Performance analysis of a robust proportional–integral–derivative control technique for the auto-focusing mechanism of an optical surface profile measurement system

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Abstract: A control system is regarded as the vital component of a process loop. For centuries, various control techniques have been employed to control actuators or process parameters in an effective manner to obtain the desired response. The field of control engineering has seen phenomenal changes in the past two decades or so, epitomized by the rise of ever-more sophisticated modern intelligent control techniques. However, despite such massive improvements in control techniques, century-old proportional–integral–derivative (PID) controllers are still widely used in industry and research alike owing to the simplicity of fabricating and tuning such devices. Many studies in the literature have shown that a particular variant of the PID controller, namely a robust PID, is even more effective in achieving better performance of the closed-loop system. This paper investigates the performance enhancement of the response of the auto-focusing mechanism of a surface profile measurement system using a robust PID controller as opposed to a controller tuned using the conventional Ziegler–Nichols (ZN) method. It can be observed through the reported results that the use of the former technique helps to achieve the highly desired closed-loop response of fast settling time, reasonable overshoot, loop disturbance rejection, and high system bandwidth of the focusing mechanism.

Keywords: Robust control technique, surface profile measurement system, auto-focusing mechanism, optical DVD reader, controller tuning, ZN method

1 INTRODUCTION

The research work presented in this paper deals with the design and tuning of a robust controller for the auto-focusing mechanism of a surface profile measurement system. The surface profile measurement system reported in this paper is used to measure inconsistent surfaces such as machined timber. In general, timber samples contain varying grain sizes that make up the surface, difference in moisture content across the length, cracks on the surface due to anatomical reasons, machining inaccuracies, and so on. These characteristics result in a property of non-uniform light reflectance and introduce uncertainty to the measurement. To ensure consistent error-free measurement, the auto-focusing probe needs to be able to adapt to the changes in measurement condition quickly. For this reason, it is imperative that a suitable control system is designed and incorporated in the focusing mechanism of the surface profile measurement system.

In order to ensure smooth and efficient running of any given industrial process or system within a given operating range and environment, a properly designed and tuned controller is an absolute necessity. The control system required in each case depends on several factors ranging from plant dynamics to rejection of load disturbance [**1**]. Open-loop, feedback, robust, and adaptive controllers are widely used in research and industry to control various systems [**2**]. Open-loop controllers are the simplest ones, but suffer from a high level of uncertainty

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introduced in the controlled variable in the presence of disturbances. A simple feedback controller is a more practical alternative, striking a balance between complications in implementation and improvement in system response under load disturbances. Robust controllers are much more resilient to disturbance in the control loop, but require higher computing power within the controller. Adaptive controllers are currently widely researched as they offer the opportunity to realize fully autonomous systems at the expense of a very high requirement of machine intelligence.

The feedback controllers fall into two main modes – the discontinuous and continuous [**3**]. The focus of the current paper is to explore a continuous mode controller for an auto-focusing system, thus only this controller mode will be discussed. The simplest forms of a continuous controller are the proportional (P) controller, integral (I) controller, and the derivative (D) controller. In practice, in a given industrial control scenario, only one controller mode is insufficient to control the process efficiently and within given requirements. Thus, the composite controllers of PI, PD, and proportional–integral–derivative (PID) are widely used. According to Astrom and Hagglund [**4**], despite massive advances in control systems over the last 50 years, 97 per cent of the surveyed controllers in industry are essentially PID controllers. This influence of PID controllers is due to the simplicity in designing, tuning, and implementing them in a process loop.

Simply determining that a system requires a PID controller is not sufficient. A proper controller tuning mechanism depending on the plant dynamics, performance requirements, and tuning tools available, is also required. A badly tuned controller degrades the performance of the process and ultimately contributes to wastage and loss of quality. However, choosing the right tuning technique for a particular process is far from easy. O'Dwyer [**5**] has reported a total of 1134 tuning rules for PI and PID controllers. Therefore, it can easily be seen that tuning a controller to meet operating performance requirements is a daunting task.

Ever since the paper on controller tuning was published by Ziegler and Nichols [**6**] in 1942, the Ziegler–Nichols (ZN) tuning method has been the most widely known and used tuning technique in control engineering. Originally proposed for pneumatic actuators, this empirical technique provides a very simple hands-on generic method of tuning feedback controllers for many processes. Thus, for over 65 years, the ZN method has been synonymous with controller tuning all over the world. The present paper will look into comparing this popular technique with a robustly tuned PID controller for specific application in an auto-focusing profilometer.

The paper is structured in the following manner: a brief introduction to the surface profile measurement system used for the research work is provided; then the plant model for the auto-focusing system is identified using an empirical method; design and tuning of the PID controller using two different methods is investigated; a comparison is drawn between the performance of the robust controller and that of a controller tuned using the ever popular ZN method; and finally some concluding remarks are given.

It is worth mentioning here that this paper focuses on simulated performance analysis of the two tuned controllers using the MATLAB® and SIMULINK® software packages. This forms the preliminary part of the research work to develop a fully-fledged low-cost surface profile measurement system. The actual controller performance analysis will be carried out in the latter part of the research project and will be reported in a later publication.

2 SURFACE PROFILE MEASUREMENT SYSTEM

Surface profile measurement systems have been widely used in various process industries for the evaluation of finished surfaces and tool wear. A great deal of research is being conducted to develop systems to evaluate a number of engineering surfaces, e.g. wood [**7**], computer numerically controlled (CNC) turned metals [**8**], optical lenses [**9**], turbine blade assembly [**10**], and so on.

One of the most common and primitive methods of surface quality or defect detection is the visual inspection approach [**11**]. In many industries this method, to this day, remains a valid procedure for surface inspection. This is a very subjective way of looking into product finishes and in most cases fails to ensure a reliable and consistent minimum standard of end product finish. To overcome the limitations imposed by visual inspection, a range of contact and non-contact accurate measurement techniques have emerged over the years.

The research work discussed in this paper involves the use of a low-cost DVD player laser pick-up (HOP1000 from Hitachi) as the surface profile measurement sensor in a non-contact manner. The construction of the sensor is shown in the schematic diagram of Fig. 1. The basic principle of the system is to sense the surface profile with the help of the focusing method. Depending on the change in

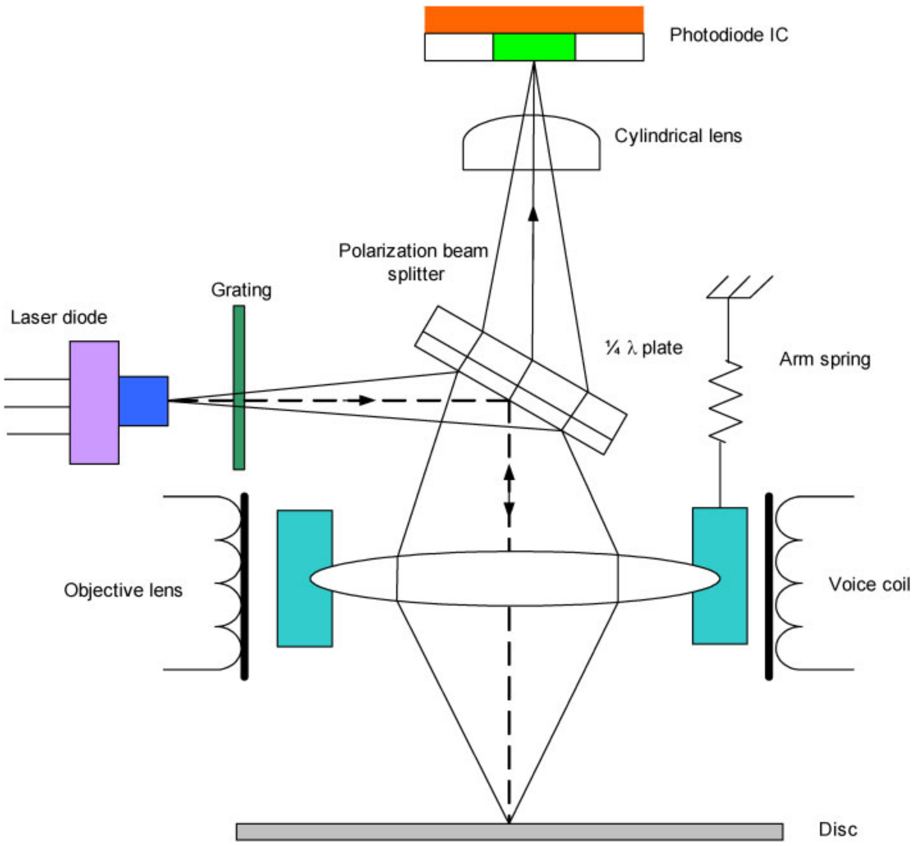

Fig. 1 Schematic diagram of the components of a DVD optical reader

surface profile of the object under test, the focal length of the lens varies. This method is termed the focusing technique and has been used experimentally to test some engineering surfaces, e.g. compact disks (CDs), mirrors, etc. [**12**, **13**]. The details of this method and various opto- and electromechanical components are not discussed here, as these are beyond the scope of this paper.

It can be seen from Fig. 1 that the voice coil motor (VCM) is responsible for moving the objective lens, which in turn helps the system to obtain focus on the surface of the disk or, in this case, the surface under test.

The schematic diagram of the closed-loop system profile measurement system is shown in Fig. 2. The detailed measurement scheme is discussed by Islam *et al.* [**14**], including the function of each block within this closed-loop system.

In order to control the movement of this VCM, an effective control system along with actuator drive is needed. To design and tune the controller, first and foremost, it is necessary to understand the character-

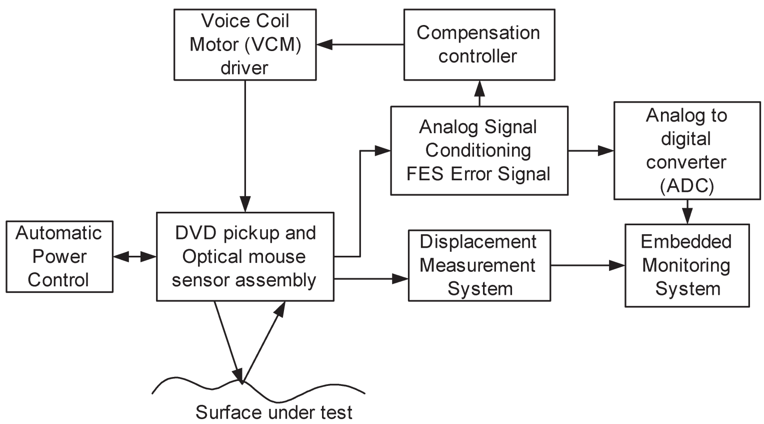

Fig. 2 Integrated profile measurement system

istics of the plant (i.e. the VCM actuator). Section 3 discusses how identification of the system has been carried out in order to select the appropriate controller.

3 SYSTEM IDENTIFICATION

The VCM system of the objective lens actuator was excited with a sinusoidal waveform of a fixed voltage by the analogue signal generator, TG120 (from Thurlby Thandar Instruments). This input voltage was set to 2V peak to peak, where the maximum input to the system has been earlier identified as 6V peak to peak [**14**]. A laser Doppler vibrometer (LDV) VH300 (from Ometron) was used to measure the movement of the objective lens. A four-channel digital oscilloscope, TDS 2004B (from Tektronix) was used to measure simultaneously the input and output of the VCM system.

The frequency of the input at the fixed amplitude was varied from 0 Hz to 1 kHz and for each frequency the amplitude vibration and vibration of the lens were measured. The frequency response curve of the system was drawn using the commonly used Bode plot.

If $G(j\omega)$ is the transfer function of the system in the frequency domain, then

$$G(j\omega) = |G(\omega)|e^{j\varphi(\omega)} \tag{1}$$

where $\varphi(\omega)$ is the phase and $|G(\omega)|$ is the magnitude.

The logarithmic gain of the system becomes

$$G_{\mathrm{dB}} = 20\log_{10}|G(\omega)| \tag{2}$$

which is expressed in decibels (dB). This logarithmic gain and the phase of the system were plotted and the results are shown in Fig. 3.

From the first observation, it is obvious that the resonant frequency at which the output of the system peaks is at 48 Hz. However, this information alone is not sufficient to identify the system.

Further analysis shows that the magnitude plot of the system has two asymptotes. The low-frequency asymptote gives the direct current (d.c.) gain of the system [**2**]. It can be observed that the high-frequency asymptote drops off at −40 dB per decade and the high-frequency phase asymptote is −180°. These two observations support the fact that the system has two more poles than zeros. When compared with the Bode plots of different order systems given in reference [**2**], it can be seen that the transfer function of the VCM can be given by

$$G(s) = \frac{K}{(s\tau_1 + 1)(s\tau_2 + 1)} \tag{3}$$

Further analysis shows that the system in equation (3) can be simplified to

$$G(s) = \frac{G_{\mathrm{dc}}\omega_{\mathrm{n}}^2}{s^2 + 2\varsigma\omega_{\mathrm{n}}s + \omega_{\mathrm{n}}^2} \tag{4}$$

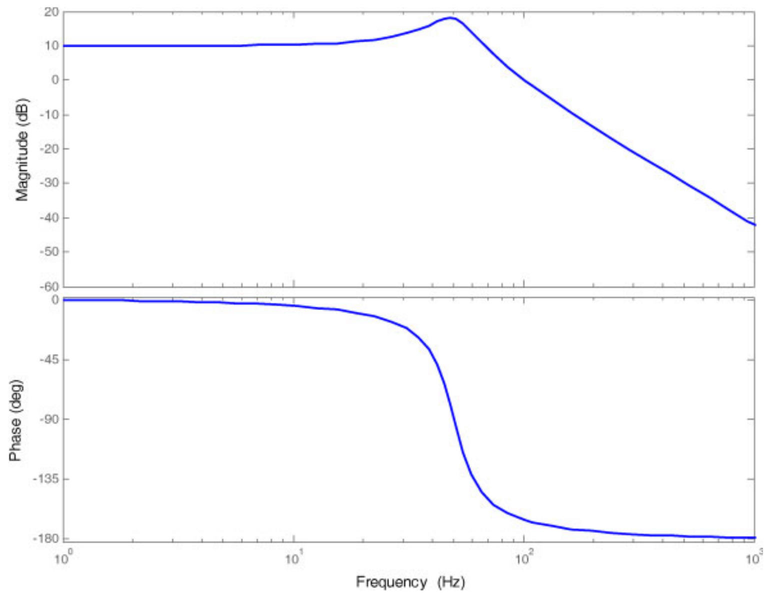

Fig. 3 Bode diagram of the VCM system

Thus, this is a second-order system with a damping ratio of ζ and natural frequency of ω_n.

After substituting the values for the aforementioned parameters, equation (4) comes to

$$G(s) = \frac{312.08e3}{s^2 + 125s + 98.7e3} \tag{5}$$

The −3dB bandwidth of the system, shown in Fig. 3 stands at 75 Hz. However, with the resonant frequency at 48 Hz, the maximum practical bandwidth of the system is less than 48 Hz. For industrial applications, the bandwidth of the system needs to be much higher than this, as pointed out by Islam *et al.* [**14**].

4 CONTROLLER DESIGN AND TUNING

The order and transfer function of the system has been determined in the previous section. Based on this information, selection of the type of controller and its parameters is discussed next.

According to Astrom and Hagglund [**1**], PID control is sufficient when the process to be controlled is of second order. They have shown that there are no benefits gained by using a more complex controller in this case. Thus, the chosen controller for the present auto-focusing system will be the PID type.

A generic simple feedback controlled loop is shown in Fig. 4. In this case, the controller is a PID. The structure of this PID controller can be described mathematically as in equation (5)

$$u(t) = K\left[e(t) + \frac{1}{T_i}\int_0^t e(\tau)\,d\tau + T_d\frac{de(t)}{dt}\right] \tag{6}$$

where u is the control variable and e is the control error ($e = y_{sp} - y$).

It can be seen from Fig. 4 that there are two variables l and n in the feedback, which correspond to load disturbance and measurement noise respectively. In a practical control scenario, it is common to have substantial measurement noise. Prefiltering techniques are usually employed to eliminate this noise from the control loop [**3**].

Load disturbance is also a source of concern when designing and tuning a controller for a given process. In the case of the profile measurement system, load disturbance is unavoidable because the focusing mechanism will be subjected to surface irregularities and needs to adjust to this disturbance in a fast and efficient manner in order to minimize measurement error.

This is an important criterion which needs to be taken into account when tuning the controller for this specific application. Hence, the robust PID controller, which exhibits the desired performance in the presence of significant plant uncertainty [**2**], is probably more appropriate than ZN for the auto-focusing mechanism of this profile measurement system. This assumption will be explored in more detail in the results and discussion section 5 after both the ZN and robust tuning methods have been introduced.

4.1 Ziegler–Nichols (ZN) tuning method

There are two ways a controller can be tuned using the ZN method. One is the process-reaction method and the other is the ultimate cycle method [**4**].

In the process-reaction method, a transient disturbance is introduced to the system by a small, manual change of the controlling variable using the final control elements. The response of the controlled variable is then measured. Then, various parameters such as the lag time, process reaction time, and variable change are measured, as shown in Fig. 5.

According to this process-reaction method

L = lag time in minutes

$$N = \frac{\Delta C_p}{T} \tag{7}$$

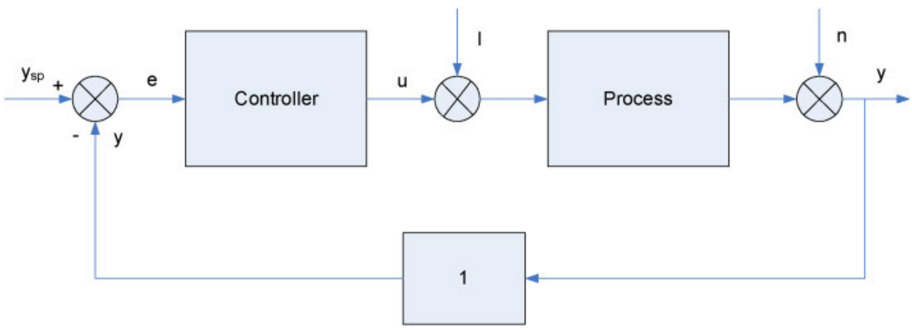

Fig. 4 Block diagram of the feedback loop

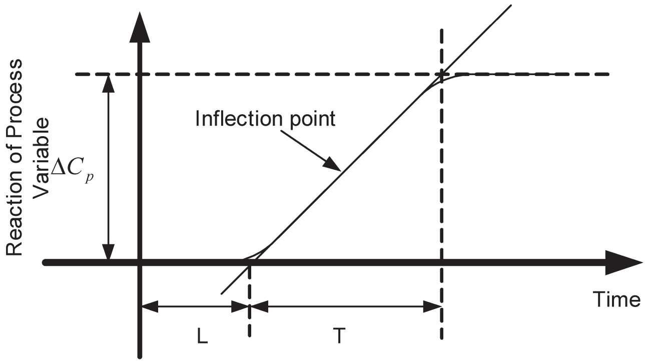

Fig. 5 Process reaction curve

where N = reaction rate (per cent/min), ΔC_p = variable change i(per cent), and T = process reaction time (min).

The appropriate proportional gain, integration time and derivative time for the three-term PID controller can then be found from the following equations

$$K = 1.2\frac{\Delta P}{NL} \tag{8}$$

$$T_\mathrm{I} = 2L \tag{9}$$

$$T_\mathrm{D} = 0.5L \tag{10}$$

However, this technique of open-loop process reaction curve does not yield very good closed-loop performances [**15**]. Also, the parameters are all measured in minutes and this method had been proposed for a pneumatic system some 60 years ago. Thus, it is very difficult to obtain the process reaction parameters from a curve with fast settling mechanisms like the one described in this paper.

A more appropriate solution is the ultimate cycle method. In this technique, all the controller gains are set to their minimum apart from the proportional gain. Then the system is excited by a transient disturbance (usually through a step input). The proportional controller is so adjusted until the closed-loop system is in steady oscillation. This gain is termed the critical gain (K_cr) of the system and the period of oscillation (measured in minutes) is called the critical period (T_cr).

The proportional gain (K), integration time (T_I), and derivative time (T_D) for the three-term PID controller can then be found from the following equations

$$K = 0.6K_\mathrm{cr} \tag{11}$$

$$T_\mathrm{I} = \frac{T_\mathrm{cr}}{2} \tag{12}$$

$$T_\mathrm{D} = \frac{T_\mathrm{cr}}{8} \tag{13}$$

The response of the closed-loop system tuned using this ZN ultimate cycle method is shown and discussed in the next section.

The aforementioned parameters for tuning the controller were derived for a specific application and especially for pneumatic actuators, as discussed earlier. Thus, in order to obtain highly flexible and responsive controllers which can counteract the inherent non-linearity and disturbance within a given load, parameter optimization techniques are often employed. This technique results in a multilevel tuning method. Research carried out by Horsley *et al.* [**16**], Huang and Wang [**17**], and more recently by Mhaskar *et al.* [**18**] show how the closed-loop performance of the plant can be enhanced using the optimization techniques.

Although this type of controller optimization is not new (as pointed out above), research has shown that the robust tuning method proposed and implemented in this paper is a novel contribution for the DVD profilometer measuring timber surfaces.

The method for tuning and optimizing the PID controller using techniques discussed by Dorf and Bishop [**2**] is discussed in the following subsection. As mentioned before, this optimized controller is termed a robust PID controller owing to its resilience in the presence of significant plant disturbance and ability to maintain desired output.

4.2 Robustly tuned PID controller

There are several criteria that determine the performance of a controller [**19**]. These quantitative measures of the performance of the system are termed performance indices. These indices are chosen such that emphasis is given to the important system specifications. The general form of the performance integral is

$$I = \int_0^T f[e(t), r(t), y(t), t]\,\mathrm{d}t \tag{14}$$

where f is the function of the error, input, output, and time. The upper limit of the integral T is a finite time chosen in such a way that the integral approaches a steady-state value. This value is usually chosen to be the settling time.

In order to reduce the contribution of the large initial error to the value of the performance integral, as well as to emphasize errors occurring later in the response, the following ITAE (integral of time multiplied by absolute error) index has been proposed [**2**]

$$\text{ITAE} = \int_0^T t|e(t)|\,\mathrm{d}t \tag{15}$$

According to Dorf and Bishop [**2**], the coefficients that will minimize the ITAE performance criterion for a step input have been determined for the general closed-loop transfer function as

$$T(s) = \frac{b_0}{s^n + b_{n-1}s^{n-1} + \cdots + b_1 s + b_0} \tag{16}$$

where b_0, b_1, $b_2,\ldots,$ b_{n-1} = normalized transfer function coefficients.

The transfer function of a PID controller is as follows

$$G_c(s) = K_1 + \frac{K_2}{s} + K_3 s \tag{17}$$

Thus, the closed-loop transfer function of the VCM system along with this controller $G_c(s)$ can be found to be

$$T_1(s) = \frac{G_c G(s)}{1 + G_c G(s)} \tag{18}$$

From equation (15), the optimum coefficients of $T(s)$ with this controller $G_c(s)$ based on the ITAE criterion for a step input can be derived as [**2**]

$$S^3 + 1.75\omega_n s^2 + 2.15\omega_n^2 s + \omega_n^3 \tag{19}$$

Thus, comparing equations (17) and (18), the optimum value of the controller coefficients, K_1, K_2, and K_3 are determined based on the ITAE criterion. These coefficients differ slightly from the three terms discussed in equations (10), (11), and (12). The relationships between them are as follows

$$P = K_1 = K \tag{20}$$

$$I = K_2 = \frac{K}{T_I} \tag{21}$$

$$D = K_3 = KT_D \tag{22}$$

where P = proportional gain, I = integral gain, and D = differential gain of the controller.

Detailed discussions on the optimization techniques and derivation of the optimized terms are beyond the scope of this paper but can be found in reference [**20**]. The discussed ITAE performance index produces an excellent transient response to step input, which can be seen from the results reported in section 5.

5 RESULTS AND DISCUSSION

This section compares the performance of the PID controller tuned using the ZN method with that using the robust method. The controller parameters obtained using the two methods are shown in Table 1. The Bode plots presented in the paper are generated with a sampling rate of 1 Hz, while the performance curves of settling time and response to load disturbance are sampled at 100 μs.

Bode diagrams in Fig. 6 show the uncompensated system response (i.e. the displacement of the VCM) along with the closed-loop system response with a ZN-tuned controller in place. It is easily noticeable that the resonant frequency of the system has been changed and the resonant frequency of the closed-loop system stands at 300 Hz. This is a substantial improvement from the natural system response, which has been shown to be 48 Hz. The −3 dB bandwidth of the closed-loop system can be measured at 500 Hz, although practically, due to the resonant peak, the maximum bandwidth of the system is limited to much less than 300 Hz. The value of 100 Hz can be deduced to be the practical bandwidth of the system.

However, with the implementation of an ITAE-based robust PID controller, the bandwidth of the closed-loop system increases dramatically. Bode plots in Fig. 7 show both the uncompensated and robustly controlled system responses. The robust

Table 1 Tuning parameters for ZN and robustly tuned PID controllers

ZN tuning	Robust tuning
$P = 12$	$P = 8.8\text{e}3$
$I = 9600$	$I = 82\text{e}6$
$D = 0.00375$	$D = 0.355$

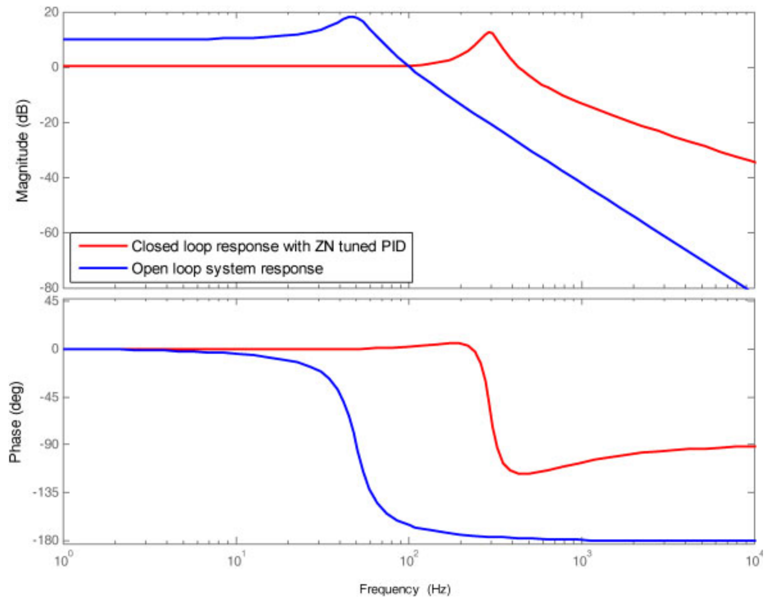

Fig. 6 Open- and closed-loop frequency responses with ZN tuning

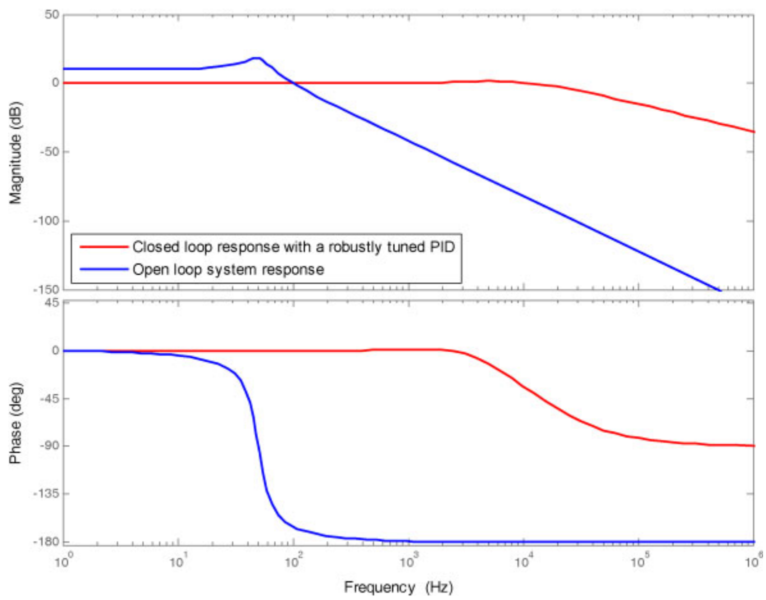

Fig. 7 Open- and closed-loop frequency responses with robust tuning

controller yields a closed-loop system bandwidth of 21 kHz, which is a marked improvement to the open-loop bandwidth of 48 Hz or even to the ZN-tuned controller bandwidth of 100 Hz.

Apart from the bandwidth, the performance of a closed-loop system can also be evaluated by the settling time and percentage overshoot values. The closed-loop system tuned with the ZN method is shown in Fig. 8. The settling time of the system (taking the time in which the response settles to 20 per cent of the final value), is found to be 15 ms. The percentage overshoot of the system stands at about 60 per cent of the final value.

The response of the closed-loop system to a step input can be seen from Fig. 9. The percentage overshoot of the system stands at 17 per cent, while the settling time is reduced to only 0.2 ms. This marks a reduction in settling time of almost 99 per cent.

It can be envisioned that the auto-focusing mechanism of the optical measurement system will undergo random load disturbance due to the nature of the sample being tested. Thus, a good load rejection characteristic of the control system is required in order to maintain focus on the surface.

A disturbance in the closed loop was created that was 80 per cent of the initial step input. The results

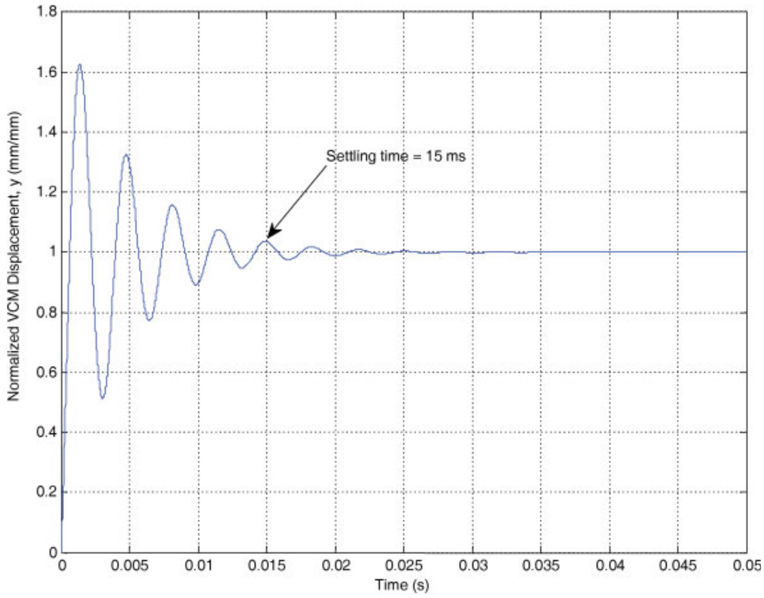

Fig. 8 Closed-loop response to step input with ZN tuning

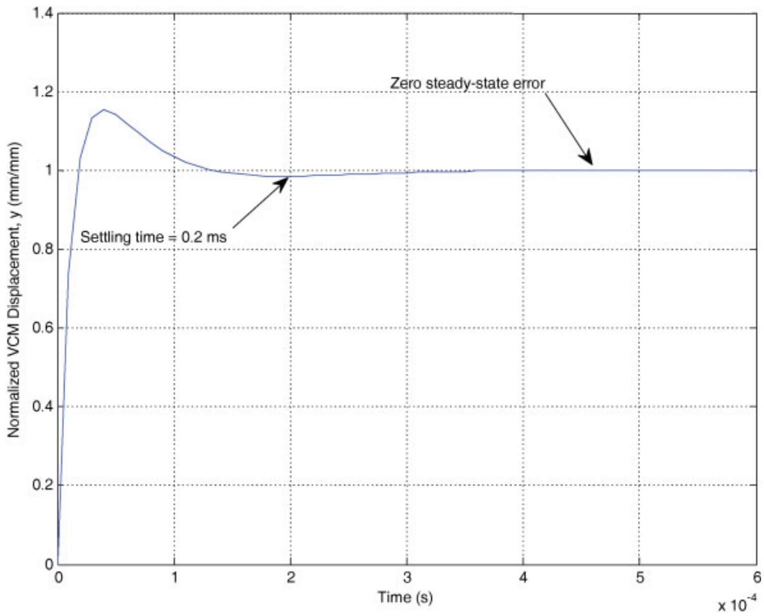

Fig. 9 Closed-loop response to step input with robust tuning

of load disturbance rejection by the ZN-tuned and robustly tuned loops are shown in Figs 10 and 11 respectively. From Fig. 10 it is apparent that a load disturbance is introduced at time 40 ms after the VCM system has settled down. The amplitude of the disturbance is as previously mentioned, 80 per cent of the final value. It can be observed that the system is somewhat destabilized at the introduction of the disturbance. However, for an 80 per cent disturbance, only a 5 per cent change in system output is observed. Thus, it can be said that the ZN-tuned closed-loop system offers very good load rejection capabilities.

Figure 11 shows the response of the same system with a robust controller in place. It can be seen that a load disturbance of 80 per cent of the final value has been introduced after 10 ms of the initial step input.

Despite a very high value of load disturbance, it can be seen that the output of the closed-loop system does not change after the initial step input. Thus, it can be said that the controller has excellent robust load disturbance rejection capabilities.

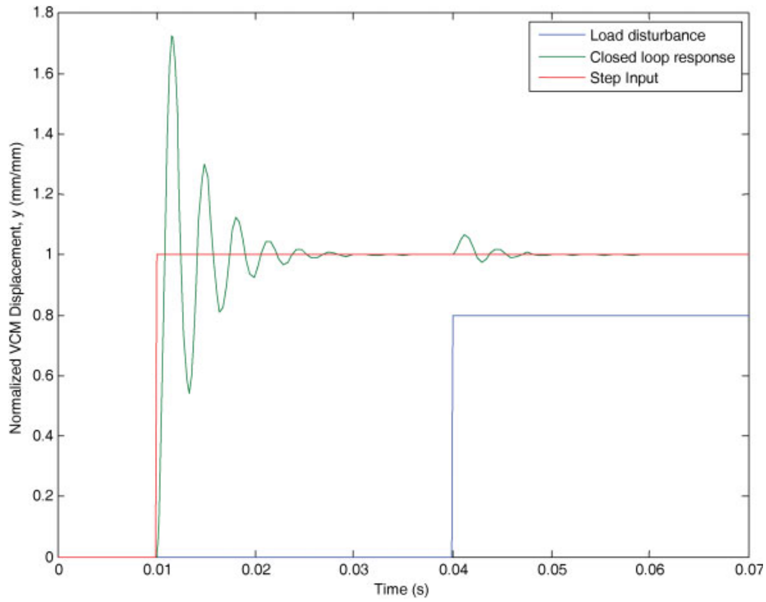

Fig. 10 Effect of load disturbance in ZN tuning

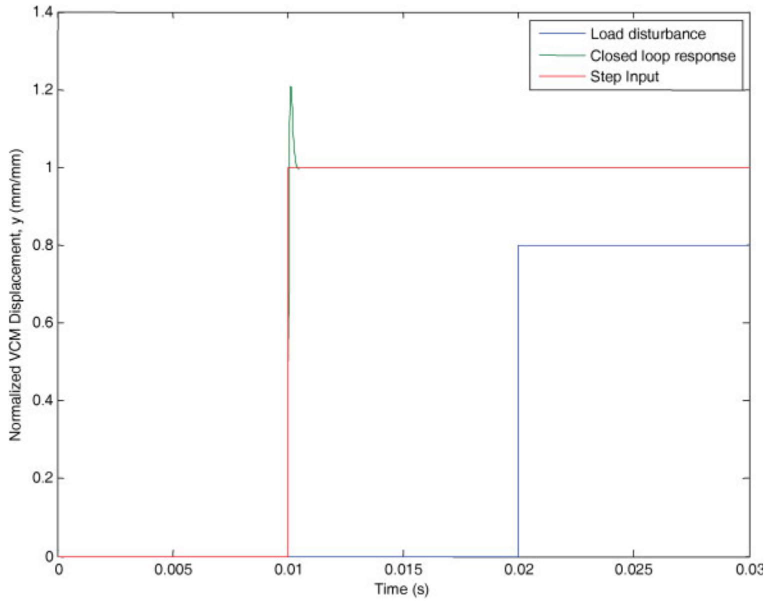

Fig. 11 Effect of load disturbance in robust tuning

The surface profile measurement scheme shown in Fig. 2 comprises a displacement measurement system. There is also an optical non-contact system providing encoder reading to the main microcontroller carrying out the profile measurements [**14**].

Within the proposed set-up, a pulse signal is generated whenever the encoder reading reaches a certain threshold, which in this case is set at 100 μm. Calculations show that the required frequency response of the closed-loop system needs to be at 20 kHz for measurements to take place at 2 m/s, which is close to the acceptable industrial profile measurement speed [**7**].

The closed-loop system tuned with the ZN method was supplied with a square wave input of 1 kHz, which resembles a measurement rate of 100 mm/s. The response of the system along with the input can be observed in Fig. 12. As shown by the frequency response curve of Fig. 6, the ZN-tuned system is not able to provide satisfactory output at 1 kHz, owing to its bandwidth limitation of 100 Hz.

Figure 13 reports the response of the system tuned using the robust optimization technique to the same 1 kHz square wave input. It can be observed that, although the overshoot stands at 30 per cent, a fast

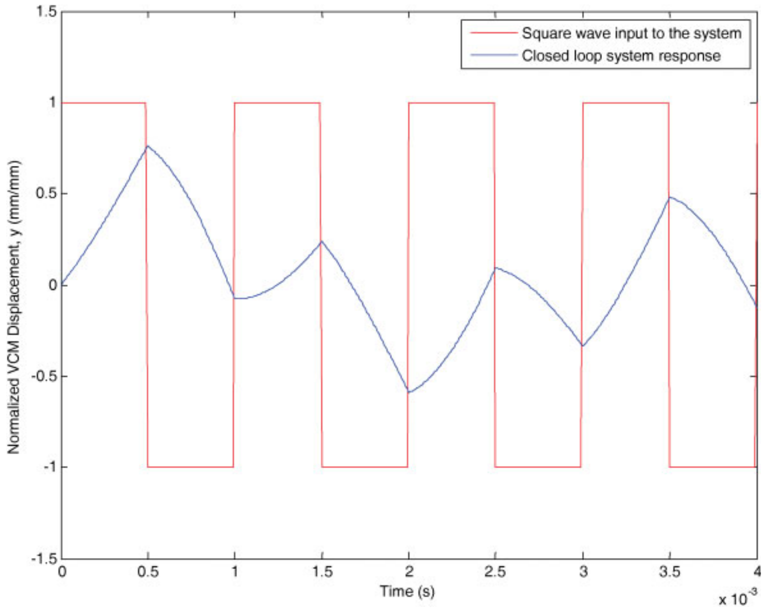

Fig. 12 Response to square wave input at 1 kHz with ZN tuning

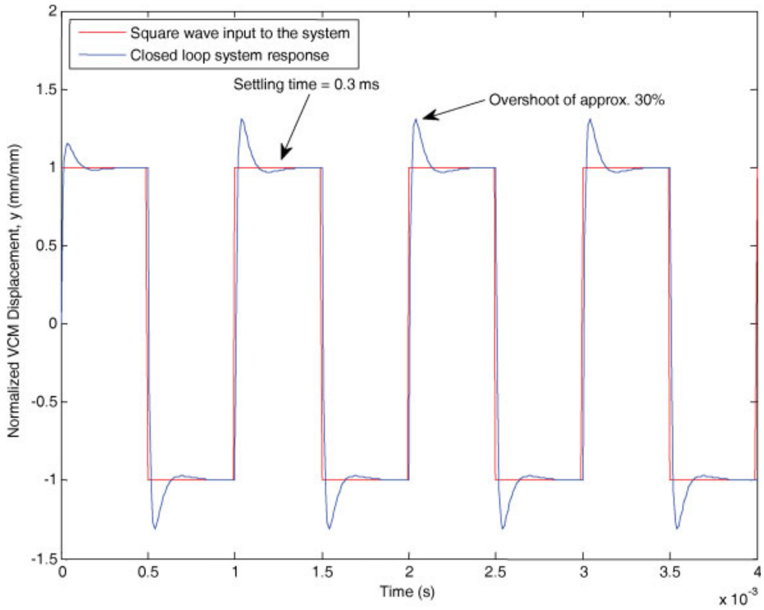

Fig. 13 Response to square wave input at 1 kHz with robust tuning

settling time of 0.3 ms along with a generally acceptable response is obtained from this closed-loop system.

6 CONCLUSION

A comparative analysis of two different tuning methods for a PID controller has been discussed in this paper. The controllers are tuned for a specific application of the auto-focusing mechanism of a surface profile measurement system. It can be seen from the results that the most commonly used ZN method of tuning a controller provides deficient performance for this particular system. It fails to meet the most important transient response requirements of low overshoot and fast settling time. However, a controller tuned using this technique fares reasonably well with load disturbance in the process loop.

On the other hand, a more analytically tuned controller based on the ITAE performance criterion exhibits an excellent transient response. It also shows great robustness by completely rejecting load disturbances. Although this sort of tuning requires much more complicated computational and analytical manipulations, the improvement in overall performance outweighs this drawback. Thus, it can be concluded that a robust controller is a more suitable control solution for such an auto-focusing mechanism within a profile measurement system.

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