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QUANTITATIVE FEEDBACK THEORY-BASED ROBUST CONTROL FOR A SPINDLE OF LATHE MACHINE

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Abstract- This paper presents theoritical and experimental design of a robust control based on quantitative feedback theory (QFT) approach to control deepness variations on cutting process for a spindle of lathe machine. The dynamics of a spindle is uncertain and varying due to deepness variation on cutting process. Practical design steps are explained in which QFT based conditions are assembled to obtain a compensator and pre-filter gain to control a spindle. The robust controller show an advantage control the system under various cutting conditions. The performances of the proposed controller are evaluated in terms of input tracking capability of the spindle responses and speed responses of a spindle incorporating deepness on cutting process. Experimental results show that QFT based robust control provides the improvement of robustness and performances.

Index terms: experimental, lathe, QFT, spindle, robust control.

I. INTRODUCTION

Efficiency improvement is the evidences of technology development in manufacturing processes. The increasing accuracy, precision and reliability of the process results are extremely challenge. The development and manufacturing of highly dynamic machine to fill these needs, the researchers performed a variety of innovations especially in manufacturing. The performances of manufacturing scopes in the future require increasing demand of high quality precision products with highly quality and quantity. These supports the desire for the design of suitable control techniques which are stability boundary for dynamics characteristics. The main goal of modelling for a spindle of lathe machine is to achieve an accurate model representing the actual system behaviour. It is important to recognise the moving nature and dynamic characteristics of the system and construct a suitable mathematical framework. The lathe machine structures consist of spindle system which operates a relating to a quality of final products and the overall productivity and efficiency of the machine tool itself. The spindle of a CNC machine was rotated by the main motor, holds the cutting tool, which cuts the work piece, the cutting forces are generated which effects the spindle accuracy directly [1]. Modelling of a lathe spindle has been widely established.

Martin and Ebrahimi [2] investigated the model of milling action which investigated the relationship of the whole machine tool dynamics which close the loop between a spindle and axis feed subsystem. The strategy for precision turning of shafts on CNC centre lathes has been presented [3]. Shivakumar et.al [4] presented a software (Ansys) to analysis the model of lathe spindle was supported by two bearings with different spans and using alloy steel material. Safety factor, equivalent alternating stress and life of a spindle was found by strain and stress analysis and which a good agreement with the analytical results [1]. Reddy and Sharan [5] also explored the finite element method to model and design of a lathe spindle for analysing static and dynamic charateristics. Lin et.al [6] also conducted a study of dynamic models and design of spindle bearing systems.

The research on feeding processes scope on a lathe machine with adaptive method was performed [7]. Robust system utilization as a controller in the area of dynamic payload changes was investigated. Furthermore Takehito [8] investigated state predictive LQG controllers to achieve the stability and better performances of the vehicle over the computer network. Gao and Liu [9] developed method constructs a classifier by sampling positive and negative samples and then to improve the robustness of these tracking methods, a novel object tracking algorithm is proposed based on support vector machine and weighted multi-sample sampling method.

The dynamics load of a spindle especially on the feeding processes, these motivated to conduct inovations with a robust system. Therefore on this study arranged a spindle speed control design with a robust control system method using QFT method. Tu et. al [10] explained and implemented the robust control for driving a mill on the highest possible feedrate without damaging the cutter and spindle of Boeing which has significantly reduced spindle failures in their extremely demanding high speed machining production lines. Van Dijk et.al [11] explored a robust control approach using synthesis to solve the most important process parameters (depth of cut and spindle speed) were treated as uncertainties to guarantee the robust stability.

This paper presents the design and development of a robust controller based on QFT approach for a spindle of lathe machine. It is found that the QFT approach has not been explored for control of a spindle of lathe machine where the system dynamics have uncertainties due to the variation of cutting process.

Using the robust controller to identify a compensator and pre-filter gains can be used for all cutting loads with satisfactory responses. To cast this control design problem into the QFT frame-work, the transfer functions of the system with various cutting loads are obtained by carrying out system identification. Subsequently, the dynamic model is represented into formulation which leads to the formulation of system requirement into QFTs representation that can accommodate the convex model. A set of robust compensator and pre-filter gains are then obtained by solving the QFTs with desired specifications. For performance assessment, QFT controller for control of a spindle in terms of spindle velocity and robustness to cutting load variations. Experimental results show that better robustness and system performance are achieved with QFT-compensator pre-filter controller despite using a single set of compensator and pre-filter gains. The rest of the paper is structured as follows: Section 2 provides a brief description of a spindle system considered in this study. Section 3 describes the modelling process of the system based on system identification. Development of robust control based QFT approach is presented in Section 4. Section 5 presents the controller implementation. Experimental results are presented in Section 6 and the paper is concluded in Section 7.

II. THE SPINDLE OF LATHE MACHINE

A typical representation for a spindle of lathe machine is shown in Figure 1. The rig consists of three main parts: a spindle, sensors and a processor. The spindle is rotated by the main motor, holds the cutting tool, which cuts the work piece, then the cutting forces are generated which effects the spindle accuracy directly.

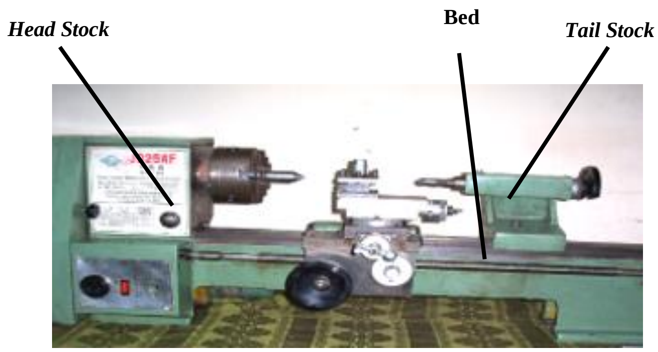


Figure. 1 The Experimental Setup of a Spindle [13]

The plant is focused on a spindle of lathe machine. The spindle consists of DC motor with input of 38 volt and 0.48 kw. Maximum angular velocity with route per minute is 800 rpm. The input and output of the lathe spindle in this study are is voltage and angular velocity (rpm) respectively. The sensor is a shaft encoder at the motor. The setting point of a spindle is lowest position related to a maximum of specimens diameter based on a lathe performance. The position sensor can be read by analog to digital converter (ADC), hencefort the data can be read by computer. The computer will give the operation to drive a motor DC driver.

This study investigates the system identification of a lathe spindle incorporating deepness of cutting process. The performances of the system identification are evaluated in terms with and without cutting process. The method constructs the system identification for a lathe spindle selectively. In this section, it is indicated that the identification system is consisted of the program based on Matlab. The interface of the identification system application uses identification tools on Matlab.

Based on the deepness of cutting process with single input single output (SISO) of a spindle used pseudo random binary sequence (PRBS) method. PRBS signal was generated by shift register with feedback both simulation and experimental work. The maximum length of sequence is 2N-1 where N is the number of cells in the shift register. For this study used cells N number of 10 cells, thus generating PRBS signal can be obtained of 1023 [12].

III. DYNAMIC MODEL OF A SPINDLE

Modelling of a spindle in this study based on the system identification as detailed in M. Khairudin [13]. Due to high non-linearity of the dynamic model, formulation of a robust control algorithm is difficult, and thus a simple and practical method is required. The transfer function model of a spindle without cutting process can be written as:

$$G_1(s) = \frac{9.925\ s^2 + 1794\ s + 1.424e005}{s^3 + 137.5\ s^2 + 2.487e004\ s + 1.648e005} \tag{1}$$

Other wise for the spindle with depth of cut of 0.2 mm can be found as:

$$G_2(s) = \frac{6.191\ s^2 + 1049\ s + 7.6e004}{s^3 + 90.74\ s^2 + 1.908e004\ s + 9.356e004} \tag{2}$$

and the transfer function of the spindle with depth of cut of 0.5 mm can be found as:

$$G_3(s) = \frac{6.459\ s^2 + 1158\ s + 8.873e004}{s^3 + 119.3\ s^2 + 2.184e004\ s + 1.163e005} \tag{3}$$

The general form of the third order system will be used in this study can be written as:

$$G_3(s) = \frac{b_5 s^4 + b_4 s^3 + b_3 s^2 + b_2 s + b_1}{a_6 s^5 + a_5 s^4 + a_4 s^3 + a_3 s^2 + a_2 s + a_1} \tag{4}$$

The responses of the motor spindle in time domain can be found at Figure 2. In this study, aluminum bar will be the specimens were cut on lathe with initial diameter of 20 mm. To investigate the effects of deep of cutting on the dynamic characteristics of the system, a lathe spindle with various deep of cutting was examined. Figure 2 shows experimental results of spindle speed responses of the lathe spindle without cutting and with depth of cut of 0.2 and 0.5 respectively.

It is noted that the steady-state angular positions for spindle speed decrease with increasing depth of cut. The results also show that the transient responses of the system are affected by the variations of depth of cut. It is noted with increasing depth of cut, the system exhibits lower steady state.

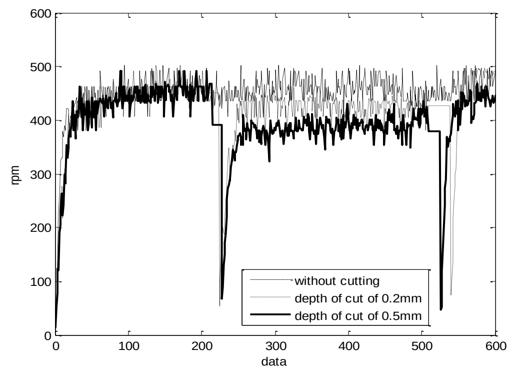


Figure. 2 Experimental measurement of spindle speed

IV. DESIGN OF QFT BASED ROBUST CONTOL OF A SPINDLE

The QFT approach is considered as a basis for determining the compensator and pre-filter gains since this approach can provide a high degree of robustness and design flexibility as detailed in M. Khairudin [16]. To obtain the parameter uncertainties of spindle model in the form of a SISO system, the first step is collecting all the plant inputs and outputs of j and i respectively for the variation of deepness of cutting process based on spindle conditions. Consider an uncertain third-order system of a spindle as

$$G(s) = \frac{b_3 s^2 + b_2 s + b_1}{a_4 s^3 + a_3 s^2 + a_2 s + a_1} \qquad (5)$$

which the parameter uncertainties are:

$$6.191 \le b_3 \le 9.925$$

$$1049 \le b_2 \le 1794$$

$$7.6e004 \le b_1 \le 1.424e005$$

$$90.74 \le a_3 \le 137.5$$

$$1.908e004 \le a_2 \le 2.487e004$$

$$9.356e004 \leq a_1 \leq 1.648e005 \qquad (6)$$

Tracking specifications desired, the spindle to be desired that a speed N (t) to the input D (t) has the characteristic response time as

$$5 \leq t_R \leq 30\ s$$

$$t_p \leq 20\ s$$

$$t_s \leq 30\ s, \text{ with criteria } of\ \ 2\ \% \qquad (7)$$

$M_p \leq 0.3$, with maximum allowable overshoot is 20% of steady state. In order to obtain a transfer function for the upper bound, which has a response time specification as:

$$t_R \geq 5\ s$$

$$t_p \leq 20\ s$$

$$t_s \leq 30\ s, \text{ with criteria of } \ 2\ \%$$

$$M_p \leq 0.3$$

$$t_s = 4T = \frac{4}{\sigma} = \frac{4}{\zeta\omega_n} \rightarrow 30 = \frac{4}{\sigma} \rightarrow \sigma = 0.1333\ /\ s \qquad (8)$$

With maximum overshoot M_p as

$$M_p = e^{-\left(\frac{\sigma}{\omega_d}\right)\pi} \rightarrow 0.3 = e^{-\left(\frac{0,4186}{\omega_d}\right)} \qquad (9)$$

$$ln(\ 0.3\ ) = -\left(\frac{0.4186}{\omega_d}\right) \rightarrow \omega_d = \frac{0.4186}{1.20397} = 0.3477\ \ \frac{rad}{s}$$

$$Tg\ \phi = \left(\frac{\omega_d}{\sigma}\right) \rightarrow \phi = tan^{-1}\left(\frac{0.3477}{0.1333}\right) \rightarrow \phi = 1.204\ \ rad \qquad (10)$$

The next step is to test the rise time t_r :

$$t_R = \frac{\pi - \phi}{\omega_d} = \frac{3.14 - 1.204}{0.477} = 5.568\ \ s \quad \text{(is covered)} \qquad (11)$$

To test a peak time t_p :

$$t_p = \frac{\pi}{\omega_d} \rightarrow t_p = \frac{3.14}{0.3477} \rightarrow t_p = 9.0308 \ s \text{ (is covered)} \quad (12)$$

To reach a transfer function, it needs to calculate ζ and ω_n

$$\sigma = \zeta\omega_n \rightarrow \omega_n = \frac{\sigma}{\zeta} = \frac{0.1333}{\zeta} \quad (13)$$

$$\omega_d = \omega_n \sqrt{1-\zeta^2} \rightarrow \omega_d^{\ 2} = \omega_n^{\ 2}\left[1-\zeta^2\right] \quad (14)$$

$$0.12089 = \frac{0.017769}{\zeta^2}\left[1-\zeta^2\right] \rightarrow \zeta = 0.35797 \rightarrow$$

$$\omega_n = 0.37237 \ \frac{rad}{s}$$

So the transfer function can be obtained as

$$G(s) = \frac{\omega_n^{\ 2}}{s^2 + 2\zeta\omega_n s + \omega_n^{\ 2}} \quad (15)$$

In order to obtain the transfer function for the upper bound as

$$G(s) = \frac{0.13866}{s^2 + 0.2666 \ s + 0.13866} \quad (16)$$

Using the same technique, it can be obtained the transfer function for lower bound.

$$G(s) = \frac{0.0287}{s^2 + 0.2666 \ s + 0.0287}$$

Step input response for the lower and upper bound can be found at Figure 3.

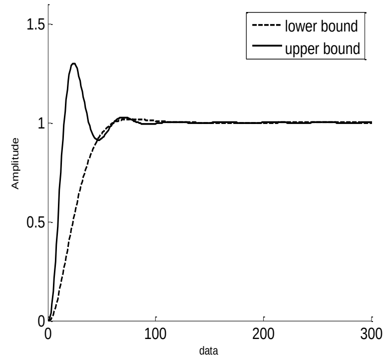


Figure. 3 The lower and Upper bound Step Response

Figure 4 presents the frequency responses of several transfer functions that the location can not covered by the desired range. Design compensator and pre-filter are intended to reach the all frequency responses of the plant that located in the range. The compensator and pre-filter have been designed.

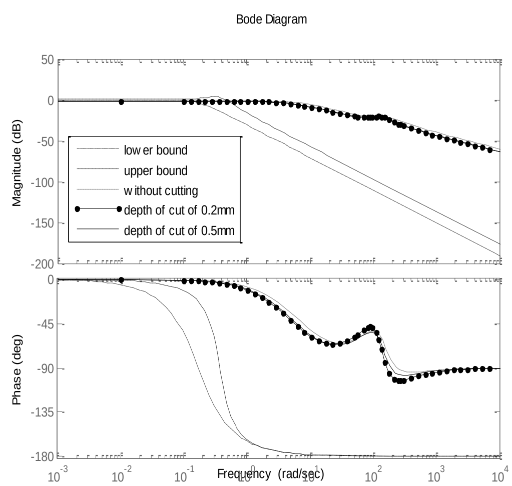


Figure. 4 Bodeplot for uncertainty of dynamic system

Prior to design a template is the first step to generate a template of the parameter uncertainty of plant that is obtained through the system identification process. The next process is the selection of frequency (array frequency), the selected frequency array will be used to generate a template and bound calculation. It is based on a template form that bound will change if these templates form also changed. Array Frequency ~~are~~ selected from the smallest to largest frequency. In this study, the array frequency for the plant $G(s)$ is [0.001; 0.01; 0.1; 1; 1.5; 20; 100] rad/s.

The next step is to determine a bound or constraint in the design of the compensator and pre-filter. A bound is an implementation of the system specification in the frequency domain. Robust margin bound specified of the plant $G(s)$ can be found as

$$\left|C\right| < 1.2 \ \ which \quad \omega > 0$$

By taking μ = 1.2, this shows that the desired lower gain margin can be written as

$$1 + \frac{1}{\mu} = 1 + \frac{1}{1.2} = 1.833 \ \ dB$$

and the desired lower phase margin can be obtained as

$$\begin{aligned} &= 180^{0} - \theta \\ &= 180^{0} - arcCos\left[\frac{0.5}{\mu^{2}} - 1\right] \\ &= 49.25^{0} \end{aligned}$$

Lower gain margin of 1.833 dB is a limit of boundary critical point q of (0 dB, -1800). For this design, these areas in the curve of 1.833 dB should not be crossed by the curve for the nominal plant to be designed. Lower phase margin of 49.250 shows a minimum phase margin is allowed. To design the compensator, generated bound is the worst case bound, which the worst constraints can be generated. The obtained design is minimal design that will meet all the criteria specified bound. Through designing with the bound, it will be covered automatically as well for all bound. In other way, the plant $G(s)$ for the frequency range of [0.001; 0.01; 0.1; 1; 1.5; 20; 100] rad / s, the generated bound is an intersection between a robust tracking bound and robust margin bound. To cover these bound through generating all the existing bound for the same frequency. The initial design of the compensator or controller through designing the initial value of $G(s)$ = 1. The curve is a curve of nominal loop $L_0(s) = P_0(s) * G(s) * H(s)$ with a value of $H(s)$ of 1. The components were added to $L_0(s)$ to obtain the results of the compensator design $K(s)$ and pre-filter $F(s)$. The final compensator can be obtained as :

$$K(s) = \frac{0.3606\, s^3 + 421.5986\, s^2 + 9179.618\, s + 21149.1079}{s^3 + 145.677\, s^2 + 127.02339\, s + 0.28127}$$

The components were added to $L_0(s)$ to obtain the results of the compensator design $K(s)$ and pre-filter $F(s)$. Using the same process, the final pre-filter can be obtained as :

$$F(s) = \frac{1.5785\,(s + 17.23)}{s^2 + 13.303\, s + 27.1979}$$

V. CONTROLLER IMPLEMENTATION

The results of compensator and pre-filter were obtained from the spindle system, assuming a system behavior and changes in spindle model parameter uncertainty caused by the changes in spindle conditions, which included without load, the load of cutting depth of 0.2 mm, and 0.5 load mm. Figure 5 presents the block diagram of control system.

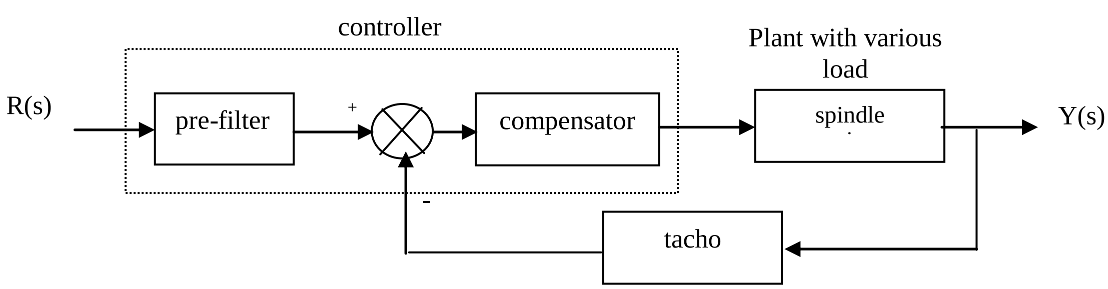


Figure 5. The block diagram of control system

Figures 6 shows most of the spindle's responses are still below the desired gain margin, although the results of the design for the spindle $G(s)$ there is a surge gain and is only valid for a certain frequency range. It is noted that the desired design still cover the specifications robust margins. Analysis of the design results for the pre-filter $F(s)$ shown in Figure 7. It appears that for the range of frequencies below of 0.2 rad / sec has the intensities close to 0 db, but for above of 0.2 rad / sec occur variations intensities. Frequency range between 0.1 to 10 rad / sec still covered under the design specifications. Generally, it is noted that the frequency response design results almost mostly located in the upper and lower limits of the desired specifications, so that it meets the criteria of a robust tracking bound though this only takes place in a particular frequency range.

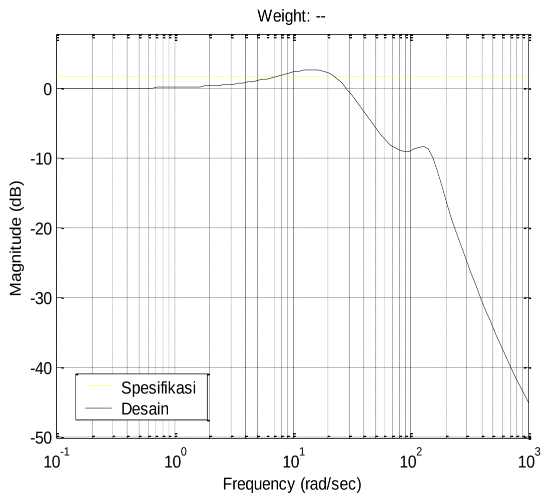


Figure 6. Robust gain margin chek $G(s)$

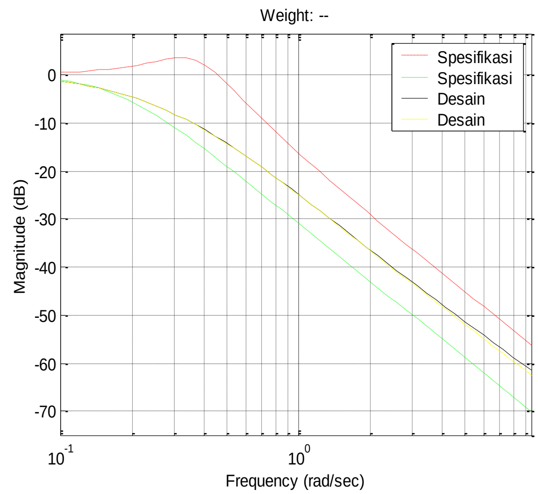


Figure 7. Robust tracking check $F(s)$

Table I presents the compensator check without pre-filter to determine whether the results of the controller design can control the spindle's response for the input. It is noted with increasing cutting depth, the steady state of spindle speed on the same speed performances but for settling time and overshoot percentages will increase for without load, load of 0.2 mm and 0.5 mm respectively.

Tabel I. Spindle response with compensator K(s) without pre-filter F(s)

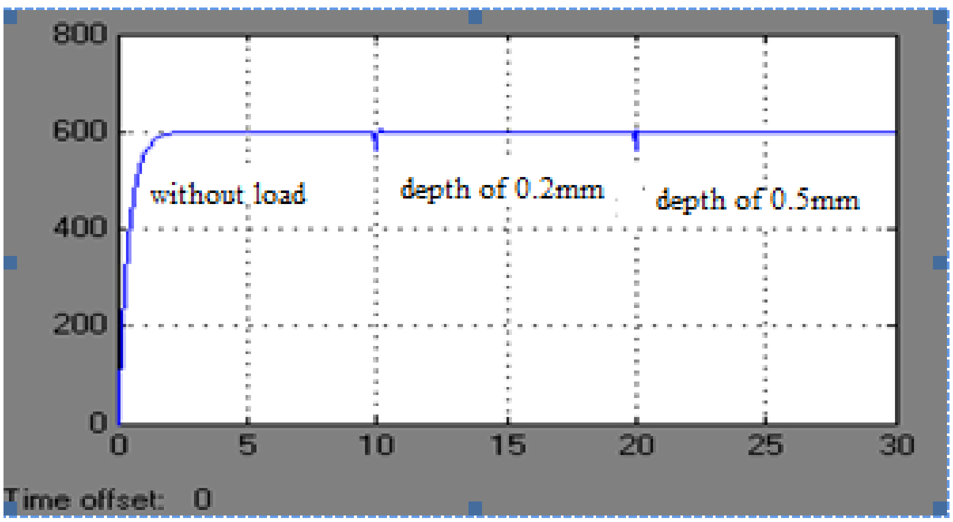
Cutting depth	Steady state (rpm)	Overshoot (%)	Settling time (ms)
Without load	600	10.50	111.2
0.2 mm	600	13.13	117.1
0.5 mm	600	13.75	133.8

In this study to obtain the tracking input conducted with inserting a pre-filter. It will be seen the similar responses though with incorporating cutting depth of a spindle. The final step for designing a QFT approach is joining the compensator and pre-filter. Tabel II describes the transient responses of the spindle of lathe machine using compensator and pre-filter. It is noted with using compensator and pre-filter will not impact to controller performances though increasing the cutting depth of spindle.

Tabel II. Spindle response with compensator $K(s)$ and pre-filter $F(s)$

Cutting depth	Steady state (rpm)	Overshoot (%)	Settling time (s)
Without load	600	0	2393.2
0.2 mm	600	0	2393.5
0.5 mm	600	0	2393.7

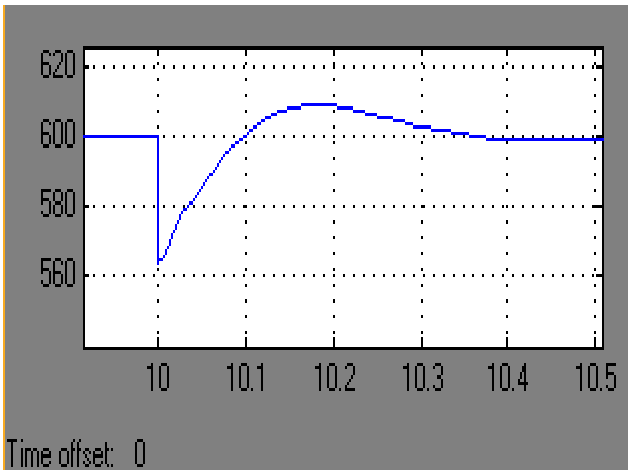
For robustness from the disturbances, the simulation conducted when the system performed on the steady state, then given the external disturbances. The disturbance in this case is the various of depth cutting to spacement which held by the spindle. Figure 8 shows the robustness performance of the spindle with the various of depth cutting. The change of various depth of cutting conducted of 10 s periodicly.



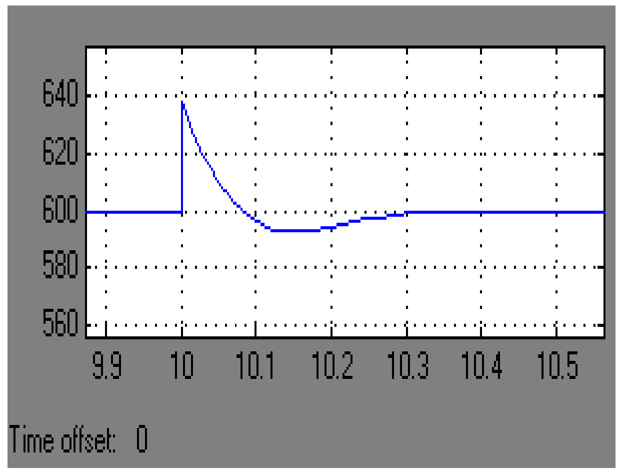
(x axis=time (s), y axis=speed (rpm))

Figure 8. The robustness performance of the spindle

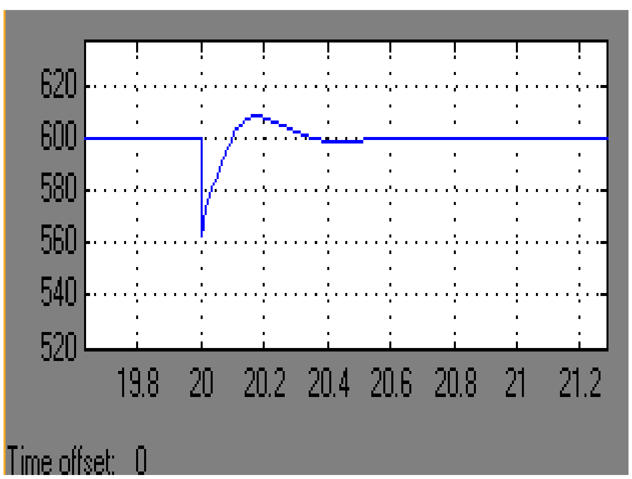
To clarify the processes of moving the cutting depth in Figure 8 can be seen in Figure 9. Figure 9 shows that the controller can able to control the spindle speed to changes in parameters and to obtain robustness. The first moving of cutting depth changes from without load move to cutting depth of 0.2 mm. The second moving followed by cutting depth of 0.5 mm. The occurrence of oscillation at turn of the first load (ranging from 564.05 to 608.5) for 0.565 seconds. While at turn of the second load oscillation occurs (from 564.03 to 608.5) for 0.565 seconds. It is due to the switch.



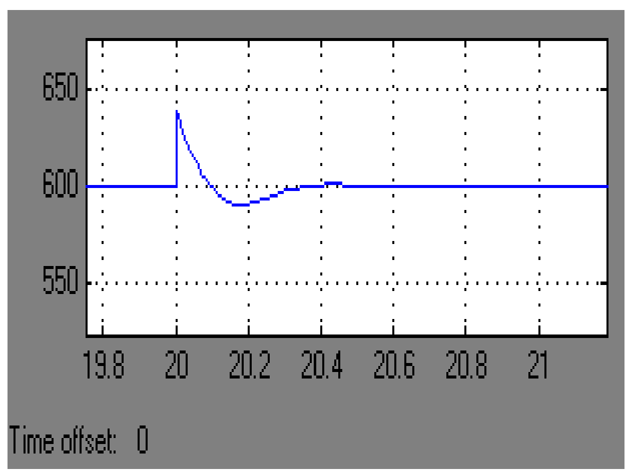
a. moving from without load to cutting depth of 0.2mm



b. moving from cutting depth of 0.2mm to without load



c. moving from cutting depth of 0.2mm to 0.5mm



d. moving from cutting depth of 0.5mm to 0.2mm

Figure 9. The robustness of spindle

The simulation is performed to assess the dynamic model and system responses at rise time (t_{r1} and t_{r2}), settling time (t_s), peak time (t_p) and overshoot percentage (M_p) and disturbance.The t_{r1} is the time required to change from steady state until it reaches a state under the influence of disturbance. The t_{r2} is the time taken from the beginning of the influence of the disturbance up to 100% of setpoint, the time required to reach the highest point overshoot (t_p), disturbance is the magnitude of the response changes from steady state output caused by load changes. The performances of the system can be found at Table III.

Tabel III. Spindle performance with moving load

NO	MOVING LOAD	t_{r1} (ms)	t_{r2} (ms)	t_p (ms)	M_p (%)	t_s (ms)	*DISTURBANCE* (%)
1.	without load to depth of 0.2mm	169.4	565.1	97.2	1.5	440.1	7.33
2.	depth of 0.2mm	169.6	565.2	97.2	1.5	440.1	7.37
3.	depth of 0.2mm to 0.5mm	175.3	565.6	0.01	6.46	300.5	8.01
4.	depth of 0.2mm to without load	175.3	565.6	0.01	6.46	300.5	8.01
5	without load to depth of 0.5mm	169.4	565.1	97.2	1.5	440.1	7.33
6	depth of 0.5mm to without load	175.3	565.6	0.01	6.46	300.5	8.01

The design of robust control based QFT approach in this study is able to cover the spindle speed conditions incorporating depth of cutting which consists of without load to depth of 0.2 mm, then was proceed load changes to depth of 0.5 mm. It appears that the responses remain on the same steady state conditions. Although only a few moments to change the responses. This case is implied the switch aspect that takes a while. There is a few time is unstable when conducting the switch to make the responses.

VI. EXPERIMENTAL RESULTS

Compensator and the pre-filter were designed to be implemented using a personal computer (PC). The technical steps to implement the controller consist of the first step is the compensator and the pre-filter transfer function have to be transformed into a discrete form. Secondly, the discrete form has to be transformed into a differential equation form.

The discrete form of the compensator and pre-filter has been designed are:

$$K(z) = \frac{0.3606z^3 + 2.146z^2 - 4.427z + 1.953}{z^3 - 2.085z^2 + 1.182z - 0.09721}$$

$$F(z) = \frac{0.02597 \quad z - 0.0197}{z^2 - 1.802 \quad z + 0.8083}$$

Then the differential equation of compensator can be obtained

$$z^3 Y(z) - 2.085z^2 Y(z) + 1.182 \; zY(z) - 0.09721Y(z) =$$
$$0.3606z^3 X(z) + 2.146 \, z^2 X(z) - 4.427 \; z \, X(z) + 1.953X(z)$$

$$Y(z) - 2.085z^{-1} Y(z) + 1.182 \; z^{-2} Y(z) - 0.09721z^{-3} Y(z) =$$
$$0.3606X(z) + 2.146 \, z^{-1} X(z) - 4.427 \; z^{-2} X(z) + 1.953z^{-3} X(z)$$

$$Y(z) = 2.085z^{-1} Y(z) - 1.182 \; z^{-2} Y(z) + 0.09721z^{-3} Y(z) +$$
$$0.3606X(z) + 2.146 \, z^{-1} X(z) - 4.427 \; z^{-2} X(z) + 1.953z^{-3} X(z)$$

$$Y(k) = 2.085Y(k-1) - 1.182Y(k-2) + 0.09721Y(k-3) +$$
$$0.3606X(k) + 2.146 \; X(k-1) - 4.427 \; X(k-2) + 1.953X(k-3)$$

Using the same process, the differential equation of the pre-filter can be obtained :

$$z^2 Y(z) - 1.802 \; z \, Y(z) + 0.8083Y(z) = 0.02597 \; zX(z) - 0.0197X(z)$$

$$Y(z) - 1.802 \; z^{-1} Y(z) + 0.8083z^{-2} Y(z) = 0.02597 \; z^{-1} X(z) - 0.0197z^{-2} X(z)$$

$$Y(z) = 1.802 \; z^{-1} Y(z) - 0.8083z^{-2} Y(z) + 0.02597 \; z^{-1} X(z) - 0.0197z^{-2} X(z)$$

$$Y(k) = 1.802 \; Y(k-1) - 0.8083Y(k-2) + 0.02597X(k-1) - 0.0197X(k-2)$$

The compensator and pre-filter were implemented using a PC. The PC calculated control signals based on the input voltage and motor speed. The data obtained from the motor speed tacho generator. The data of rotation speed is converted into voltage changes. The compensator and pre-filter algorithm calculate the magnitude of the control signal based on data from tacho generator. Figure 10 shows a diagram of automation rotation for spindle of lathe machine.

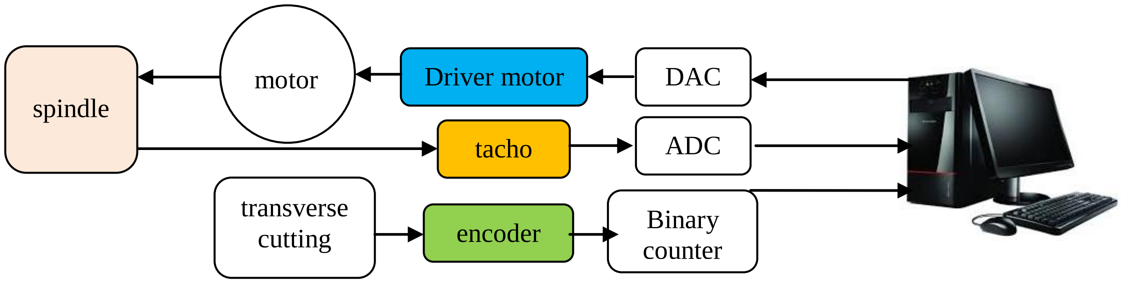
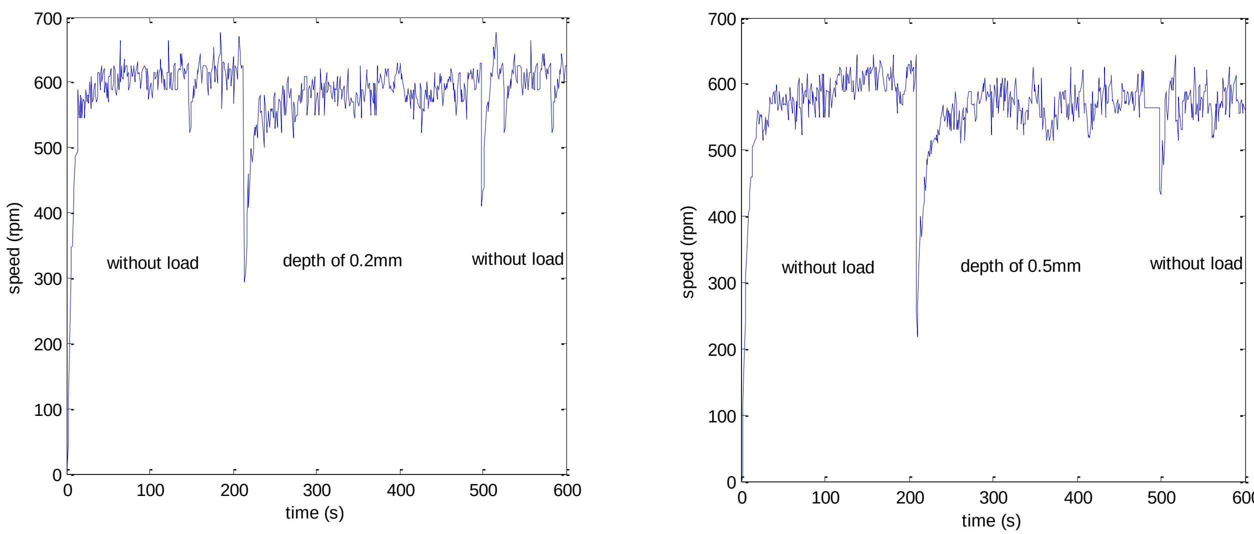


Figure 10. Diagram of automation rotation for spindle of lathe machine

The experimental results carried out for changes the condition from without cutting to a cutting depth of 0.2 mm and contrarily, load changes from without cutting to a depth of 0.5 mm and contrarily, and then load changes from without cutting to a depth of 0.2 mm continued to a depth of 0.5 mm and contrarily. Figures 11 and 12 show the experimental responses of the system.



a. Moving from without load to depth of 0.2mm and contrarily

b. Moving from without load to Experimental results of 0.5mm and contrarily

Figure 11. Experimental results of incorporating depth cutting from without load

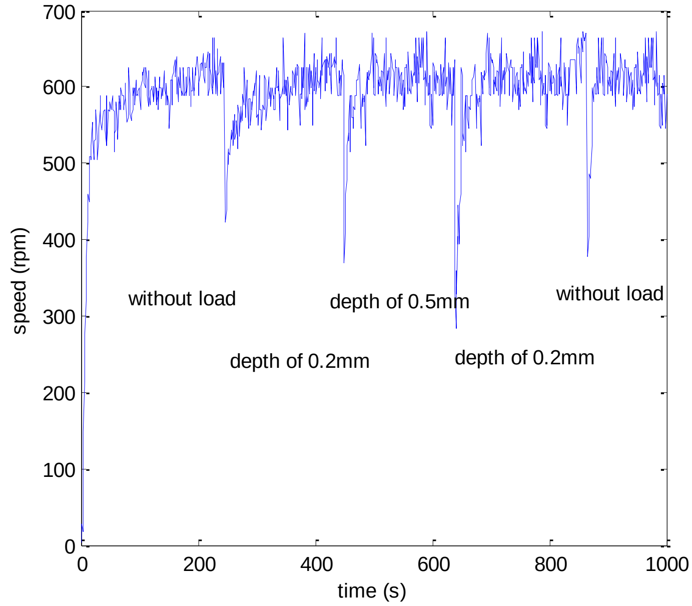


Figure 12. Experimental results of moving from without load to Experimental results of 0.2mm continued to Experimental results of 0.5mm and contrarily

VII. CONCLUSION

The development of a robust compensator and pre-filter control of a lathe spindle based QFT approach with varying deep of cutting has been presented. A set of linear model has been developed by taking through system identification of a lathe spindle approach. Practical design steps have been presented where the QFT approach has been employed to obtain robust compensator and pre-filter gains to control the spindle of lathe machine under various depth of cutting conditions. Robustness in the performance of the controller has been evaluated. Experimental results envisaged that despite using the same sets of compensator and pre-filter gains, the proposed QFT control provides better control performance for various depth of cut conditions.

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