



# Active vibration control and real-time cutter path modification in rotary wood planing<sup>☆</sup>



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## ARTICLE INFO

### Article history:

Received 1 July 2016

Revised 22 March 2017

Accepted 13 June 2017

Available online 3 July 2017

### Keywords:

Wood machining

Eddy current sensor

Piezoelectric actuator

Vibration control

Cutter path optimisation

LQG controller

## ABSTRACT

Forced structural vibration and cutting tool inaccuracy have been identified to be the primary causes of surface defects in rotary wood planing. This paper presents the development of a control strategy used to compensate for the effects of both vibration and cutting tool inaccuracy on planed wood surface finish. The solution is based on active vibration control and real-time modification of the cutting tool trajectory using an optimal Linear Quadratic Gaussian tracking controller. A small-scale mechatronic wood planing machine, which has an actively controlled spindle unit, has been designed for practical investigation of the proposed technique. Experimental results show that the applied compensation increased the dynamic performance of the machine and the quality of the surface finish produced.

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## 1. Introduction

Rotary planing is one of the most important machining processes in the value-added woodworking industry [1,2]. The details of the process are well reported in the literature [2–5]. Due to the kinematics of the process, waves known as cuttermarks are formed on the machined wood surfaces. The surface waviness is the key parameter used in the woodworking industry to determine the quality and production efficiency of the machine. A high quality surface finish is classified by cuttermarks width typically less than 1.5 mm, and cuttermarks width greater than 2.5 mm is typically considered to be a lower quality surface finish [3]. The uniformity of the cuttermarks is also one of the main parameters used to determine the quality of the surface finish.

It has been established that forced vibration and cutting tool inaccuracy are the primary causes of surface quality degradation in wood planing [3]. Although the defects can be amplified by the type and the condition of the wood material, they are primarily caused by the machine deficiencies [3,4]. It is estimated that defects could reduce processing yield in wood planing by 27% [1]. Cutting tool inaccuracy occurs due to the difficulty in grinding all the cutting edges in a cutterhead to a common radius. The use of

the best available precision tools and techniques would still produce a total indicated run-out (TIR) typically within the range of 5–10  $\mu\text{m}$  [5]. The TIR is the difference between the largest radius cutter and the smallest radius cutter on a cutterhead. A TIR value of less than 1  $\mu\text{m}$  is required to produce an acceptable multi-knife finish but this is not technically feasible at any reasonable price via mechanical methods [3].

The current solution to the problem of cutting tool inaccuracy in the woodworking industry is to apply a process known as jointing. The term is used to describe the dressing of cutting tools in order to true all the cutting edges to a common radius [3]. Non-jointed cutting edges produce significant repeating patterns of non-uniform cuttermarks on the machined workpiece. The disadvantage of jointing is that it removes or reduces any back clearance angle from the cutter leading to accelerated tool wear. Tool wear does not only increase tooling costs, but it also increases the normal cutting forces generated during the machining process, which leads to an undesirable burnishing action on the timber surface.

In addition to cutting tool inaccuracy, structural vibration is another major cause of defects in rotary wood planing. Vibrations can arise from multiple sources such as mass imbalances, and bearing misalignments. Vibration results into eccentric running of the cutterhead relative to the workpiece thereby altering the dynamic cutter run-outs of the cutting edges. In cases where jointing is applied, the jointing device could also modify the cutter tracking orbit due to the jointer vibration, which is excited by cutterhead imbalance [5]. Also, the cutting forces generated due to the relative

<sup>☆</sup> Reviews processed and recommended for publication to the Editor-in-Chief by Associate Editor Cesare Fantuzzi.

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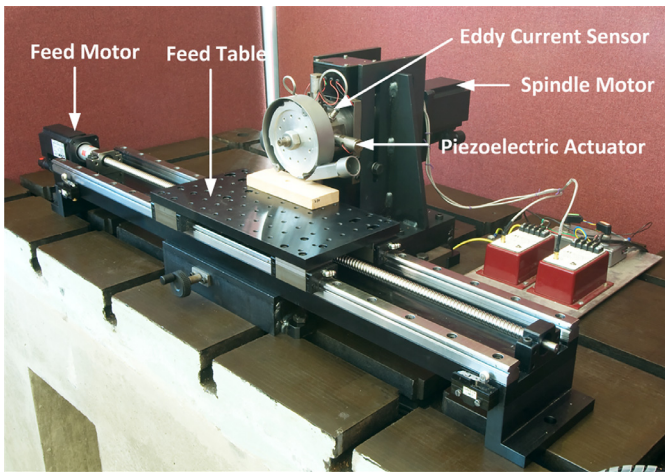


Fig. 1. Small-scale mechatronic wood planing machine.

motion between the cutter and the workpiece could also lead to vibration.

Some ideas on how to improve the quality of planed wood surfaces through real-time adjustments of the cutting tool trajectories have been reported in the literature. The first one is the introduction of horizontal cutterhead movement during the machining process [6,7]. Similar approach that introduces vertical cutterhead movement has also been reported [8]. The aim is to produce surface finish with reduced waviness heights. Results show that these methods produce cuttermark heights below what is obtained from the conventional machining. However, these techniques still do not compensate for the effects of vibration and cutter inaccuracy. Real-time optimisation of the cutter path has also been used to compensate for cutting tool inaccuracy [9,10]. The technique used is to adjust the cutterhead vertical position in such a way that all the cutting edges are at the same radius when engaged with the workpiece. Tool path optimisation is not limited to wood planing as the technique has also been used in other wood machining operations to minimise cutting force and tool wear [11].

The adjustment of cutter path has been extensively studied in wood planing but vibration control has received less attention. Work reported so far on vibration control in wood machining has focused on circular sawing and milling [12–18] and other wood machining processes [19]. Work on the improvement of the dynamic performance of a wood planer spindle using piezoelectric actuators has also been reported in [20].

Although cutting tool inaccuracy compensation and active vibration control in rotary wood planing have been studied independently [9,10,20] there is a greater challenge in solving the two problems concurrently. Therefore, the aim of this paper is to develop an appropriate control strategy that is capable of combining real-time tool trajectory adjustment with dynamic process disturbance compensation. The proposed controller is a combination of feedforward and feedback control loops. Experiments are performed on a small-scale mechatronic wood planing machine in order to investigate the effects of the compensation on the surface finish quality.

## 2. Mechatronic wood planing machine

The work reported in this paper is based on an experimental test facility designed to explore different techniques that can be used to improve the performance of industrial wood planing machines (Fig. 1). The small-scale mechatronic wood planing machine is instrumented with two non-contact eddy current sensors and four piezoelectric actuators for controlling the spindle in the plane

Table 1  
Angular positions and the static run-outs of the four cutters.

Cutter	Static run-out ( $\mu\text{m}$ )	Angular position (counts)
1	$y_1 = 0$	$\theta_1 = 60$
2	$y_2 = -12$	$\theta_2 = 560$
3	$y_3 = -2$	$\theta_3 = 1060$
4	$y_4 = 1$	$\theta_4 = 1560$

perpendicular to its rotational axis. The eddy current sensors are used to measure the spindle displacement in the X-Y plane. The actuators are capable of moving the spindle up to a peak-to-peak amplitude of  $36\mu\text{m}$ . The spindle is also equipped with a rotary encoder for measuring its angular position so that its displacements can be synchronised with its rotational angle. The encoder generates 2000 counts per revolution and an index pulse once every revolution. The index pulse is used to establish an absolute rotational angle of the spindle. The eddy current sensors are aligned with the actuators in axial direction but rotated by  $45^\circ$  with respect to the actuators as shown in Fig. 2. Therefore, the sensor readings are always converted from the sensors coordinate system ( $x_s, y_s$ ) to the actuators coordinate system ( $x_a, y_a$ ).

Cutting tool inaccuracy compensation through periodic vertical displacements of the spindle has already been explored and tested on the machine [9,10]. The compensation was applied to a four-knife cutterhead based on the static run-outs of the cutters and their absolute angular positions on the spindle.

The four-knife cutterhead mounted on the spindle has a mass of 293g and a nominal diameter of 120 mm. The cutters are numbered from one to four in the clockwise direction. The measured static run-outs and the angular positions of the cutters at their lowest points are given in Table 1. As seen in the table, the TIR of the cutterhead is about  $13\mu\text{m}$  (cutter 2 and 4).

The run-outs were measured after the cutterhead had been mounted on the spindle in order to eliminate the effect of any misalignments between the centre of the cutterhead and the spindle centre. The main shortcoming of the approach explored in [9,10] is that the dynamics of the machine are not taken into account. Vibration could cause the dynamic run-outs of the cutters to be significantly different from the statically measured values. The resultant effect of superimposed vibration on the surface finish quality depends on its amplitude and phase. Therefore, an approach which optimises the cutter path based on the static cutter measurements and also compensates for the effects of vibration is required for reliable operations of the machine.

## 3. Controller design

The proposed control system for achieving the aim of this paper is a combination of disturbance feedforward and feedback control loops. A simple feedback control mechanism alone may not meet extra high tracking performance requirements, especially in the presence of large disturbances and when there is a long delay between the plant output measurements and the firing of the actuators. Significant improvements in the tracking performance can be obtained if some of the process disturbances are measured and fed forward into the control loop, so that corrective actions can be initiated in advance before they affect the system's response. The feedback loop consists of a Linear Quadratic Gaussian (LQG) tracking controller with integral action. The LQG controller is used because it provides the best possible performance using the least amount of control effort [21]. Apart from its optimality, it is also stable and robust to process disturbances and measurement noise.

An LQG controller can be used either as a regulator or a set-point tracker. When it is used as a regulator, the control objective is to drive the plant output to zero. A regulator-type LQG controller

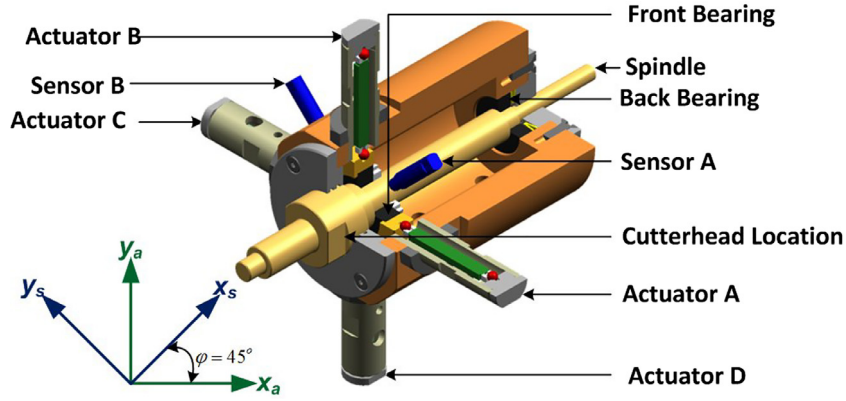


Fig. 2. Actuators and sensors arrangement on the smart spindle unit.

would be suitable for implementing vibration control system only because the objective is to drive the spindle to state of equilibrium [20]. In this paper, the controller is used as a set-point tracker because the control objective is to drive the error between the actual spindle output and a desired reference signal to zero.

The operation of the controller is such that when the spindle unit is to be displaced, a proportional amount of control effort is generated by the feedforward loop based on a priori knowledge of the static knife run-outs, static gain of the piezoelectric actuators and the anticipated effects of measurable disturbance inputs (vibration). Large tracking errors could result due to inaccurate knowledge of the dynamic gain of the piezoelectric actuators, the hysteresis of the piezoelectric actuator, and any unknown disturbances acting on the machine. The feedback information from the eddy current sensors is then used to generate any increase or decrease in the combined controller output that is required to minimise the error between the reference signal and the actual spindle position. The overall controller design is presented as follows:

### 3.1. Optimal tracking controller design

Consider a linear time-invariant system described by the following discrete-time state-space equations [21]:

$$x(k + 1) = Ax(k) + Bu(k) \quad (1a)$$

$$y(k) = Cx(k) \quad (1b)$$

where:

- $x$  is the state vector
- $u$  is the input vector
- $y$  is the output vector
- $A$  is the state matrix
- $B$  is the input matrix
- $C$  is the output matrix

The requirement is to design a control system with satisfactory transient response and the steady state output of the plant should be equal to an arbitrary reference signal. This requirement can be greater achieved if the control law makes use of both the nominal state vector and the integral of the error between the reference and actual output. Therefore, the integral of the error between the spindle position  $y(k)$ , and the reference set-point value  $r(k)$ , is used as an additional state variable to form an augmented state vector. The output of the integrator is given as:

$$i(k) = \sum_{k=0}^N r(k) - y(k) = \sum_{k=0}^N r(k) - Cx(k) \quad (2)$$

The augmented state vector  $\hat{x}$  is given as follows:

$$\hat{x}(k) = \begin{bmatrix} x(k) \\ i(k) \end{bmatrix} \quad (3)$$

The state equation for the augmented system is given as:

$$\hat{x}(k + 1) = \underbrace{\begin{bmatrix} A & 0 \\ -C & 0 \end{bmatrix}}_{\hat{A}} \hat{x}(k) + \underbrace{\begin{bmatrix} B \\ 0 \end{bmatrix}}_{\hat{B}} u(k) \quad (4a)$$

$$y(k) = \underbrace{[C \quad 0]}_{\hat{C}} \hat{x}(k) \quad (4b)$$

An optimal control law that includes feedback of both the nominal states variables and the added state variable is formulated as follows:

$$u(k) = -K\hat{x}(k) = K \frac{x(k)}{i(k)} \quad (5)$$

The gain  $K$  is calculated so that the control law minimises the objective function given by Eq. (6). The objective function defines the trade-off between tracker performance and control effort.

$$J(0, N) = \sum_{k=0}^N \hat{x}^T(k).Q.\hat{x}(k) + u^T(k).R.u(k) \quad (6)$$

where  $Q$  is the state weighing matrix and  $R$  is the control cost matrix. The augmented state feedback gain matrix  $K$  is calculated from the following equation:

$$K = R^{-1}.\hat{B}^T.M \quad (7)$$

where  $M$  is the solution to an algebraic Riccati equation, given by

$$\hat{A}^T.M.\hat{A} - M - \hat{A}^T.M.\hat{B}.(\hat{B}^T.M.\hat{B} + R)^{-1}.\hat{B}^T.M.\hat{A} = 0 \quad (8)$$

$$K = [K_x \quad K_i] \quad (9)$$

The final control law, which includes the reference signal feedforward, is written as:

$$u(k) = -[K_x \quad K_i] \frac{x(k)}{i(k)} + K_r r(k) \quad (10)$$

where:

- $K_x$  is the nominal state feedback gain
- $K_i$  is the steady state error integral gain
- $K_r$  is the reference signal feedforward gain

### 3.2. Optimal observer design (Kalman filter)

A full-state feedback cannot be used to design a control system because all the state variables cannot be realistically measured. A Kalman filter is required to estimate the state vector, based upon the output measurement  $y(k)$  and known input  $u(k)$ . The goal of the Kalman filter is to provide an estimated state vector  $x_e(k)$  so that the estimation error  $e(k) = x(k) - x_e(k)$ , is brought to zero in the steady state. The Kalman observer design is adapted from [21]. Consider a stochastic noisy plant which has the following discrete state-space representation:

$$x(k+1) = Ax(k) + Bu(k) + Fv \quad (11a)$$

$$y(k) = Cx(k) + z \quad (11b)$$

where:

- $v$  is the process noise vector, which may arise due to modelling errors such as neglecting non-linearities or higher frequency dynamics;
- $z$  is the sensor measurement noise vector arising from sensor imperfections.
- $F$  is the stochastic noise coefficient matrix, which is also being estimated by the system identification algorithm.

The state-equation of the Kalman filter is written as follows:

$$x_e(k+1) = Ax_e(k) + Bu(k) + L[y(k) - Cx_e(k)] \quad (12)$$

where  $L$  is the Kalman filter gain matrix. The Kalman filter is formed by minimising the covariance of the estimation error,  $R_e = E[e(k), e^T(k)]$ . The estimation covariance matrix  $R_e$  can be evaluated by solving the following algebraic Riccati equation:

$$0 = A.R_e + R_e.A^T - R_e.C^T.Z^{-1}.C.R_e + F.V.F^T \quad (13)$$

where  $A$ ,  $F$  and  $C$  are the plants state coefficient matrices,  $V$  is the process noise spectral density matrix, and  $Z$  is the measurement noise spectral density matrix. The Kalman filter gain matrix is calculated as follows:

$$L = R_e.C^T.Z^{-1} \quad (14)$$

### 3.3. Optimal linear quadratic Gaussian compensator design

Having used the so-called principle of separation to design the optimal tracking controller and the Kalman filter, the two are combined to form an optimal compensator for the plant. The final state-space realisation of the optimal compensator for tracking a reference signal is given by:

$$x_e(k+1) = A - BK_x - LC + LK_x)x_e(k) + Ly(k) \quad (15a)$$

$$u(k) = -K_x x_e(k) - K_i i(k) + K_r r(k) \quad (15b)$$

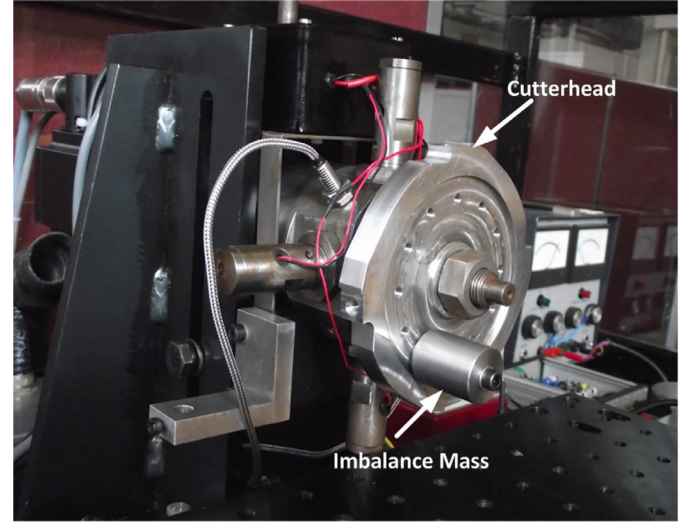
The feedforward gain,  $K_r$ , the controller weighting matrices,  $Q$  and  $R$ , and the Kalman filters spectral noise densities,  $V$  and  $Z$ , are the tuneable design parameters for the LQG tracking compensator. Hence, the desired closed-loop systems performance is obtained by suitably selecting the values of  $K_r$ ,  $Q$ ,  $R$ ,  $V$  and  $Z$ .

### 3.4. Vibration disturbance feedforward compensation

The feedforward loop requires knowledge of the signature of the vibration disturbance in advance. The procedure used to gather that information is presented in this section. Some tests have been performed in order to estimate the amount of vibration caused by imbalance forces in the system. The spindle is rotated at different speeds and its vibration in the vertical plane is measured using

**Table 2**  
Spindle vibration with and without the cutterhead.

Speed (RPM)	Amplitude ( $\mu\text{m}$ ) Spindle Only	Amplitude ( $\mu\text{m}$ ) Spindle + Cutterhead
600	0.31	0.31
1000	0.32	0.33
1500	0.35	0.36
2000	0.36	0.37
3000	0.38	0.39
4000	0.41	0.42



**Fig. 3.** Cutterhead and the attached imbalance mass of 116g.

the eddy current sensors. The vibration amplitudes are shown in Table 2.

The results in Table 2 show that there is no significant vibration in the system. This is desirable for a good wood surface finish. However, vibration is needed in order to test the performance of the control system. Vibration disturbance is introduced into the test rig by attaching a mass weighing 116g to the cutterhead at a radius of 35 mm from its centre (Fig. 3). The spindle is rotated at the desired speed and its vertical vibration is measured using the eddy current sensors. This is done without any inputs from the piezoelectric actuators so that any displacement away from equilibrium would be due primarily to the mass imbalance or other sources of vibration acting against the stiffness of the piezoelectric actuator and flexural hinges. While sampling the eddy current sensors, the angular position of the spindle is also measured using the rotary encoder mounted on the spindle. This provides information about the position of the resultant mass imbalance.

A sample plot of the spindle displacement against its angular position is shown in Fig. 4. The signals were acquired using the eddy current sensors and the rotary encoder for duration of 60s while rotating the spindle at the speed of 1500rpm. The vibration amplitude is estimated by computing the average of the positive peaks. The angular position of the spindle at the positive peaks, which is the position of the resultant mass imbalance, is also determined through averaging. The spindle vibration is re-generated in real-time using the following equation:

$$y(t) = A \cos[\theta(t) - \theta_a] \quad (16)$$

where:

- $y(t)$  is the instantaneous vertical position of the spindle ( $\mu\text{m}$ )
- $A$  is the estimated vibration amplitude ( $\mu\text{m}$ )
- $\theta(t)$  is the instantaneous angular position of the spindle (rad)

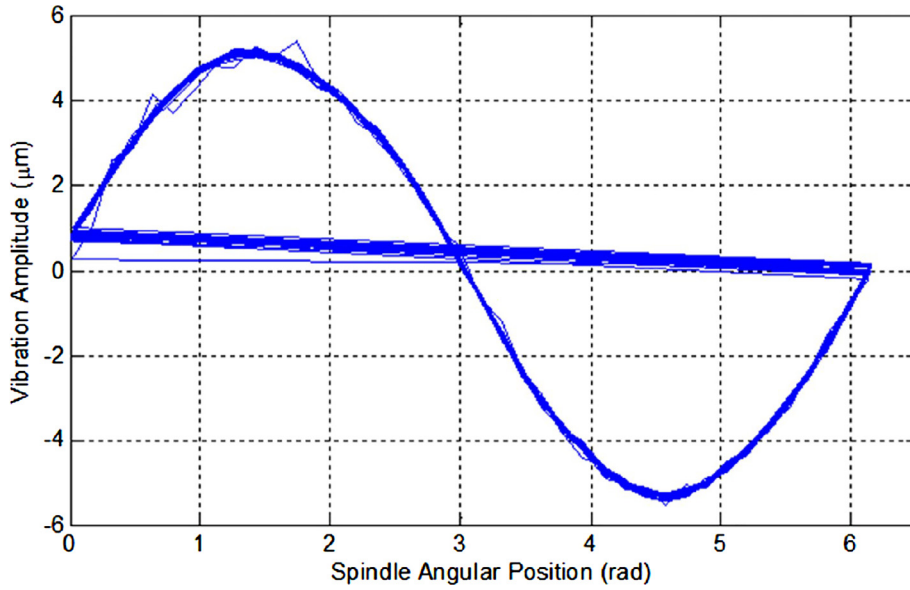


Fig. 4. Graph of displacement against angular positions of the spindle.

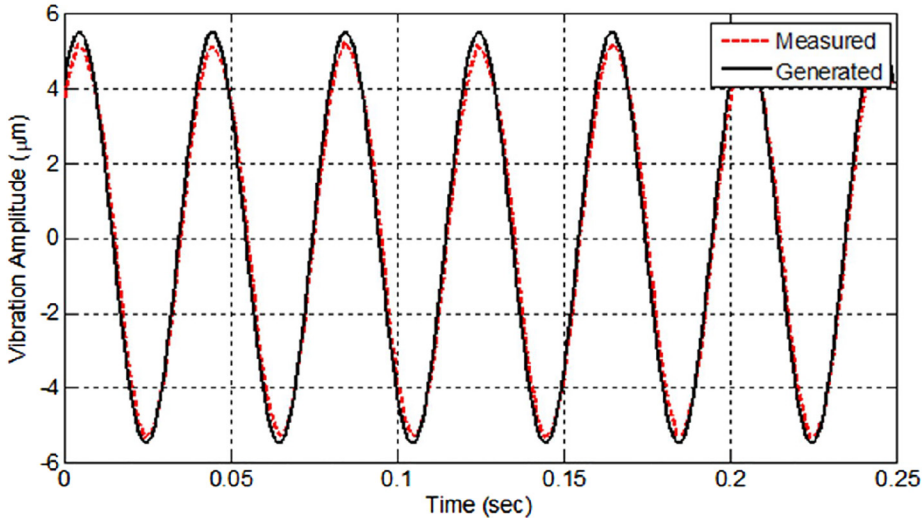


Fig. 5. Real-time measured and re-generated spindle vibrations.

- $\theta_a$  is the estimated angular position of the resultant imbalance mass (rad)

A sample plot showing a real-time measured vibration and the re-generated vibration of the spindle is shown in Fig 5. It should be noted that the vibration amplitude needs to be determined for each rotational speed. The position of the mass imbalance may also need to be re-determined if the cutterhead is removed and then mounted back on the spindle. The vibration signal needs to be converted to the equivalent piezoelectric actuator voltage. The static gain of the piezoelectric actuators is used to calculate the vibration disturbance input. The gain is determined from the voltage-displacement relationship between the piezoelectric actuators and the spindle. The equation is given as follows:

$$U_d(t) = y(t)/G_s \tag{17}$$

where:

- $u_d(t)$  is the disturbance input (V)
- $G_s$  is the static gain of the piezoelectric actuator ( $\mu m/V$ )

The objective is to introduce a signal which is 180° out of phase with the process disturbance vibration. The final control law is ob-

tained as follows:

$$u(k) = -K_x x(k) - K_i i(k) - K_r r(k) - u_d(t) \tag{18}$$

#### 4. Experimental tests and results

The controller is designed in Simulink and Stateflow on a host computer. A real-time rapid prototyping environment from Mathworks known as xPC Target is used to control the wood planing machine. The xPC Target enables direct execution of the Simulink and Stateflow models on a target computer for real-time testing. The xPC Target computer is equipped with a data acquisition board through which it takes spindle orbit and angle of rotation data from the planing machine, and also applies control signals to the piezoelectric actuators. The testing environment is shown in Fig. 6

##### 4.1. System identification

The LQG feedback controller requires a state-space mathematical model of the plant. Experimental system identification technique is used to obtain the model of the test rig because it is

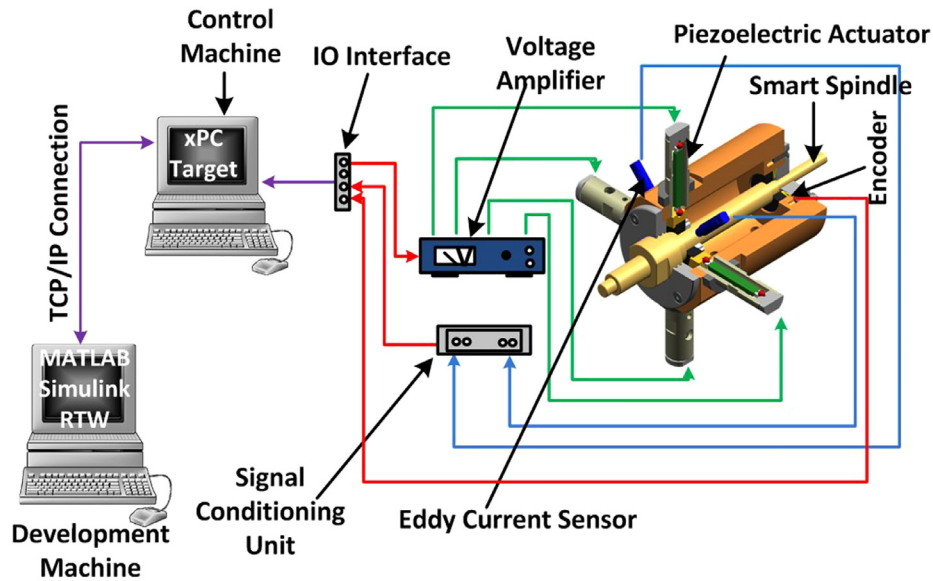


Fig. 6. Schematic of the active wood machining control environment.

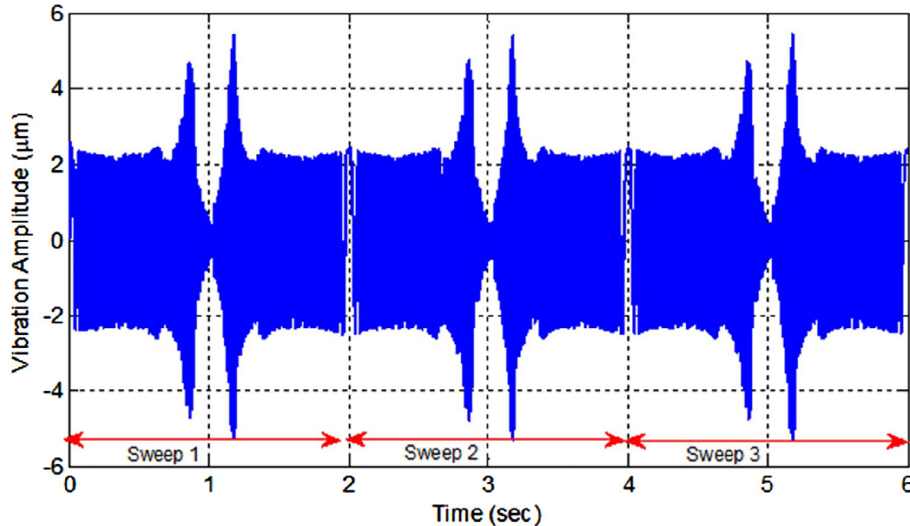


Fig. 7. Time-domain response of the spindle to a sine-sweep.

based on observed input-output data from the system [22,23] and it does not require the details of the physical principles guiding the behaviour of the machine. Also, the parameters of the test rig components such as damping, spindle dimensions, actuator stiffness, and amplifier impedance and the underlying mathematical relationships are not required. Frequency response testing is used to determine the dynamic characteristics of the test rig and also generate data for the system identification procedure.

A sine-sweep signal of 25V ranging from 0 to 1 kHz at the rate of 0.5 kHz/s is applied to the piezoelectric actuators on the vertical axis of the spindle (B and D in Fig. 2), and the acquired spindle response is shown in Fig. 7. MATLAB System Identification Toolbox is used to estimate the state-space model matrices from the measured input-output data. Since the experimental system identification approach does not assume any prior first principles model of the plant, the entire procedure can be repeated whenever required.

The identification of the spindle unit in the horizontal direction is not that critical because the cutting tool error compensation and the vibration control system are implemented on the vertical axis only. The system identification was carried out while the spindle

was stationary since it has been demonstrated in [24] that rotation does not generally have any significant influence on the dynamics of a spindle.

#### 4.2. Controller performance test

The tests performed in order to determine the performance of the controller were in three stages.

**Case 1 Cutting tool inaccuracy compensation** The spindle was made to track an arbitrary reference signal. The signal is the one required to compensate for inaccuracy in a four-knife cutterhead having three proud knives. The spindle speed was increased from 1000 rpm to 4000 rpm without the imbalance mass attached to the cutterhead, so there was no vibration disturbance acting on the spindle. The comparison between the reference spindle position and the actual spindle position at 3000rpm is shown in Fig. 8. The efficiency of the controller is measured based on the objective of the active machining system, which is to keep the spindle at desired vertical positions when the cutters are within the angular positions where they produce visible cuttermarks on the



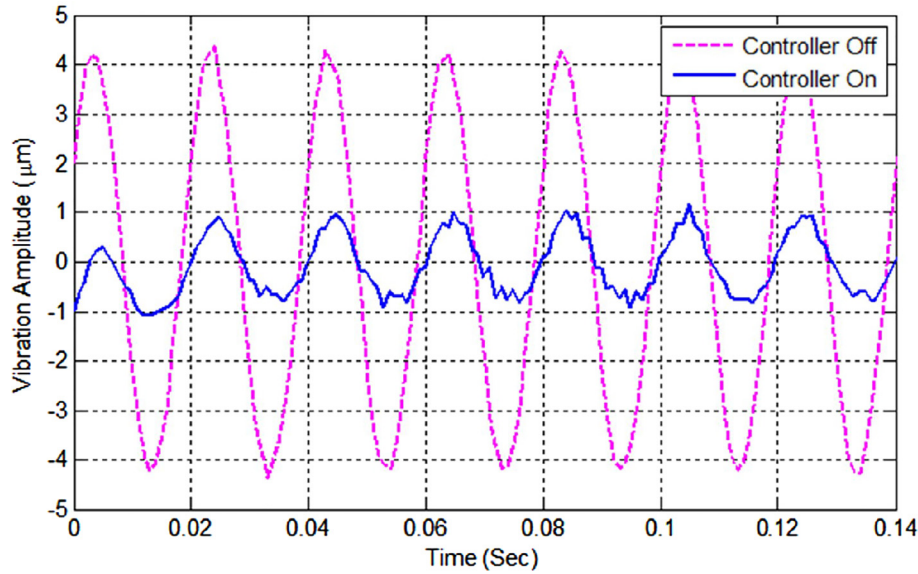


Fig. 9. Controlled and uncontrolled vibration of the spindle at 3000 rpm.

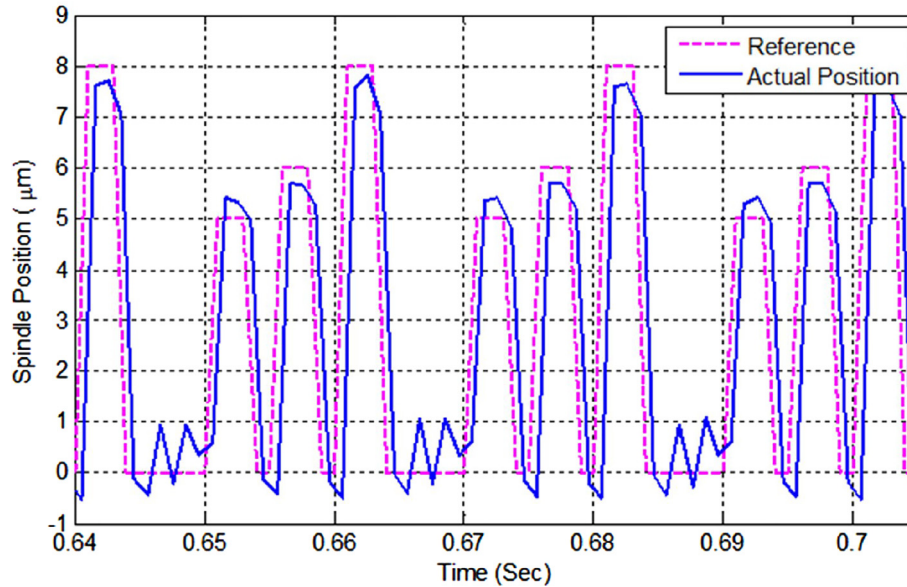


Fig. 10. Tracking response of the spindle with imbalance mass at 3000 rpm

It is expected that the enhanced dynamic performance of the spindle would lead to improved surface quality provided that the effects of other factors such as the timber material properties are negligible.

#### 4.3. Surface finish quality evaluation

In order to evaluate the effects of the cutterhead displacements on surface finish quality, some wood samples were machined with and without compensations. Firstly, a sample was machined without any compensation at the cutting speed of 1000rpm and feed speed of 133mm/s. These parameters will produce a 2mm waviness pitch for a four-knife cutterhead [3,5]. The machining was done without the mass attached to the cutterhead so vibration disturbance effect is negligible. The obtained surface profile measured using the alicona InfiniteFocus instrument is shown in Fig. 11.

Another wood sample was machined using the same cutting parameters but with the compensation applied. Cutter two and

three were displaced downward by  $12\mu\text{m}$  and  $2\mu\text{m}$  respectively, and cutter four was displaced upward by  $1\mu\text{m}$ . The surface profile obtained is shown in Fig. 14.

As seen in Fig. 11, the surface profile is a three-knife finish instead of a four-knife finish due to the  $13\mu\text{m}$  TIR of the cutterhead. This means that the cuts made by the shortest knife have been completely removed by the other longer knives. Although there are some variable differences in the heights of corresponding cuttermarks from one revolution to the other, the overall waviness pattern is in close agreement with the static run-out measurements. The four-knife finish in Fig. 12 was obtained because the compensation was able to minimise the effects of the cutting tool inaccuracy. The standard deviation of the widths of the cuttermarks is used as the quality assessment parameter. The standard deviation is calculated as follows [10]:

$$\sigma = \sqrt{\left(\frac{1}{N} \sum_{i=1}^N (S_i - \bar{S})^2\right)} \quad (20)$$



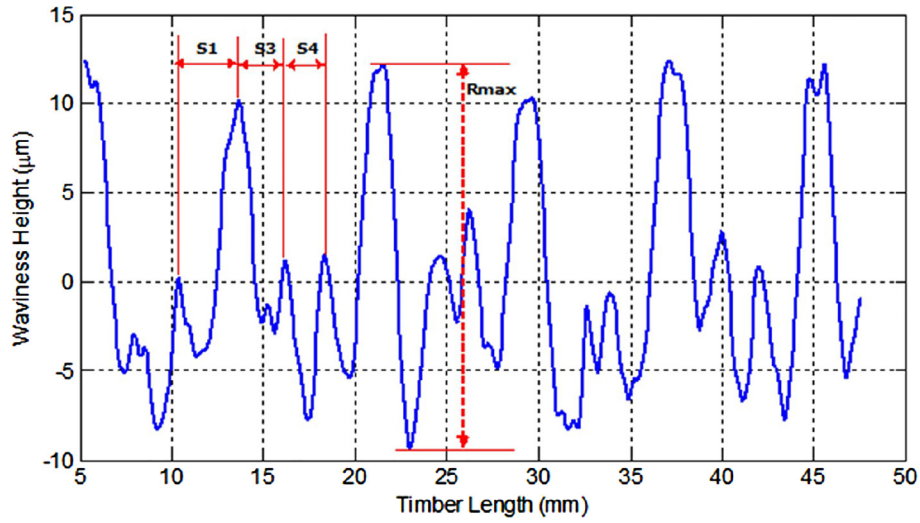


Fig. 11. Surface profile of a sample machined without compensation.

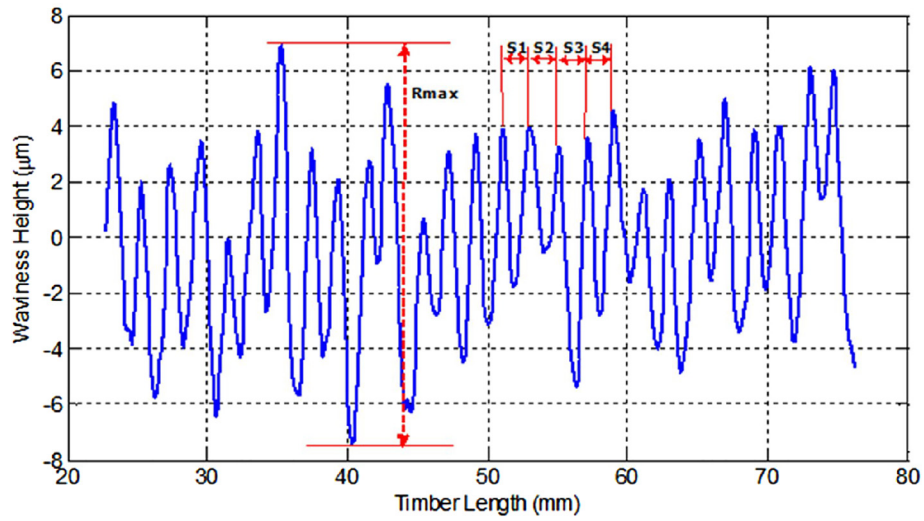


Fig. 12. Surface profile of a sample machined with cutting tool inaccuracy compensation.

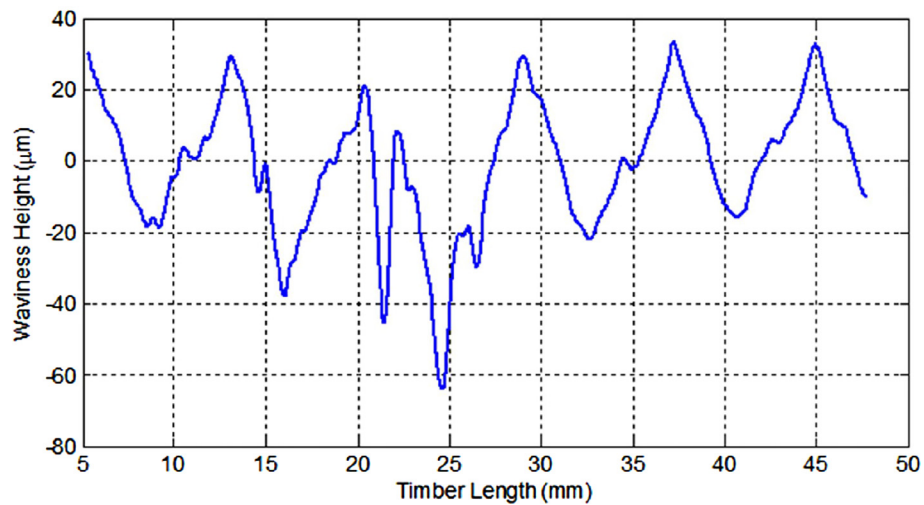


Fig. 13. Surface profile of a sample machined under vibration disturbance but without compensation.

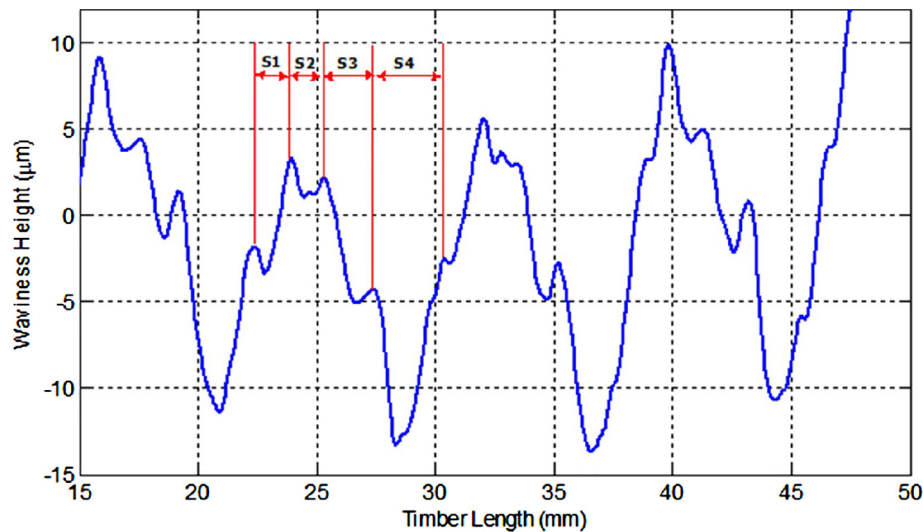


Fig. 14. Surface profile of a sample machined under vibration disturbance but without compensation.

Table 6

Surface finish quality evaluation for cutting tool inaccuracy compensation.

Surface finish	Rmax ( $\mu\text{m}$ )	Cuttermarks Width (mm)				$\sigma$ (mm)
		S1	S2	S3	S4	
Uncompensated surface finish	20.95	3.37	0.00	2.50	2.12	1.26
Compensated surface finish	13.82	1.90	2.15	2.08	1.88	0.13

where:

- $N$  is the number of cuttermarks used
- $\bar{S}$  is the desired waviness pitch (2 mm)
- $S_i$  is the width of the  $i$ th cuttermark

The surface finish with the lower standard deviation has a higher quality because the widths of the individual cuttermarks are close to the desired pitch. The surface finish quality evaluation data for Fig. 11 and Fig. 12 are shown in Table 6. Based on these values, the compensated surface finish has a higher quality rating than the uncompensated surface.

A set of cutting tests were also performed with an imbalance mass of 4060g-mm attached to the cutterhead at the same angular position as knife four. The spindle was rotated at the speed of 1500rpm and the imbalance mass produce a forced vibration of about  $7\mu\text{m}$  amplitude at the cutterhead position. This effectively increased the dynamic TIR of the cutterhead from  $13\mu\text{m}$  to  $27\mu\text{m}$ .

A wood sample was machined without any compensation at the feed speed of 200mm/s in order to aim at 2mm waviness pitch. The resulting surface profile is shown in Fig. 13. Although the cuttermarks appear to be relatively uniform, the waviness pitch is about 8mm instead of the desired 2mm. This indicates that the profile is a single-knife finish. Based on the grading system used in the woodworking industry, cuttermarks pitch greater than 2.5mm is a low quality finish.

Another wood sample was machined under the same conditions but the cutterhead inaccuracy and vibration compensation was applied during the machining operation. The resulting surface profile is shown in Fig. 14.

As seen in Fig. 14, the surface profile is a four-knife finish having a once per revolution vibration defect condition [3]. The applied compensation reduced the vibration amplitude to  $3\mu\text{m}$  based on the measured outputs of the eddy current.

## 5. Conclusion

This paper presents an active method used to compensate for the effects of cutting tool inaccuracy and spindle vibration in rotary wood planing. The technique is based on real-time periodic optimisation of the cutting tool trajectory coupled with the rejection of the vibration disturbance acting on the spindle. A control system that is based on optimal LQG controller and feedforward disturbance rejection was used. Experiments were performed on a mechatronic wood planing machine in order to evaluate its dynamic performance and also determine the effectiveness of the technique on the quality of the surface finish. The results show that the technique provides up to 75% increase in the dynamic performance of the machine. Also, the surface profiles obtained from the compensated machining operations have higher quality than the ones machined without compensation.

## Acknowledgement

The authors acknowledge support from the EPSRC Centre for Innovative Manufacturing in Intelligent Automation in undertaking this research work under grant reference number EP/IO33467/1.

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