C_{fu} = calorific value of fuel	T_{fu} = temperature of fuel
M_{air} = mass flow rate of air	
M_{fu} = mass flow rate of fuel	

The dynamics of the combustion process are extremely fast in comparison to the dynamics of evaporation. For the purposes of modeling, the dynamics of the combustion process have been neglected.

Most sections of the boiler - economisers downcomers and superheaters, are heated by convection i.e. by hot flue gases flowing over them. The mass flow rate of the combustion gases is needed to calculate how much heat is transferred to these sections of the boiler.

For gaseous fuels, the mass flow rate of gas through the furnace is found by summing the mass flow rate of the incoming fuel and air.

$$m_g = m_{air} + m_{fu} \quad (3.3)$$

Where M_{air} = mass flow rate of air (kg / s)

M_{fu} = mass flow rate of gas (kg / s)

For solid and liquid fuels, the mass flow rate of gas can be assumed equal to the mass flow rate of the incoming air only.

Excess air is usually expressed as a percentage increase over the stoichiometric air requirement.

The equation for excess air by volume is:

$$\text{Excess Air} = \frac{\text{Volume Actual air} - \text{Volume Stoichiometric Air}}{\text{Volume Stoichiometric Air}} * 100\% \quad (3.4)$$

Hence for deriving mathematical model, internal structure and functions of subsystem of the boiler should be studied. This mathematical model of the boiler is derived from basic mass and energy balance equations. The boiler is modeled for various sections of it.

Furnace heat balance equation for combustion is given in equation

$$CFwF + hAwA + hGwG - Qir - Qis - wEGRs \left(1 + \frac{y}{100}\right) hEG = VF \left(\frac{d}{dt}\right) (\rho EGhEG) \quad (3.5)$$

Furnace Mass balance equation for combustion

$$wF + wA + wG - WEG = VF \left(\frac{d}{dt}\right) \rho EG \quad (3.6)$$

The Mass and heat balance equation of riser is give in equations below

$$Wd - wr = Vr \left(\frac{d}{dt}\right) (\rho rhr) \quad (3.7)$$

$$Qr + wdhw = wrhr + Vr \left(\frac{d}{dt}\right) (\rho rhr) \quad (3.8)$$

In the similar manner the mass and heat balance equations of boiler drum is given in equations.

$$We + (1 - x)wr - wd - we = \left(\frac{d}{dt}\right) (mdl) \quad (3.9)$$

$$Wehe + (1 - x)wrhvw = wdhw - wechv + \left(\frac{d}{dt}\right) (mdlhw) \quad (3.10)$$

The mass and heat balance equations of boiler super heater is given in equations

$$wv - ws + wa = Vs \left(\frac{d}{dt}\right) (\rho s) \quad (3.11)$$

$$Qs + wvhv = wshs - (ha - hf)wa + Vs \left(\frac{d}{dt}\right) (\rho shs) \quad (3.12)$$

Furnace Dynamics

To develop the mathematical model so as to simulate the dynamic behavior of the furnace, we take into account the working chart in figure below. The input quantity, which varies in time, is the fuel mass flow rate m_f , and the output value is the pressure inside the furnace, p_f . The model includes the mass, energy, and momentum balances, the heat transfer from hot flue gases to water and steam model and the flue gas flow through the boiler model.

Combustion heat balance:-

$$V_f \frac{d(\rho_g \cdot h_g)}{d\tau} = m_a' \cdot h_a + m'_{fb} \cdot LHV - m'_{fg} \cdot h_g - Q'_{f} - Q'_{l} \text{ [KW]} \quad (3.13)$$

Combustion mass balance:-

$$V_f \frac{d\rho_g}{d\tau} = m'_{fb} + m'_a - m'_{fg} \left[\frac{kg}{s} \right] \quad (3.14)$$

Flue gas flow through the boiler:-

$$m'_{fg} = k_f \cdot p_f \left[\frac{kg}{s} \right] \quad (3.15)$$

Furnace gas pressure:-

$$P_g = R_g \cdot \rho_g \cdot T_g \left[\frac{N}{m^2} \right] \quad (3.16)$$

Combustion dynamics:-

$$m'_{fb} = m'_f \left(1 - e^{-\frac{\tau}{T_f}} \right) \quad (3.17)$$

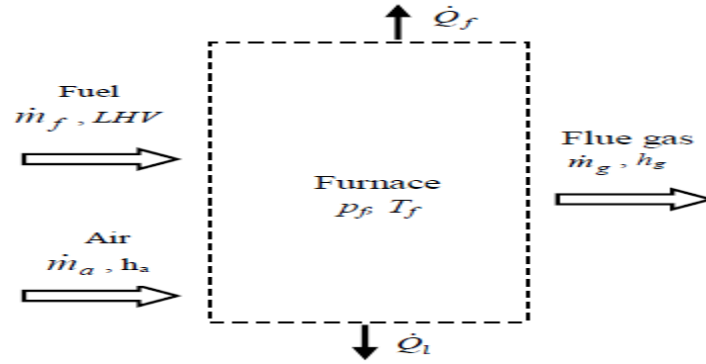


Fig 3.1:-Physical model of the boiler furnace [2, 6, 9]

Where:-

VF [m³] furnace volume;

ρ_g [kg/m³] flue gas density;

$f m$ [kg/s] fuel mass flow rate;

$fb m$ [kg/s] burning fuel mass flow rate;

$a m$ [kg/s] air mass flow;

$g m$ [kg/s] exhaust gas mass flow;

LHV [kJ/kg] fuel lower heating value of the fuel;

$f Q$ [kW] heat transferred by radiation to vaporizer

$$Q'f = CR * S * (Tg^4 - Trt^4) \approx CR * S * Tg^4 \quad (3.18)$$

Q_f [kW] Lost heat through furnace walls;

T_g [k] flue gas temperature;

H_g [kJ/kg] flue gas enthalpy

$$hg = cp_g * (Tg - Tref) + href \quad (3.19)$$

T_{ref} [K] references exhaust gas temperature;

Cp_g [kJ/kg K] flue gas specific heat at constant pressure;

h_a [kJ/kg] air specific enthalpy;

cp_a [kJ/kg K] air specific heat at constant pressure;

T_a [K] air temperature;

$ref h$ [kJ/kg] flue gas reference enthalpy;

R_g [kJ/kg K] gas constant for flue gas;

cR [kW/m²·K⁴] the coefficient of mutual heat transfer between flame and wall;

k_f [m·s] a friction coefficient.

3.2. Mathematical Modeling of Boiler

In Industrial Boiler, Air is flowed by FDFs (Forced Draft Fans). Then it is preheated with exhaust gases. After measuring air flow, it is guided to the boiler. Combustion occurs in the boiler and exhaust gases heat the water which flows through the boiler's walls. The exhaust gases are passed through the II type shape boiler by suction of IDFs (Induced Draft Fans). IDFs guide exhaust gases to the atmosphere through a chimney [11]. A damper which is installed on boiler is used to control the flow of exhaust gases. Fig given below it illustrates block diagram of the combustion system of the boiler.

Flow of input fuel should be increased to provide the desire power. So, the set point of fuel flow changed and controller1 C_1 will open the valve of fuel pipe and increase the fuel flow. This effect is shown by $F1(s)$. As fuel flow increased, air flow must be increased either to have a more efficient combustion, Thus the set point of air flow changes. Because of generated error, controller 2 C_2 will open the dampers of FDFs and so it increases the Air flow. This effect is shown by $G1(s)$. Since increase in fuel flow and air flow will cause increasing pressure of boiler, opening valve of fuel pipe by C_1 and opening dampers of FDFs by C_2 will increase pressure of boiler. These two effects are shown by $F_2(s)$ and $G_2(s)$ respectively. As the pressure of the boiler becomes more than set point, controller 3 C_3 opens the dampers of IDFs which cause decreasing in the pressure $H1(s)$. Increase of pressure of the boiler will decrease fuel and air flow that is shown by $H2(s)$ and $H3(s)$. The system is multiple inputs multiple outputs (MIMO) and it is too complicated to be controlled. In this article the system is simplified to a single input single output (SISO) system to control pressure of the boiler. The simplified system is shown in Fig. 4.2, in which $F2(s)$ and $G2(s)$ are considered as disturbances on the system.

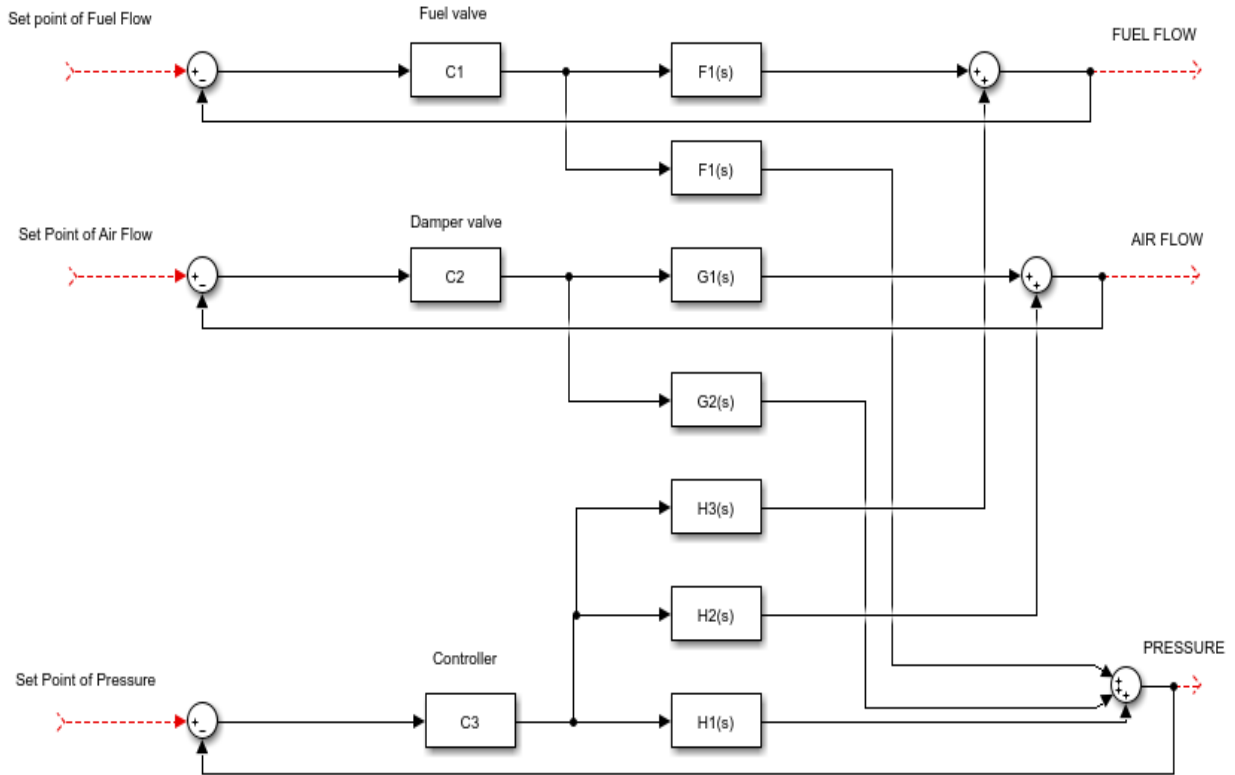


Fig 3.2:- Block Diagram of Combustion control of Industrial Boiler

Using terms and equations introduced in, finally we will derive uncertain LTI transfer functions that model the dynamic of boiler pressure. Then we consider the Air flow and Fuel Flow of the Boiler as Disturbance of the system. That means the system will reduced to 2 DOF Plant

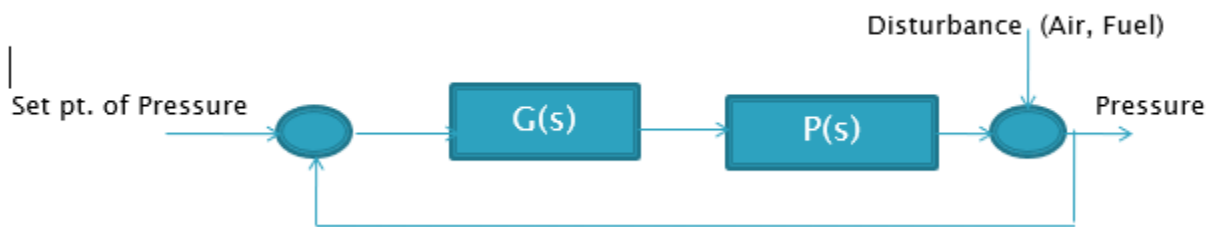


Fig 3.3: - Reduced SISO Model of Power Plant Boiler

The boiler model is developed on the basis of physical laws, previous efforts in boiler modeling, known physical constants, plant data, and heuristic adjustments. The resulting fairly accurate model is LTI, 2nd order, and includes Air flow and fuel flow as output disturbance, measurement noise models (Pellegrinetti & Bentsman, 1996).

The methods to obtain the correct model of any steam boiler plant are not readily found in open literature and are often specific to a particular system. This is particularly true on industrial environments where the signals given by any system cannot be predicted until they are working on place because they can be very affected depending on the conditions of where the system is located, the standard work of the company and the users. The boiler model which is used in this thesis contains LTI 2nd order, Output disturbance (Air flow and Fuel flow) are going to be managed (Pellegrinetti & Bentsman, 1996). The mathematical Models are an approximation of real systems and contain imperfections by several reasons:-

- ✚ Use of low order descriptions
- ✚ Un-modeled dynamics
- ✚ Obtaining Linear Models for specific operating (Working with poor Performance outside of this working point), etc.

Finally, the transfer functions of whole system $P(s)$ as 2 DOF will be given by the following transfer function since low order models are often used to avoid modeling cost. The main idea is the first nonlinear QFT technique was to replace the nonlinear system by set of LTI system for a set of acceptable outputs.

$$P(s) = \frac{D}{S(TS + 1)} \quad (3.20)$$

Where D and T are Uncertain Parameter of the Plant with interval of $D = [1 \ 3]$ and $T = [5 \ 70]$

3.2.1. Robust control of the boiler system

In the general control configuration given below where P is generalized Plant and K is generalized controller.

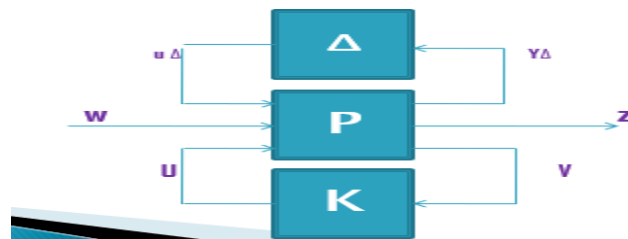


Fig 3.4:- General control configuration

The overall objective is to minimize some norms of transfer function from w to z . Therefore, based on the information in v , the controller design problem is to find the controller K which produces the control signal U that counteracts the influences of w on z by minimizing the closed loop norm $\|T_{wz}\|_{\infty}$. The matrix Δ is a block diagonal matrix that includes all possible perturbations to the systems. It is usually normalized such that $\|\Delta\|_{\infty} \ll 1$. Suitable Practical approach such as DK-iteration can be used to find μ -Optimal controller. Weighting function of H_{∞} are W_P , W_u and W_m . Where W_P is the Performance Weight which is large enough at low frequency where suitable nominal performance is desired. W_m is the upper bound function on possible multiplicative uncertainties which is large enough at high frequency where large model uncertainties may occur. The system sensitivities function S , is the transfer function between output Y and disturbance d and tracking error e and reference input r .

$$S = (1 + PG)^{-1} \quad (3.21)$$

Since $\|S\|_{\infty}$ is the good tool to measure the disturbance rejection and tracking performance of the closed loop system, small value of it is appropriate. Normally for Robustness and measurement noise we require that T is small for high frequency. So, with an arbitrarily small deviation from the steady state, due to the disturbance, and with a sensitivity close to zero, the control system is more independent of the plant uncertainty. Obviously, in order to achieve an increase in $|C(j\omega)|$ is necessary to increase the crossover frequency for the system. So, to achieve arbitrarily small specifications implies to increase the bandwidth of the system.

3.3. QFT Design for Robust control of Boiler

The QFT approach developed by Horowitz works on extended classical control theory and utilizes well established tools such as Bode and Nichols plots.

Compare with the other robust control strategies, QFT have following benefits:

- ✚ One robust controller design for entire frequency range.
- ✚ The controller design purely based on the specifications defined by the user and it is possible to define control structure before the design process
- ✚ It is possible to determine performance specifications early which are going to be achievable in the design process.

Robust Control of Industrial Boiler Using Quantitative Feedback Theory

The QFT basic operation is to convert specification of the design process on closed-loop response and plant parameter variations into performance bounds and robust stability specifications on open-loop transmission of the nominal system.

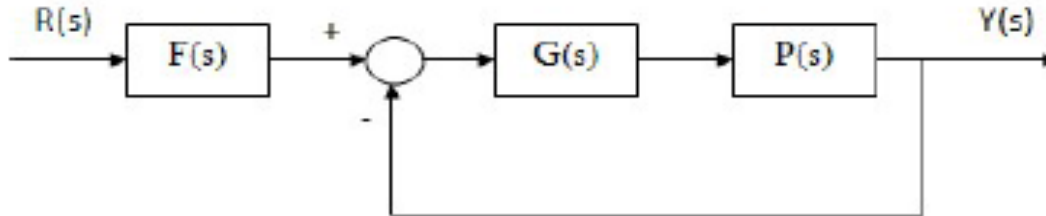


Fig 3.5:- Block diagram for QFT Control System [4]

The QFT basic objective is to design controller $G(s)$ and pre-filter $F(s)$ to meet various stability and performance specifications for all $P(s) \in P$. QFT deals with desirable but incompatible features of the controller design. It acts like an extension work of classical design methods in order to design loop shaping concept. In QFT design Nichols Chart is used to perform loop shaping concept. Complex numbers are represented in terms of their magnitudes and phases in the Nichols chart. Every complex number has represented in Cartesian form as (x, y) and a representation of polar form as (r, θ) . The Nichols chart coordinates are represented in the form of $[\theta, 20\log(r)]$. The range of the coordinates is from $-\infty$ dB to $+\infty$ dB. In the design phase, the use of Nichols chart is applicable to the finite range of magnitudes. Due to the frequency response analysis used by QFT method, design compensators guarantee that the performance specifications for each SISO or MIMO problem are satisfied.

The QFT mechanism have simple and transparent design concept by accepting various tradeoffs from the designer to achieve the closed loop system performance specifications.

A linearized model is considered as a plant dynamics to represent the nominal model of the system which is to be controlled. The system parameter variations are taken into account at every frequency of interest (ω_i), to generate plant templates according to the parameter variations, which represent the frequency response of the family of parameter variation systems at a fixed frequency.

In the loop shaping phase, the controller $G(s)$ is designed by adding gain, poles and zeros until the nominal loop $L_0=G_0G$ lies near to its bounds. A robust controller will be evaluated if it meets specified bounds and it has the minimum high frequency gain.

The general controller structure in QFT controller design is expressed by the transfer function in Eq. below:

$$P(s) = Kp \frac{Zeros(s)}{Poles(s)} = \frac{\left(Kp \left(\frac{s}{z1} + 1 \right) \left(\frac{s}{z2} + 1 \right) \dots \left(\frac{s^2}{\omega ni^2} + 2\zeta i \frac{s}{\omega ni} + 1 \right) \dots \right)}{s^n \left(\frac{s}{p1} + 1 \right) \left(\frac{s}{p2} + 1 \right) \dots \left(\frac{s^2}{\omega nj^2} + 2\zeta j \frac{s}{\omega nj} + 1 \right)} e^{-Ts} \quad (3.22)$$

Where, Kp represents gain of the controller, z_i represents real zero, p_j represents real pole, ' r ' represents order of the integrator, ωni represents no. of complex zeros and ωnj represents no. of complex poles.

3.3.1. Plant Definition Window

In QFT toolbox design the first step is defining the plant with respect to uncertainty parameters, where we use the frequency vector of the acceptable plants are given

$$\Omega = [0.01, 0.05, 0.1, 0.5, 1, 2, 3, 4, 5, 10, 20, 30, 40, 50, 100, 500]$$

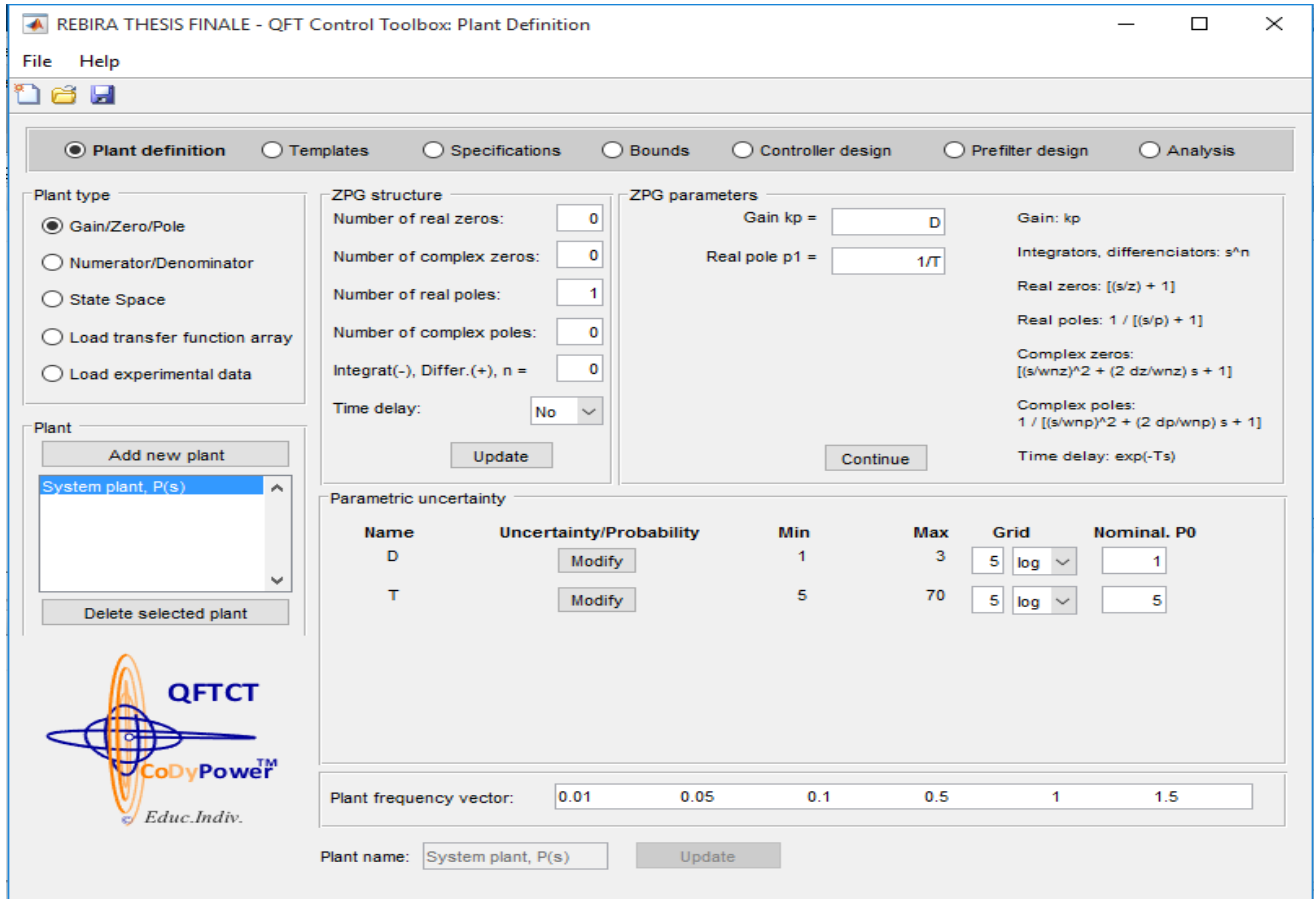


Fig 3.6:- Plant Definition

3.3.2. Generation of Templates Window

The important step in QFT controller design is generation of templates.

Template can be represented on the Nichols chart as a frequency response of the plant with parametric variations at a particular frequency. Fig. 4.6 represents the template generation process. The range of frequencies can be defined by me according to design considerations and experience and some times in the trial and error manner.

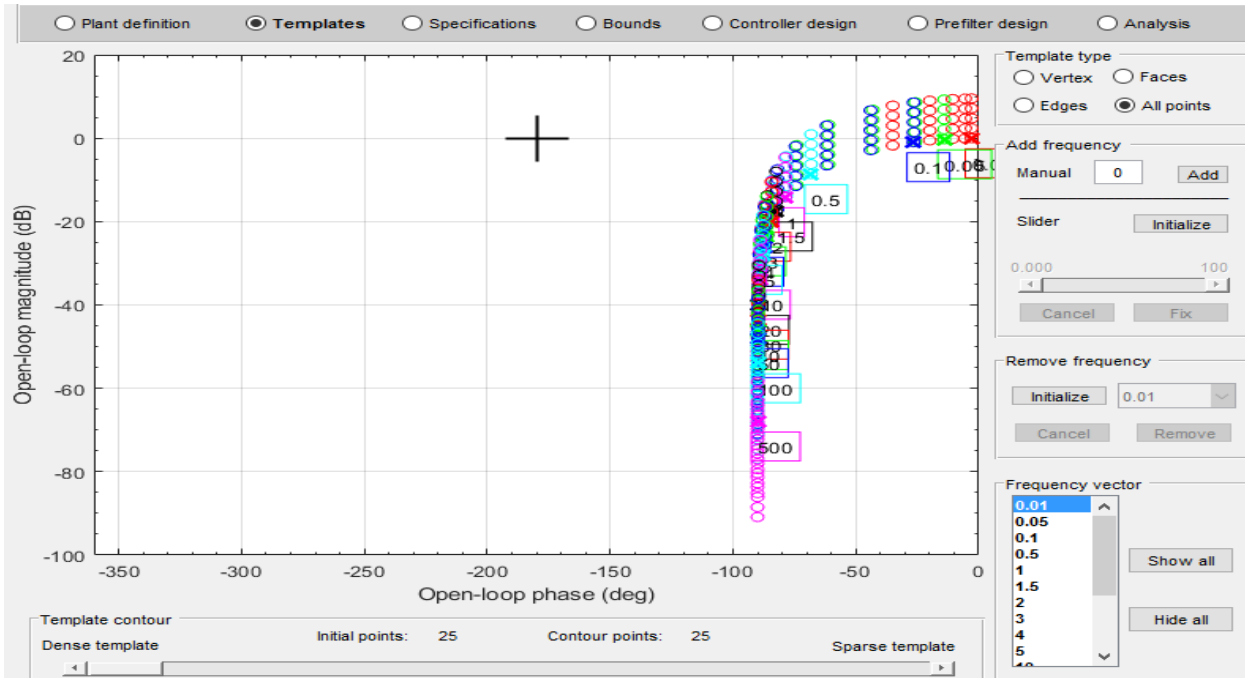


Fig. 3.7:- Generated plant templates at each frequency

In above some bounds are shown for the x-axis; note that the shape of the bounds doesn't depend on any initial controller whatsoever. Plotting all the bounds is of no use because then nothing is visible anymore. Notice that the bounds have continuous lines on the top meaning the final open loop design has to stay above the indicated line, and dotted lines on the bottom meaning the open loop design has to stay below the indicated line.

Some important design objectives which necessitate trade-offs in feedback control are:

1. Performance, good disturbance rejection: needs $T \approx I$ or $S \approx 0$.
2. Performance, good command following: needs $T \approx I$ or $S \approx 0$.
3. Mitigation of measurement noise on plant outputs: needs $T \approx 0$ or $S \approx I$.

The requirement for acceptable performance is for any reference $r(t)$ between $-R$ and R and any disturbance $d(t)$ between -1 and 1 , keep the output $y(t)$ within the range $r(t) - 1$ to $r(t) + 1$ (at least most of the time), using an input $u(t)$ within the range -1 to 1 .

$$d(t) = \sin\omega t \quad 3.22$$

With $e = y - r$ we then have: For any disturbance $d(\omega) \leq 1$ and any reference $r(\omega) \leq R(\omega)$, the performance requirement is to keep at each frequency ω the control error $e(\omega) \leq 1$ using an input $u(\omega) \leq 1$. It could also be argued that the magnitude of the sinusoidal disturbances should approach zero at high frequencies.

While this may be true, we really only care about frequencies within the bandwidth of the system, and in most cases it is reasonable to assume that the plant experiences sinusoidal disturbances of constant magnitude up to this frequency. Similarly, it might also be argued that the allowed control error should be frequency dependent.

3.3.3. Controller Design Specifications Window

Controller design using QFT is carried out by formulating several of the system's frequency specifications, according to the requirements. In this study, three types of specifications are used: robust stability, robust disturbance attenuation, and Robust tracking specifications.

So to determine the QFT controller specification first we have to take the following Assumption

- i. The ideal form of PID controller and pre-filter structure are selected for the design:

$$Gx(s) = Kp \left(1 + \frac{1}{Tis} + Tds \right) \quad (3.23)$$

$$Fy(s) = \frac{1}{\left(\frac{s}{pf1} + 1 \right) \left(\frac{s}{pf2} + 2 \right)} \quad (3.24)$$

Where $X = \{KP; Td; Ti\}$ is the vector of controller design parameters and $Y = \{pf1; pf2\}$ is the vector of pre-filter design parameters with the following uncertain parameter.

- ii. Then Initial domains of these parameters are considered as
 $KP = [0; 30]; Td = [0; 20]; Ti = [0; 15]; pf1 = [0; 20]; pf2 = [0; 20]$.
- iii. In addition to this, we have chosen existing controller as an initial guess of controller and Pre-filter parameters.
 $Kp_0 = 12.6; Td_0 = 0.31; Ti_0 = 2.83; pf1_0 = 3.7; pf2_0 = 8$.
- iv. The initial uncertainty to calculate specification value will be $Do=1, To=5$.

Robust Stability Specification

When the nominal open loop is given as $L(\theta, j\omega) = P(\theta, j\omega)G(j\omega)$ and $H(s) = 1$ and $F(s) = 1$, a robust stability specification is expressed as:

$$|TR(\theta, j\omega)| = \left| \frac{L(\theta, j\omega)}{1 + L(\theta, j\omega)} \right| = \frac{P(j\omega)Gx(j\omega)}{1 + P(j\omega)Gx(j\omega)} \leq \delta_1(s) = W_s = 1.2 \quad (3.25)$$

Where $\delta_1(\omega)$ is the maximum magnitude of the closed-loop tracking transfer function $Y(j\omega)/R(j\omega)$.

In this case we entered as a constant, $W_s = 1.2$, and toolbox gave us the GM and PM. Figure below shows us the Nichols chart with the stability specifications we used.

$$PM = 180^\circ - \frac{180^\circ}{\pi} \arccos\left(\frac{0.5}{W_s} - 1\right) \quad (3.26)$$

Then from the relationship between robust stability constant with Gain margin and Phase margin the toolbox will generate **GM=5.26dB** and **PM=49.24** degrees.

Sensitivity Specifications

In this case we have to calculate the Disturbance at Plant output specification which is to be minimized. In this project the disturbance across plant output will be classified in to Air flow and fuel flow so as to get the desired Pressure value.

$$|T3(j\omega)| = \left| \frac{1}{1 + P(j\omega)G(j\omega)} \right| \leq \delta_3(\omega), \omega \in \Omega_3 \quad (3.27)$$

So from above general equation of determining the value of disturbance across output plant we get third order transfer function.

$$S(s) = \frac{5s^3 + s^2}{5s^3 + 4.9s^2 + 12.59s + 4.4522} \quad (3.28)$$

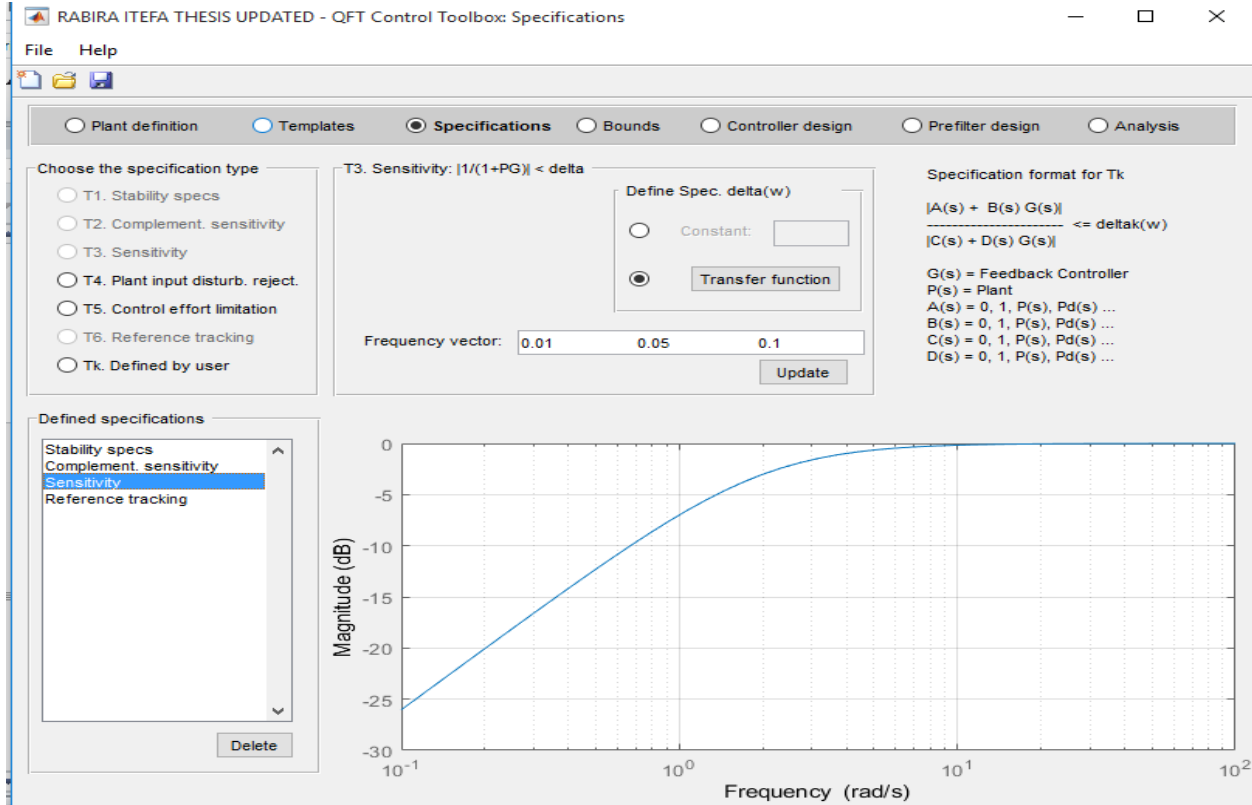


Fig 3.8: - Sensitivity specification

Complementary Sensitivity Specifications

In contrast, the system Complementary sensitivity Transfer function, is the transfer function between output y and reference input r to achieve suitable nominal performance and robust stability,

$$T = PG(I + PG)^{-1} \quad (3.29)$$

$$T(s) = \frac{P(s)G(s)}{1 + P(s)G(s)} = 1 - S(s) = \frac{3.9s^2 + 12.59s + 4.4522}{5s^3 + 4.9s^2 + 12.59s + 4.4522} \quad (3.30)$$

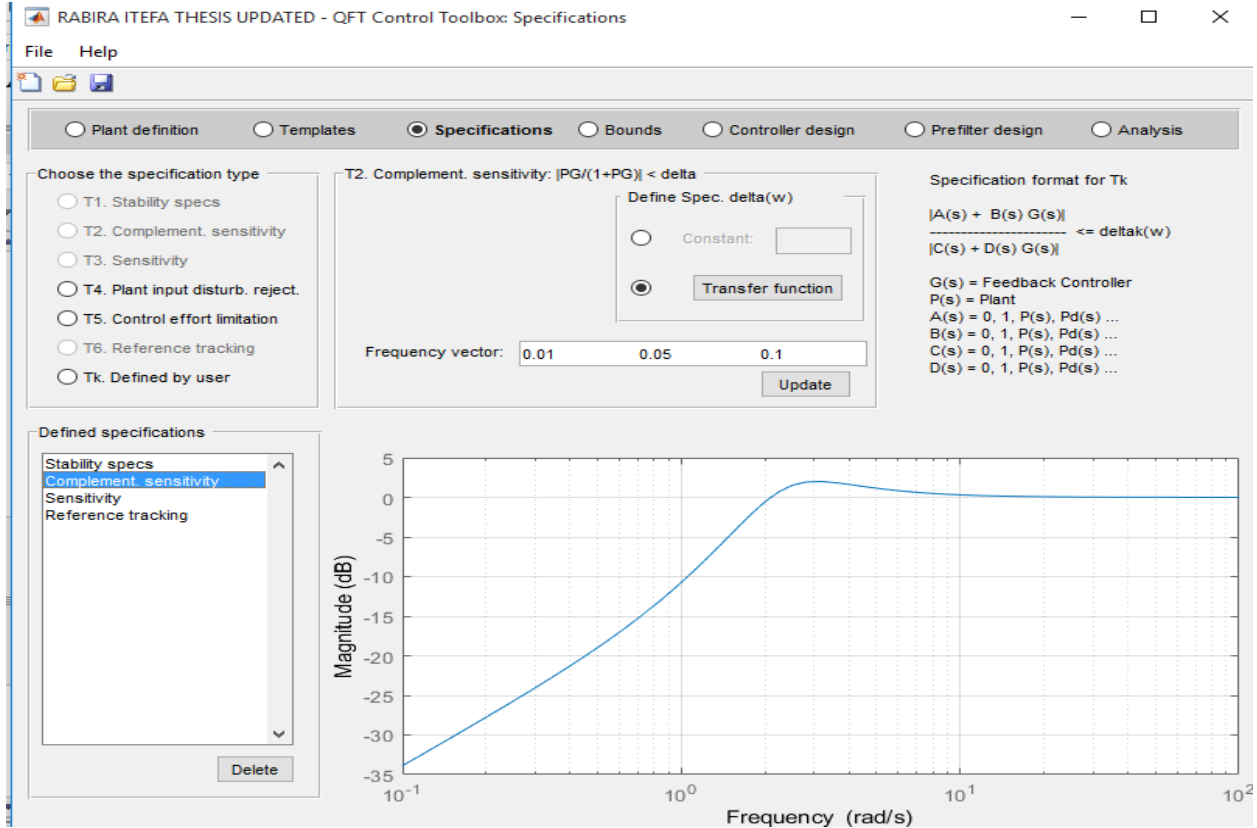


Fig 3.9:- Complementary sensitivity specification

As complementary sensitivity starts to decrease at the closed loop bandwidth, the largest possible value of T dictates the band width.

Weighting functions for parametric uncertainties

From an engineering point of view, it is extremely difficult to give an explicit and versatile formula or to set direct rules on how to choose H_∞ weighting functions, due to the enormous variety of different practical problems. However, many complex systems can often be considered as a combination of equivalent and typical low-order plants. Providing some definite rules for some of these plants will certainly be of great interest to control engineers.

To begin with, consider a standard mixed sensitivity S/R/T design (Postlethwaite et al., 1987) shown above. $W_p(s)$, $W_u(s)$, and $W_m(s)$ are weighting functions for the sensitivity matrix function $S(s)$, the control signal sensitivity matrix function $R(s)$, and the complementary sensitivity matrix, respectively, where $S = (1 + PG)^{-1}$, $R=GS$, $T = PG(1 + PG)^{-1}$ and P is the plant, and G is the controller.

The mixed sensitivity

$$\|N\|_{\infty} = \max \sigma(N(j\omega)) < 1 \quad (3.31)$$

$$N = \left\| \begin{array}{c} W_p S \\ W_u K S \\ W_m T \end{array} \right\|_{\infty} \quad (3.32)$$

Whereas these three weighting function are Tuning knobs of the control Engineers to balance Disturbance rejection and Tolerance to low frequency and high frequency modeling errors.

It is impossible to make both $\|S\|_{\infty}$ and $\|T\|_{\infty}$ small with together. Instead of this we determine the mixed sensitivity norms of N will be minimized as after selecting the form of N and the weights, the H_{∞} optimal controller is obtained by solving the problem $\min \|N(K)\|_{\infty}$ where K is a stabilizing controller.

The plants that will be discussed in this section are a first-order lag (PT_1), a first-order plus dead time process (PT_{1t}), a second-order system with real poles and dead time (PT_{2t}), and second-order oscillatory system (PT_2S).

For a plant without integrator, the following form of Weighting transfer function is suggested:-

$$W_m(s) = \lambda \frac{s + \rho^{\mu}}{s + \tau 1} \quad (3.33)$$

Where $\mu \in \{0, 1\}$. When $\mu=1$, $1/\lambda=1/(W_m(\infty))$ which represents the amplification factor of disturbance at high frequency that is disturbance rejection ability of the system is at high frequency. Parameter μ will be set to zero when high frequency disturbance is not considered.

$$W_m = \frac{25s + 1}{s + 1} \quad (3.34)$$

The next step of the H-infinity design process is to obtain the sensitivity weighting function. The minimum nominal sensitivity requirement and the sensitivity weighting function will be defined. The weighting function was obtained by tuning a standard equation presented in (Skogestad and Postlethwaite, 2005) so that its magnitude is less than the requirement at all frequencies

$$W_p = \frac{\frac{s}{Ms} + Ws}{s + Ws * \epsilon s} \quad (3.35)$$

We get finally:-

$$W_p = \frac{\frac{s}{1.05} + 1.5}{s + 1.5 * 10^{-4}} \quad (3.36)$$

The above weight implies that we require an integral action ($W_p(0) = 10000$) and allow a de-amplification of disturbances at high frequencies by a factor four at most ($W_p(\infty) = 0$).

The input/tuning weighting functions were set to unity.

$$W_u = 1 \quad (3.37)$$

Reference Tracking

The reference tracking problem is given a reference signal, the output of the plant must track the reference within some bounds in the presence of plant uncertainty.

The controller is designed to reduce the closed-loop uncertainty and ensure stability, and a pre-filter is designed to shape the absolute reference to output response.

And Robust tracking specifications is shown below.

$$|A(j\omega)| \leq |Fy(j\omega) \frac{P(j\omega)Gx(j\omega)}{1 + P(j\omega)Gx(j\omega)}| \leq |B(j\omega)| \quad (3.38)$$

Where $A(j\omega)$ is the lower tracking specification and $B(j\omega)$ is the upper tracking specification. From the given plant with uncertainty and Ideal (initial) controller we determined the lower and Upper Bounds of the specifications.

$$A(j\omega) = \frac{0.658(s + 24)}{(s + 2 + j1.1376)(s + 2 - j1.1376)} \quad (3.39)$$

$$B(j\omega) = \frac{746.76}{(s + 3)(s + 4)(s + 8)(s + 69)} \quad (3.40)$$

At each $s=j\omega$.

From above upper and lower bound tracking the zero of $A(j\omega)$ i.e. ($s=-24$) and two fast poles in $B(j\omega)$ i.e. ($s=-8$ and $s=-69$) have been included to ensure that the magnitude frequency responses of $A(j\omega)$ and $B(j\omega)$ diverge each other at high frequency.

The frequency of interests ω have been selected as $\omega = \{0.05, 1, 2, 3, 4, 5, 10, 20, 50\}$ rad/s.

Robust Control of Industrial Boiler Using Quantitative Feedback Theory

Then entering those transfer function of the lower and upper bound specification in toolbox we generate the following graph where all the plant will be acceptable only between those bounds.

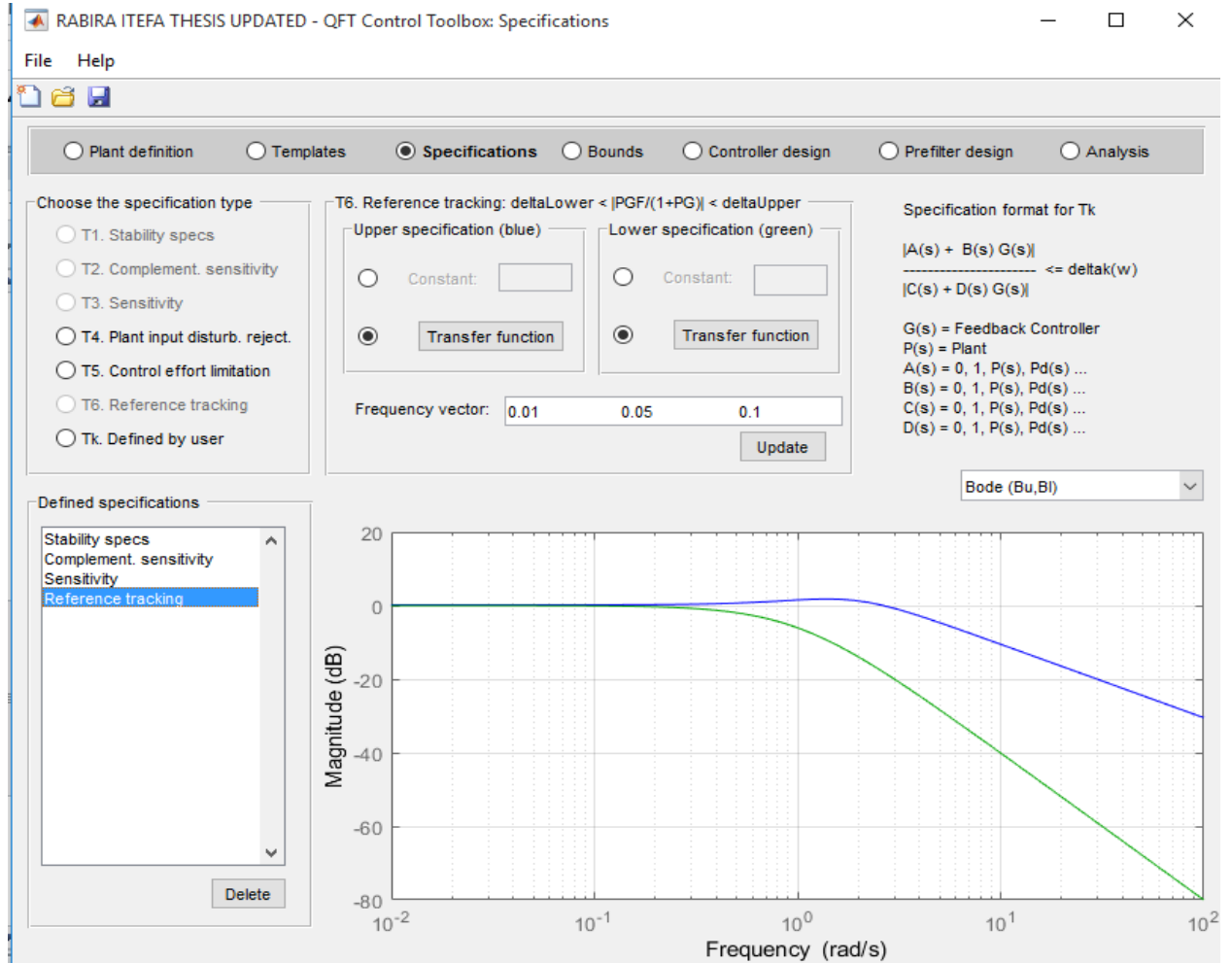


Fig 3.10:- Reference Tracking Specification

3.3.4. Generation of the bounds window

The next step in QFT is translating the templates into bounds on the Nichols plot. For this a nominal plant model is necessary. Which of the plants is chosen to be the nominal one is irrelevant, it can even be one that is outside the parameter space. The nominal plant corresponds in each template to a certain point. From demands on stability and for instance disturbance rejection certain circles and lines will appear (M-lines). Generation of bounds is necessary for disturbance rejection also.

The QFT toolbox provides eleven types of bounds that can be generated. It is pointed out that stability requirement like Gain margin and Phase margin are easy to indicate on the Nichols Chart. So by requiring a PM or GM, the M-line to stay out of can be calculated directly. This shows again another advantage of the Nichols Chart.

The bounds are calculated by moving each plant template around the Nichols Chart to satisfy the control specification. At each point, the nominal plant is marked on the Nichols Chart. This is done by hand.

Stability Specification Bound

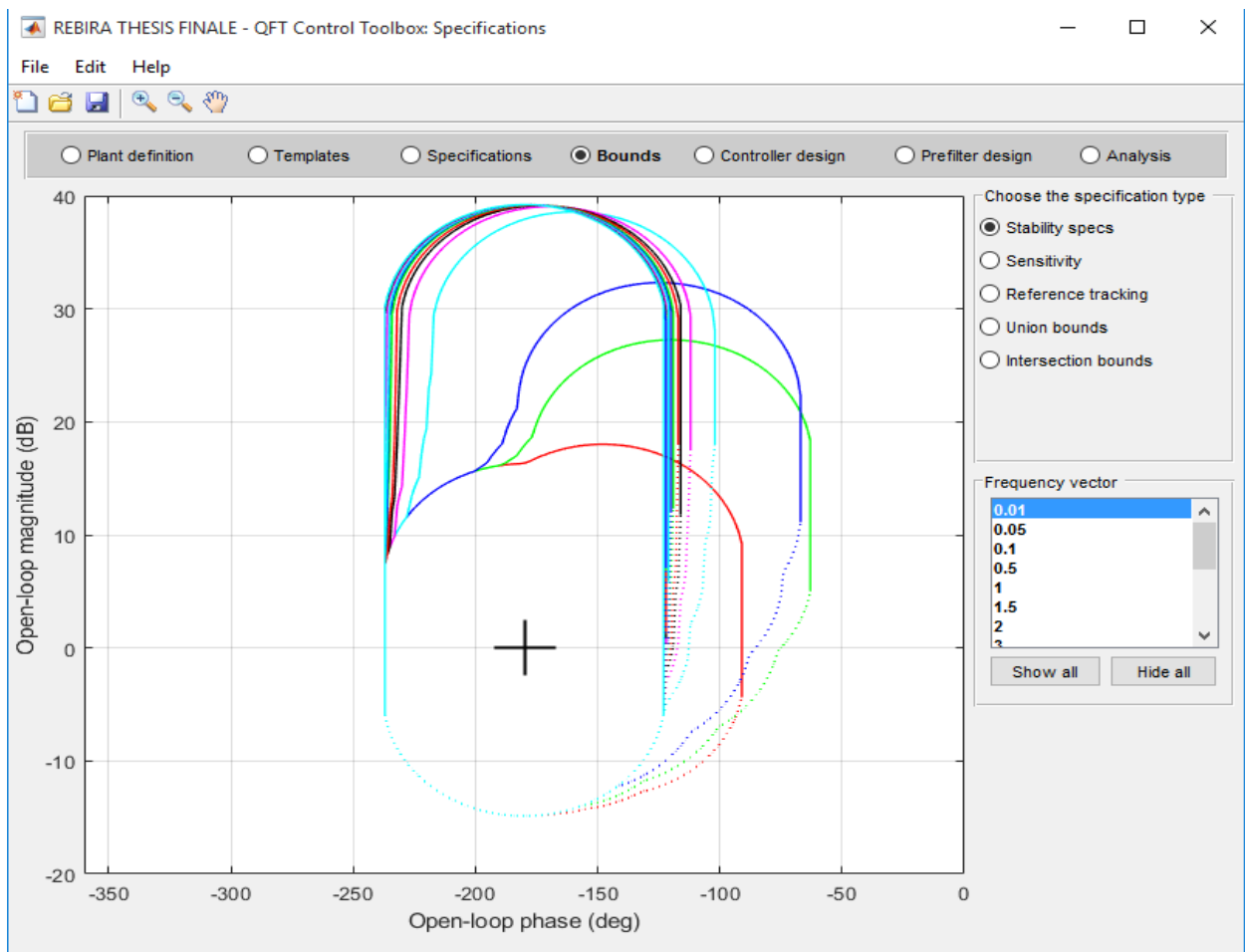


Fig 3.11: - Stability specification bound

- ❖ In this bound window each frequency ω determine the region of $C(i\omega)$ for which the specification can be satisfied for all processes P for some $F(i\omega)$ and we plot the regions $C(i\omega)P_{nom}(i\omega)$ in the Nichols diagram(Horowitz Bound)

Choose the initial controller to be $G(s) = 1$. Then $L_n = P(s) G(s)$.

The plot of L_n , along with the bounds needs to satisfy the performance specification for $w < 0.1$. This means that the $w = 0.1$ point on the loop transmission must lie above the performance bound. The gain of L_n affects its vertical position.

Sensitivity Bound

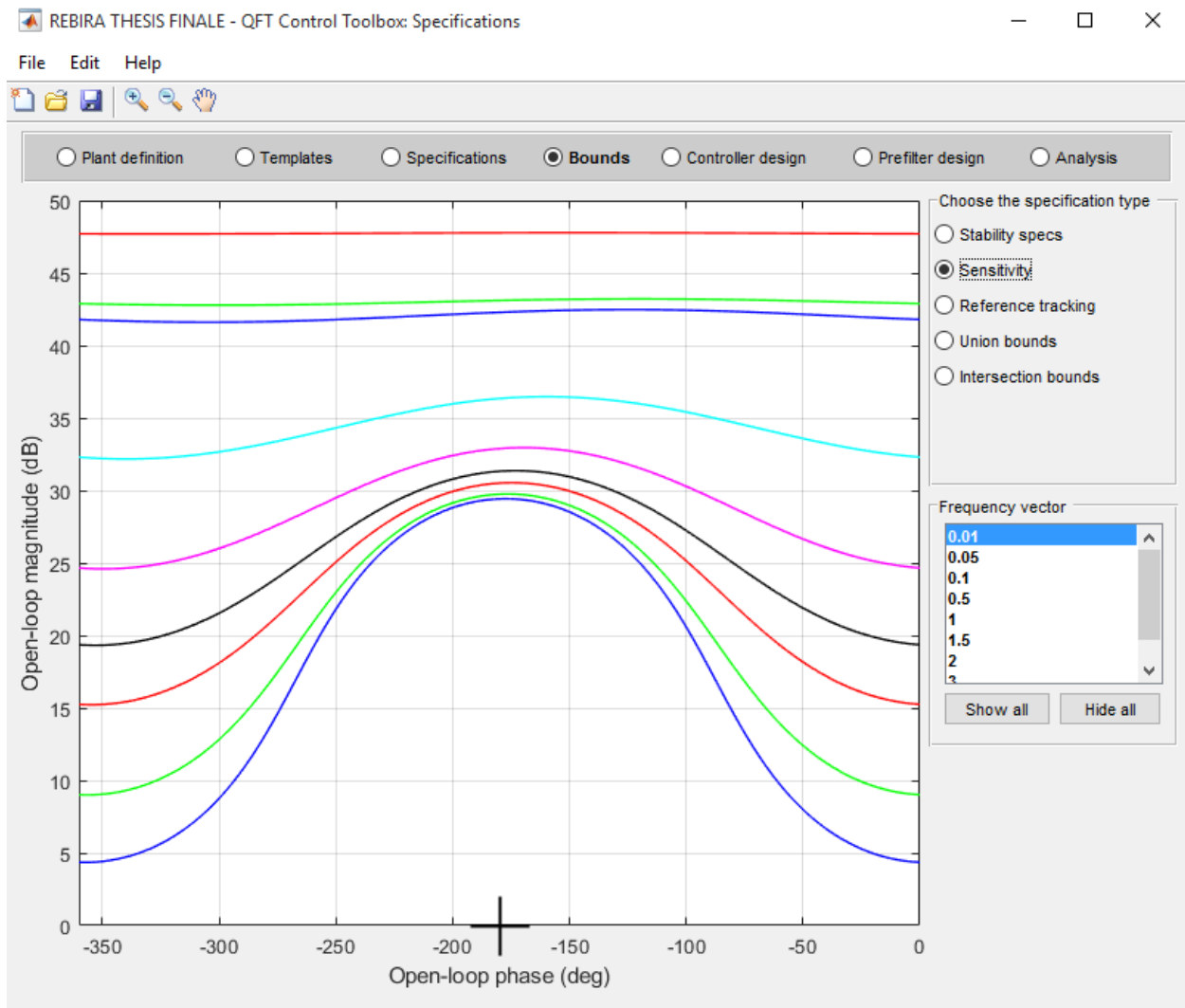


Fig 3.12: - Sensitivity Specification bound

Reference Tracking Bounds

In the case of tracking bounds, the phase of the nominal open-loop system is first fixed, and the open-loop gain is varied until the maximum variation in closed-loop gain over the corresponding uncertainty template is equal (within a small specified gain tolerance) to the “spread” in closed-loop specifications $\delta(\omega_i)$.

Particular care must be taken when the template encloses the critical (-1) point, since uncertainty templates are normally specified by points at their boundary. (This condition may be easily checked by a simple routine that estimates the angle variation of the vector connecting the critical point to a point on the boundary of the template).

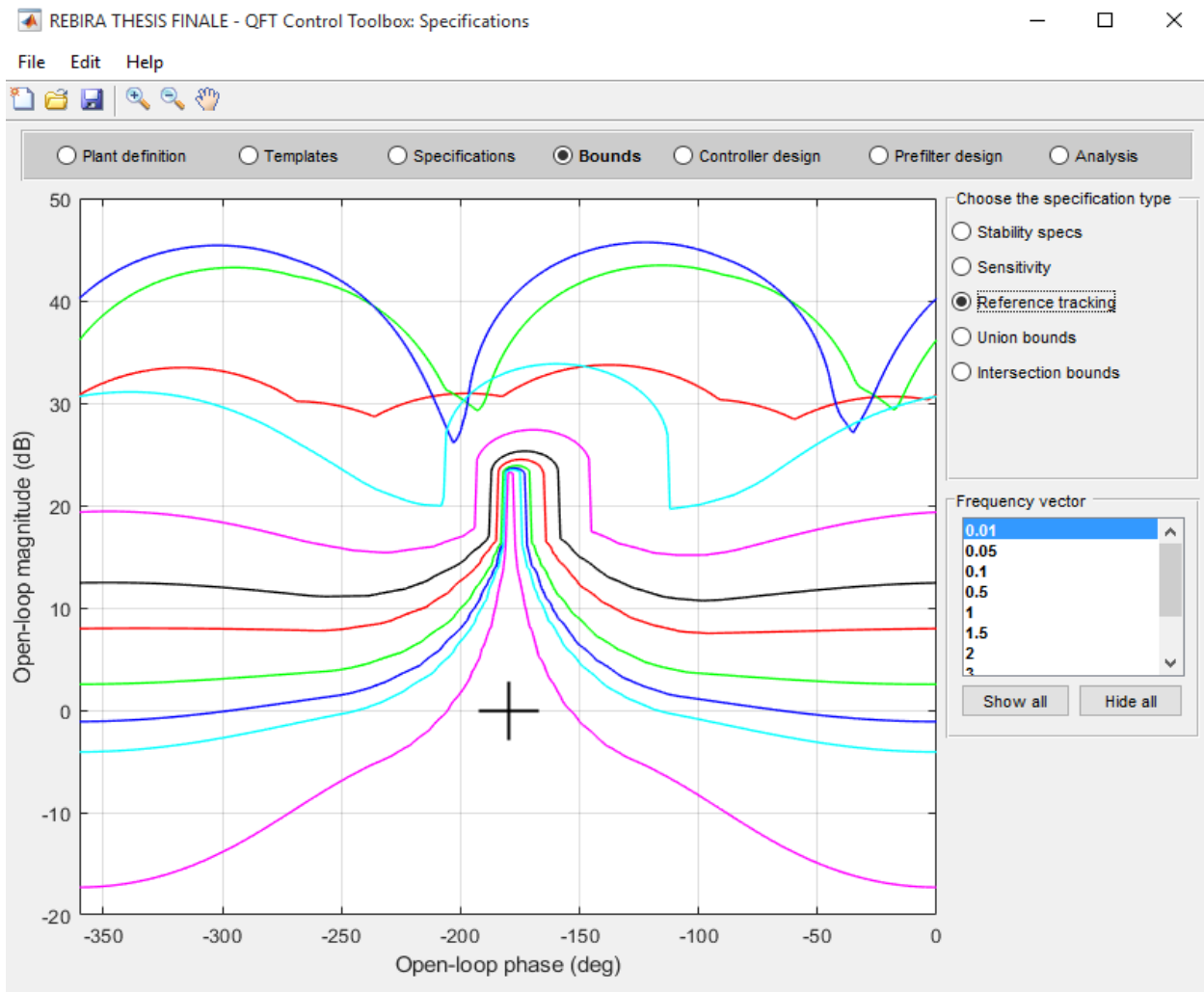


Fig 3.13:- Reference Tracking Bounds

Union Bounds

Viewing the grouped bounds is often used to compare different specifications and quickly identify competing bounds that a nominal loop cannot satisfy simultaneously. In general, when the problem involves more than one set of bounds, one should compute the worst case bound of all sets. It is much simpler to work with a single, worst case bound (i.e., the intersection of all bounds) than with a collection of many bounds.

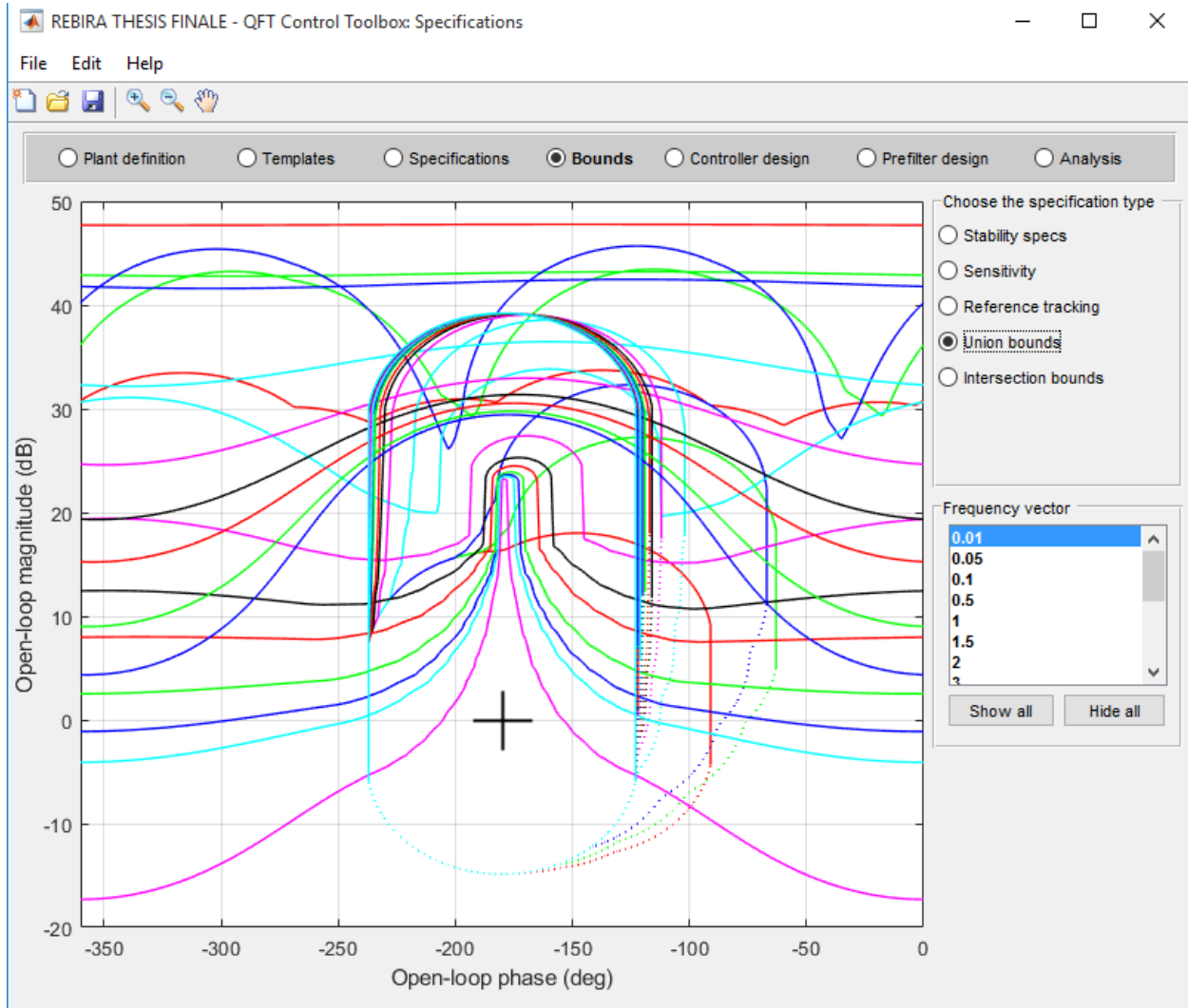


Fig 3.14:- Union all Specification Bounds

Intersections Bounds

Intersection of bounds is determined and the worst case of all bounds is shown in Fig. 4.14 given below. The composite or intersection bound for each value of frequency ω_i is composed of those portions of each respective bound (tracking and disturbance) that are most restrictive. When there are intersections between four bounds, the outmost of the four boundaries becomes the perimeter. If there are no intersections, then the bound with the largest value or with the outermost boundary dominates. This is the final bound taken for the design of the feedback controller.

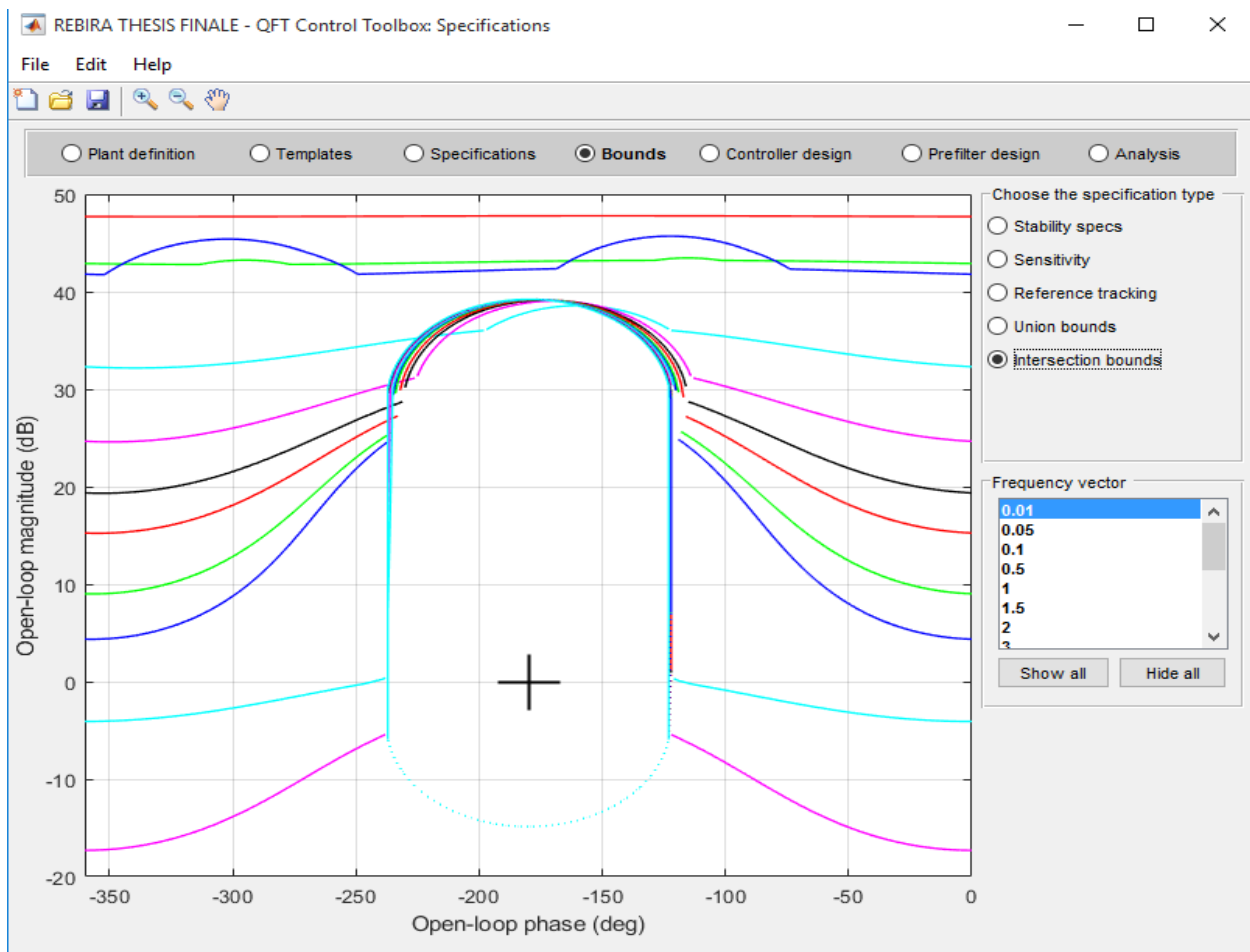


Fig 3.15: - Intersection (Worst Case) Bound

3.3.5. Controller Design (Loop Shaping) Window

Having computed all bounds, the next step in QFT design involves the design (loop shaping) of a nominal loop function that meets its bound. The nominal loop is a product of the nominal plant and the controller to be designed. The nominal loop has to satisfy the worst case of all bounds.

In the loop shaping procedure, bear in mind that the optimum designs in an LTI system is the minimization of the high-frequency loop gain while satisfying the performance bound. Gain has a significant effect on the high frequency response and it is important to minimize the high frequency loop gain in order to minimize the effect of sensor noise whose spectrum, in general, lies in the high-frequency range.

So far we have seen that in order to attain the desired closed loop specifications the following requirements must be fulfilled:-

- ❖ $L(j\omega)$ must not penetrate the U-contour to ensure stability and minimum damping.
- ❖ The nominal open loop must lie on or above the tracking and disturbance bounds on the NC (for each corresponding frequency) to achieve robust tracking performance and disturbance rejection. If it happens to exist in the left region of the NC, it must also lie on or above the 0 dB bound to ensure stability.

Open loop shaping can be achieved using compensator design. A compensator or controller is a dynamic system that is used in cascade with the plant to achieve the desired open loop characteristics. Once we introduced the information about the plant and the control specifications, and the templates and bounds have been calculated, the next step involves controller design or loop shaping. This involves changing the gain and adding the poles and zeros, either real or complex, until the nominal loop lies near its bounds, more specifically above the solid-line bounds.

Having computed stability and performance bounds, the next step in a QFT design involves the design (loop shaping) of a nominal loop function that meets its bounds. The nominal loop is the product of the nominal plant and the controller (to be designed). The nominal loop has to satisfy the worst case of all bounds. The Toolbox includes an interactive design environment for loop shaping.

Often, we know a priori of certain poles and zeros that the controller must have, e.g., an integrator, or have designed a controller using other methods. This is the purpose for the initial controller input argument C_0 .

Generally speaking, loop shaping involves adding poles and zeros until the nominal loop lies near its bounds and results in nominal closed-loop stability.

- ✚ When we design QFT controller the Integrator is added to improve the low frequency response and Better meet tracking Bounds.
- ✚ The gain is then increased to satisfy the low frequency bounds, yielding the bottom left plot.
- ✚ At this point, the low frequency requirements are met but the close loop is unstable while it is again corrected by another classical controller like PID.

The nominal loop to be designed is $L_0(s) = P_0(s) G(s)$ where $P_0(s)$ is the nominal plant. Within the design environment, although we loop shape the plot of $L_0(s)$ during design we are in essence loop shaping the controller $G(s)$. Therefore, in any design step, the zeros and the poles we are working with are our controller elements $G(s)$. PID controllers, also known as three term or process controllers, are one of the most common type of controllers used commercially. The transfer function of a PID controller is the following:

$$G(s) = Kp + \frac{Ki}{s} + Kds \quad (3.41)$$

From Controller design we showed that some steps of design of feedback controller $G(s)$ =PID in the Nichols chart (Loop shaping).

The derivative time from $Td=0.4$ to 20.

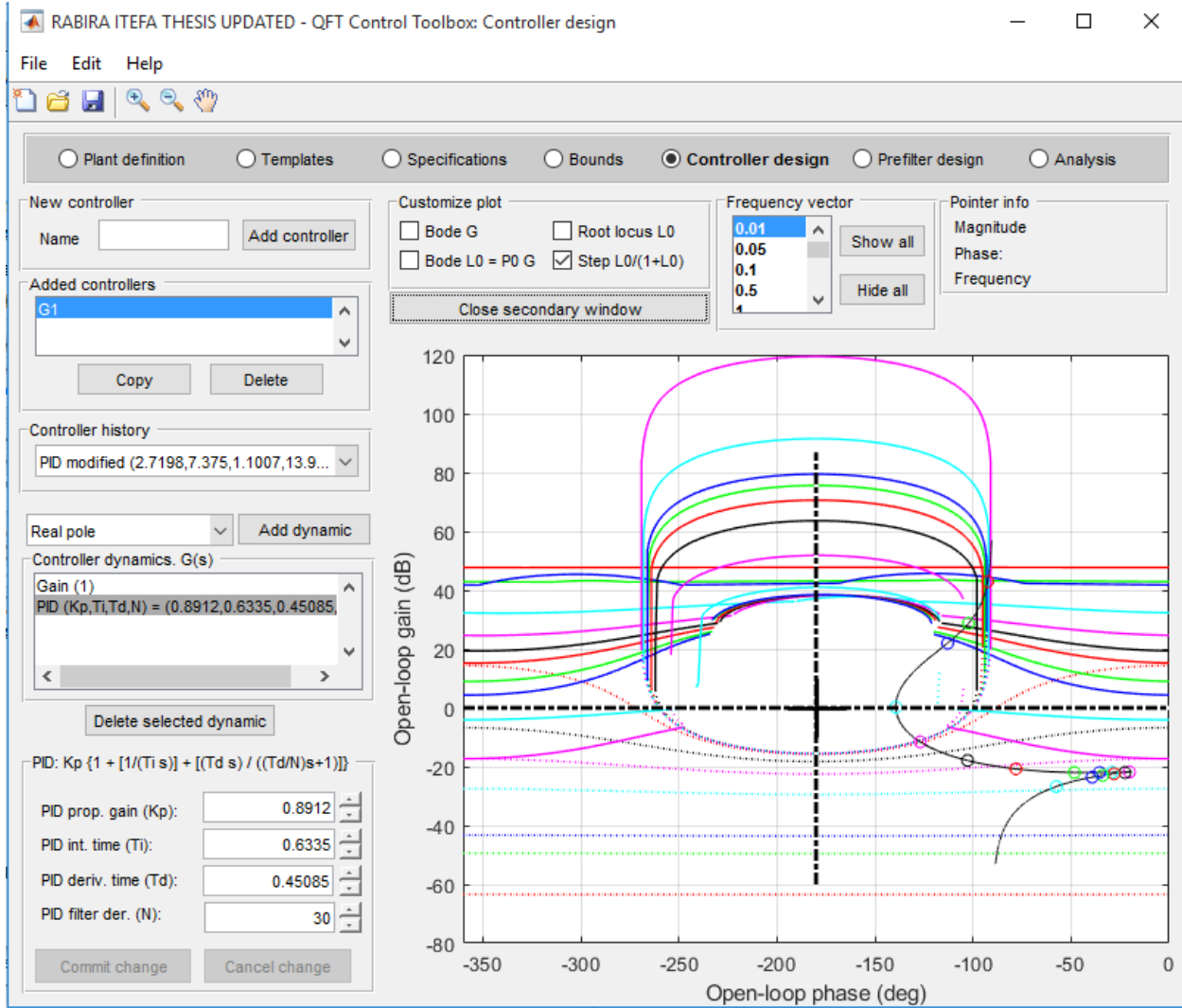


Fig 3.16: - Controller design window at $T_d=0.45$ seconds

This section is devoted to QFT method in order to design a controller for the power plant pressure.

From the design we have to synthesize the suitable controllers and pre-filters such that:

- ✚ The closed-loop system is stable;
- ✚ It can track desired inputs
- ✚ Cross-coupling effects can be reduced by using suitable robust disturbance rejection bounds

Robust Control of Industrial Boiler Using Quantitative Feedback Theory

The QFT design objective of my Thesis achieves the following:

- Representing the characteristics of the plant and the desired system performance specifications in the frequency domain.
- Using these representations to design a compensator (controller).
- Representing the nonlinear plant characteristics by a set of LTI transfer functions that cover the range of structured parametric uncertainty.
- Representing the system performance specifications by LTI transfer functions that form the upper B ($j\omega$) and lower A ($j\omega$) boundaries for the design.

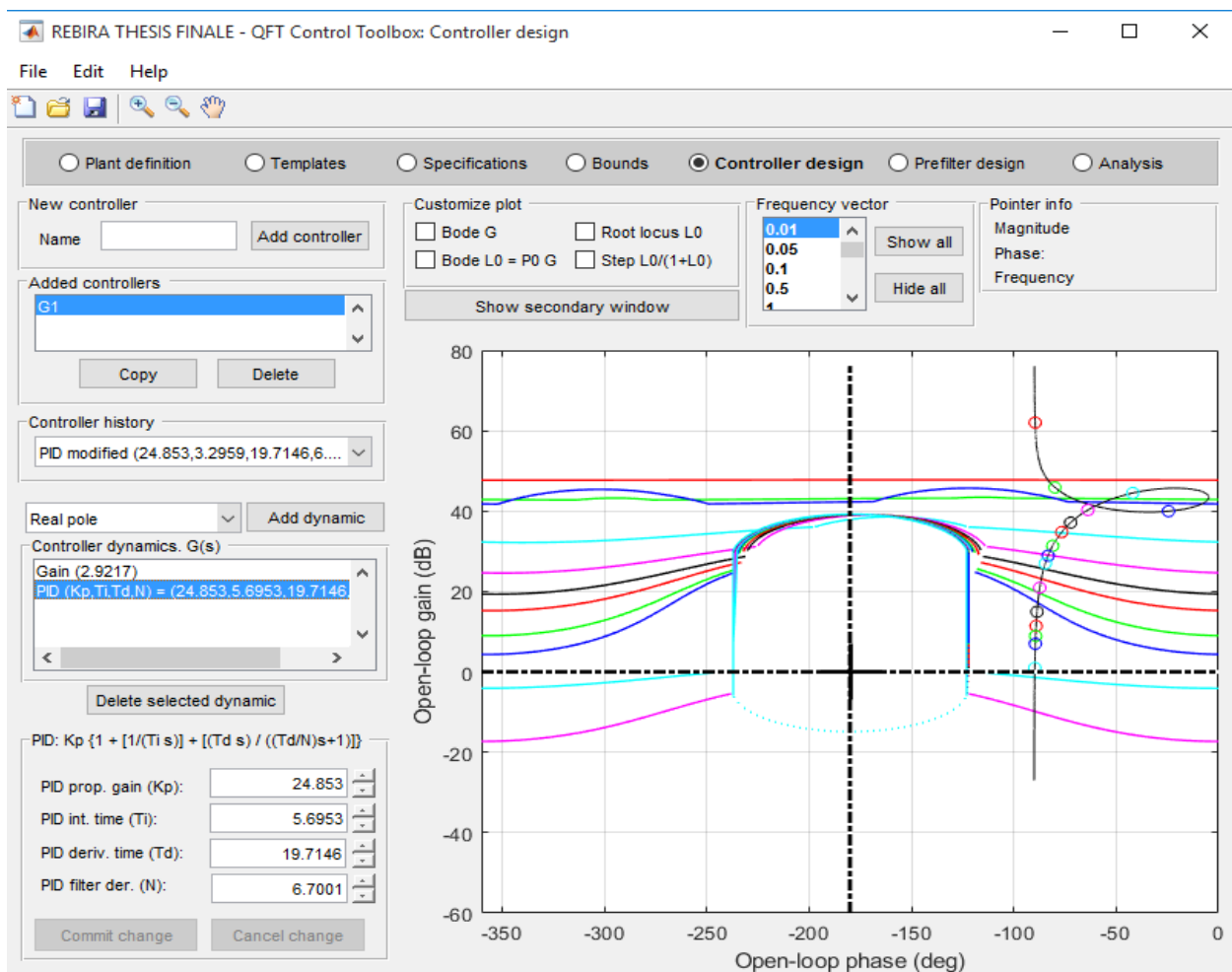


Fig 3.17:- Controller Design.

Robust Control of Industrial Boiler Using Quantitative Feedback Theory

From above figure, note that in the left region of the NC (region of phases from -360° until the first phase at which the U-contour starts forming) the nominal open loop (if it exists in this region) must lie on or above the bound at 0 dBs (straight line) to ensure stability. It is also a good practice to consider the U-contour as a performance bound. While at high frequencies due to the dominance of the derivative term, PID is introduced. Note also that the shaped optimal open loop lies on or above the bounds, with four frequencies of interest ($i=1, 3, 4, 6$) lying exactly on the bounds.

The performance specifications or NVH (noise, Disturbance across Fuel and Air Flow) specifications depends plant to plant varied, and is a measure of smoothness. In this case study we assume a maximum error that is computed by $\max(\text{Output Pressure} - \text{Reference Pressure})$ to be 50bar. This value indicates the band around the Boiler reference pressure within which the Boiler pressure is maintained. In the case of low Boiler pressure, this band is critical as a large drop from reference pressure could cause inefficient Combustion.

From Graphical approach of fig 3.18 controller settings are as follows. $KP = 24.853$, $TI=5.6953$, $TD = 19.7146$, Filter Coefficient (N) = 7.

$$G_{pid}(s) = kp \left\{ 1 + \left[\frac{1}{Ti s} \right] + [Td s / \left(\left(\frac{Td}{N} \right) s + 1 \right)] \right\} \quad (3.42)$$

$$G_{pid}(s) = 24.853 \left(1 + \left(\frac{1}{5.6953s} \right) + \frac{19.7146s}{2.816s + 1} \right) \quad (3.43)$$

3.3.6. Pre-filter Design Window

The design of an appropriate nominal open loop $L_o(j\omega)$ guarantees only that the variation ('spread') of the magnitude response of the control ratio $|TR(j\omega i)|$ is within the allowed specifications (i.e. less or equal than $dR(j\omega i)$) [Ref.1]. The role of a *pre-filter* in a control system is to place:-

$$LmT(j\omega) = \frac{L(j\omega i)}{1 + L(j\omega i)} \quad (3.44)$$

Within the given specifications in the frequency domain, that is, the variation of $|TR(j\omega i)|$ must lie within the bounds $A(j\omega)$ and $B(j\omega)$. Figure shows the bounds $A(j\omega)$ and $B(j\omega)$ with the variation of $|TR(j\omega i)|$ for the considered in equation with parameter range of $D=\{1,2,3\}$ and $T=\{5,10,20,70\}$. Note that in order to find the maximum spread of the variation of $|TR(j\omega i)|$, the minimum and maximum values for each uncertainty parameter must be included, i.e. the 'best' and the 'worst' case-combination.

Pre-filter Design procedures

1. First, the CL system responses without the pre-filter are determined by taking various combinations of the uncertainty parameters of the plant (including at least the maximum and the minimum values of each of the parameters to ensure maximum spread).

Initially when we assume the ideal pre-filter as unity i.e. $F(s) = 1$, the response in terms of magnitude and phase is out upper bound and lower bound specification.

Where the Blue dotted line is the desired specification and the black one is when Pre-filter applied to track within the acceptable region.

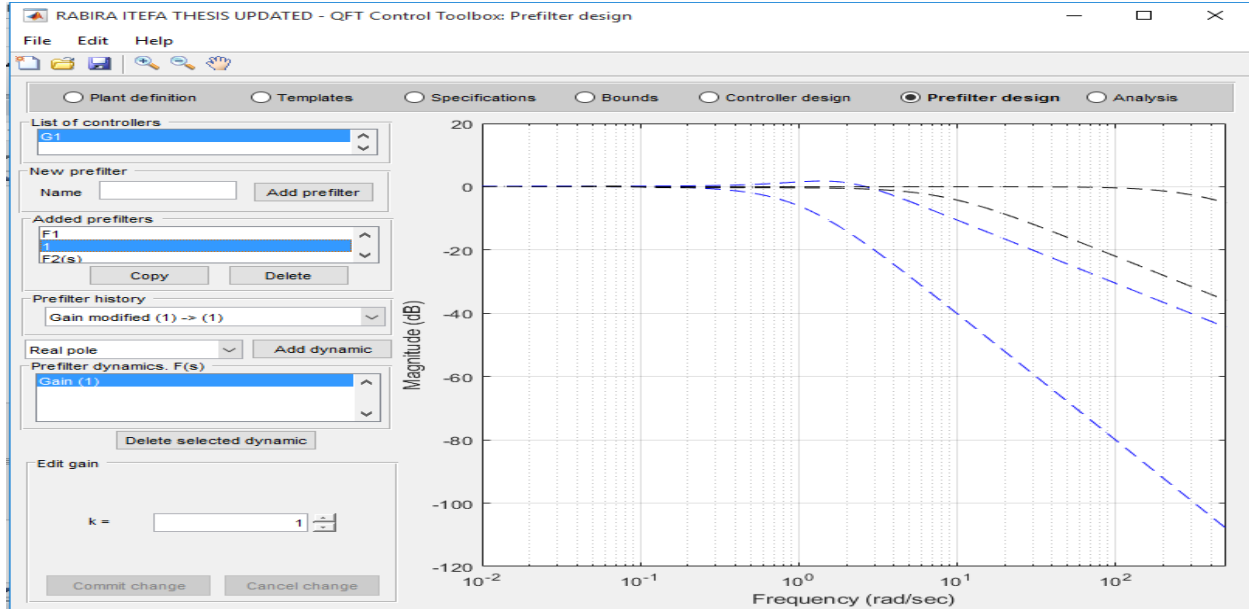


Fig 3.18: - When Pre-filter is $F(s) = 1$

2. From the responses determined in step 1, the maximum and minimum bounds are obtained (their difference gives the maximum spread). Figure above shows the desired range of bounds and the maximum spread of the CL system responses.
3. From steps 1 and 2 above, we obtain the differences $[Bu - LmTmax]$ and $[Bl - LmTmin]$. These differences represent the maximum acceptable bound and the minimum acceptable bound of the magnitude frequency response of the pre-filter. Figure 4.2.B shows the range in within the bound of the pre-filter must lie in order to satisfy that the CL system responses will lie within the given specifications.
4. Using straight-line approximations (usually by inspection from the graphs), $F(s)$ can be synthesized such that $LmF(j\omega)$ will lie within the range of allowable bounds from step 3.

Moreover for step forcing functions,

$$\lim_{s \rightarrow 0} F(s) = 1 \quad (3.45)$$

is enforced so that the s.s.e to step inputs is zero.

5. The pre-filter $F(s)$ obtained from above procedure ensures that the CL system responses lie within the specified range shown in figure above ($B(j\omega) - A(j\omega)$), for all combinations of the uncertain parameters.

Robust Control of Industrial Boiler Using Quantitative Feedback Theory

The frequency response of $F(s)$ at certain frequencies can vary more than at other. This can be verified from figure 3.3. Where we can see that at frequencies $\omega 4, 5$ the CL system responses have a greater range of adjustment because the specifications are not tight.

In addition in frequencies $\omega 1, 2, 6, 7$ the specifications are very tight and therefore the pre-filter bounds are limited (i.e. only one straight-line approximation can be chosen).

From the Pre-filter designed Using the specification will be a second order pre-filter suffices; this has the following transfer function:-

$$F(s) = \frac{Ag}{\left(1 + \frac{s}{\omega c1}\right)} \quad (3.46)$$

Where, Ag is an adjustable gain (usually 1), $\omega c1$ is the first cut-off frequency, respectively. Higher order pre-filters can be used in order to give a more accurate result. Nevertheless, higher order pre-filters are more complex and thus the minimum-order possible pre-filter is desired.

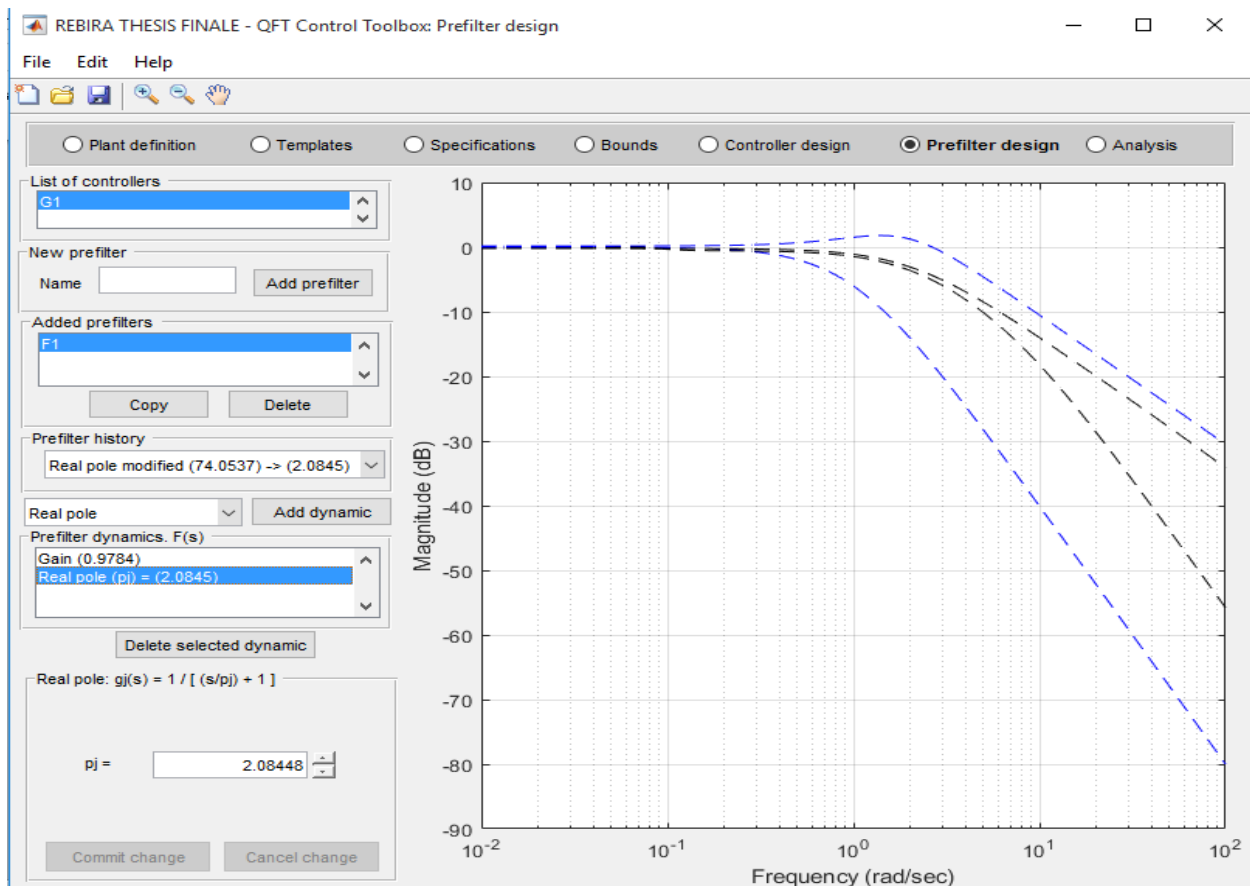


Fig 3.19:- When we add gain of 0.9784 and real pole of 2.0845 to Pre-filter

From Graphical approach we generate the following pre-filter:-

$$F(s) = \frac{0.9784}{s + 2.0845} \quad (3.47)$$

And when adding the pole to track by taking the zero to further to acceptable Upper bound and lower bound specification. At this point we have successfully completed the design of controller $G(s)$ to meet margin specs and reduce closed-loop tracking variations. Now we are ready to complete the design.

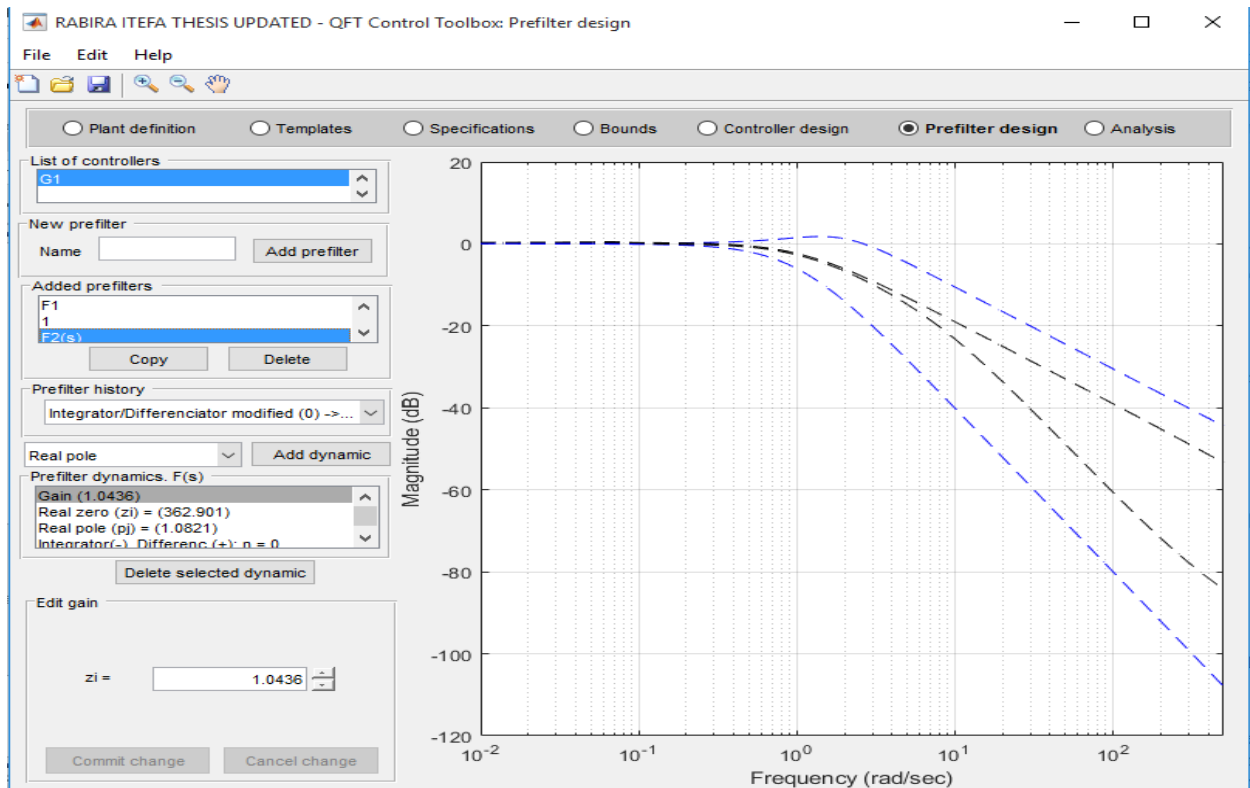


Fig 3.20:- Adding real pole of 1.0821 and real Zero of 362.9 to pre-filter

Then from graphical approach of pre-filter design we generate first order transfer function.

$$F(s) = \frac{1.0436(s + 362.9)}{s + 1.0821} \quad (3.48)$$

In summary, we conclude that the role of $F(s)$ is to shape the input tracking energy but not affect closed-loop robustness properties. This is necessary because shaping the loop transmission to satisfy the QFT bounds only guarantees that the overall variation in the closed-loop magnitude will be less than the specified tolerance.

3.3.7. Analysis Windows

Once the controller design is done, the Analysis window is active in the QFT toolbox. The analysis is performed in both, the frequency domain and time domain. The frequency domain analysis panel helps us to do an analysis of the closed-loop system response of the control system with respect to a specification defined in the specification window. In figure below the dashed line is desired specification and solid line the worst case of the control system over the plant uncertainty at each frequency.

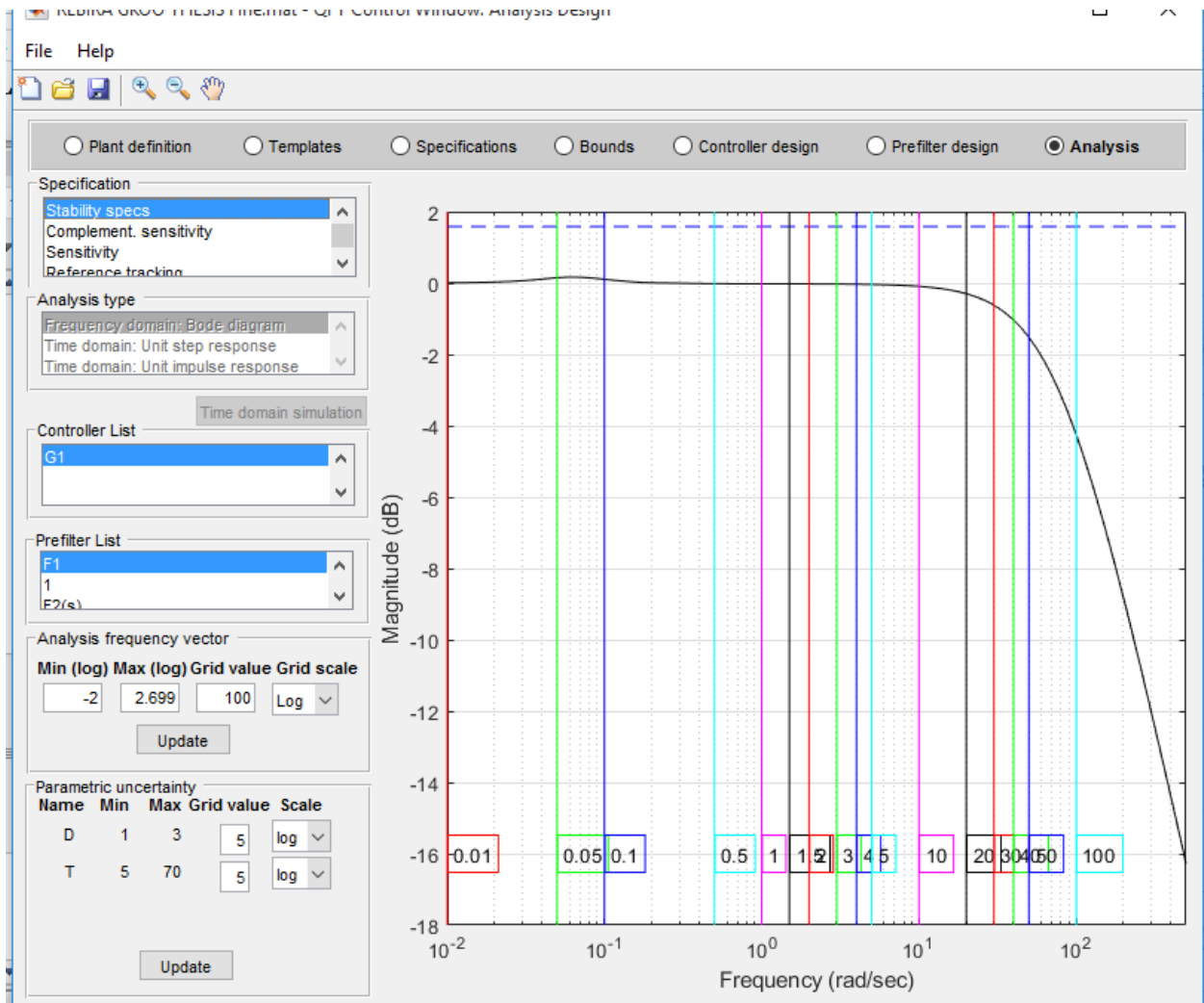


Fig 3.21:- Frequency domain of stability specification after pre-filter

From above graph we conclude that evaluating the margin performance shows that we are meeting the specification (since the bounds were satisfied during loop shaping and the shape of the plant template is fixed at the high-frequency range).

$$\max_{p \in P} \left| \frac{PG}{(1 + PG)}(j\omega) \right| \quad (3.49)$$

Sensitivity

Figure 3.22 shows the frequency domain analysis for output disturbance rejection. The dotted line represents the upper boundary of the specs and we see from the figure that all the plants in the worst case scenario are within the bounds for frequency ranges 0.01 rad/s to 5 rad/s. On the other hand, from higher frequency rad/s we have seen that it is out of specifications.

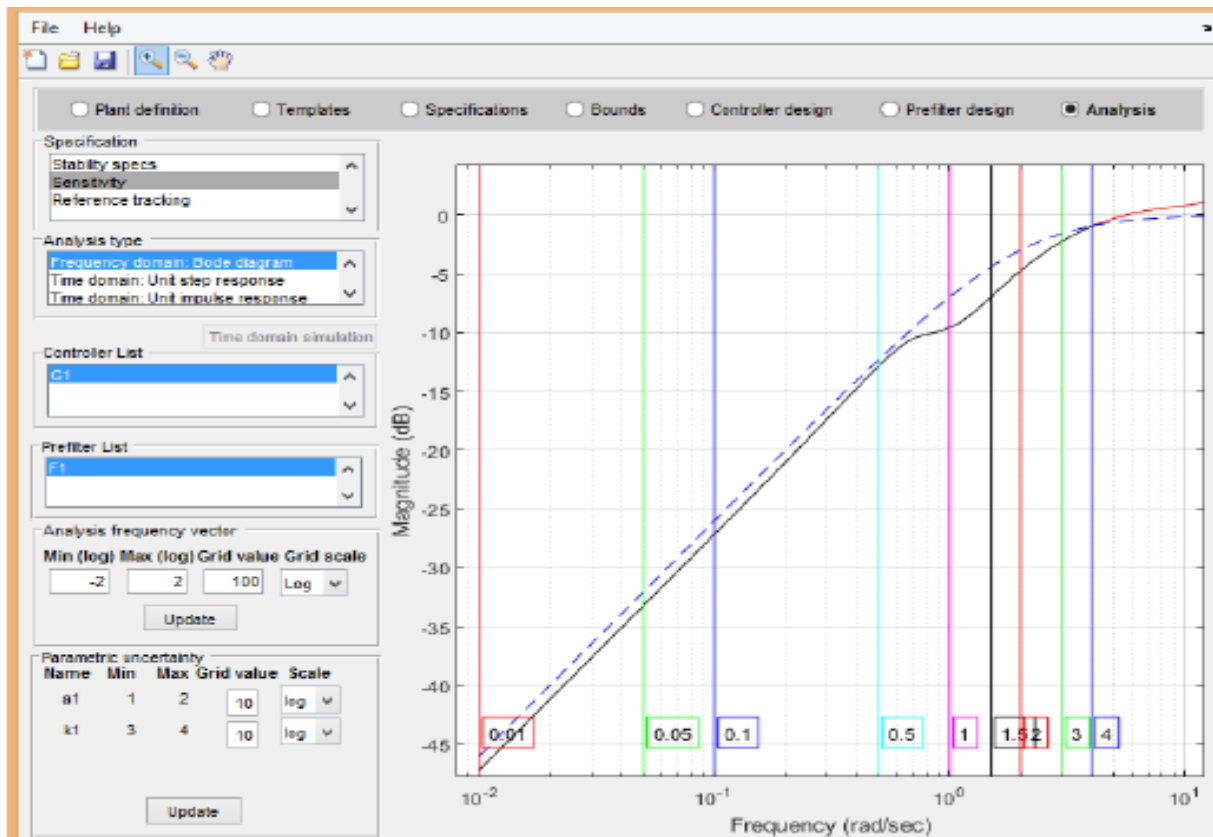


Fig 3.22:- Sensitivity Frequency domain specification

Figure 3.21 and 3.22 shows respectively that the stability and sensitivity specifications of the closed loop system are met for all the plants within the uncertainty. The solid lines (Worst case within the uncertainty) are below the specification (Dashed lines).

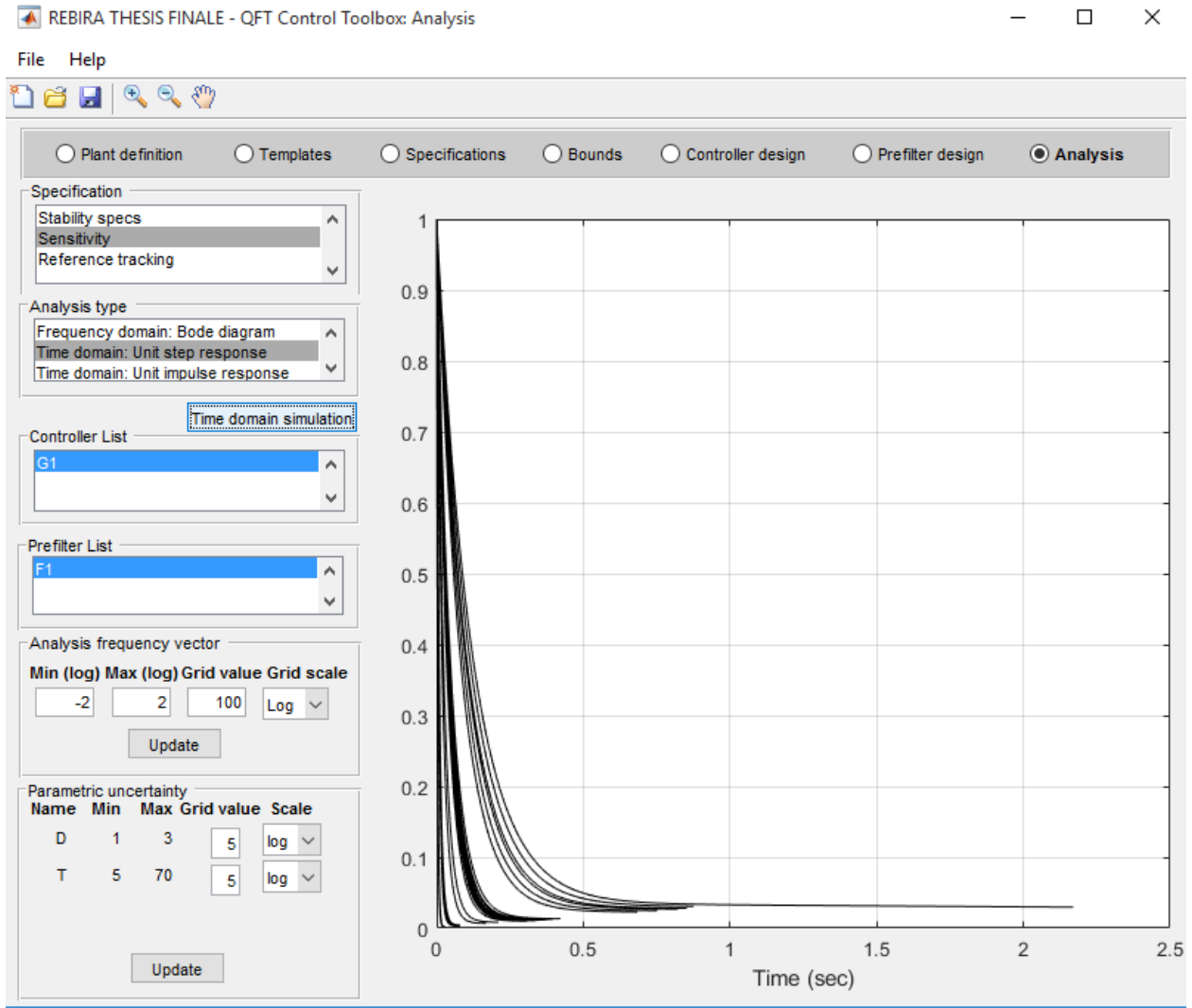


Fig 3.23:- Sensitivity Time domain specification

The time domain panel analyses the time response of the control system, with many plants over the plant uncertainty. The toolbox lets us apply a unit step and unit impulse signal. Figure below shows the time domain analysis for a unit step where 10 plants are analyzed. The number of plants analyzed (number of lines plotted) depends on the values introduced in the “Parametric uncertainty” panel in the plant definition window.

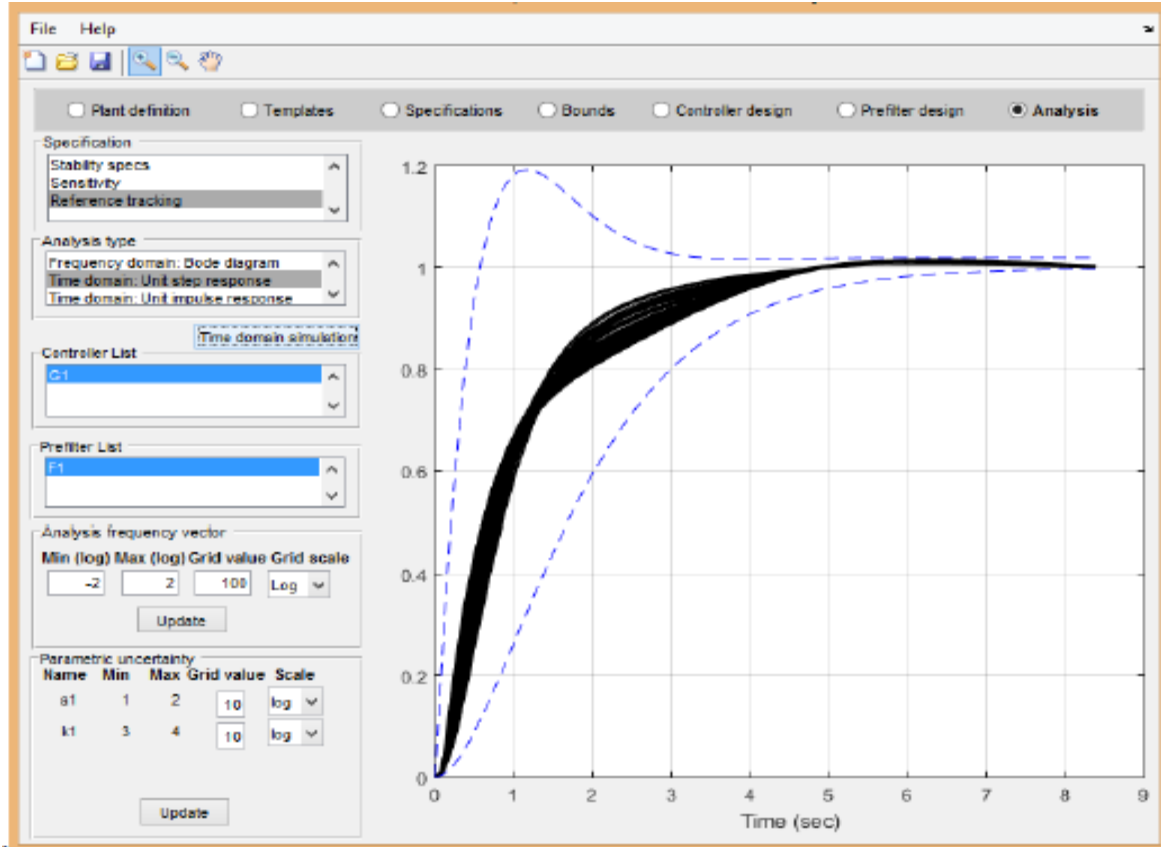


Fig 3.24:- Reference tracking time domain specification

Figure 3.24 show that the reference tracking specification of the closed loop system is met for all the plants within the uncertainty. The figure shows that the response for $Lo(s)F(s)/[1+Lo(s)]$ is within the given desired upper and lower bound specification with the reference signal value is $r(s)=$ unitary step. The number of plant analyzed (number of line plotted) depends on the values introduced in the uncertainty parameter value of D and T in the Pressure transfer function of the Boiler Plant. The solid lines (20 plants) are between the upper and lower specification (dashed lines).

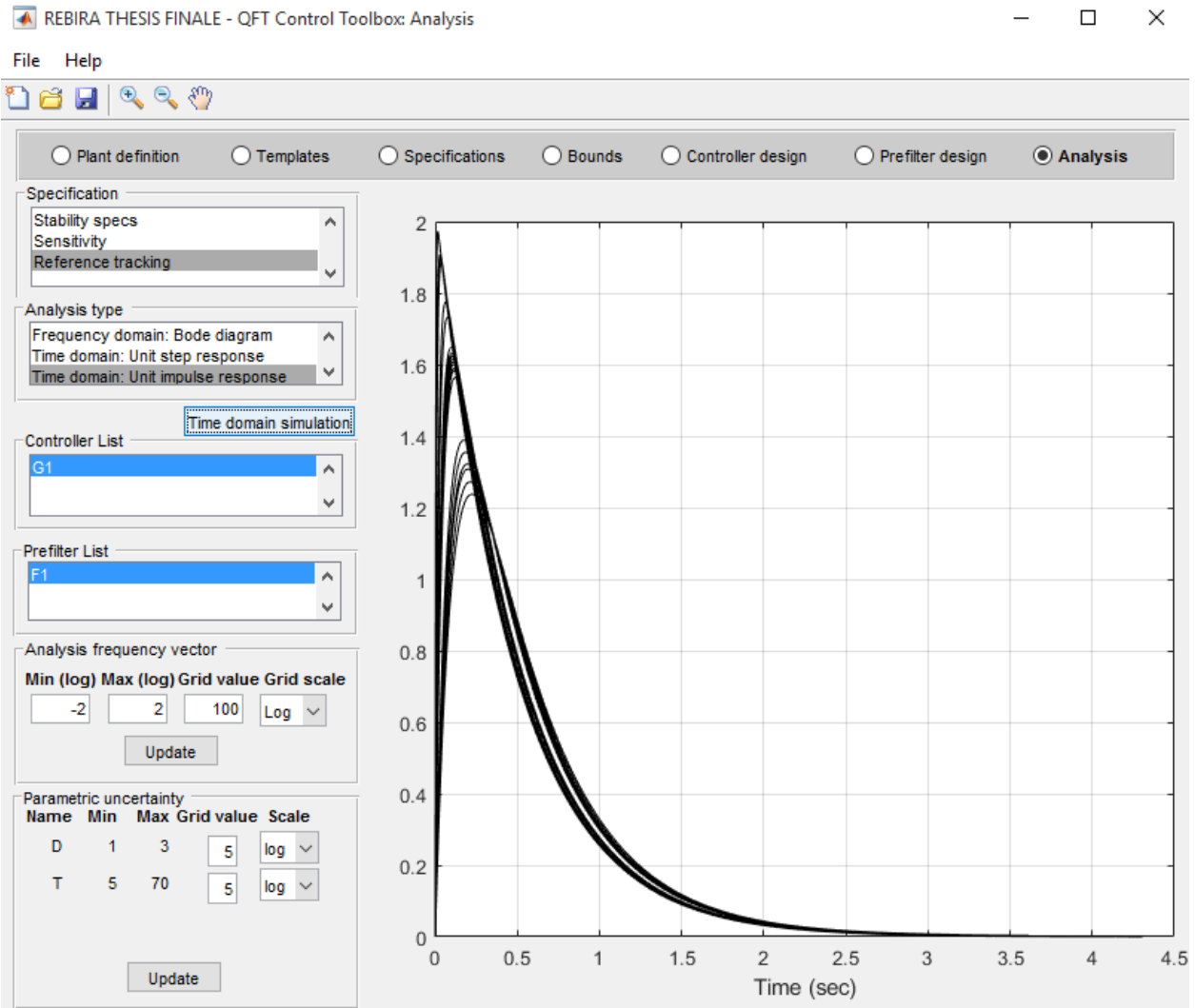


Fig 3.25:- Impulse response of Reference tracking specification

The above fig 3.25 show that, the close loop time domain impulse response of the Plant where the tuned response satisfied within the reference tracking specification.

In this figure we analyzed that the nominal close loop response ($P_o(s)/(1+L_o(s))$), is within upper and lower bound specification for all frequencies where the disturbance $d(s)$ is unitary impulse. The number of plant analyzed (Number of line plotted) depends on the values introduced in the parametric uncertainty panel.

CHAPTER FOUR

SIMULATION STUDIES & ANALYSIS OF RESULTS

4.1. System Data

From our design point of view the data we covered for QFT design of robust control of Industrial Boiler is given as uncertainty parameters with the range values described in section 3.2. In this simulation we carried out the working frequency ranges of $\Omega = [0.01, 0.05, 0.1, 0.5, 1, 2, 3, 4, 5, 10, 20, 30, 40, 50, 100, \text{ and } 500]$. Depending on these parameters we define each control specification to generate controller and pre-filter which meets various specifications our simulations carried out the uncertainty parameter which have the range of $D = [1 \ 3]$ and $T = [5 \ 70]$. This uncertainty is used in all our controller specification calculations and simulations.

4.2. Analysis of Results

The simulations are analyzed from Nichols chart that is used to design for reference tracking since it contains the loci of the complementary sensitivity. The nominal loop gain that satisfies the reference tracking constraint is gain that occurs at the handle of the template when the maximum change of the complementary sensitivity loci intersecting the template is less than or equal to the specifications for some nominal loop phase angle.

An additional constraint may be a 3dB robust stability margin for parametric uncertainties. This means that the system must have a worst-case gain margin of 10.69dB and phase margin of 41.46° . The design uses the Inverse Nichols chart as the domain which is a graph similar to the Nichols chart except that the loci are of the sensitivity function. The gain cross-over frequency ω_{gc} is the frequency where the nominal loop has a gain of 0dB. It is an important frequency because it gives the designer knowledge about the bandwidth of the loop. The robust stability bounds are calculated at frequencies “near” the gain cross-over frequency. In order to obtain the nominal loop robust stability, bound the designer must shift the template around the 3dB locus in the Inverse Nichols chart such that the template contacts the 3dB locus. While performing this operation, the nominal bound is obtained by delineating the points over which the handle falls.

When the bound generation is complete, an intersection of the bounds must be done. The nominal loop must then be shaped to satisfy the graphical constraints. The resulting controller provides the required feedback to reduce the closed-loop uncertainty to levels that allow the final pre-filter shaping to meet the absolute tracking specifications.

The final step is to design the pre-filter. As stated earlier, the feedback controller is responsible for reducing the closed-loop uncertainty and the pre-filter shapes the final reference to output response. As bounds were found for the loop gain that allowed $G(s)$ to be derived, bounds on the pre-filter $F(s)$ can also be found.

Manually, this can be achieved by placing the template on the nominal open-loop gain L_o , with the handle of the template coinciding with the nominal loop transfer functions at the respective template frequency. The pre-filter upper bound at some frequency ω_i is given by

$$|A(j\omega_i)|_{dB} - |T_{max}(j\omega_i)|_{dB} \quad (4.1)$$

Where $|T_{max}(j\omega_i)|$ is the maximum value of the intersection of the complementary sensitivity loci with the plant template at ω_i . Similarly, the lower bound is given by:-

$$|B(j\omega_i)|_{dB} - |T_{min}(j\omega_i)|_{dB} \quad (4.2)$$

The pre-filter $F(s)$ is designed to be minimum phase and to lie within these bounds. QFT generates controllers that are robust stable and easy to implement, making it the perfect choice for industrial applications. The simulations verified that the designed controllers do indeed perform according to the control and performance specifications.

The design required the use of analytical linear time invariant models from which the frequency response had to be calculated.

From the above controller design we generate the close loop step response with nominal plant and $G(s)$.

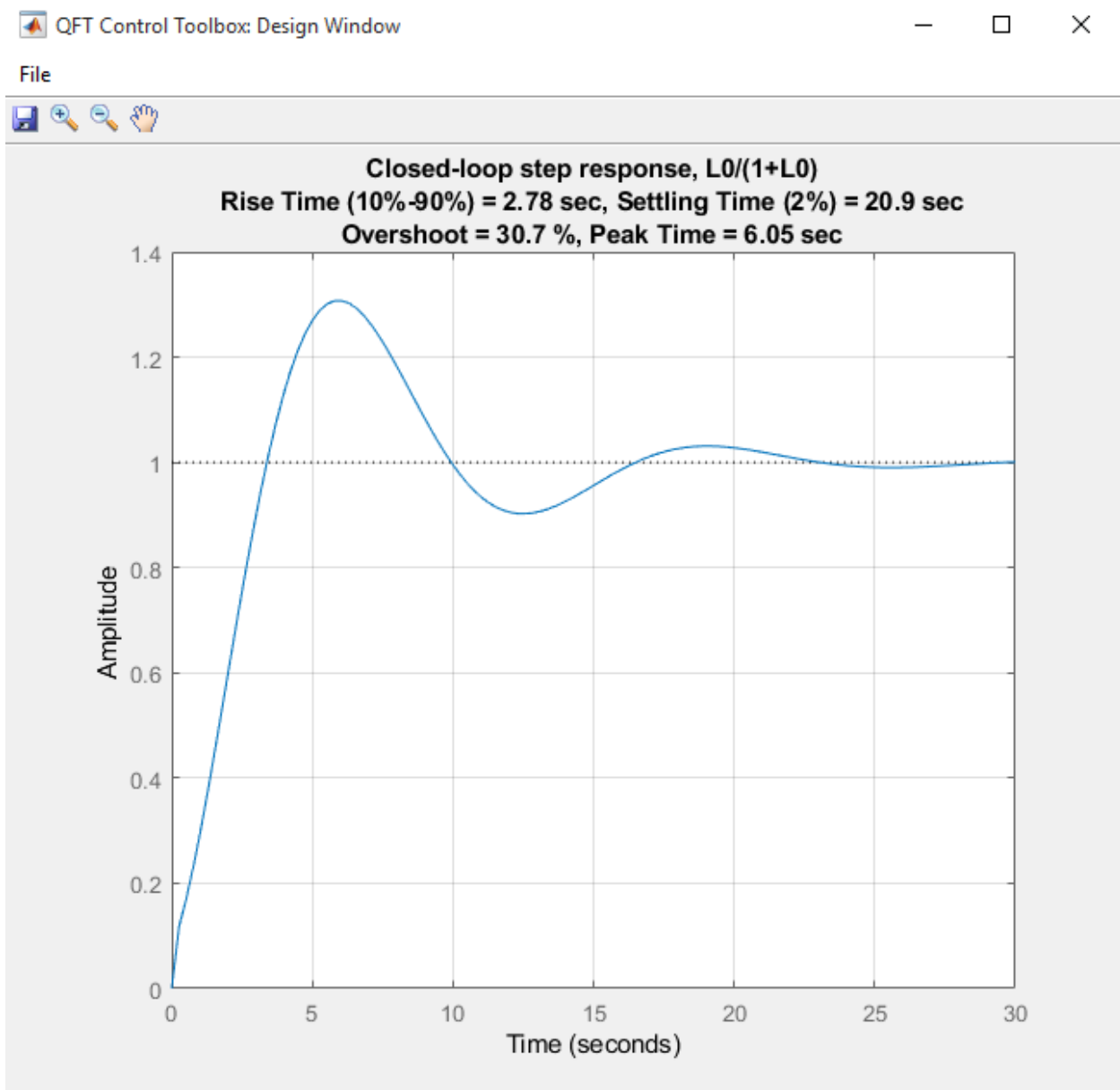


Fig 4.1: - Response Design of $G(s)$, derivative time $T_d=0.45$ seconds.

From above close loop step response with nominal plant and $G(s)$ we conclude the time domain specification which has the value of rise time of 2.78 seconds, Settling Time of 20.9 seconds, Peak Time of 6.05 seconds and Overshoot value of 30.7%.

These values told us the Plant still need some tuning of PID parameter to decrease the overshoot response.

So we have to increase derivative value to decrease the overshoot response of close loop.

No	Parameter	Value
1	Rise time	2.78sec
2	Settling time	20.9sec
3	Overshoot	30.7%
4	Peak Time	6.05sec

Table 4.1:- Response parameters of Closed loop at initial controller with $T_d=0.45$ sec

So far, the response time of the control system has remained fixed while you have changed the transient-behavior coefficient. These operations are equivalent to fixing the bandwidth and varying the target minimum phase margin of the system. If you want to fix both the bandwidth and target phase margin, you can still change the balance between reference tracking and disturbance rejection.

As in the PID controller case, the initial PID design balances reference tracking and disturbance rejection. In this case as well, the controller yields some overshoot in the reference-tracking response, and suppresses the input disturbance with a longer settling time.

Reducing the effect of parameter uncertainty by shaping the open-loop frequency responses so that the Bode plots of the J closed-loop systems fall between the boundaries $A(j\omega)$ and $B(j\omega)$, while simultaneously satisfying all performance specifications.

To improve the response we have to add derivative action to the controller gives PID Tuner more freedom to achieve adequate phase margin with the desired response speed.

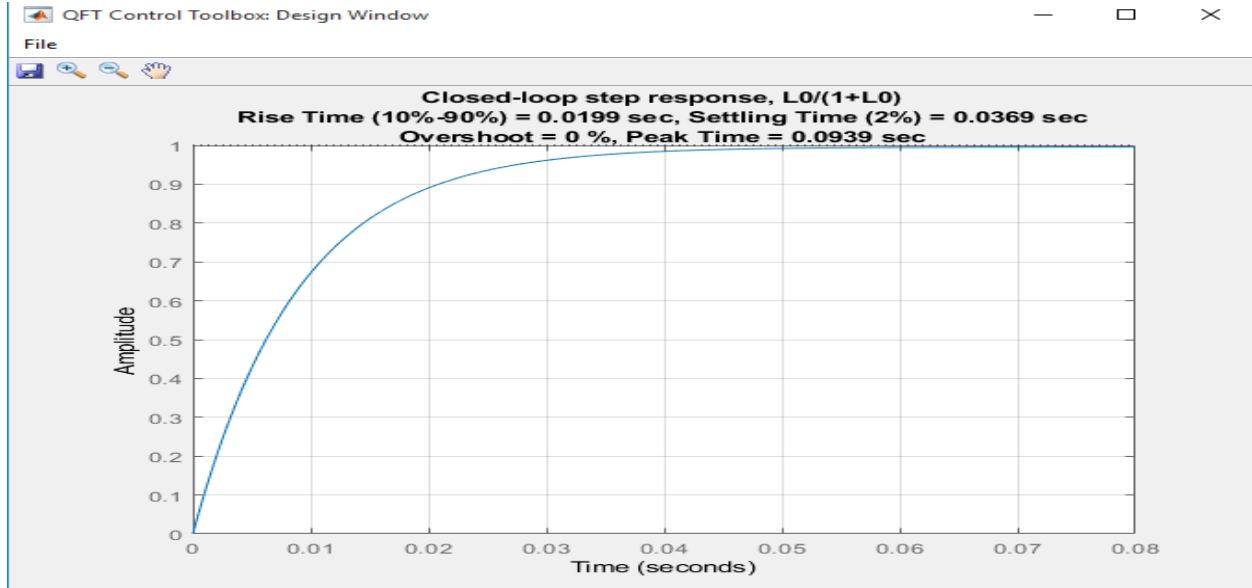


Fig 4.2: - Close loop step response of the plant at $T_d \approx 20$

Specifically the gained parameters are as follows

No	Parameter	Value
1	Rise time	0.0199sec
2	Settling time	0.0369sec
3	Overshoot	0%
4	Peak Time	0.0939sec

Table 4.2:- Response of QFT controller at Derivative time $T_d=20$

From above controller the closed loop step response is generated at a given derivative time when it decreases the overshoot response to get the desired close loop response. The responses with this balanced controller are now displayed as the Block response, and the controller tuned with a focus reference-tracking is the Tuned response. The plots show that the resulting controller tracks the reference input with considerably less overshoot and a faster settling time than the balanced controller design.

A PID controller is capable of fast disturbance rejection without significant increase of overshoot in set-point tracking and also they are useful to mitigate the influence of changes in the reference signal on the control signal.

The aim is to adjust the three gain factors (proportional, integral and derivative control) according to the dynamics of the plant, so that both the degree of error reduction (if not error elimination) and the dynamic response will be acceptable. In the frequency domain the PID controller introduces phase lag to the nominal open loop (reaching almost -90°) at low frequencies due to the dominance of the integral term and phase lead (reaching almost 90°) at high frequencies due to the dominance of the derivative term. In intermediate frequencies the introduced phase is either negative (due to integral term) or positive (due to derivative term), the proportional term having 0° phase introduces only gain. It is then clear that the resulting PID controller achieves good approximation at low frequencies, especially the integral action, so we can expect that the resulting PID controller will retain the disturbance rejection performance of the high-order controller.

After the bounds on the loop gain, $L(j\omega)$, are determined, then shape the compensator or controller so that $L(j\omega_i)$ fulfills the bounds. In this case we have done by adding real pole 2.0845 of filter to $G(j\omega)$ so that $L(j\omega)$ lies above $B(\omega_i, \phi)$. $L(j\omega)$ must also satisfy the stability bounds. The optimum design of the compensator is obtained when the loop gain value of 0.9784, $L(j\omega)$, lies on the bounds at low frequencies and decreases in magnitude as fast as possible in the high frequency range, i.e. $L(j\omega)$ has minimum gain and bandwidth.

According to the above simulation, robust stability, nominal performance and robust performance are satisfied and also for tracking performance for an arbitrary reference signal. But, what is the most important and brilliant difference between classical controller and QFT ones is control effort signal which is in an optimal range by controller designed in this thesis. It means in order to have a specified pressure we will need less fuel and energy consumption.

As mentioned, QFT is used to design a robust controller for pressure of the boiler. After modeling parametric uncertainties, parametric uncertainties are represented used to analyze nominal performance, robust stability and robust performance of the uncertain system. By using Quantitative feedback theory QFT, an optimal robust controller is designed which results in robust performance of the system against parametric uncertainties. The basic design steps can be summarized as linearization of the boiler pressure dynamics, design of suitable robust disturbance rejection bounds by minimization of Sensitivity Function and nonlinear simulation.

My Thesis indicates that an increase of accuracy in tracking problem has a direct relationship with reduction of the cross-coupling effect between component of boiler by designing suitable disturbance rejection bounds, reduction of settling time in tracking bounds for associated linear system and improvement of associated linear uncertain system modeling. The result is a robust design which is insensitive to plant variation.

- ✓ There is one design for the full envelope that is there is no need to verify plant's inside templates.
- ✓ Any design limitations are apparent up front.
- ✓ In comparison to other multivariable design techniques there is less development time for a full envelope design.
- ✓ One can determine what specifications are achievable early in the design process.
- ✓ One can redesign for changes in the specifications quickly.

So from our controller we generated the system stability of Nominal close loop calculated from the specification will be as follows.

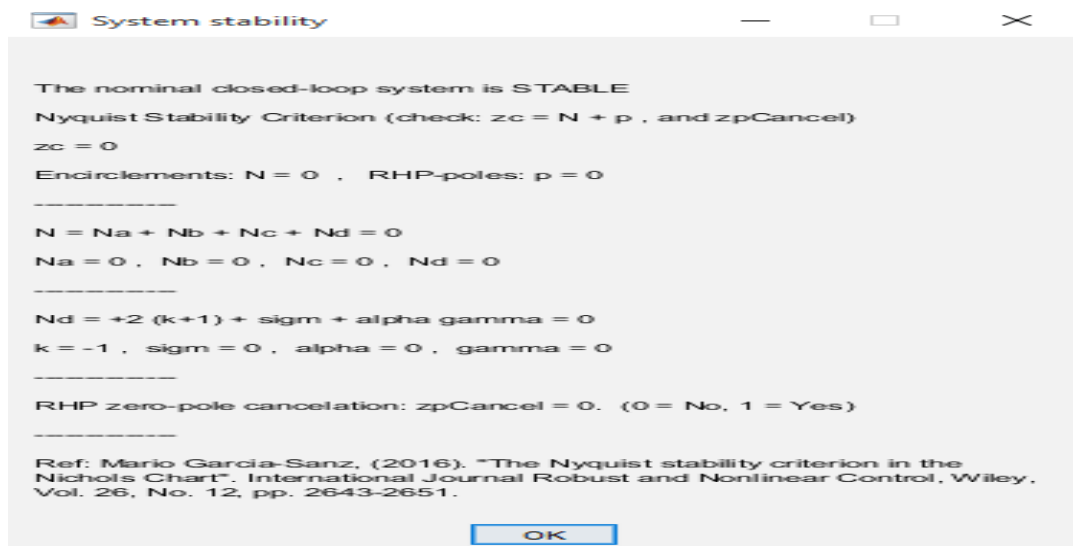


Fig 4.3: - System stability of the designed controller

From Fig 4.1 we generate the Reference tracking specification by Increasing the transient-behavior coefficient to 0.70 nearly eliminates the overshoot, but results in sluggish disturbance rejection.

You can try moving the Transient behavior slider until you find a suitable balance between reference tracking and disturbance rejection for your application. How much the slider affects the balance depends on the plant model.

From fig 3.20 of pre-filter design we analyse the pre-filter calculates automatically the worst upper and lower close loop response case of $L_o(s)F(s)/[1+L_o(s)]$ over the plant uncertainty. This case should be between the upper and lower reference tracking function to meet the specifications. From the figure the dashed blue line represents the upper and lower tracking specification and the dashed black line represents the maximum and minimum case of $L_o(s)F(s)/[1+L_o(s)]$ of close loop response. In the first part of this study we focused a Robust QFT toolbox was used to design the controller with a set of specifications.

✚ Robust stability is enforced by ensuring that:

- No unstable pole-zero cancellations occur b/n the plant and the controller (for every $p \in \mathbb{CP}$)
- The nominal open-loop frequency response $L_o(j\omega) = G(p_o; j\omega)P(j\omega)$ (defined for any $p_o \in \mathbb{CP}$) does not cross the -1 point (i.e. the (-180; 0)degrees point on Nichol's chart) and makes a total number of (anti-clockwise) encirclements around it equal to the number of unstable poles of $L_o(s) = G(p_o; s)K(s)$, and

- That no (perturbed) open-loop response crosses the -1 point, i.e.

$$-1 \notin \bigcup G(k\omega)P(\omega) \text{ where } \omega \in \mathbb{R} \quad (4.24)$$

During simulation we found out that varying the Fuel flow and Air flow would help us meet our specifications The robust performance control is essentially a tradeoff between Minimizing $\|W_p S_o\|$ i.e. Performance and $\|W_u T_o\|$ i.e. Robustness where the value of $|W_p|$ and $|W_m|$ are less than one at any frequency which we gained at equation 3.16 and 3.17

In the frequency domain analysis the dashed lines represents the desired specification $\delta_i(\omega)$ and the solid line represents the worst case of control systems over the plant uncertainty at each frequency. From the fig 3.21 the solid line is not a transfer function, but it is the worst case among all the transfer function over the plant uncertainty at every frequency.

Generally, from my thesis finally we analyze that the design feedback $C(i\omega)$ is to reduce effects of uncertainty and disturbances, considering structure of process variations and pre-filter $F(i\omega)$ to shape reference response.

CHAPTER FIVE

CONCLUSIONS, RECOMMENDATIONS AND FUTURE WORKS

5.1. Conclusions

Due to the presence of uncertainty in the pressure dynamic of the boiler, the application of robust control method for achieving high accuracy in tracking is inevitable. The dynamic model of the boiler may associate with parametric uncertainties. As mentioned, QFT is used to design a robust controller for pressure of the boiler. After modeling parametric uncertainties, parametric uncertainties are represented used to analyze nominal performance, robust stability and robust performance of the uncertain system.

The basic design steps can be summarized as linearization of the boiler pressure dynamics, design of suitable robust disturbance rejection bounds by minimization of Sensitivity Function and LTI simulation. After the bounds on the loop gain, $L(j\omega)$, are determined, then shape the compensator or controller so that $L(j\omega_i)$ fulfills the bounds. In this case we have done by adding real pole 2.0845 of filter to $G(j\omega)$ so that $L(j\omega)$ lies above $B(\omega_i, \phi)$. $L(j\omega)$ must also satisfy the stability bounds. The optimum design of the compensator is obtained when the loop gain value of 0.9784, $L(j\omega)$, lies on the bounds at low frequencies and decreases in magnitude as fast as possible in the high frequency range, i.e. $L(j\omega)$ has minimum gain 0.45 and small bandwidth .

By applying QFT robust control, system tracks the desire reference inputs in a less time and with smoother time responses. It is observed from the simulation results that the average overshoot is 0%, rising time is 0.0199 seconds and the settling time is 0.0369 seconds and Peak time is 0.0939 seconds with QFT controller. This work covers all the steps followed to use the Quantitative Feedback Theory (QFT) to study the behavior of a steam boiler system in an industrial environment. The obtained results are exposed and explained with the aim of describing robust stability and performance for the control of the plant. Both optimal robust controllers guarantee robust performance of the system against the uncertainties and result in desired time responses of the output variables.

5.2. Recommendations

Even though assumptions and approximations are made to linearize the mathematical equations describing the air, fuel flow and pressure controller are not satisfactory to the higher degree of accuracy condition. Therefore the robust control can be optimized with design MIMO systems of Industrial Boiler considering all inputs and output by extending to cascaded loop systems and multi-loop systems using a sequential loop closure approach to gain the exact experimental control systems for further improvements in the performance of robust and performance controller for boiler combustion control. This is already validated by simulation only. So any one can take this design and it can be implemented by prototypes.

5.3. Suggestions for future works

As continuation of this thesis work, it is suggested to carry out the following:

1. The performance of pressure of boiler system can be examined under parameter variation, like as that of air flow and fuel flow. So anyone can consider all parts of Boiler combustion elements.
2. In this study, the dynamics of the system doesn't consider the noise like temperature. So anyone can develop systems by considering Noise by temperature.
3. In addition to this, by taking only combustion furnace as on design, it may risk to get overall boiler pressure efficiency. So anyone design MIMO systems of Industrial Boiler considering all inputs and output by extending to cascaded loop systems and multi-loop systems using a sequential loop closure approach to gain the exact experimental
4. And finally, one anyone can implement QFT based robust control of industrial Boiler using hardware.

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