# Feedback Compensation of Tool Deflection in a High-Speed Machining AMB Spindle

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***Abstract*—This paper describes a method for compensation of tool tip deflection in a high-speed, magnetically levitated, machining spindle. The proposed method utilizes measurements from the AMB sensors to estimate forces and calculate the resulting tool deflections. The deflections are then compensated online with a feedback signal that changes the loaded tool tip's position. The modeled results show significant performance improvements, which are experimentally validated.**

## I. Introduction

High-speed machining is an emerging technology in industry, with advantages such as faster material removal rate and less tool wear. With increased rotational speeds, traditional rolling element bearings inherently impose limitations and require constant maintenance. To overcome these issues, active magnetic bearings (AMBs) are employed.

The AMB is a device that allows for levitation of a ferromagnetic body using electromagnetic forces. In this way, mechanical friction is absent and there is no wear of components. The system is also lubrication free and suitable for clean environments. In addition, for machining application, AMBs allow for a larger rotor diameter, thus increasing the rotor's stiffness.

The AMB system is naturally unstable as electromagnets can provide only attractive forces. To stabilize it, active feedback control is used. Actively controlled systems provide additional benefits such as programmable stiffness and damping characteristics for specific frequencies. In that way, lower vibration can be achieved at the required cutting speeds.

One of the possible machining applications is a boring operation. Boring requires a long and slender rotor for holding the tool bit inside the workpiece cavity. Therefore, a boring setup is a flexible system. When boring, the radial force acting on the tool causes its deflection and deviation from the desired tool path. Increasing the cutting speed to the high-speed region reduces the force without sacrificing the material removal rate but cannot totally eliminate the error.

The issue of compensating tool deflection has received significant attention in the literature. For example, it was considered for miniature ball mills in [1] in which the authors used an experimentally tuned model for offline compensation of the deflection. The offline compensation approach with trajectory correction for the flexible tools is also presented in [2, 3] in which force measurements are used to calibrate and estimate tool forces for the specific cutting conditions. This requires additional hardware and extensive experiments before actual machining. It was demonstrated by Auchet et al. [4] that it is possible to measure cutting forces with magnetic bearings.

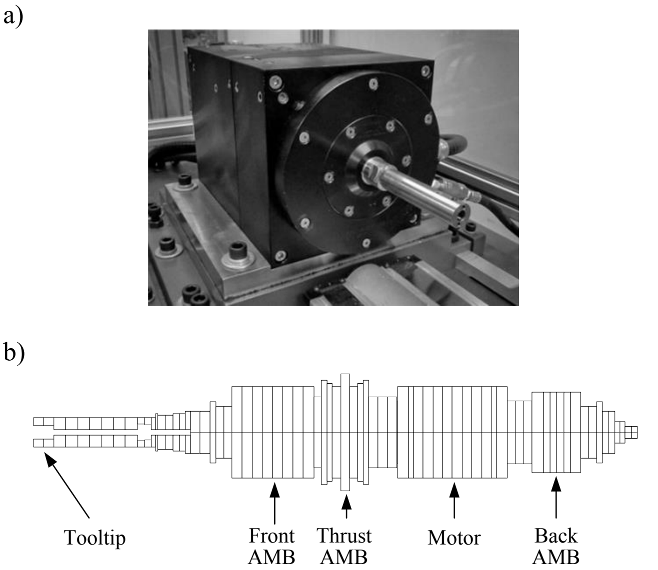


Figure 1. AMB Spindle a) photograph and b) finite element discretization.

Using AMBs, no additional hardware is required and measurements are available in real time.

This article presents a method to compensate tool deflection based on available current and rotor position measurements from the AMBs. To eliminate error due to tool tip deflection, the tool is shifted according to the estimated deflection. The shifting is achieved by using the bearings' airgap with various methods such as the one developed by Eckhardt and Rudolf [5].

## II. System Description

The system used for experimental validation of the tool compensation method is presented in Fig. 1. The photograph is shown at the top and the corresponding rotor finite element model (FEM) is shown at the bottom. This system is an industrial grade high-speed machining spindle for single point boring, levitated with AMBs. The maximum speed of 50,000 rpm is provided by a 10 kW induction electrical motor whose rotor is fully levitated by two radial and one axial magnetic bearing. The static capacity for the front, back and axial bearings is 1400 N, 600 N, and 500 N, respectively.

For research, the industrial system is augmented to be controlled by a dSPACE rapid control prototyper. It includes DS1005 PPC processor board, DS2001 A/D and DS2101 D/A

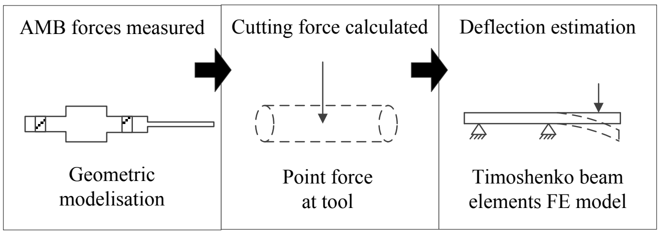


Figure 2. Estimation of the tool deflection

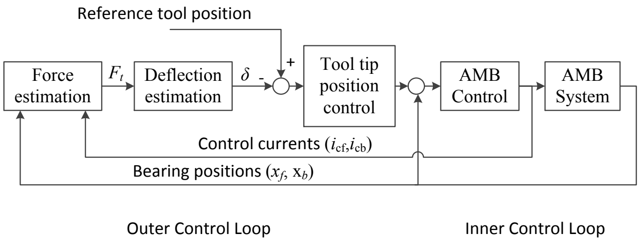


Figure 3. Compensation of the tool deflection

converters. The levitation and compensation control studied is implemented with sampling time 12.5 kHz.

The linear model of the actuator is obtained with bias current linearization with radial AMB force $f$ characterized as

$$f = k_x x + k_i i, \tag{1}$$

where $k_x$ is position stiffness, $k_i$ is current stiffness, $i$ is control current, and $x$ is displacement from the center point. The stiffness values are experimentally identified by levitating the system and applying a set of static loads.

The rotor is discretized and modeled with the FEM. The obtained model is truncated to include only the first three flexural modes and tuned to the measured frequency response [6]. The frequency response of the open-loop system is obtained by exciting it with a sinusoidal current signal, one axis at a time, and measuring positions at all AMB sensors. Finally, the full response is obtained as described by [7]. The obtained rotor model accurately represents the system, with the first natural frequency of 1070 Hz. This frequency is higher than the maximum rotational frequency, thus the system is subcritical.

## III. DEFLECTION COMPENSATION

In the previously discussed works [1- 3] the tool deflection is compensated by changing the tool path before the cutting process. To get an accurate estimate of the forces acting on the rotor, preliminary experimental measurements were used with specific conditions and particular material. With magnetic bearings, the forces can be estimated online [4] and that information is used to compensate for the tool deflection.

This work assumes that the reduced rotor model accurately represents the system. Thus, the contribution of the flexible modes above the third one is negligible. Another assumption is that the only external disturbance acting on the rotor is the cutting force which is applied to the tool tip.

The deflection is estimated using several modeling steps as proposed by Dépincé and Hascoët [2]. First, the geometric model of the system is used to find the relationship from tool load to reaction force at the bearing. Then, the corresponding cutting force is calculated based on the bearing forces. Finally, the deflection is estimated using the FE model of the flexible rotor. The general order is presented in Fig. 2. At the second step, the point force acting on the tool is estimated based on AMB control currents and position measurements with the use of Eq. (1).

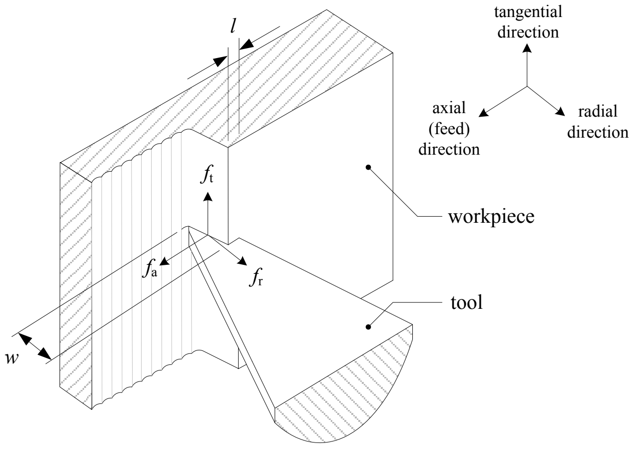


Figure 4. Diagram of the cutting tool imbedded in the workpiece while machining

With the knowledge of tool deflection, the rotor position is corrected. The correction is done based on the rigid body motion of the rotor by shifting the AMB setpoints. That correction causes an increase in the acting force, which is sensed through control currents and bearing positions. In that way, closed-loop deflection compensation is achieved. The scheme describing the process is presented in Fig. 3. The inner control loop consists of a typical AMB stabilizing control. The outer control loop performs the tool deflection compensation.

### A. Rotor model

The flexible rotor model derived from the FEM is described as follows

$$\mathbf{M}\ddot{\mathbf{x}} + (\mathbf{D} + \Omega\mathbf{G})\mathbf{x} + \mathbf{K}\mathbf{x} = \mathbf{f}, \tag{2}$$

where $\mathbf{M}$, $\mathbf{D}$, $\mathbf{G}$, and $\mathbf{K}$ are the mass, damping, gyroscopic and stiffness matrices, $\mathbf{f}$ is a vector of external forces acting on the rotor, $\Omega$ is the rotational speed, and $\mathbf{x}$ is a vector of modal coordinates. Equation (2) is transformed to the state space form of an unconstrained, or free-free ($_{\text{ff}}$) rotor [6]

$$\begin{aligned} \dot{\mathbf{x}} &= \mathbf{A}_{\text{ff}}\mathbf{x} + \mathbf{B}_{\text{ff}}\mathbf{u}, \\ \mathbf{y} &= \mathbf{C}_{\text{ff}}\mathbf{x}. \end{aligned} \tag{3}$$

This model includes two rigid rotor modes and the first three flexible modes. When the initial conditions and external forces acting on the system are known, the model represented by Eq. (3) allows for estimation of the position of the center of mass and any other point of the rotor such as the tool location. The inclusion of flexible modes takes into account the deformation under load. Thus, the vector $\mathbf{u}$ contains the forces acting on the rotor and the corresponding points of force application are described by matrix $\mathbf{B}_{\text{ff}}$. The tool tip displacement is provided by applying transformation to the state vector with matrix $\mathbf{C}_{\text{ff}}$. To estimate rotor deflection, an accurate force description should be provided.

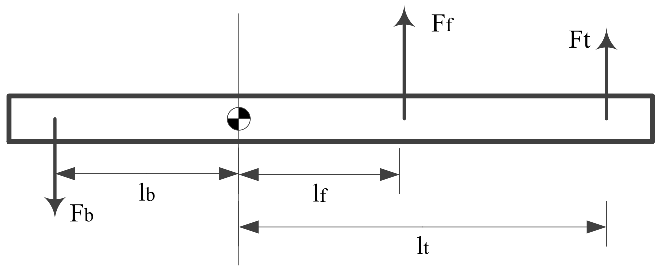


Figure 5. Scheme of forces acting on the rotor

*B. Forces acting on the rotor*

The rotor is supported by two bearings that counteract gravitational force and any disturbance force acting on the system. The gravitational force can be estimated with high precision by weighing the rotor before the assembly and then updating the value when a cutting tool and tool holder are added. The estimation of the center of mass can be done both experimentally and by FEM modelling of the rotor.

To estimate the forces of the magnetic bearings, the simple linearized representation (1) is used. This is a widely used model obtained from a nonlinear bearing description [8]. The displacement from the center point is provided by sensors that are present in most AMB systems and the control current is known from the control system.

As the bearing force model is linearized under the assumption of small displacements from the bearing magnetic center, it is important to keep the rotor near the bearing magnetic center during operation. In that way inaccuracies are minimized. There are several procedures described in the literature to position the rotor in the magnetic center, for example an iterative method [9].

The main issue of the article is cause by the radial cutting force at the tool tip. The system under consideration is a single point boring spindle, thus the machining force is decomposed into components in the directions axial, radial, and tangential to the rotor. The process of cutting and corresponding forces is illustrated in Fig. 4.

The axial component of the cutting force is addressed by the relatively simple dynamics of the thrust bearing. The tangential component of the cutting force is overcome by the motor. The radial component however may cause significant tool deflections perpendicular to the workpiece surface.

Without loss of generality only one plane is considered; the results are readily applied to the second plane by substituting the corresponding measurements (displacements and control currents). With the previously stated assumptions, the system under steady state conditions corresponds to the one described in Fig. 5. The balance of moments gives

$$\begin{aligned} &Fl_{\mathrm{ff}} + F_{\mathrm{b}} l_{\mathrm{b}} + F_{\mathrm{t}} l_{\mathrm{t}} = 0 \\ &F_{\mathrm{t}} = \frac{-F_{\mathrm{b}} l_{\mathrm{b}} - F_{\mathrm{f}} l_{\mathrm{f}}}{l_{\mathrm{t}}}, \end{aligned} \tag{4}$$

where $F_{\mathrm{f}}$, $F_{\mathrm{b}}$, and $F_{\mathrm{t}}$ are the forces of front bearing, back bearing, and tool tiprespectively , $l_{\mathrm{f}}$, $l_{\mathrm{b}}$, and $l_{\mathrm{t}}$ are distance from the center of mass to the front bearings, back bearing, and the tool tip respectively.

The mass of the rotor can be taken into account in two ways. The first is by including it as a term in Eq. (4). The second is by subtracting the gravitational component from the bearing forces. The latter approach is more useful for practical implementation when this component is readily available after initial levitation of the system. In addition, this approach simplifies the equations.

Considering that only control currents and sensor signals are available, the full force vector acting on the system is obtained by combining Eqs. (1) and (4)

$$\mathbf{u} = \begin{bmatrix} f_f \\ f_b \\ f_t \end{bmatrix} = \begin{bmatrix} k_{\mathrm{if}} & 0 & k_{\mathrm{xf}} & 0 \\ 0 & k_{\mathrm{ib}} & 0 & k_{\mathrm{xb}} \\ -\frac{l_{\mathrm{f}}}{l_{\mathrm{t}}} k_{\mathrm{if}} & \frac{l_{\mathrm{b}}}{l_{\mathrm{t}}} k_{\mathrm{ib}} & -\frac{l_{\mathrm{f}}}{l_{\mathrm{t}}} k_{\mathrm{xf}} & \frac{l_{\mathrm{b}}}{l_{\mathrm{t}}} k_{\mathrm{xb}} \end{bmatrix} \begin{bmatrix} i_{\mathrm{cf}} \\ i_{\mathrm{cb}} \\ x_f \\ x_b \end{bmatrix}, \tag{5}$$

where $i_{\mathrm{cf}}$ and $i_{\mathrm{cb}}$ are currents and $x_{\mathrm{f}}$ and $x_{\mathrm{b}}$ are displacements in front and back bearings respectively. The obtained force vector in Eq. (5) is directly substituted into Eq. (3) yielding the tool deflection as

$$\begin{aligned} &\dot{\mathbf{x}} = (\mathbf{A}_{\mathrm{ff}} + \mathbf{B}_{\mathrm{act}} \mathbf{K}_{\mathrm{x}} \mathbf{C}_{\mathrm{act}}) \mathbf{x} + \mathbf{B}_{\mathrm{act}} \mathbf{K}_i \mathbf{I}_{\mathrm{c}} + \mathbf{B}_{\mathrm{t}} F_t, \\ &\delta = \mathbf{C}_{\mathrm{t}} \mathbf{x}, \end{aligned} \tag{6}$$

where $\mathbf{B}_{\mathrm{act}}$, $\mathbf{C}_{\mathrm{act}}$, $\mathbf{B}_{\mathrm{t}}$, and $\mathbf{C}_{\mathrm{t}}$ are the components of the $\mathbf{B}_{\mathrm{ff}}$ and $\mathbf{C}_{\mathrm{ff}}$ matrices which correspond to the magnetic actuator ($_{\mathrm{act}}$) and tool ($_{\mathrm{t}}$) locations and $\delta$ is the tool position. The rest of the matrices are defined as

$$\mathbf{K}_{\mathrm{x}} = \begin{bmatrix} k_{\mathrm{xf}} & 0 \\ 0 & k_{\mathrm{xb}} \end{bmatrix}, \quad \mathbf{K}_{\mathrm{i}} = \begin{bmatrix} k_{\mathrm{if}} & 0 \\ 0 & k_{\mathrm{ib}} \end{bmatrix}, \quad \mathbf{I}_{\mathrm{c}} = \begin{bmatrix} i_{\mathrm{cf}} \\ i_{\mathrm{cb}} \end{bmatrix}.$$

The tool force $F_{\mathrm{t}}$ is obtained by substituting Eq. (1) into Eq. (4) resulting in

$$F_t = \left( -\frac{l_{\mathrm{f}}}{l_{\mathrm{t}}} k_{\mathrm{if}} \right) i_{\mathrm{cf}} + \left( \frac{l_{\mathrm{b}}}{l_{\mathrm{t}}} k_{\mathrm{ib}} \right) i_{\mathrm{cb}} + \left( -\frac{l_{\mathrm{f}}}{l_{\mathrm{t}}} k_{\mathrm{xf}} \right) x_f + \left( \frac{l_{\mathrm{b}}}{l_{\mathrm{t}}} k_{\mathrm{xb}} \right) x_b.$$

As dynamics is neglected the estimation is done at steady state as follows

$$\delta = \mathbf{C}_{\mathrm{t}} (s\mathbf{I} - \mathbf{A}_{\mathrm{ff}} - \mathbf{B}_{\mathrm{act}} \mathbf{K}_{\mathrm{x}} \mathbf{C}_{\mathrm{act}})^{-1} \left( \mathbf{B}_{\mathrm{act}} \mathbf{K}_i \mathbf{I}_{\mathrm{c}} + \mathbf{B}_{\mathrm{t}} F_{\mathrm{t}} \right)|_{s \to 0}, \tag{7}$$

where $s$ is a Laplace variable, and $\mathbf{I}$ is a unit matrix. Equation (7) yields to the transformation matrix, which is based on the assumptions discussed above, estimates the deflection of the tool under load.

There are several other sources of external forces acting on the rotor in high-speed systems. One is a magnetic pull from an electrical motor that appears when the active part of the rotor is shifted from the center point. Assuming that under small displacements, the resulting lateral motor force is small compared to the machining force and should not introduce significant error.

Another source of excitation is unbalance, which has a great importance in such applications. Although balancing requirements for machining spindles are high, the presence of residual unbalance cannot be avoided. However, unbalance has attracted a great amount of attention from the research community and there are techniques for estimation of these forces [10, 11]. Thus, they can be separated from the tool tip forces.

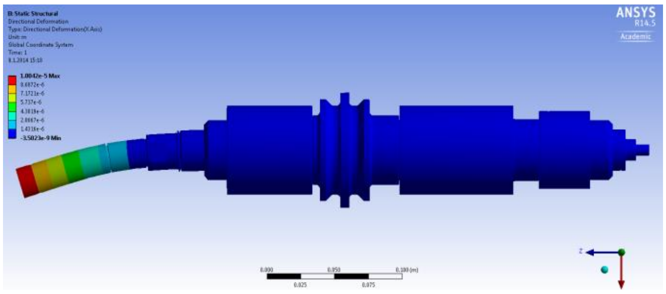


Figure 6. AMB spindle deformation under applied force

TABLE I
TOOL TIP POSITION MEASUREMENTS

| Tool reference, µm | Load | Compensation | Tool tip position, µm |
|---|---|---|---|
| 0 | NO | NO | 0 |
| | YES | NO | -7.037 |
| | YES | YES | -3.077 |
| | NO | YES | 0.276 |
| 10 | NO | NO | 0 |
| | YES | NO | -7.472 |
| | YES | YES | -3.542 |
| | NO | YES | -0.369 |
| 25 | NO | NO | 0 |
| | YES | NO | -7.725 |
| | YES | YES | -3.822 |
| | NO | YES | 0.077 |
| 50 | NO | NO | 0 |
| | YES | NO | -6.809 |
| | YES | YES | -3.168 |
| | NO | YES | -0.490 |
| 100 | NO | NO | 0 |
| | YES | NO | -7.05 |
| | YES | YES | -3.825 |
| | NO | YES | -0.668 |

### *C. Tool tip control*

The solution provided by Eqs. (3) and (5) gives an estimation of the tool tip position with the given assumptions. These equations correspond to the blocks "Force estimation" and "Deflection estimation" in Fig. 3. To eliminate the difference between the actual and desired tool positions, a control block should be applied that is in addition to the typical AMB control system.

The main priority of the bearing control is to keep the system stable. Thus, to change the tool tip position, several approaches can be applied.

The first one is to offset the rotor equally in both bearings according to the estimated deflection. In that way, a rigid (translational) mode is used. The main benefit of this approach is that it is inherently built-in in AMB systems and requires only supplying commands to the existing control system. The drawback is that the rotor may be moved out from the linearization center point. Thus, the error is increased and stability of the system is decreased.

The second approach is to apply a control system designed to control the position of the tool tip [12]. In that way the drawbacks of the previous method are alleviated but some effort on applying another control system is necessary.

The benefit of the proposed tool tip deflection compensation approach is that it is linear and can be included

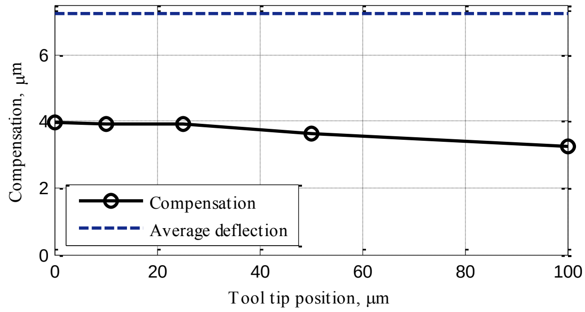


Figure 7. Compensation with respect to tool tip position

into any linear control system that is usually used for AMB control. The solution of Eq. (7), with some state space algebra according to scheme in Fig. 3, is combined with an existing controller into a single state space representation. Thus, the implementation of the proposed technique does not require any changes in control structure but only supplying new controller matrices.

## IV. EXPERIMENTAL RESULTS

To experimentally validate the results of deflection compensation, the outer feedback loop is implemented in dSPACE for the industrial high-speed spindle. To attenuate noise the signals for the tool deflection blocks are filtered with a low pass filter. In addition to the AMB sensors, two additional eddy current sensors are mounted at the tool tip location. The measurements from the additional sensors are only logged and they are not connected to the feedback control system.

To achieve constant force, an external load is applied to the rotor at the tool tip location. A load of 3.2 kg freely hanging weight is attached with a wire. This weight provides a static force with magnitude greater than the force expected under cutting conditions. The direction of the load corresponds to that of a control axis of the AMB system for convenient measurements.

The same experiment is modeled with ANSYS® FEM software. The system is simply supported at the bearing locations and a force is applied at the tool tip position. The results of FEM analysis are presented in Fig. 6, with a tool deflection value of 8.93 µm.

The experiment is performed in the following order. At first, the load is applied to the system and the position of the tool tip is measured. Next, the compensation for the deflection is applied and the tool tip position is again measured. Then, the load is removed, compensation is disabled, and the results are recorded. Finally, the compensation is enabled without the applied load and the results are measured. This type of experiment is repeated for several tool tip reference positions. The data is presented in Table I with normalization to the case without load and without compensation.

The total tool deflection can be estimated as the difference between cases without load and with load. For all experiments the average value is 7.219 µm. The error when compared to the ANSYS® results is about 25%, which is explained by the complicated nature of the rotor with shrink fits and AMB laminations. These complex elements and nonlinearities are not accounted for in the ANSYS FEM model. An additional source of error is that the proposed method neglects sensor/actuator non-collocation.

With compensation enabled the tool deflection is reduced on average by 3.732 μm. This value depends on the tool tip reference position as presented in Fig. 7. It is seen that with the greater deviation from the center point, the compensation degrades. This is the result of electromagnetic model linearization. The linearization with bias current is valid for small deviation and control currents less than bias current. As the rotor moves further from this point, the force estimations become less accurate. In general the difference between estimated and actual tool deflection comes from the rotor model simplification and measurement noise. The measurements are available only at the bearings locations and small deviations propagate significantly to the tool tip position.

There is an almost indistinguishable difference between the experiments without load. When compensation is turned on it tries to compensate some small amount, less than one micrometer. It is the result of deviation from the linearization point and gravitational force that bends the rotor.

## V. Conclusions and Further Work

In this article a method for compensation of tool deflection is presented for spindles supported on AMBs. The approach utilizes force estimation from magnetic bearings and calculates deflection with the reduced-order rotor model. The deflection is compensated for by correcting the rotor position at the bearing locations.

The experimental results demonstrated improvement in tool tip position under load by 52%. In addition, experiments showed a dependence on the model linearization which limits the accuracy at large displacements.

The proposed approach is simple to implement and can be done through appropriate replacement of the control law. Thus, no hardware and no software changes are necessary for the system. However, to obtain reasonable accuracy for industrial implementation, a full set of cutting experiments is necessary. In addition, the unbalance force should be separated from the cutting force, which requires on-line unbalance identification before cutting.

The future work will concentrate on implementation of nonlinear estimator allowing better accuracy for the full operating range. Another direction is toward increasing the bandwidth of the compensation and repeating experiments at some rotational speed with actual cutting.

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