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# SİMULTANEOUS TRACKING AND DRIVE-BASED VIBRATION SUPPRESSION OF CNC MACHINE TOOL FEED SYSTEMS

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## Keywords

*CNC feed systems, reference tracking, vibration suppression.*

## Abstract

*Computer numerical controlled (CNC) machine tools require higher performances due the continuous development of high-precision and high-speed capabilities. Objective in the control of the feed drive systems is to accurately track the specified reference trajectory for the position of the workpiece. However, cutting force disturbances or structural vibrations during the machining process may cause unacceptable tracking errors. Therefore, it is crucial to design a control system which ensures accurate tracking on reference trajectories and simultaneously suppressing undesirable vibrations. Escalating requirements on high acceleration and high speed leading to the larger sized and lightweight feed drive systems, also pose elevated risks about excitation of vibrations during the operation. Hence, machining performance depends on the dynamic properties and characteristics of the feed drive system as they constitute an important part of CNC machine tools. Transformation from rotary to linear motion using ball-screw feed drives are widely applied in machine tool industry for the time being due to their high mechanical efficiency and lower cost. However, during the high-speed machining process, the probable vibrations of the elastic ball-screw drive system dynamics will have an adverse affect on the machining accuracy of the machine tool and also on the stability of the control system. We establish an expedient lumped parameter model of the ball-screw feed drive system considering several axial and rotational degrees of freedom. Incorporating this model with the electrical drive and the position/vibration control system we build a complete simulation model. We show that our position/vibration controller with feedforward and feedback control can grant high positioning accuracy with simultaneous suppression of possible vibrations during machining.*

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# CNC TEZGÂH BESLEME SİSTEMLERİNDE EŞ ZAMANLI YÖRÜNGE TAKİBİ VE TAHRİK-TABANLI TİTREŞİM BASTIRMA

**Anahtar Kelimeler**

*CNC besleme sistemleri, yörünge takibi, tahrik-tabanlı titreşim bastırma.*

**Öz**

*Bilgisayar sayısal kumandalı (CNC) tezgâhlar, yüksek hassasiyetli ve yüksek hızlı yeteneklerin sürekli gelişimi nedeniyle daha yüksek performans gerektirir. Besleme tahrik sistemlerinin kontrolündeki amaç, iş parçasının konumu için belirlenen referans yörüngesini doğru şekilde izlemektir. Ancak, talaş kaldırma sırasında oluşan kesme kuvveti bozucuları veya yapısal titreşimler kabul edilemez izleme hatalarına yol açabilir. Bu nedenle, referans yörüngelerinde doğru izleme sağlayan ve aynı zamanda istenmeyen titreşimleri bastıran bir kontrol sistemi tasarlamak kritiktir. Yüksek ivme ve yüksek hız gereksinimlerindeki artış, daha büyük boyutlu ve hafif besleme tahrik sistemlerine yol açmakta ve çalışma sırasında titreşimlerin uyarılmasına ilişkin riskleri artırmaktadır. Dolayısıyla, talaşlı imalat performansı, CNC tezgâhlarının önemli bir parçasını oluşturan besleme tahrik sisteminin dinamik özelliklerine ve karakteristiklerine bağlıdır. Döner hareketin bilyalı vida mekanizmasıyla doğrusal harekete dönüştürülmesi, yüksek mekanik verimlilikleri ve düşük maliyetleri nedeniyle günümüzde takım tezgâhı endüstrisinde yaygın olarak kullanılmaktadır. Ancak, yüksek hızlı işleme sırasında bilyalı vida tahrik sisteminin elastik dinamiklerinden kaynaklanan olası titreşimler, hem makinenin işleme doğruluğu hem de kontrol sisteminin kararlılığı üzerinde olumsuz etki yaratabilir. Bu çalışmada, çeşitli eksenel ve dönel serbestlik derecelerini dikkate alan uygun bir yığılmış parametre modeli geliştirilmiştir. Bu modeli elektrik tahriki ve konum/titreşim kontrol sistemi ile birleştirerek tam bir simülasyon modeli oluşturduk. Konum/titreşim denetleyicimizin ileri besleme ve geri besleme denetimiyle, işleme sırasında oluşabilecek titreşimleri aynı anda bastırırken yüksek konumlama doğruluğu sağlayabildiğini gösteriyoruz.*

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## 1. INTRODUCTION

Machining technology with high speed has been extensively used in manufacturing industries to improve productivity (Schulz and Moriwaki, 1992). The development of high speed machining technology requires higher compliance for CNC machine tools. However, vibrations during high speed machining can reduce the dimensional accuracy and the surface finish of the workpiece, shorten the useful life of the cutters and reduce productivity. Therefore, vibrations of the CNC machine tools must be closely monitored and controlled (Altıntaş, Brecher, Pritschow and Uriarte, 2011). Dynamical performance of the feed drive system limits the machining efficiency and precision of CNC machine tools (Zhang J., Du, Zhang G. and Zhao, 2016). Due to their high transmission efficiency, high axial stiffness, and long life, ball-screw drive systems are widely used in CNC machine tools (Du, Feng, Li, Wang, Yu and Zhang, 2018). However, during the high-speed machining process, the probable vibrations of the ball-screw drive system will have an adverse affect on the machining accuracy of the machine tool and also on the stability of the control system. Moreover, high feed speeds in a limited screw stoke requires high accelerations, which will unavoidably increase the vibrations and decrease the machining accuracy (Altıntaş, Erkorkmaz and Zhu, 2000). Hence, it is necessary to explore the characteristics of vibration and appropriate vibration control techniques of the ball-screw feed drive systems to enhance the performance during machining.

Ball-screw feed drive systems' components are complex and elastic. To facilitate the analysis and control, the continuous mechanical systems with infinite degrees of freedom which are represented by partial differential equations are discretized into discrete systems with finite degrees of freedom, which results in lumped parameter models (Kim and Chung, 2006) or finite element models (Zaeh, Milberg and Oertli, 2004). Hybrid models may also be used (Pislaru, Ford and Holroyd, 2004). In general finite element models are more accurate than lumped models in determining the eigenmodes of the system. Usually, some simplifying assumptions are made in the process of modeling. Moreover, damping and stiffness parameters are not precisely known in advance hence, this uncertainty will have an adverse affect on the accurate control of the feed drive system. System identification and modal analysis techniques can be used to build more realistic models to determine the parameters and elastic dynamics of the feed drive system (Zaghbani and Songmene, 2009).

Due to the ease of use, easy adjustment of parameters and their high adaptability traditional controllers are widely used (Cho, Jeon and Le, 2009). As such, ignoring structural resonances that may be caused by flexible eigenmodes, many traditional controller designs are based on the ball-screw drive mechanical system's rigid body dynamics. Hence this approach limits the achievable bandwidth (Pritschow, 1998). The primary traditional control approaches are so-called damping method (Chung, Smith and Tlusty, 1997), pole placement method Jones and Ulsoy, 1999), input shaping method (Hyde and Seering, 2002), notch filter method (Smith, 1999), and active feedforward control method (Tung and Tomizuka, 1993). Tung and Tomizika (1993) used a zero phase error tracking controller (ZPETC) which is based on the elimination of all unstable poles and stable zeros. Erkorkmaz and Kamalzadeh (2006) propose an adaptive sliding mode controller (ASMC) based on the the ball-screw drive system rigid body dynamics. They designed a notch filter based on the torsional vibration information obtained from the end of the screw. Elastic modes of the ball-screw system are also considered in modern control methods to increase the bandwidth of the controller. Robustness is maintained by active feedback. Hence, better disturbance attenuation performance and wider closed-loop bandwidth can be attained, and the sensitivity of the closed-loop system to parameter variations can be diminished. Symens, Swevers and van Brussel (2004) considered time-varying stiffness and uncertain model parameters into account while modeling the ball-screw drive system and subsequent design of the controller. Authors used robust control method based on gain-scheduling, that can change the controller parameters based on the time-varying characteristics of vibration of the drive system. Fan S., Fan D., Hong and Zhang (2012) designed a tracking controller with a dual-loop composed of a robust disturbance observer and controller to solve vibration suppression and trajectory tracking problems. Hanifzadegan and Nagamune (2014) designed a robust linear parametrically varying (LPV) controller for different operating conditions of the system to work in a gain-scheduling strategy.

Controllers designed by modern control methods, in general, have high order and/or they are non-linear, and it is usually difficult to implement them in practical applications. In this paper we use traditional control methods for simultaneous trajectory tracking and vibration suppression of feed drive systems. We provide expedient lumped models for both electrical and mechanical parts of the drive system. Our mechanical model also embodies flexible modes of the drive so that the controller we design can suppress possible vibrations provided that they are within its bandwidth. We show that our position/vibration controller with feedforward and feedback control enables very high positioning accuracy with simultaneous vibration suppression during machining.

The paper is organized as follows: Modeling of electrical and mechanical parts of the drive system is presented in Section 2. Smooth trajectory generation to mitigate command excited vibrations and vibration suppression by the feed drive controller are in Section 3. Model based numerical simulation results are in Section 4. Finally we summarize the conclusions of our study in Section 5 and we provide suggestions for further research.

## 2. Electromechanical modeling of the feed drive system

The relevant components of a typical ball-screw assembly is shown in Figure 1. In this configuration, servomotor which is fixed to the machine with the bearings, transmit the torque onto the ball-screw shaft through an elastic coupling. Linear guideways are used to constrain the motion of the machine table in axial direction. Rotational motion is transformed into linear motion by the ball-screw transmission mechanism.

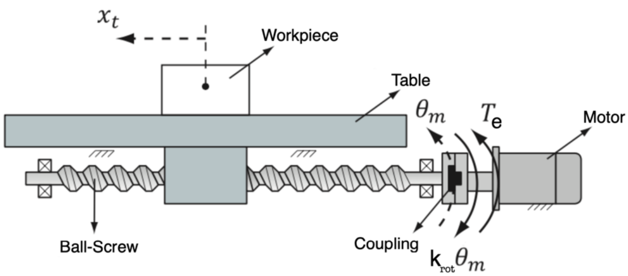


**Figure 1**. Ball-screw drive system

### 2.1 Electrical drive system

A servo motor operates as part of a closed loop feed drive system providing torque and velocity as commanded from a controller utilizing a feedback device to close the loop. Voltages  and  are received by the AC servo motor, which are then converted by the rotor and stator field interactions into torque. For each phase of the motor, Kirchhoff voltage law requires . Here,  is the magnetic flux linkage can be expressed as , where  is the current and  is the inductance. Due to the variation of inductance by rotation, Park transform is generally used to transform the equations into -frame which is rotating with the rotor. Equation for the Kirchhoff voltage law above can be transformed into the -frame applying the Park transform to flux linkages  and to stator currents , where  is the electrical rotor speed (Petruzella, 2010).

$$V_d = Ri_d + \frac{d\lambda_d}{dt} - \omega_e \lambda_q \qquad (1a)$$

$$V_q = Ri_q + \frac{d\lambda_q}{dt} + \omega_e \lambda_d \qquad (1b)$$

Note that the electrical speed  is related to the rotor's mechanical speed  through the equation , where  is the number of poles of the motor. Magnetic flux linkages on the  and -axes are related with the current  and the current  respectively.

$$\lambda_d = L_d i_d + \lambda_{PM} \qquad (2a)$$

$$\lambda_q = L_q i_q \qquad (2b)$$

Substituting Equations (2) into (1a) and (1b) respectively, gives

$$\frac{di_d}{dt} = \frac{1}{L_d}\left[V_d - Ri_d + \omega_e L_q i_q\right], \qquad (3a)$$

$$\frac{di_q}{dt} = \frac{1}{L_q}\left[V_q - Ri_q - \omega_e (L_d i_d + \lambda_{PM})\right]. \qquad (3b)$$

Motor torque is given by (Petruzella, 2010)

$$T_e = \frac{3}{2}\left(\frac{p}{2}\right)\left[\lambda_d i_q - \lambda_q i_d\right]. \tag{4}$$

Substituting Equations (3) into (4) and rearranging yields

$$T_e = \frac{3}{2}\left(\frac{p}{2}\right)\left[\left(L_d - L_q\right)i_d i_q + \lambda_{PM} i_q\right]. \tag{5}$$

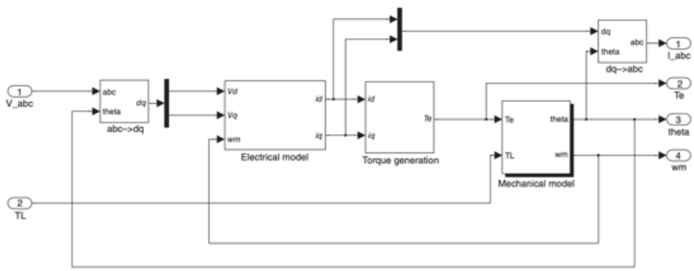


**Figure 2.** Three-phase AC servo motor Simulinkmodel

Model of the electric motor with the mechanical drive system dynamics which is derived in the next subsection is shown in Figure 2.

To vary the speed, the voltage and the frequency of the AC waveforms provided to the motor is adjusted by the AC motor drive. The pulse width modulation (PWM) method is used to transform DC to AC in the inverter section. AC drive also amplifies low level command signals generated by the controller to higher power signals (voltage/current) needed to operate the motor. Simulinkmodel of AC servo vector control is shown in Figure 3.

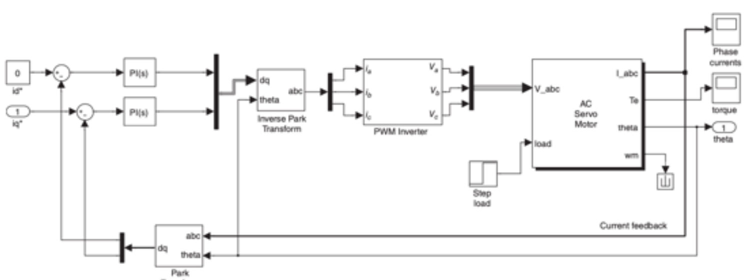


**Figure 3.** Simulinkmodel of AC servo vector control block given in Figure 2

## 2.2 Mechanical drive system

In general, a ball-screw drive mechanical dynamics can be characterized with flexural elastic, axial and rotational elastic eigenmodes. The excitation of the flexural elastic modes and their effect on the motion of the feeder greatly depends on the operating conditions on the drive system as well as the fabrication and mounting tolerances. The lumped parameter reduced degree of freedom model of ball screw feed drive system can simplify calculations and facilitate the controller design while keeping the few low-order modes of the system. Figure 4 shows a lumped parameter model for the ball-screw feed drive system.

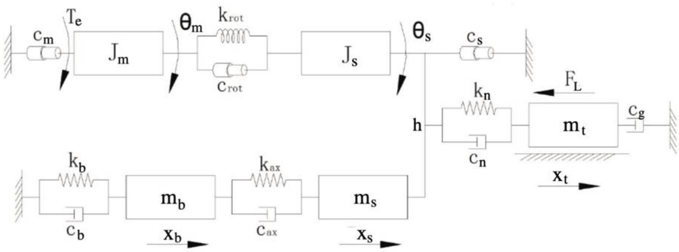


**Figure 4**. Lumped mechanical model for the feed drive system

Lumped parameter model explicitly includes the influence of the shaft on the axial and rotational modes of the feed drive system. Hence, the shaft is separated into a rotational and axial branch while the dynamical coupling is attained by the constraint equations. Stiffness of the bearing, stiffness of the coupling and the stiffness of the shaft can be combined to an overall axial stiffness k_ax and to an overall rotational stiffness k_rot values, since all components in the model are expressed by discrete springs and dampers. The parameters in the model are the servo motor torsional damping c_m, the inertia of servo motor $J_m$, the equivalent torsional stiffness $k_{rot}$, equivalent torsional damping c_rot, the screw shaft side total equivalent inertia $J_s$, the axial stiffness of base $k_b$, the axial damping of the base $c_b$, the mass of base $m_b$, the equivalent axial stiffness $k_{ax}$, the equivalent axial damping $c_{ax}$, screw shaft side total equivalent mass $m_s$, screw shaft side damping $c_s$, the stiffness of ball-screw nut $k_n$, ball-screw nut damping $c_n$, the mass of the work table $m_t$, and the axial damping of the guide $c_g$. Rotational motion of the screw shaft is transformed into the linear motion of the work-piece table is accomplished using a ball-screw mechanism. The transmission ratio of the ball-screw mechanism, $h=l/(2\pi)$, is defined as the traveling distance l of the work table for one revolution of the shaft.

Total equivalent inertia of the screw $J_s$ is comprised of the inertia of the coupling $J_c$, the inertia of the screw $J_{sw}$, and the mass of the table $m_t$ transformed to the rotational motion of the screw. Therefore $J_s=J_{sw}+J_c+h^2\, m_t$. Rotary inertia of the screw with its shaft length $l_{sw}$, equivalent diameter $d_{sw}$, and with material density ρ can be approximated by $J_{sw}=(\rho\pi/32)\, d^4_{sw}\, l_{sw}$.

Total equivalent mass of the screw m_s is composed of screw mass m_sw, coupling mass m_c, and the servomotor rotor mass m_m. Hence $m_s=m_m+m_c+m_{sw}$, where $m_{sw}=(\rho\pi/4)\, d^2_{sw}\, l_{sw}$.

Screw shaft uses a thrust bearing on the servomotor side to provide axial support. Hence, the axial stiffness $k_{ax}$ of the ball-screw feed system is comprised of thethrust bearing axial stiffness $k_{brg,}$ and the axial stiffness of the screw shaft $k_{swax}$ Therefore $k_{ax}=(1/k_{brg}+1/ksw_{ax})^{-1}$, where $ksw_{ax}=E\pi d^2_{sw}/4l_n$. Here E is the modulus of elasticity, and $l_n$ is the ball-screw length at table position.

The torsional stiffness $k_{rot}$ of the ball-screw feed system is comprised of the torsional stiffness of the screw $ksw_{rot}$ and the torsional stiffness of the coupling $k_c$. Therefore $k_{rot}=(1/k_c+ksw_{rot})^{-1}$. Torsional stiffness of the screw can be approximated by $ksw_{rot} = G\pi d^4_{sw}/32l_n$. Here G denotes the shear modulus of elasticity.

The contact stiffness $k_n$ of the screw nut can be expressed as $k_n=0.8K_R\,(F_L/0.1C_L)^{1/3}$ (Shimizu S., Sharma and Shimizu H., 2007). Here K_R is the nut reference rigidity, F_L is the axial load, and $C_L$ is the basic dynamic load.

Using Lagrangian formulation, equations of motion can be obtained from

$$\frac{d}{dt}\left(\frac{\partial T}{\partial \dot{\mathbf{q}}}\right) - \frac{\partial V}{\partial \mathbf{q}} + \frac{\partial R}{\partial \dot{\mathbf{q}}} = \mathbf{Q} \qquad (6)$$

where generalized coordinates $\mathbf{q}$ and generalized forces $\mathbf{Q}$ are $\mathbf{q} := (\theta_m\ \theta_s\ x_b\ x_s\ x_t)^{\mathrm{T}}$, and $\mathbf{Q} := (T_e\ 0\ 0\ 0\ -F_L)^{\mathrm{T}}$ respectively. Kinetic energy $T$, potential energy $V$, and the dissipation function $R$ of the system are

$$\begin{aligned}
T &= 1/2\, J_m\dot{\theta}_m^2 + 1/2\, J_s\dot{\theta}_s^2 + 1/2\, m_b\dot{x}_b^2 + 1/2\, m_s\dot{x}_s^2 + 1/2\, m_t\dot{x}_t^2 \\
V &= 1/2\, k_{rot}(\theta_m - \theta_s)^2 + 1/2\, k_b x_b^2 + 1/2\, k_{ax}(x_s - x_b)^2 + 1/2\, k_n(x_t - x_s - h\theta_s)^2 \\
R &= 1/2\, c_m\dot{\theta}_m^2 + 1/2\, c_{rot}\left(\dot{\theta}_m - \dot{\theta}_s\right)^2 + 1/2\, c_s\dot{\theta}_s^2 + 1/2\, c_b\dot{x}_b^2 + 1/2\, c_{ax}(\dot{x}_s - \dot{x}_b)^2 \\
&\quad + 1/2\, c_n\left(\dot{x}_t - \dot{x}_s - h\dot{\theta}_s\right)^2 + 1/2\, c_t\dot{x}_t^2. \qquad (7)
\end{aligned}$$

Substituting Equations (7) into Equation (6), equations of motion for the lumped mass mechanical model of ball screw drive system can be expressed in the form

$$M\ddot{\mathbf{q}} + C\dot{\mathbf{q}} + K\mathbf{q} = \mathbf{Q} \qquad (8)$$

where matrices $M$, $C$, and $K$ are

$$M = \mathrm{diag}(J_m, J_s, m_b, m_s, m_t)$$

$$C = \begin{pmatrix} c_m + c_{rot} & -c_{rot} & 0 & 0 & 0 \\ -c_{rot} & c_{rot} - c_s - h^2 c_n & 0 & -hc_n & hc_n \\ 0 & 0 & c_{ax} + c_b & -c_{ax} & 0 \\ 0 & hc_n & -c_{ax} & c_{ax} + c_n & -c_n \\ 0 & -hc_n & 0 & -c_n & c_n + c_g \end{pmatrix}$$

$$K = \begin{pmatrix} k_{rot} & -k_{rot} & 0 & 0 & 0 \\ -k_{rot} & k_{rot} - h^2 k_n & 0 & -hk_n & hk_n \\ 0 & 0 & k_{ax} + k_b & -k_{ax} & 0 \\ 0 & hk_n & -k_{ax} & k_{ax} + k_n & -k_n \\ 0 & -hk_n & 0 & -k_n & k_n \end{pmatrix}$$

## 3. Vibration suppression and trajectory tracking

Vibration suppression and trajectory tracking are two main goals for the control of high-speed machining feed drive systems. Vibration suppression is to mitigate the structural and feed-drive vibrations due to cutting force disturbance and due to fast changes of feed speed, where trajectory tracking requires high accuracy and fast speed for workpiece position tracking. There are several challenges to meet these requirements. First, high tracking speed is restricted to avoid the excitement of flexible modes of the of the ball screw structure. Second, the application of the cutting force close to one of the natural frequencies of the drive or structure will lead to impermissible vibrations. Third, the dynamical parameters of the system alter during the machining process due to variable table load and variable flexural stiffness of the screw shaft. Wear of the feed drive system components will be accelerated by the vibrations and it can lead to closed-loop instability if they are transmitted the servo control system. The discontinuities in the trajectory motion profiles that generate position, velocity, acceleration and jerk commands for the drive system may also excite the natural frequencies of the feed drive and/or machine tool structure. High reactive forces between the stationary parts of the machine and the moving mass caused by the drives with high acceleration and speeds can also lead to undesired transient vibrations. Furthermore, interaction between the servo controller, the current loop and feed drive's mechanical response may also cause vibrations. Hence the controller must be properly designed to cope with the instability and robustness problems.

### 3.1 Trajectory generation

The reference motion commands to feed drives with high acceleration and jerk creates a primary source for excitation of the machine tool and feed drive vibrations.

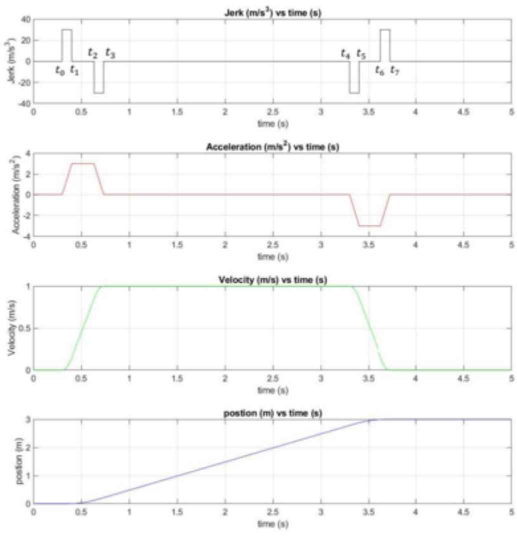


**Figure 5**. Third order trajectory planning

Moreover, If the trajectory has discontinuities due to commands at discrete control intervals, they will possess wide frequency content. Consequently, inertial forces due to high reaction forces between the moving mass and stationary parts of the machine may excite the resonant modes of the machine and cause undesired transient vibrations. Objective of machine tool trajectory tracking is to follow the reference motion precisely while avoiding the saturation of the drive motor and power amplifier. We assume that a trajectory is planned over a distance , given bounds on jerk, acceleration and velocity, and that, within these bounds, the trajectory is time optimal. For instance, the bound on jerk can be related to rise time behavior of a non-ideal power amplifier as commonly found in electromechanical systems generating the actuation force. Therefore, maximum current change is bounded resulting in a maximum change of actuation force and a maximum change on acceleration, hence a bound on jerk. A symmetrical trajectory can be determined by a constant jerk period, a constant acceleration period, and a constant velocity period. In our work bounds on jerk, acceleration, and velocity are 30 m/s, 3 m/s, and 1 m/s respectively for a third order displacement of 3 m. The resulting motion profiles are given in Figure 5.

### 3.2 Drive-based vibration suppression

Drive based vibration suppression proposed in Dietmair and Verl (2009) is a practical yet effective method to increase the damping ratio for a specific eigenmode in robotic arms and machine tool feed drives. It is based on modification of the velocity loop in a cascade control structure with a position loop to increase the damping ratio.

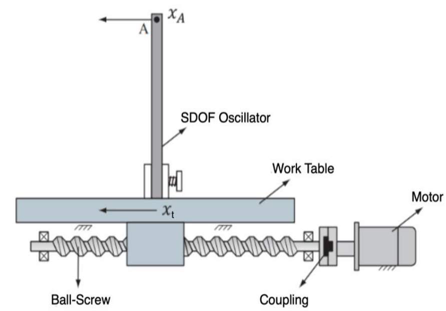


**Figure 6.** Oscillator mounted on a ball-screw drive

The vibrations due to inertial forces caused by the quick positioning of the work table or due to dynamic forces exerted by the machine tool on the workpiece can be modeled by a vibrating flexible beam mounted on the work table as shown on Figure 6. The dynamics of this bar can be expressed by

$$\frac{x_A(s)}{x_t(s)} = \frac{\omega_n^2}{s^2 + 2\zeta\omega_n s + \omega_n^2} \qquad (9)$$

in which  indicates the vibration at the tip point of the oscillator and  represents the work table displacement. Drive based vibration suppression concerns the algorithms which enhance the damping of a specific eigenmode which would lead to a vigorous vibration attenuation of the flexible screw shaft. Figure 7 (Hosseinabadi, 2013) shows the ball-screw drive system's block diagram in closed loop, when a cascade control law is implemented. Here  and  denote the equivalent mass and damping of the system as seen by the work table respectively.

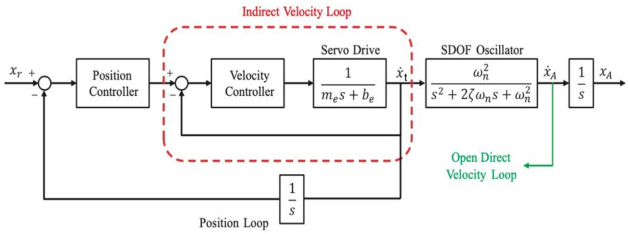


**Figure 7**. Cascade control structure

The indirect velocity loop uses the angular velocity of the motor measured by a shaft mounted rotary encoder (Altıntaş et al, 2011) to ensure a fast system response during the velocity command tracking. A rotary or linear encoder is utilized in position feed back for accurate positioning of the wok table . Oscillator's vibration can be included in the simulation model by combining its dynamics in the control loop simulation of the ball-screw drive system. In order to mitigate the vibration of the oscillator, 90 phase shift with unity gain at the dominant eigenfrequency  is implemented on the feedback path of the velocity measured at the tip point of the oscillator . This velocity can be measured by an accelerometer attached to the screw shaft bearing which can be assumed to be attached to the point A in Figure 6 in our simulation. Provided that the vibration of the oscillator occurs at frequency  within the bandwidth of the loop, this indirect velocity loop can be approximated by its DC gain  as shown in Figure 8 (Hosseinabadi, 2013).

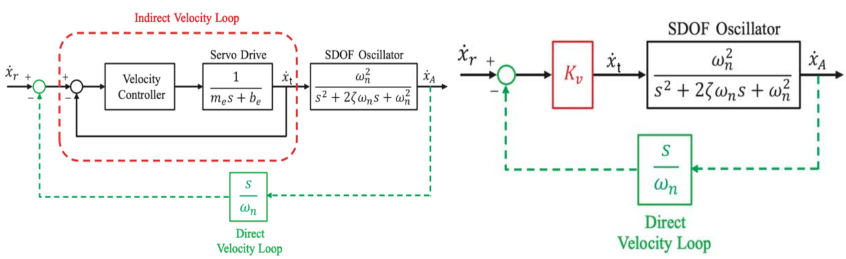


**Figure 8.** Velocity loop with oscillator and active damping

It can easily be noticed from the block diagram that the closed loop transfer function of the outer direct velocity loop is

$$\frac{\dot{x}_A(s)}{\dot{x}_r(s)} = \frac{K_v \omega_n^2}{s^2 + 2(\zeta + K_v/2)\omega_n s + \omega_n^2} \cdot \qquad (10)$$

Equation (10) reveals that the damping ratio, hence the vibration suppression property is enhanced. For systems with multiple dominant eigenmodes, a network of active damping can be designed in which each eigenmode is segregated by cascading tuned band-pass filters prior to the phase-shifter (acceleration feedback on the velocity feedback path). Summation of the several phase shifter outputs comprises the feedback to the direct velocity loop.

## 4. Numerical Simulations and Results

In this section we determine the eigenfrequencies and eigenfrequency sensitivities of the screw drive system and present the simulation results for simultaneous tracking and vibration suppression.

### 4.1 System Data

Mechanical and electrical numerical data for the ball-screw drive machine tool feed system are given in Table 1 and Table 2.

**Table 1**. Mechanical data

| Parameter | Symbol | Value |
|---|---|---|
| Rotor mass (kg) | $m_m$ | 12.9 |
| Coupling mass (kg) | $m_c$ | 1.78 |
| Table mass (kg) | $m_t$ | 250 - 500 |
| Base mass (kg) | $m_b$ | 4390 |
| Rotor inertia (kg·m$^2$) | $J_m$ | $8.25 \times 10^{-3}$ |
| Coupling inertia (kg·m$^2$) | $J_c$ | $1.79\times 10^{-4}$ |
| Coupling torsional stiffness (N·m/rad) | $k_c$ | $1.9 \times 10^3$ |
| Screw density (kg/m$^3$) | $\rho$ | 7850 |
| Screw modulus of elasticity (GPa) | E | 200 |
| Screw shear modulus of elasticity (GPa) | G | 80 |
| Thrust bearing axial stiffness (N/m) | $k_{brg}$ | $1.0 \times 10^8$ |
| Nut dynamic load (N) | $C_L$ | 50.5 |
| Nut reference rigidity (N/m) | $K_r$ | $6.37 \times 10^8$ |
| Base axial stiffness (N/m) | $k_b$ | $1.5 \times 10^8$ |
| Screw diameter (m) | $d_{sw}$ | $3.6 \times 10^{-2}$ |
| Screw pitch length (m) | $\ell$ | $2.2 \times 10^{-2}$ |
| Screw length (m) | $l_{sw}$ | 3.0 |
| Screw length at table position (m) | $l_n$ | [0.5 - 2.5] |

**Table 2.** Servomotor data

| **Parameter** | Symbol | Value |
|---|---|---|
| Rated power (kW) | $P$ | 4.4 |
| Rated torque (N·m) | $T_{e_r}$ | 18.6 |
| Rated speed (rad/s) | $\omega_{m_r}$ | 157.1 |
| Number of poles | $p$ | 4 |
| Stator resistance per phase (Ohm) | $R$ | 1.44 |
| Direct and quadrature inductance (mH) | $L_d, L_q$ | 8.15 |
| Permanent flux linkage (Wb) | $\lambda_{PM}$ | 0.21 |
| Proportional gain of current loop | $K_p$ | 10 |
| Integral gain of current loop | $K_i$ | 1 |

### 4.2 Determination of eigenfrequencies and eigensensitivity

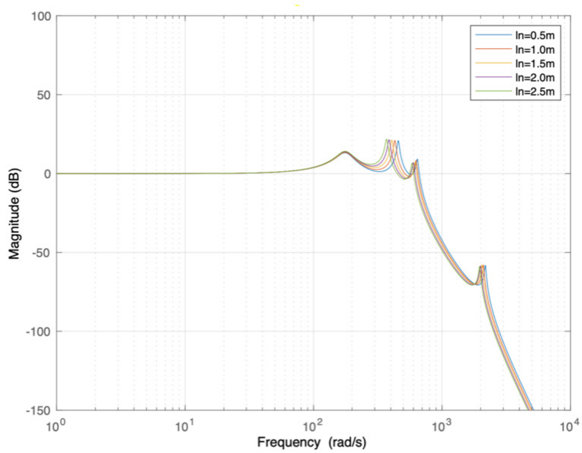


**Figure 9.** Frequency response of the ball-screw drive at several table positions

Frequency response of the drive system with 250 kg table mass (with the workpiece) for several table positions is given in Figure 9. There are two axial (base and screw) modes, two rotational (screw and motor) modes, and a rigid body mode. Dominant mode of vibration is the second peak in the plot, corresponding to first axial eigenfrequency of the screw, which is in agreement with the previous work in the literature (Frey, Dadalau and Verl, 2012). It can be noticed from the figure that maximum change of this eigenfrequency from its nominal value (m) is 12 in the table position interval  m.

**Table 3.** Eigenfrequencies of the ball-screw drive mechanical system at several positions

| **Table position** (m) | **Eigenfrequencies** (rad/s) | | | |
|---|---|---|---|---|
| | 1st | 2nd | 3rd | 4th |
| 0.5 | 177.8 | 457.5 | 644.8 | 2178.0 |
| 1.0 | 177.7 | 430.1 | 629.9 | 2101.2 |
| 1.5 | 177.5 | 407.3 | 615.0 | 2044.1 |
| 2.0 | 177.3 | 388.0 | 600.7 | 2000.3 |
| 2.5 | 177.1 | 371.3 | 587.1 | 1965.7 |

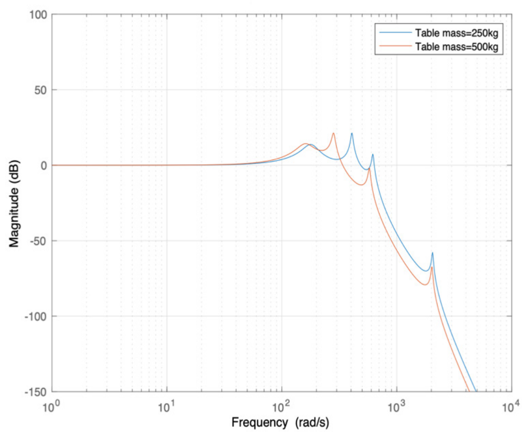


**Figure 10.** Frequency response of the feed drive mechanical system for different table masses at =1.5m

Figure 10 illustrates the shift in eigenfrequencies for 250 kg and 500 kg table mass at 1.5 m nominal table position. Considerable reduction in first axial mode eigenfrequency can be noted from the figure that the axial mode eigenfrequency is decreased to 282.1 rad/s from 407.3 rad/s, which is more than 30.

### 4.3 Simulation Results

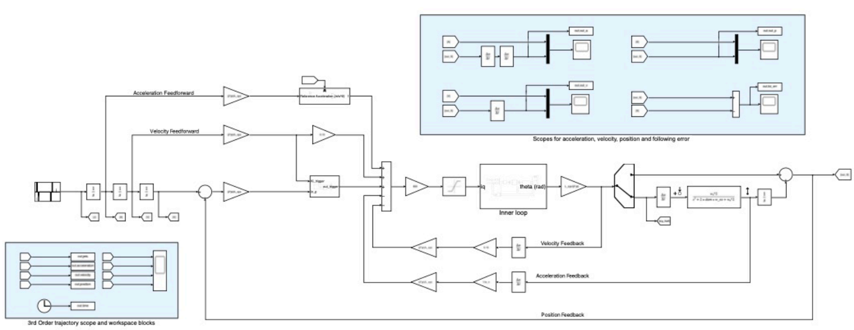


**Figure 11.** Simulink system model with inner loop given in Figure 3

Feed drive system is a cascaded velocity/position/acceleration closed-loop controlled system with feedforward and feedback gains as shown in Figure 11. Controllers have been tuned to obtain fast response with minimum tracking error and without overshoot in point-to-point move on a trajectory. Cascaded loops' controller gains are tuned commencing from the innermost loop through outwards. The feedforward used in velocity can eliminate the trajectory tracking error. However, if its gain  gain is set too high, overshoot will occur due the large error spikes created in the deceleration/acceleration zones of the velocity profile. Setting the  identical to the  was our beginning point, which has lead us to acceptable results. Tuning the PID position feedback position controller in Figure 11 involves gain to be placed before saturation block.  gain is placed on the velocity feedback path. We limit the current the integrator output (integrator clamping) to be in the interval (-5.8 A - 5.8 A). In our work, we use the acceleration feedforward only if an eigenmode of the system is excited. Tracking error is negligible without acceleration feedforward and even without acceleration feedback when there is no vibration on the system.

If an excited eigenmode is detected by the mounted accelerometer, acceleration feedback (phase shifter) and acceleration feedforward is activated. The resulting tracking accuracy obtained by applying an impulse on the mechanical system dynamics is obtained and the result is shown in Figure 12. It is clear that the rms value of the tracking error is less than 10 microns during the transient response, which is acceptable for most of the modern high-speed machining applications.

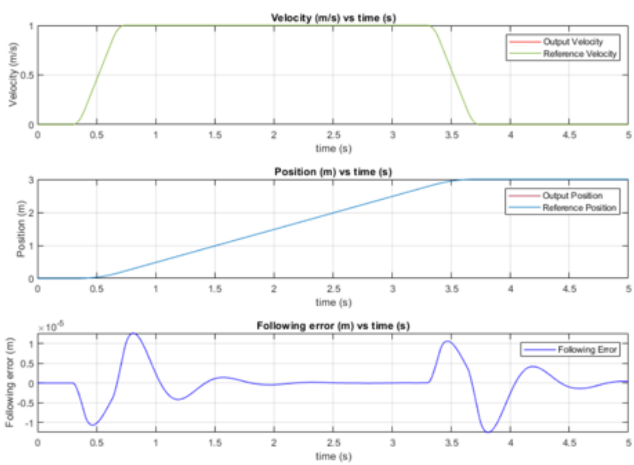


**Figure 12.** Simulink system model with inner loop given in Figure 3

## 5. Conclusions and Recommendations for Future Work

In this paper we have studied the simultaneous tracking and drive-based vibration suppression of CNC machine tool feed systems. We have derived an electromechanical model of the feed drive including several flexible modes of the drive.

Our expedient lumped parameter model comprises of the predominant axial and rotational eigenmodes to analyze the influence of flexibility on tracking accuracy and on vibrations during the operation. We have generated a third order smooth trajectory and we have shown that a third-order smooth trajectory with bounds on velocity, acceleration and jerk does not excite the eigenmodes with a properly designed feedforward control path. Moreover, we have also tested the resulting tracking accuracy in case of a predominant eigenmode is excited by the cutting/ milling forces or by any other disturbance or impulsive action due to the non-linearities in the drive system. Incorporating a properly designed acceleration feedforward and acceleration feedback (phase shifter), it has been shown that transient vibration amplitude can be attenuated to a value less than 10 m rms.

Main parameters that may affect the stability and performance of the feed drive control system are the alternating work table position and the different work-piece mass. Regarding the former, we have demonstrated that for different positions along the feed drive shaft dominant eigenfrequency of the drive system (first axial mode) varies in between 12 from the nominal value of our model. The effect of the latter is more prominent in our numerical example. In fact, the dominant eigenfrequency varies by more than 30  from its nominal value for two-fold increase in the work-piece mass. It can be concluded that if the work-piece mass is not constant, classical PID feedback and feedforward control techniques may not be satisfactory for vibration suppression. As such, implementation of modern control techniques such as robust H or LPV control can provide guaranteed robust stability and acceptable performance not only over all parameter variations during the operation but also for variant work piece mass to be machined.

## Declarations and Conflict of interest

The author declares that this study adhered to research and publication ethics. No conflict of interest is declared by the author.

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