Backlash Detection in CNC Machines Based on Experimental Vibration Analysis

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Abstract*—Exploiting backlash free mechanisms and gearboxes, the stiffness of servo axes in CNC machines can be improved. In fact, the rigidity of these axes will be increased, so that different types of feed-forward control can be implemented. Furthermore, backlash free mechanisms facilitate designing foolproof systems without challenging with the instability and nonlinear behavior of backlash. However, any mechanical failure and looseness causes backlash which in turn may lead to severe vibrations. So, backlash detection and exploiting efficient preventive maintenance (PM) to prevent interrupt in the production line is of interest. In this paper, first a simple model for mechanical system of a CNC servo axis, and its control system will be presented. This reduced order system for speed control will be detailed. Next, the bandwidth of vibration frequencies due to backlash is estimated and behavior of a servo axis with various backlashes is simulated. Then, to encapsulate the role of backlash in different conditions, performance response of five mechanically different axes with different servo gains in various CNC machines are empirically investigated. These experimental frequencies of vibration obtained in completely different CNC machines (small, medium and heavy size) are compared with those estimated. Simulation and experimental results show that the frequency of vibration in a servo axis with backlash is not affected by the value of backlash, while the position control gain dictates this frequency. Finally, an experimental equation will be developed that estimates this frequency for various CNC machines.

***Keywords*—Backlash, CNC machines, Vibration Analysis.**

I. INTRODUCTION

Backlash causes inaccurate motion, and extraordinary vibration which imposes severe limitations on the quality of control. This will damage the power transmission train parts, thus various techniques have been suggested for backlash detection and estimation, [1-2]. Simple models of mechanical backlash found in the control system literature ignore the impact behavior that dominates motion at low amplitudes. As a result, such models can not characterize the low-amplitude limit cycle oscillations that occur in closed-loop control systems. Therefore, a more realistic model of mechanical backlash should incorporate visco-elastic contact forces between the two masses, [3]. On the other hand, for backlash compensation, high-performance controllers require high-quality measurements of the current state of the power train, [4-5]. Information about the size of the backlash is also needed. In the case of known backlash, a backlash compensating controller can guarantee exact output tracking. When the backlash characteristics are unknown, adaptive laws have been proposed to update the controller parameters and to guarantee bounded input-output stability, [6-7].

Introducing a smooth inverse function of the backlash and using that in the controller design with back stepping technique, a backlash compensator has been proposed, [8]. For the design and implementation of this controller, no knowledge is assumed on the unknown system parameters. Backlash compensators have been also proposed using dynamic inversion by the fuzzy logic, [9]. The classification property of fuzzy logic systems makes them a natural candidate for the rejection of errors induced by the backlash, which has regions in which it behaves differently. An adaptation algorithm can be developed to estimate the backlash parameter online, and under certain conditions a fuzzy compensator with the adaptation algorithm guarantees that the backlash output converges to the desired trajectory, [10]. Extending the dynamic inversion technique to discrete-time systems by using a filtered prediction, and using a neural network for inverting the backlash nonlinearity in the feedforward path, an algorithm has been proposed for backlash compensation that guarantees bounded tracking errors, and also bounded parameter estimates, [11].

In industrial CNC machine tools, to have an accurate finished surface, both exact positioning and smooth movement with low contour deviations are demanded. On the other hand, progress in providing high speed milling and grinding spindles (with the aim of exploiting ceramic/ hydrostatic bearings), and improvement in cutting materials have facilitated producing high quality finished surfaces with high speed servo axis. Backlash free mechanisms (like double nut screws, double pinion gearbox, etc.) have perfect linear dynamics, and enable us to fulfill all aforementioned demands. However, in the case of small backlash that is rising from worn mechanical parts, an uncontrolled nonlinearity will impose to the system that distorts the dynamic behavior of the axis and disturbs its expected smooth movement. In this paper, first a reduced order model of the mechanical system of a servo axis will be introduced, and an ordinary CNC control system will be reviewed. The response of speed and current control for the mechanical vibration will be discussed. Next, based on the experimental results of five mechanically different axes (with different servo gains), a simple model of backlash is illustrated and the bandwidth of vibration frequency due to backlash is estimated. Then, the effect of value of backlash on a servo axis is simulated and the experimental frequencies of vibration in various CNC machine tools (small, medium and heavy size) are compared with the estimated frequencies.

II. Axis Mechanical Model

Backlash between mechanical parts may take place due to failure or looseness, wear, etc., which imposes nonlinearity to the system. Basically, in a critical condition backlash leads to vibration when desired acceleration is zero (speed is constant) and movement is not affected by the gravity. The result would be wasted surface finish, broken cutting tool, damaged mechanical elements of axis, which in turn increases backlash. Therefore, detection, and control of backlash becomes significantly essential especially for CNC machine tools that the default assumption is a backlash free mechanical mechanism.

In a CNC machine, each axis composed of different mechanical parts like coupling, gears, lead screw, etc. that have different mechanical properties. All of these elements can be modeled as a set of springs and dampers, while the more elements we take the more resonance frequencies will exist. However, in most cases the servo axis can be modeled based on the lower resonance frequency, and the other elements can be ignored or assumed to be rigid. In Fig.1 a reduced order of mechanical model is illustrated where:

τ_l : load torque

τ_d : reaction of axis on motor

τ_m : motor torque

J_m : moment of inertia of the motor including its gears or coupling

J_l : moment of inertia of the load

v_1 : velocity of axis obtained from motor side

v_2 : velocity of axis directly obtained

k_s : stiffness of axis

c_s : damping of axis

k_r : ratio between motor speed and axis speed

δ : amount of backlash

ω_m : speed of motor

Equation of motion for this model (without backlash) is:

$$\tau_m = (J_m + \frac{J_l}{k_r^2} \times M(s)).s.\omega_m - \frac{M(s)}{k_r}\tau_l \quad (1)$$

Where: $M(s) = \dfrac{1 + 2\xi(s/\omega_n)}{1 + 2\xi(s/\omega_n) + (s/\omega_n)^2}$,

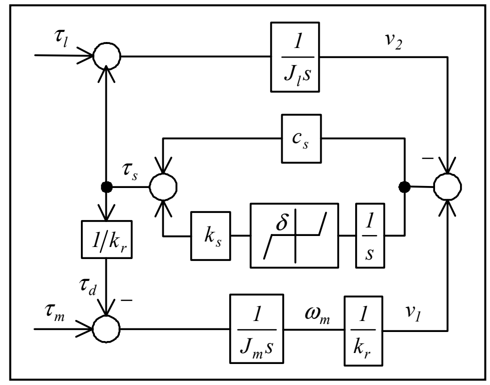

Figure 1. Axis model block diagram

$\omega_n = k_s/J_l$ and $\xi = c_s/(2\sqrt{k_s J_l})$

III. Axis Control System

The software of the investigated CNC machine tools is Siemens SINUMERIC 840D that implements a cascade control system including three main parts: position controller (P), speed controller (PI) and current controller (PI), where actuator is a permanent magnet synchronous motor (AC servo motor). Fig.2 shows the simplified block diagram of a servo axis closed loop control system. Various parameters and signals in this block diagram are defined as:

x_d : setting position from interpolator

x_{act} : axis actual position measured by direct encoder

e : following error

k_v : position control gain

k_r : ratio between motor speed and axis speed

n_{set} : setting speed of motor

n_{act} : actual speed of motor measured by motor encoder

k_{sp} : speed controller proportional gain

T_{si} : speed controller integrator time

τ_{set} : setting torque

i_{set} : setting current

i_{act} : actual (measured) current

Also the servo axis is equipped with two encoders that are known as measuring system 1(motor encoder), and measuring system 2(direct encoder). Direct encoder is directly connected to the axis and provides the exact position of the axis whether there is backlash or not. The value of direct measuring system is utilized for position feedback in position controller. The other encoder is located inside the motor and it is used as a feedback for speed controller. However, in most CNC machine tools according to the kinematics of the axis, the difference between motor encoder and direct encoder should be in a limited tolerance to assure a safe motion.

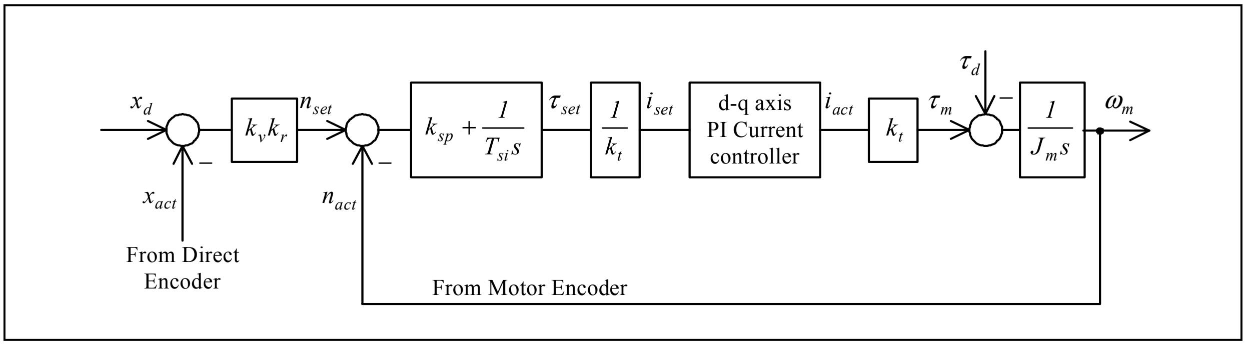

Figure 2. Closed loop position control system in a servo axis

The controllers of the cascade control are to be designed to provide higher cross frequency in each inner loop with respect to its outer loop. For tuning the servo gain of the speed and current controller an attempt should be done to keep the amplitude of speed (also current) controller at 0dB over the widest frequency bandwidth. In practice, the cross frequency of speed and current controller should reach about 200-300HZ and 500-100HZ, respectively, [12]. The magnitude of transfer function of speed and current controller in bode diagram for these bandwidths should be nearly 0db (less than 3db), [12]. Fig.3 shows the real frequency response of closed loop speed controller for a servo axis where the frequency bandwidth is about 180Hz. Experimentally the frequency of mechanical vibration is less than 50Hz and in most of the cases is less than 30HZ, so simply for our study of mechanical vibration the magnitude of speed and current controller can be replaced by 1. The phase of current controller for the frequency of vibration is zero, so by replacing the transfer function of current control loop with 1 the transfer function for the whole system can be written as:

$$\omega_m = \frac{1+k_{sp}T_{si}s}{1+k_{sp}T_{si}s+T_{si}s^2(J_m+(J_l/k_r^2)M(s))}k_vk_re+\frac{T_{si}M(s)}{k_r}s\tau_l \quad (2)$$

As $v_l = k_r.\omega_m$ so the transfer functions between v_l and e can be rewrite as:

$$\frac{v_l}{e} = k_v\left(1-\frac{T_{si}s^2(J_m+(J_l/k_r^2)M(s))}{1+k_{sp}T_{si}s+T_{si}s^2(J_m+(J_l/k_r^2)M(s))}\right) \quad (3)$$

Here, as mentioned before regarding the bandwidth of speed and current controller for the frequency of mechanical vibration (the fundamental frequency measured by FFT analysis of current and speed signals), the transfer function of (3) can be properly approximated as:

$$\frac{v_l}{e} = k_v e^{i\phi} \quad (4a)$$

where ϕ describes the phase of vibration and can be calculated by (3). For instance, by replacing the control parameters for CNC1, the transfer function for the sinusoidal signal with the fundamental frequency is:

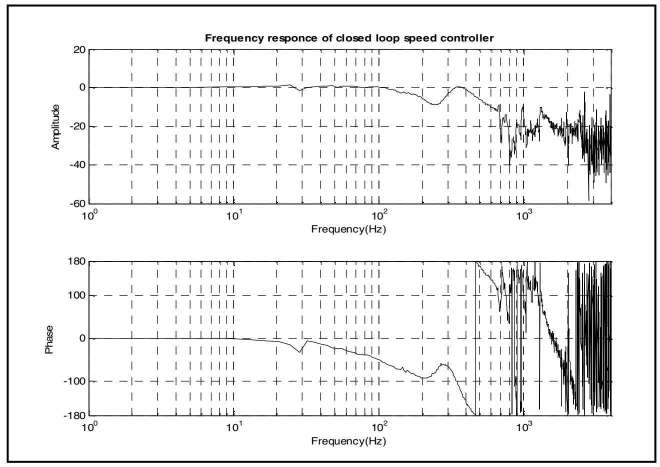

Figure 3. Frequency response of closed loop speed controller

$$\frac{v_l}{e} = k_v\left(1+\frac{6.975}{11.76j-5.975}\right) \Rightarrow \left|\frac{v_l}{e}\right| = 1.0401k_v \quad (4b)$$

According to (4), if the vibration caused by backlash is simplified as a single sinusoidal signal with the fundamental frequency, and the other elements of FFT analysis neglected, the speed of axis will be proportional to the tracking error; which is an important fact to be used in backlash detection and management.

As mentioned before, servo axis in CNC machines are equipped with two measuring systems; system 1 (motor encoder) and system 2 (axis encoder). The most important advantage of direct measuring system is accuracy in axis positioning even in the presence of backlash. However; it is not so easy to detect and measure backlash with simply comparing obtained signals of the two measuring systems during motion. In fact, due to torsion and elastic displacements in the mechanism of the axis, the installing condition, temperature variation and its effect on the linear scale (direct measuring system), transmitting accuracy of mechanism (like lead screw, rack and pinion, etc.) there exists a variable difference between the two measuring systems especially during accelerating or decelerating motion. Therefore, in most of the CNC machine tools an acceptable tolerance (about 0.5mm or 0.5deg for linear and revolute axis, respectively) is allowed for the two measuring systems just for detection of axis safe operation. Since the backlash value that causes vibration is much less than this allowed tolerance, this difference between the two measuring systems is just useful for detection of axis overload or any serious failure in mechanical parts of axis.

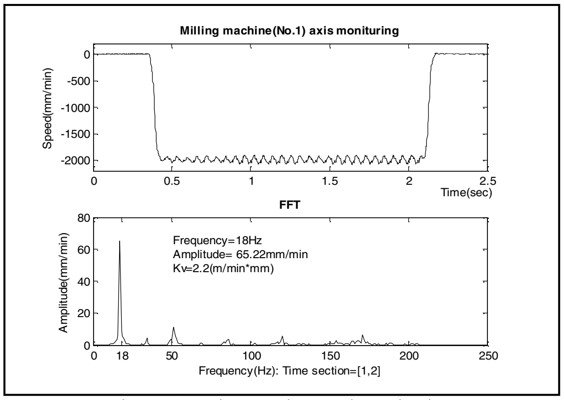

Figure 4. Linear axis speed monitoring

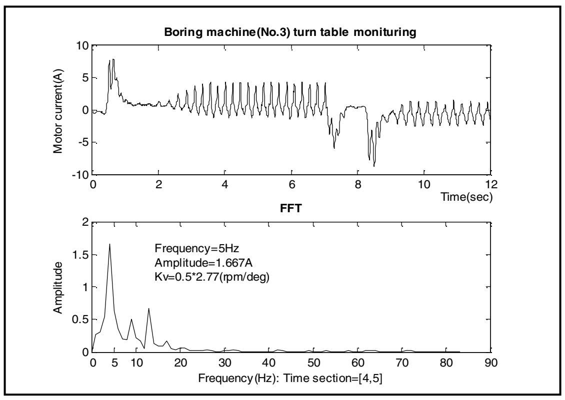

Figure 5. Rotary axis current monitoring

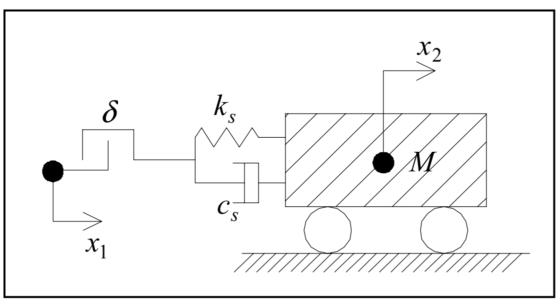

Figure 6. Axis mechanical model

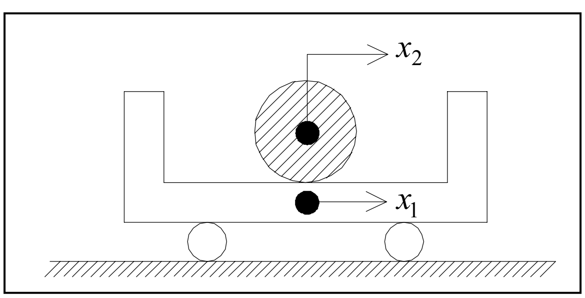

Figure 7. Backlash simulation in a simple system

IV. Backlash Model

Backlash at constant speed may lead to vibration that can be observed as fluctuations in the motor current, acceleration and momentum or speed of the axis. Such vibrations, using Fast Fourier Transformation (FFT) can be transformed to a series in which each element is composed of two characteristic values; i.e. the amplitude and the frequency. Fig. 4 and 5 show the results of FFT analysis of the speed and current for experimental measurements of two CNC machines' axes; i.e. CNC1 and CNC3. As seen in these illustrations, in both of these servo axes there is a fundamental frequency with nearly the same amplitude as the amplitude of actual signal. So, with a reasonable accuracy, backlash fluctuation can be interpreted as the fundamental element of FFT. In fact, a procedure of backlash detection and its suppression can be developed by analyzing the current/speed fluctuation (when setting acceleration is zero), based on the knowledge of amplitude and range of fundamental frequency of vibration caused by backlash. In order to develop a simple model of the mechanical system the behavior of all mechanical elements related to the high natural frequencies can be considered as rigid. The model as shown in Fig.6 is provided based on the mechanical parts in power transmitting line that interpret the lowest natural frequency. Equation of motion for this system can be written as:

$$\begin{aligned} &if(|x_1 - x_2| > \delta) \Rightarrow \\ &M\ddot{x}_2 = c_s(\dot{x}_1 - \dot{x}_2) + k_s(x_1 - x_2 - sign(\dot{x}_1)\delta) \end{aligned} \tag{5}$$

Where M is the mass inertia of the axis carriage. In Laplace space(if backlash is zero):

$$\frac{X_2}{X_1} = \frac{1 + 2\xi(s/\omega_n)}{1 + 2\xi(s/\omega_n) + (s/\omega_n)^2} \tag{6}$$

V. Frequency Bandwidth Estimation

To estimate the bandwidth of frequency that can happen for a servo axis, a simplified mechanical system is considered as shown in Fig.7. In fact, motion of the ball (x_2) is controlled with the movement of the cart (x_1), where the described servo control system is implemented. This system works based on the impact between the ball (axis) and the walls (actuator). The real condition of vibration for the speed of axis (v_2) and motor (v_1) in two different milling machines (CNC1 and CNC5) can be seen in Fig8. Also in Fig.9 theoretical expectation of backlash and its resulted impact in points A and C have been illustrated. At the time of separation (point B in Fig.9) between motor and carriage of axis (in any point of mechanism) we expect a constant and highest level of speed for the carriage (measured by direct encoder) until the next impact in point C. Impact is a very complex event involving material deformation and recovery and heat/sound generation (energy loss), However, here a simple model of impact with a constant coefficient of restitution is implemented. So instead of solving (6) for the system of Fig.6 with a complex model of impact the equation of motion (periodic motion) of the simple system shown in Fig.7 is written.

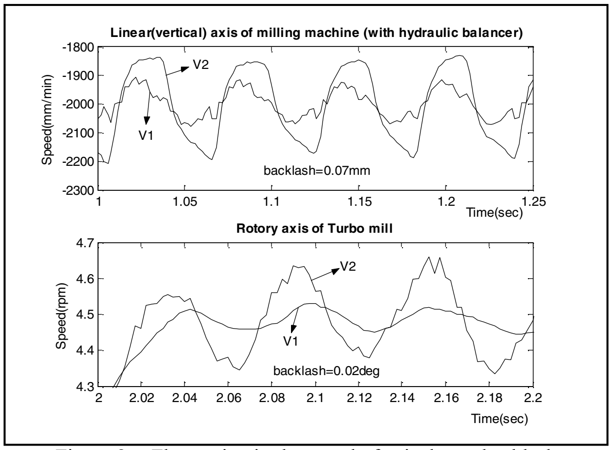

Figure 8. Fluctuation in the speed of axis due to backlash

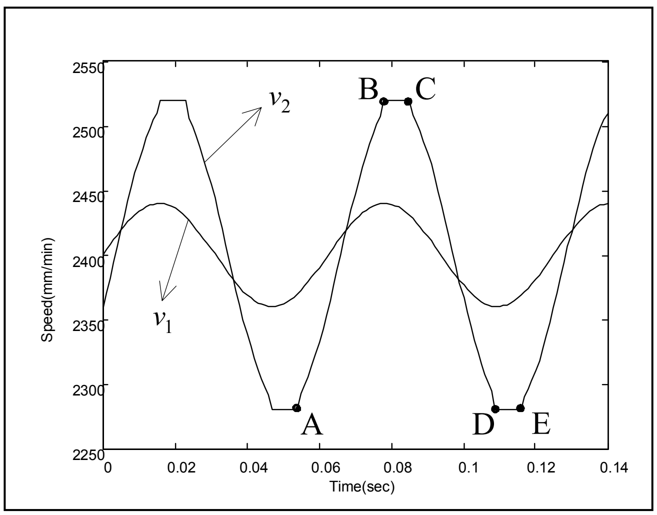

Figure 9. Expecting backlash behavior

Here, for calculating the upper limit of frequency, the effect of impulse on the driving motor is neglected and the time of contact (A-B in Fig. 8) is assumed to be zero which yields a maximum possible frequency. In a real condition as shown in Fig.8 material deformation and recovery takes considerable time. Assuming that at $t = 0$, $x_2 = 0$, so during the time before impact we have:

$$if\,(0 \le t \le \frac{T}{4}) \Rightarrow \frac{dx_2}{dt} = v_2 = const \qquad (7)$$
$$x_2 = v_2.t$$

Where T is the period of vibration. Impact will happen at $t = T/4$, and if e is defined as the coefficient of restitution, so:

$$\frac{v_2 - v_1}{v_2 + v_1} = e \qquad (8)$$

$$v_1 = k_v \times x_2 \Rightarrow v_1 = k_v.v_2.\frac{T}{4} \qquad (9)$$

Therefore, the relationship between k_v and the frequency ($1/T$) can be obtained as:

$$f = \frac{1}{4} \times \frac{1+e}{1-e} k_v \qquad (10)$$

The coefficient of restitution for steel is about 0.6 and as the energy loss in the real vibration is much higher than the assumed model (because of high ratio of mass to contact area, also friction), then the frequency of vibration in a servo axis is definitely limited as:

$$f < k_v \qquad (11)$$

Eq.(10) explicitly shows that the frequency of vibration is not affected with the amount of backlash, and it is proportional to the position control gain that will be covered by the simulation in part VI. This theoretical result also was confirmed with the experimental results of vibration analysis in CNC1 in which for different amount of backlash (0.02 - 0.07mm) the corresponding fundamental frequency was unique(18Hz).

By replacing coefficient of restitution with zero (that is certainly less than the real condition as shown in Fig. 8) the lower limit for the frequency of vibration will be obtained as:

$$f > k_v/4 \qquad (12)$$

VI. Simulation

In this part one of investigated servo axis with the presence of different backlash is simulated. Simulation is accomplished based on the machine data of the servo axis and coulomb + viscose friction is used for friction modeling. Also, based on experimental results, axis mechanical model (Fig. 6) is tuned for this axis. Fig. 10 show the axis speed fluctuation(measured by motor encoder) for speed set point of 0.026(r/s) where value of backlash is 0.01 and 0.04 (deg). These values of backlash were measured at the end contact point of mechanism where the kinematics ratio is 1258.5. Fig. 10 clearly shows that any increase of backlash does not affect the frequency of vibration but the amplitude of vibration.

VII. Experimental resullts

To generalize (11) and (12), different CNC machines with different dynamic specifications, servo gains and different backlashes were experimentally investigated. These machines are:

1. Linear axis (vertical) of milling machine driven by lead screw equipped with hydraulic balancing cylinder (CNC1).
2. Linear axis of horizontal turning machine driven with hydrostatic screw mechanism (CNC2).
3. Revolute axis (Turn table) of boring machine driven by twin pinion and gear wheel (CNC3).

4. Linear axis of boring machine driven by rack and twin pinion (CNC4).
5. Revolute axis of milling machine driven by twin pinion and gear wheel (CNC5).

Fig.11 indicates that the frequency of vibration in a servo axis is closely proportional to the position control gain and limited in the bandwidth defined by (11) and (12). As a conclusion, with an acceptable accuracy we can provide an experimental equation as below:

$$f = 0.6k_v \tag{13}$$

VIII. CONCLUSIONS

Backlash is a common trouble in all servo axis of CNC machines, defining the bandwidth of frequency of vibration in a servo axis with the presence of backlash and estimating the frequency, will be significantly helpful for both designing robust servo axis without inclusion of resonant frequency and backlash detection through the condition monitoring (CM). This was the main focus of this research paper. Developing simple models for the mechanical system of a servo axis, and the backlash itself, it was shown that the frequency of vibration in a servo axis with backlash is not affected by the value of backlash, while the position control gain dictates this frequency. Accomplishing various experiments on five different CNC machines, limitations for the bandwidth of the frequency of vibration in a servo axis were defined. Also, an experimental equation was provided that estimates this frequency with an acceptable accuracy for various CNC machines.

IX. REFERENCES

[1] Stein, J. L., and Wang, C. H., "Automatic detection of clearance in mechanical systems: theory and simulation," Proceedings of the American Control Conference, Volume 3, 21-23 June 1995, pp. 1737-1745.

[2] Lagerberg, A., and Egardt, B., "Backlash Estimation With Application to Automotive Power trains," *IEEE Transactions on Control Systems Technology*, Volume 15, Issue 3, May 2007, pp. 483-493.

[3] Gerdes, J.C., and Kumar, V., "An impact model of mechanical backlash for control system analysis," *Proceedings of the American Control Conference,* Volume 5, 21-23 June 1995, pp. 3311-3315.

[4] Jukic, T., and Peric, N., "Model based backlash compensation," *Proceedings of American Control Conference*, Volume 2, 25-27 June 2001, pp. 775-780.

[5] Campos, J., Lewis, F.L., and Selmic, R., "Backlash compensation with filtered prediction in discrete time nonlinear systems by dynamic inversion using neural networks," *IEEE International Conference on Decision and Control*, Volume 4, 12-15 Dec. 2000, pp. 3534-3540.

[6] Gang, T., and Kokotovic, P.V., "Continuous-time adaptive control of systems with unknown backlash," *IEEE Transactions on Automatic Control*, Volume 40, Issue 6, June 1995, pp. 1083 - 1087.

[7] Tao, G., and Kokotovic, P.V., "Adaptive control of system with unknown output backlash," *IEEE Transactions on Automatic Control*, Volume 40, Issue 2, Feb. 1995, pp. 326 - 330.

[8] Jing Zhou, Chengjin Zhang, and Wen, C., "Robust Adaptive Output Control of Uncertain Nonlinear Plants With Unknown Backlash Nonlinearity," *IEEE Transactions on Automatic Control*, Volume 52, Issue 3, March 2007, pp. 503 - 509.

[9] Woo, K. T., Li-Xin W., Lewis, F. L., and Li, Z. X., "A fuzzy system compensator for backlash," *IEEE International Conference on Robotics and Automation*, Volume 1, 16-20 May 1998, pp. 181 - 186.

[10] Jun Oh Jang, Pyeong Gi Lee, Hee Tae Chung, and Gi Joon Jeon, "Output backlash compensation of systems using fuzzy logic," *Proceedings of American Control Conference*, Volume 3, 4-6 June 2003, pp. 2489-2490.

[11] Seidl, D.R., Sui-Lun Lam, Putman, J.A., and Lorenz, R.D., "Neural network compensation of gear backlash hysteresis in position-controlled mechanisms," *IEEE Transactions on Industry Applications*, Volume 31, Issue 6, Nov.-Dec. 1995, pp. 1475 - 1483.

[12] Simens simodrive 611 digital/SINUMERIC840D/810D startup and optimization.

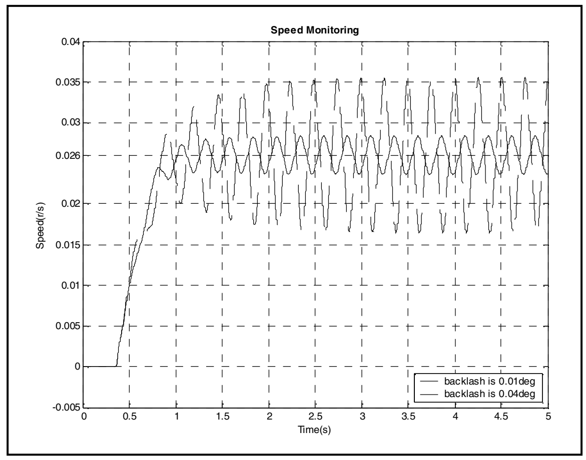

Figure 10. Effect of value of backlash on vibration

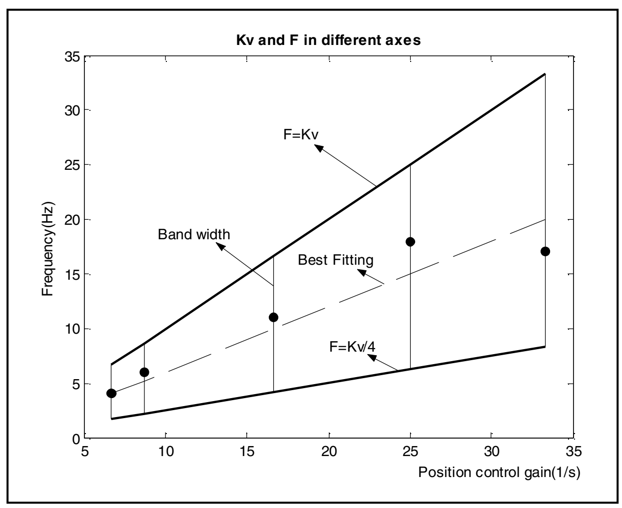

Figure 11. Frequency of vibration and the position control gain