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Abstract

The hybrid modelling technique considering distributed load, explicit damping factors and measured non-linear effects has been shown to effectively represent the dynamics of CNC machine tool feed drives. The resonant frequencies produced by the mechanical elements of the machine were simulated for an analogue feed drive with DC motors and belt-pulley transmission system. The rapid increase in the use of digital control systems and permanent magnet synchronous AC motors for modern feed drives requires an update on this modelling technique. This paper presents an investigation into the application of the technique for the digital feed drive of a Cincinnati Arrow 500 CNC machine tool equipped with a Siemens Sinumerik 840D controller. The drive utilises a direct transmission system and the field oriented control technique in the current control loop. The discrete method of approach to the current control loop is used to represent the digital states of the current, torque and speed control. The model is developed in MATLAB/ SIMULINK employing the Power System (Simpower) Blockset. Model validation in the time and frequency domain is performed using the deterministic and pseudo random binary signals from the controller’s built-in routines. The results are shown to provide a sufficient solution into the representation of the feed drive dynamic behaviour. The investigation is part of a wider study to develop an active vibration system for CNC machine tools and to develop a machine tool drive model, useful for the design, diagnosis, optimisation and condition monitoring purposes.

1 Introduction

The modern industry requires high acceleration, fast response and small tracking errors from high-speed machine tools in order to reduce cutting and non-cutting times and improve productivity. The positions, velocities and accelerations of machine tool are controlled by the machine feed drives in accordance with the commands generated by the CNC interpolator. Therefore, improving the feed drive performances represent an important factor in the process of enhancing the machine tool accuracy by reducing/eliminating the machining errors.

The modelling approach to machine tool feed drive system plays an important part in the above-mentioned process because the essential elements in machine tool dynamics can be studied and their influence on the overall machining precision can be evaluated without wasting workpieces for practical operations. The modelling and simulation techniques are used by the machine producers (during the design stage) and users (to improve the machine performance) in order to predict and study the machine behaviour under various operating conditions, at much lower costs, while enhancing performance and efficiency.

The CNC machine tools are hybrid systems, thus the hybrid modelling approach developed by Pislaru et. al. [1] is applied for the modelling the feed drives within the Arrow 500 vertical machining centre. The hybrid models with distributed load, explicit damping factors, non-linearities (backlash and Coulomb friction) are implemented in SIMULINK. The simulated results are compared with the dynamic behaviour of the machine tool feed drive to verify the accuracy and performance of model. These modelling and simulation techniques have the potential to reduce experimentation, optimise cutting parameters, and improve the quality of the machining process reflected on the machined surfaces.

The development of an accurate SIMULINK model for the Sinumerik 840D controller fitted on the machine represents an important contribution to the study of motion control for high-performance drive systems. This CNC control is a complete digital system where microprocessors command and control the axis drive motors with high precision. Such system provides a significant increase in the electrical noise immunity, better configuration flexibility, and added supports for advanced control algorithms.

The hybrid models of the digital feed drives implemented in the Cincinnati Arrow 500 machine tool are used for the study of the vibration error sources within the machine and the various elements influence on the drive performance.

2 Drive system configuration and modelling considerations

The studied feed drive is categorised into two main parts: control (electrical) part and mechanical load (Figure 1). A Sinumerik 840D digital controller is employed to control the system, including a permanent magnet synchronous AC motor as the motion source. The linear movement of the machine worktable is generated by the conversion of the rotary motion of the permanent magnet synchronous motor (PMSM) using the ballscrew assembly. The studied feed drives have feedback signals generated by incremental linear encoders and feed-forward path from the position to velocity control loops, which enables fast and an accurate positioning. The three loops for position, velocity and current control are connected in cascade (Figure 2).

Figure 1. X-axis feed drive configuration

Figure 2. Feed drive closed control loop

An accurate model of the feed drive should contain values for the various damping coefficients and non-linearities (backlash, Coulomb friction, etc.) influencing the performance of the CNC machine tool. Friction between the moving components of the machine (ballscrew–nut, guideways–worktable, etc.) can cause significant errors in precision machining. The largest portion of friction is due to complex kinematics of the ball nut and it may be sometimes desirable (provides some damping), but it is unwanted in the majority of cases (causes steady state or static error).

The model is implemented in the s-domain in MATLAB/SIMULINK to enable an efficient modelling of the mechanical load which is a combination of continuous and discrete elements. The problem associated with sampling process is considered as negligible due to the extremely fast sampling time of the controller (125 (s). The model can also be relatively easily transformed into the discrete time domain using the Discretiser blockset available in SIMULINK.

3 Control loops

The ***current control loop*** contains a controller issuing voltage commands to the power electronic system in order to maintain a required level of currents by minimising the error signal (Figure 3). This part of the control system is also referred to as the torque or acceleration control loop because it also regulates the motor electrical torque and the machine acceleration, which are directly proportional to the motor current. This loop is implemented using the *field oriented control* (FOC) method [4], which means that two current control loops are required: the *quadrature* (torque generating) and *direct* (magnetic flux producing) (Figure 3). *Clarke-Park transformation* is utilised for the conversion of the three-phase into two-phase current (*direct* (*d*) and *quadrature* (*q*) components). The *Clarke* transformation translates the three-phase variables (*R-S-T* or *a-b-c*) into two-phase time varying parameters with a static reference (*a-b* or *?-?*). The *Park* transformation converts the two-phase time varying parameters with a static reference into two-phase time invariant parameters with a rotating reference and direct-quadrature (*d-q*) axes. The later transformation can be performed only when the rotor angular position ? is known either from an integrated rotary position sensor or from a theoretical prediction. The technique is implemented using the Digital Signal Processor (DSP) technology to accurately and quickly perform the complex mathematical calculations.

Figure 3. Current Control Loop Diagram [2]

The *d-q* voltage setpoints from the PI controller feed a power electronic inverter based on the pulse width modulation (PWM) technique to energise the motor. An indirect AC power electronic converter (consisting of rectifier, DC link and inverter) (Figure 4) provides the current and voltage for the PMSM. The converter transforms the three-phase AC mains to a high voltage power supply of the DC link unit through power diodes arranged in a bridge rectifier configuration. The generated DC voltage level is approximately 1.35 times of the line-to-line mains voltage, which is further smoothed by the use of the inductor and capacitor.

The DC link voltage is used by the inverter to drive the motor by switching on and off the power electronic devices at a rate controlled by the PWM control logic unit, creating an impressed sinusoidal AC current. The power electronic converter under investigation employs a 3-arm bridge (6 pulses) PWM technique utilising IGBT inverters with 8 kHz carrier frequency. The impressed sinusoidal current is generated by varying the power on period of the ± 300 V DC link.

Figure 4. Diagram of the Power Electronic System [3]

The three-phase permanent magnet synchronous AC motors of the feed drives contain a six-pole stator, and a rotor made from the rare-earth, permanent magnet material. It also incorporates a built-in incremental encoder to indicate the instantaneous position of the rotor for the current control purpose.

The PMSM model is represented as a second-order state-space system by assuming that the rotor magnetic flux is sinusoidal. The electrical part of the model is implemented in the direct-quadrature (*d-q*) reference frame by using equations (1)-(3) whereas the mechanical part of the system describing the rotor rotational acceleration and velocity is described by equations (4)-(5).

| |(1) |

| |(2) |

| |(3) |

| |(4) |

| |(5) |

The setpoint value for the quadrature current is determined by the ***velocity control loop***. The value is calculated from the torque setpoint command issued by the loop to maintain the rotational velocity of the motor shaft at a certain value. The control loop compares the required velocity to the actual value measured by the attached rotary encoder. It then applies a proportional-integral (PI) controller on the error signal to calculate the required torque value (Figure 5).

Optional notch and low-pass filters are available in the velocity control loop to smooth out the velocity setpoint values. These filters can be used to dampen any resonance frequency in the signals which occur due to the imperfections in the mechanical configuration of the system. These frequencies need to be minimised because they could affect the performance of the velocity control loop. Additionally, optional velocity and torque pre-control functions can be used to enable the feed-forward control to increase the drive performance.

The required rotational velocity is calculated by the ***position control loop*** where it samples the linear position set values provided by the user (part program) and adjusts the worktable actual position to minimise the position signal error. The position set values originate from the controller interpolator, optional fine interpolator and advanced filtering functions. The actual position of the worktable is measured by the encoders, either directly from the linear encoder on the slide or indirectly from the rotary encoder attached to the motor.

The position controller of the machine under study is a proportional (P) type with optional feed forward controls. Its parameter is represented by the Velocity Gain (Kv factor) which is defined as the ratio between the velocity setpoint and the deviation in the position (also know as the following error).

Figure 5. Diagram of the Velocity Control Loop [2]

4 Mechanical Transmission Components

The motor rotary motion is converted into worktable linear movement through the mechanical transmission elements (Figure 6). An elastic coupling links the PMSM to the ballscrew driving the worktable whose two slides move linearly on two guideways fixed on the machine bed. High machine accuracy can be achieved when using a direct coupling with increased transmission rigidity [5]. The studied feed drive contains a backlash-free jaw-type coupling providing typically 200 to 400 percent torsional stiffness of a belt-driven system.

The ballscrew device is used due to its high efficiency, durability and accuracy. Moreover, a pre-loaded ballscrew can provide beneficial damping characteristics due to the increased rigidity and minimised chatters. This will also reduce the non-linear behaviour, i.e. backlash, although friction is still present. The ballscrew shaft is supported by back-to-back thrust bearings and single-row radial bearings. The thrust bearings are configured with 60° contact angles to provide both the radial and axial supports. The configuration constitutes the “fixed-supported” mounting for the ballscrew axial rigidity evaluation (Figure 1).

The various mechanical transmission elements oscillate during the machining process; therefore they can be considered as mass-spring combinations of second order as performed by Pislaru et al [1] and presented in Table 1.

Figure 6. Mechanical elements of the X-axis drive for the Arrow 500 machine

5 Comparison between simulated and measured results

The units of the parameters in the model are matched to those in the machine controller to enable close correlation of the measured data to the simulated results. The conversions to the appropriate unit are represented by various software blocks, such as that which converts the worktable actual position from the SI unit (m) to the position controller unit (mm) and the velocity command of the position controller from a linear (m/min) to rotary unit (rpm). The complete SIMULINK model for the position control loop can be seen on Figure 7.

Figure 7. SIMULINK Model of the Position Control Loop

The mechanical transmission components are represented in SIMULINK as lumped subsystems (motor shaft, elastic coupling, worktable) and the ballscrew as distributed element (7 subsystems with total ballscrew flexibility apportioned between each). The explicit damping coefficients are included to simulate the resonant states of the feed drive system and the nut is assumed to be at the middle of travel. The values of model coefficients are obtained and calculated from manufacturers' catalogues.

Table 1. Mathematical models of SIMULINK subsystems

Subsystem	Modelling element	Equation
Motor shaft	Inertia element	
	Rot. frictional element	
Elastic coupling	Rot. spring/damper	
Ballscrew Front (BSF)	Inertia element	
Ballscrew 1 (BS1)	Rot. spring/damper	
Ballscrew Middle (BSM)	Inertia element	
	Mass element	
Ballscrew 2 (BS2)	Inertia element	
Ballscrew End (BSE)	Inertia element	
	Rot. frictional element	
Ballscrew Nut	Linear spring/damper	

BS linear stiffness	Linear spring / damper	
Table	Coulomb friction	
	Mass element	
Bed stiffness	Inertia element	
	Rot. frictional element	

The current control loop is validated using 3% and 10% step increase of the maximum motor current (0.84 and 2.8 A respectively). The direct current setpoint is kept at zero to minimise any field weakening of the magnetic material. The model's responses show good agreement with the measured data at both input values with slight overshoot and response time difference. The model's frequency response matches the measured data up to 800 Hz (Figure 8).

The velocity control loop model is validated in time domain by introducing 200 and 500 mm/min step inputs. The simulated linear velocity data are obtained by multiplying the simulation values with the ballscrew ratio and they are found to match to the measured data closely (Figure 9). The peak at approximately 4 ms represents the mechanical system jerk and is most likely caused by the static friction on the guideways and the ballscrew support bearings. The frequency response of the velocity loop model shows 10 dB higher gain at the low frequencies up to around 400 Hz where the first peak (resonance) occurs. The model's phase response matches better with the measured data but neither of the responses can detect the higher resonant frequencies.

The validation of the position control loop is carried out by inputting a 0.5 mm step demand from the initial position of 250 mm. The measured result (Figure 10) shows an overshoot compared to the undershoot response of the simulation data although both have a second order system characteristic. The simulated response lag is most likely due to mass and stiffness parameter tolerances of the data supplied by the manufacturer. The undershoot response of the simulated response is likely from the higher damping values implemented in the model compared to the actual data.

Figure 8. Graphs of the current loop response in time and frequency domains

Figure 9. Graphs of the velocity loop response in time and frequency domains

Figure 10. Graph of the position control loop response to a 0.5 mm step input

5 CONCLUSIONS AND FURTHER WORK

The hybrid modelling technique considering distributed load, explicit damping factors and measured non-linear effects has been shown to effectively represent the dynamics of CNC machine tool equipped with digital feed drives. The technique applied discrete method of approach to the current control loop to represent the digital states of the current, torque and speed control. Model validation in the time and frequency domain was performed using the deterministic and PRBS input from the controller's built-in routines. The results were shown to provide a sufficient solution into the representation of the feed drive dynamic behaviour.

Further parametric investigation on the model's parameters is required due to the tolerance level provided by the manufacturers on their products. They include the power electronic circuitry's capacitive and resistive properties, as well as the mechanical element's stiffness, mass and damping characteristics. The jerks observed on the velocity loop response can also be represented more accurately using more advanced friction model. Future activities are also planned to model drive's advanced control algorithms and to discretise the model for real-time monitoring and diagnostic purposes.

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