Proceedings of the World Congress on Engineering 2010 Vol II
WCE 2010, June 30 - July 2, 2010, London, U.K.

The Assessment on Differences of Vertical and Horizontal Motion in 3-axis Machine Tool: Weight and Counterweight

Duy Le, Hyun-Kwang Cho, Su-Jin Kim, Gyung-Ho Kim, Chun-Hong Park

***Abstract*—In 3-axis machine center, accuracy of vertical motion is worse than the other two horizontal motions because of structure differences. These main problems, gravity effect of table weight and vibration of counterweight, are mainly discussed. Then additional components are proposed to modify the horizontal transfer function for the application on predicting motion error of vertical guide. And vibration prediction is introduced to evaluate the effect of counterweight types in order to help engineering designers to optimize machine structures.**

***Index Terms*—Vertical guide, horizontal guide, transfer function, counterweight vibration.**

I. INTRODUCTION

High precision machine centers are required in most of manufactures because the demand of high accurate components and consistency of quality are growing. The most important factor of the precision components is the accuracy of machine tools. Mainly, position errors are originated from geometric, cutting force, dynamic loading, etc. [1,2]. In order to improve machine tool accuracy, machine tool geometric errors as well as precision positioning have been being characterized and predicted for high effective machine design [3]. In 3-axis machine center, the accuracy of vertical motion is worse than the other two horizontal motions because of structure differences. Gravity effect of table weight and vibration of counterweight are mainly discussed in the current research.

The first difference is the effect of gravity to machine structure. Table weight in vertical motion causes large axial load on ball screw which is largely different to horizontal case. Misalignment of connecting chain or wire between machine head and counterweight causes moments around moving table. Moreover additional moment components - M_x, M_y and M_z - are proposed to modify the horizontal transfer function [4,5] of linear guide so that it can be applied to vertical case.

The second one is about counterweight. Although vertical system is assembled with counterweight to increase their capacity and reliability, the precision is reduced due to vibration [6,7]. Specifically, when the machine's head decelerates until stops at the desired point, it causes wavy marks on the workpiece, which reduce the surface quality during machine process. There are three kinds of counterweight: mechanical, hydraulic and pneumatic counterweight, which can be divided into two vibration models. Estimating and characterizing stiffness and damping ratio of machine parts are suggested in each model. Finally, vibration prediction is proposed to evaluate the effect of counterweight types in other to help machine designers to optimize machine structure.

Manuscript received April 7, 2010. This work is supported by the Ministry of Knowledge Economy of South Korea.
Su-Jin Kim is with the Department of Mechanical Engineering, Gyeongsang National University, Jinju, 660-701 South Korea (Phone: 82-55-751-6075; fax: 82-55-762-0227; e-mail: sujinkim@gnu.ac.kr).
Duy Le, Hyun-Kwang Cho are with Department of Mechanical Engineering, Gyeongsang National University, Jinju, 660-701 South Korea (e-mail: duyle2003@gmail.com; elgar328@nate.com).
Gyung-Ho Kim, Chun-Hong Park are with Korea Institute of Machinery & Materials, Daejon, 305-343 South Korea (e-mail: gyungho@kimm.re.kr; pch657@kimm.re.kr).

II. THE DIFFERENCES OF VERTICAL AND HORIZONTAL MOTION REVIEW

Rail deformation: There are two rail types: one is separated rails which are bolted to machine body; the other is made with the machine body itself which is assumed as super rigid. Then bolted rail type should be considered to observe the more general and larger deformation. A study case with pressure (q) - from spindle weight - acting on the rail through ball slides is 25.10^3 Pa. According to the technical guide of NSK rolling guides [8], NSK LY35 is chosen. Fig. 1 (a) shows the deformation model of bolted rail of rolling guide. Fig. 1 (b) gives the analysis result solved by Nastran in which the maximum deformation is about 0.2μm. This result is negligible or can be overcomed by using stronger rail model.

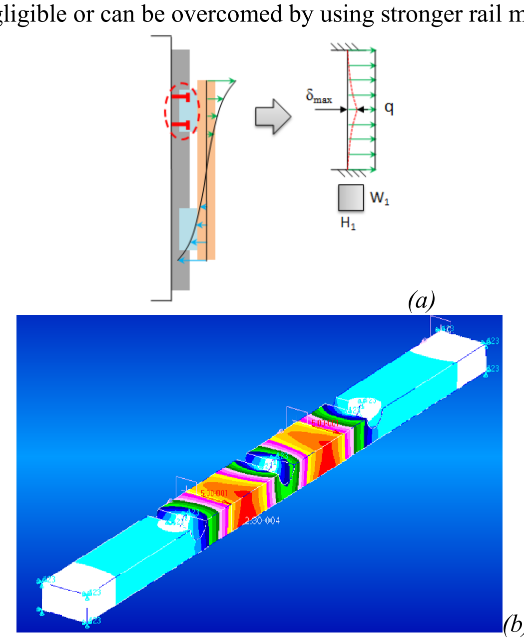


Fig. 1 (a) Bolted rail model
(b) Nastran analysis of bolted rail model

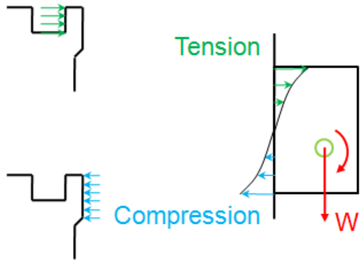
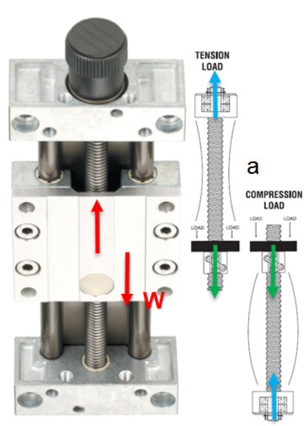


Fig. 2 Different stiffness in vertical sliding guide

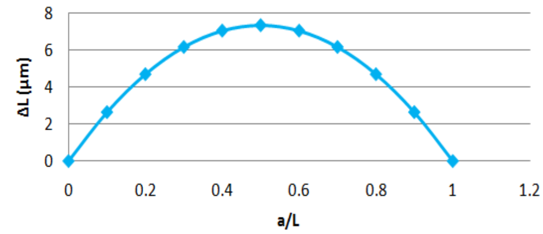
Rail Stiffness: Different from horizontal guides, vertical guides deal with big moment from spindle weight. It causes tension on top side and compression on bottom side of the contact area between guide and rail. In case of rolling guide, tensile stiffness and compressive stiffness are the same because special structure of the rails helps them to take the pressure with same area on both sides. But in sliding guide, tensile stiffness is smaller than compressive stiffness because of different areas as shown in Fig. 2.

Ball screw deformation: Ball screw in vertical case normally loads large axial force - from the spindle - on the ball nut. It causes axial displacement on the screw during the motion. Fig. 3 (a) gives an example of screw deformation in case of fix-fix assembly with 300kg load, screw length 1m, screw diameter 25mm, E = 207GPa. And Fig. 3 (b) shows the results of displacement when the ball nut moves from top to bottom of the guide. This error can be avoided by adding counterweight support or compensating on machine controller.

Backlash: In case of no counterweight, there is no backlash because the load pushes down on the nut keeping it in constant contact with the screw. If counterweight is present, backlash depends on the percentage of the counterweight as shown in Fig. 4. But nowadays these problems can be solved by using anti backlash ball screw [9] or compensating for backlash on the machine controller [10]. So, in vertical motion, backlash is not necessary to be considered as an issue. Accuracy is maintained whether the load is being raised or lowered. Another advantage of vertical motion applications is that the torque needed to lower the load is less than that required to raise it. This means there are possible opportunities for downsizing the motor. However it is always necessary to brake the screw shaft with the motor to prevent any backdriving.



(a)



(b)

Fig. 3 (a) Fix-fix ball screw displacement model
(b) Displacement and ball nut position

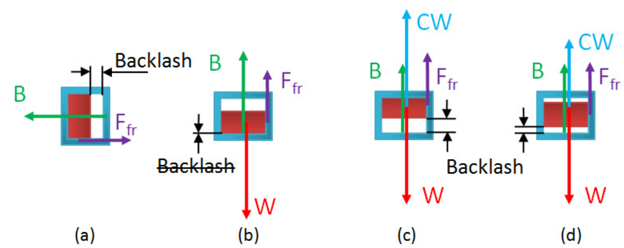


Fig. 4 Backlash in horizontal and vertical cases
(a) Horizontal case
(b) Vertical without counterweight
(c) 100% counterweight
(d) 70% counterweight

III. Vertical Model

A. Current Transfer Function

Equilibrium equations from transfer function of [4].

$$\sum \vec{F} = \vec{0} \quad (1)$$

$$\sum_{j}^{n_v}\sum_{i}^{m_v}\left(f_{z,ij} - K_z z_{ij}\right) - f_{z,r} + f_{z,e} = 0$$

$$\sum_{j}^{n_h}\sum_{i}^{m_h}\left(f_{y,ij} - K_y y_{ij}\right) - f_{y,r} + f_{y,e} = 0$$

$$\sum \vec{M} = \vec{0} \quad (2)$$

$$\sum_{j}^{n_v}\sum_{i}^{m_v}\left\{\left(f_{z,ij} - K_z z_{ij}\right)\left(R_x - X_{vci}\right)\right\} - M_{ry} = 0$$

$$\sum_{j}^{n_h}\sum_{i}^{m_h}\left\{\left(f_{y,ij} - K_y y_{ij}\right)\left(R_x - X_{hci}\right)\right\} - M_{rz} = 0$$

$$\sum_{j}^{n_v}\sum_{i}^{m_v}\left\{\left(f_{z,ij} - K_z z_{ij}\right)\left(R_y - Y_{cj}\right)\right\} - R_z\sum_{j}^{n_h}\sum_{i}^{m_h}\left(f_{y,ij} - K_y y_{ij}\right) = 0$$

They work almost perfectly in case of horizontal motion. But in the case of vertical motion they might need to be modified. That is the reason to build the separated model of forces and moments in case of vertical motion.

B. Force and Moment Modeling

Assuming ball nut is the center of system coordinates as shown in Fig. 5 (a). Counterweight force and head weight will be considered generally as **C**(C_x, C_y, C_z) and **W**(W_x, W_y, W_z). By projecting them onto each plane of the coordinates, moments and forces components are expressed:

ISBN: 978-988-18210-7-2
ISSN: 2078-0958 (Print); ISSN: 2078-0966 (Online)

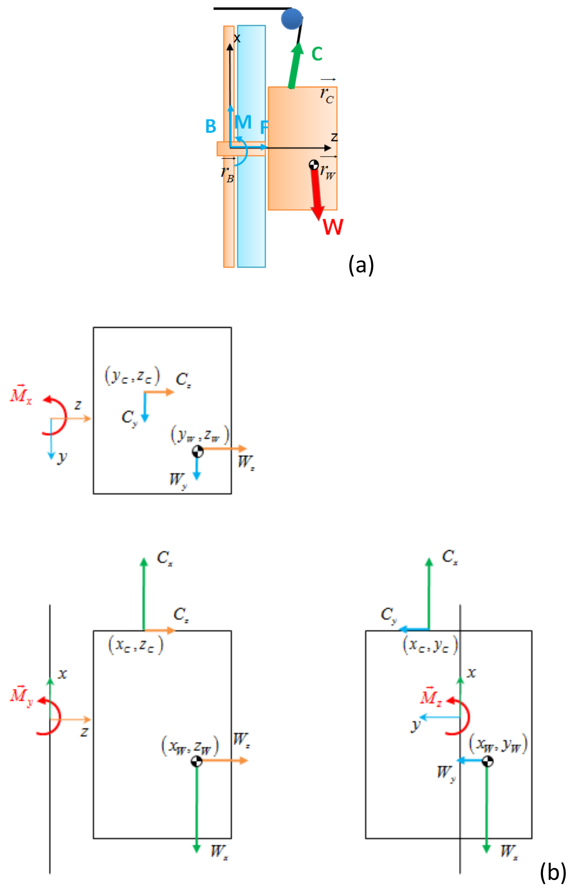


Fig. 5 Force and moment model
(a) General model
(b) Projected components

$$\begin{cases} M_x = -\left(W_y z_W + C_y z_C\right) + \left(W_z y_W + C_z y_C\right) \\ M_y = \left(W_x z_W + C_x z_C\right) - \left(W_z x_W + C_z x_C\right) \\ M_z = -\left(W_x y_W + C_x y_C\right) + \left(W_y x_W + C_y x_C\right) \end{cases} \quad (3)$$

$$\begin{cases} F_y = -\left(W_y + C_y\right) \\ F_z = -\left(W_z + C_z\right) \end{cases} \quad (4)$$

Those forces and moments are different to components in Eqns. (1) and (2). It is suggested to modify and generalize the components within those equations to improve the application of transfer function.

C. Setting error of chain or wire

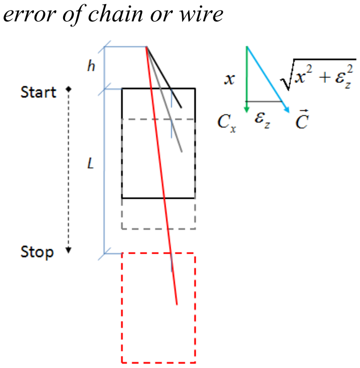


Fig. 6 Misalignment of connecting chain/wire to spindle head

In assembly process of connecting the chain or wire to the machine head, misalignment usually happens. Fig. 6 is an example that visualizes the changes of counterweight components C_x during the movement of the head, where $h \le x \le h + L$, h is initial height, L is travelling height.

Fig. 7 demonstrates the moment errors through setting error following the case:

$$\mathbf{W}(1000,0,0) \quad \mathbf{C}(C_x,0,C_z) \quad \mathbf{C} = 800N \quad h = 0.4m \quad L = 1m$$

$$x_C = 0.3m \quad x_W = -0.2m \quad y_C = 0.1m \quad y_W = -0.1m \quad z_C = 0.2m \quad z_W = 0.4m$$

At a quite large setting error of 50mm the error of M_x is less than 10N/m, M_y and M_x are less than 3%. These errors can be overcomed by increasing the stiffness of the vertical rail.

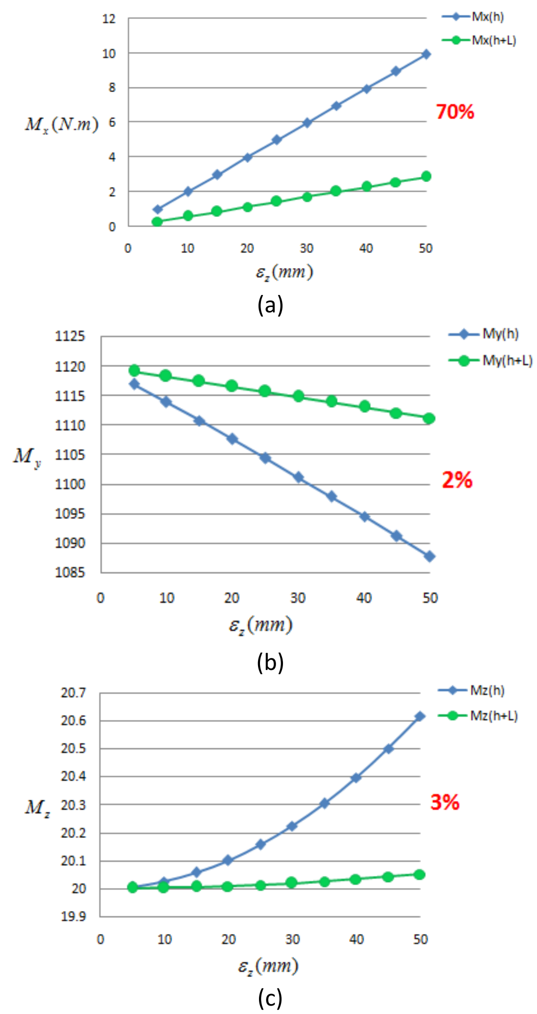


Fig. 7 Moment error and setting error
(a) M_x
(b) M_y
(c) M_z

ISBN: 978-988-18210-7-2
ISSN: 2078-0958 (Print); ISSN: 2078-0966 (Online)

IV. Counterweight and Vibration

A. Vibration Model

Machine tool with counterweight has higher load capacity and more safety during operation. But adding one more component to the machine structure means dynamic problem needs to be reconsidered. And vibration analysis is coming up. By simplifying machine center as Lin's work [6], vibration model can be carried out. Ball nut m_3 is considered as a source giving motion to machine head m_2. Machine head is connected to counterweight system m_1 by chain or steel wire with stiffness k_1 and damping c_1. Connection between the spindle and ball nut has stiffness k_2 and damping c_2. Fig. 8 (a) shows the mechanical counterweight model and vibration equation is:

$$M\{\ddot{x}\}+C\{\dot{x}\}+K\{x\}=\{F(t)\} \tag{5}$$

$$\begin{bmatrix} m_1 & 0 & 0 \\ 0 & m_2 & 0 \\ 0 & 0 & 1 \end{bmatrix}\begin{bmatrix} \ddot{x}_1 \\ \ddot{x}_2 \\ \ddot{x}_3 \end{bmatrix}+\begin{bmatrix} -C_1 & C_1 & 0 \\ C_1 & -(C_1+C_2) & C_2 \\ 0 & 0 & 0 \end{bmatrix}\begin{bmatrix} \dot{x}_1 \\ \dot{x}_2 \\ \dot{x}_3 \end{bmatrix}+\begin{bmatrix} -K_1 & K_1 & 0 \\ K_1 & -(K_1+K_2) & K_2 \\ 0 & 0 & 0 \end{bmatrix}\begin{bmatrix} x_1 \\ x_2 \\ x_3 \end{bmatrix}=\begin{bmatrix} 0 \\ 0 \\ f(t) \end{bmatrix}$$

Fig. 8 (b) presents the hydraulic or pneumatic counterweight model. Vibration equation in this case can be expressed as Eqn. (5) with $m_1 << m_2$.

B. Vibration analysis

Fig. 9 introduces a schematic model built in MATLAB Simulink. Initial parameters of this study case are:

$m_1 = 28$ (kg)
$m_2 = 38$ (kg)
$k_1 = 1.15\text{x}10^4$ (N/m)
$c_1 = 121$ (Ns/m)
$k_2 = 9.86\text{x}10^6$ (N/m)
$c_2 = 9730$ (Ns/m)

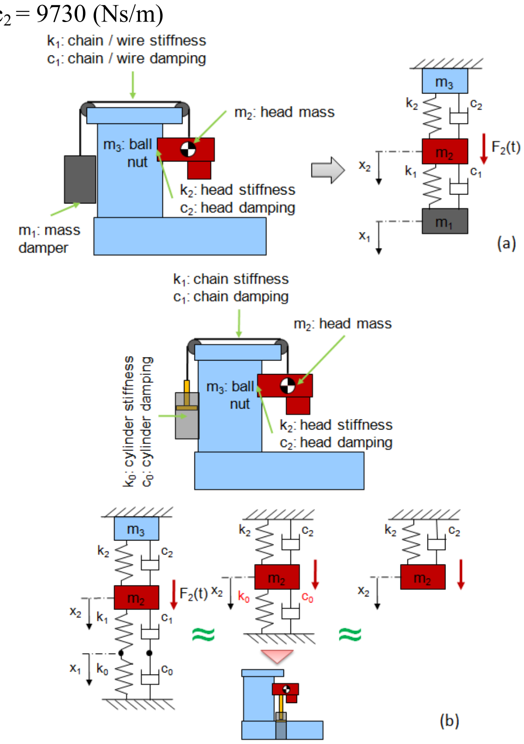


Fig. 8 Vibration model
(a) Mechanical counterweight
(b) Hydraulic and Pneumatic counterweight

One complete move of machine head to the target is normally divided in 3 stages: acceleration, constant and deceleration. Fig. 10 expresses velocity and acceleration through time during one move of the ball nut, meanwhile f(t) in Eqn. (5) should be:

$$f(t)=\begin{cases} a \text{ if } 0 \le t \le 0.5 \\ 0 \text{ if } 0.5 \le t \le 1.5 \\ -a \text{ if } 1.5 \le t \end{cases} \tag{6}$$

From that input condition displacements or position errors between spindle head and ball nut positions are plotted in Fig. 11. Fig. 11 (b) gives better results compare to Fig. 11 (a), which mean hydraulic and pneumatic counterweight and non-counterweight are more reliable than mechanical counterweight.

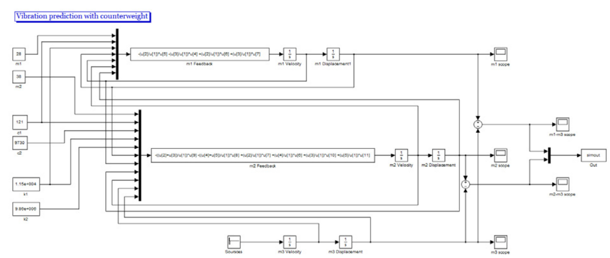


Fig. 9 Matlab simulink

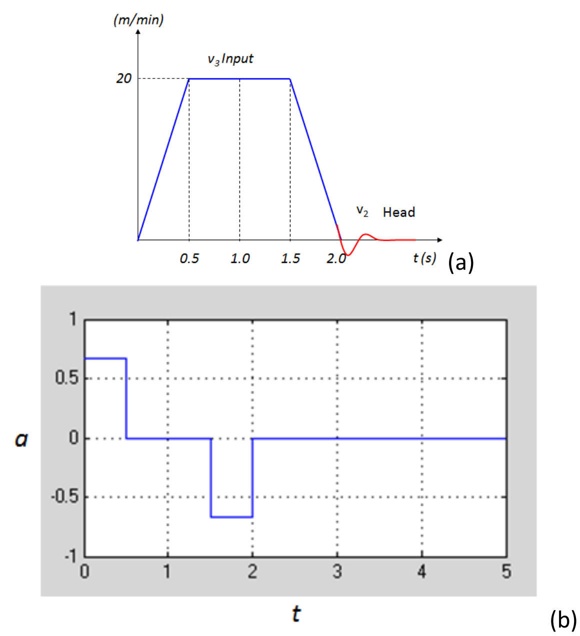
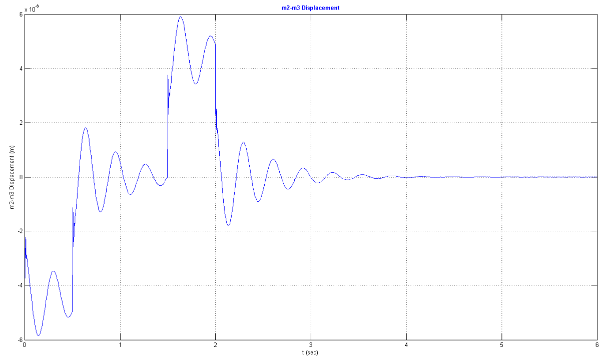
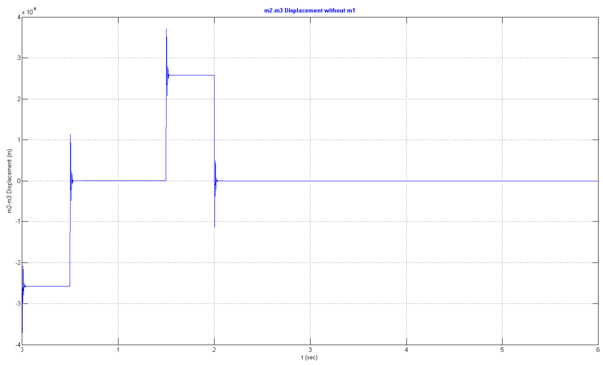


Fig. 10 Ball nut input
(a) Velocity
(b) Acceleration

ISBN: 978-988-18210-7-2
ISSN: 2078-0958 (Print); ISSN: 2078-0966 (Online)



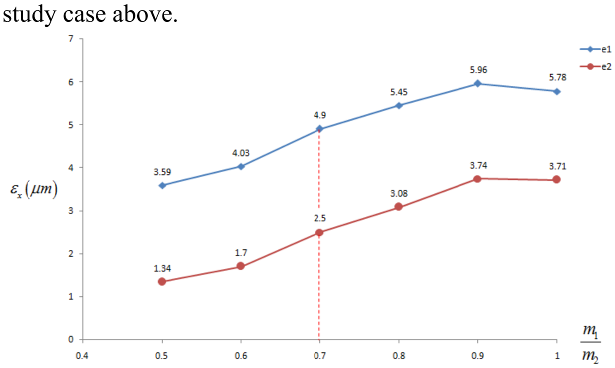
(a)



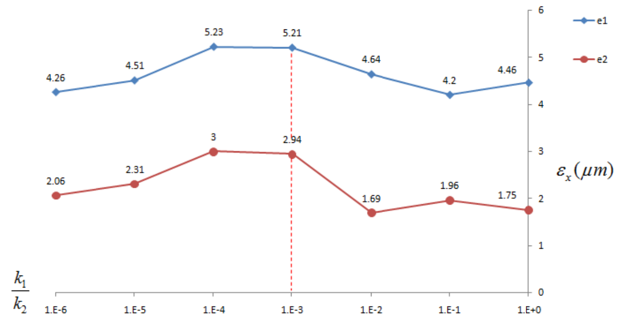
(b)

Fig. 11 Displacement of spindle due to input of ball nut
(a) Mechanical counterweight
(b) Hydraulic and pneumatic counterweigh, or without counterweight

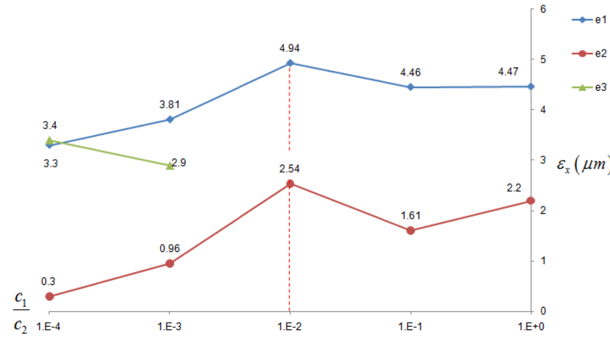
As mentioned above, after the head moves to the target point, it keeps moving up and down for a very short period before considered as permanent stop. The two first upward errors are expressed in Fig. 11. Figs. 11 (a), (b) and (c) give the relation between position errors and weight ratio of m_1 and m_2, stiffness ratio of k_1 and k_2, damping ration of c_1 and c_2, respectively. The vertical dash lines are marked for the study case above.



(a)



(b)



(c)

Fig. 11 Position errors after the spindle head reach the target
(a) With weight ratio
(b) With stiffness ratio
(c) With damping ratio

C. Chain Vibration

According to chain and sprocket assembly there is difference between velocity of machine head and counterweight system, as described in Fig. 12. The relation of velocity ratio and chain model is given by

$$1-\left(\frac{p^2}{4r_{\min}^2+p^2}\right)\le\frac{v_3}{v_1}\le 1+\frac{p^2}{4r_{\min}^2} \quad (7)$$

and shown by Fig. 13 (a).

One example is carried out to explore the force vibration of counterweight system.

$v_3 = 20$ (m/min)

r = 100 (mm)

Chain LH0822, p = 12.7 [11]

$$v_1(t)=0.001341\times\cos(165t)+\frac{1}{3}\ \text{m/s}$$

$$a_1(t)=-0.22\sin(165t)\ \text{m/s}^2$$

$$F_1(t)=-66\sin(165t)\ +3000\ \ \text{N} \quad (8)$$

From Eqn. (8) Fig. 13 (b) shows the force vibration with maximum amplitude is less than 3%.

ISBN: 978-988-18210-7-2
ISSN: 2078-0958 (Print); ISSN: 2078-0966 (Online)

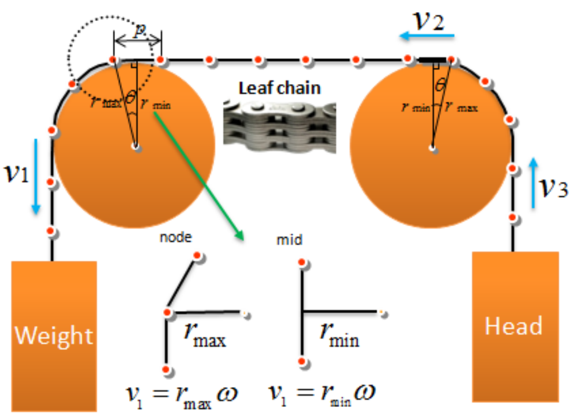


Fig. 12 Velocity difference by chain assembly

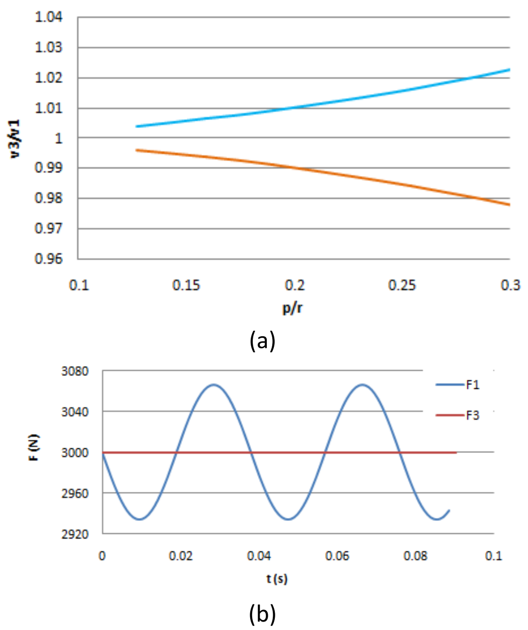


Fig. 13 (a) Velocity ratio and chain model
(b) Force fluctuation with chain LH0822

V. Conclusion

This assessment reviews the differences between vertical and horizontal motion in machine center. From the review two problems are raised and solutions are also proposed. Specific forces and moment model for vertical guide are built. But for the purpose of applying modifications to current transfer function, the model needs to be generalized. Vibration of mechanical counterweight and hydraulic or air counterweight system are predicted and evaluated.

Further works and experiments need to be solved for generalize the results into practical applications.

Acknowledgment

The authors wish to acknowledge the Ministry of Knowledge Economy of South Korea and Korea Institute of Machinery & Materials for supporting this work.

References

[1] Alexander H.Slocum, *PRECISION MACHINE DESIGN*. Prentice-Hall International, Inc, 1992.

[2] Chana Raksiri, Manukid Panichkun, "Geometric and force errors compensation in a 3-axis CNC milling machine", *Int. J. Mach Tools Manuf*, vol. 44, 2004, pp. 1283-1291.

[3] H. Vermeulen, M. Vermeulen, S. Wetzels, W. Pril, "Design for Precision: Current Status and Trends", Keynote paper, Section Precision Engineering, Eindhoven University of Technology, The Netherlands, *Annals of the CIRP*, vol. 47, Feb. 1998.

[4] Eiji Shamoto, Chun-Hong Park, Toshimichi Moriwaki, "Analysis and Improvement of Motion Accuracy of Hydro Static Feed Table", *CIRP Annals - Manufacturing Technology*, vol. 50, 2001, pp. 2285-290.

[5] Chun-Hong Park, Yoon-Jin Oh, Joon-Hee Hong, Chan-Hong Lee, "Theoretical Verification on the Motion Error Analysis Method of Hydrostatic bearing Table Using a Transfer Function", Int. J. Prec Eng and Manuf, vol. 4, 2003, pp. 64-70.

[6] S. Y. Lin, Y. C. Fang, C. W. Huang, "Improvement strategy for machine tool vibration induced from the movement of a counterweight during machining process", *Int. J. Mach Tools Manuf*, vol. 48, 2008, pp. 870-877.

[7] Ming-Hung Tsai, Ming-Chang Shih, "A Study of the Pneumatic Counterweight of Machine Tools Conventional and Active Pressure Control Method", *JSME Int. Journal*, Series C, vol. 49, 2006, pp. 890-896.

[8] *NSK LINEAR ROLLING GUIDES*. NSK Technical Journal, pp. A14-A78. Available: http://pdf.directindustry.com/pdf/nsk-precision-america/linear-rolling-guides-17324-1238.html

[9] Thomson BSA Lead and Ball Screw. Available: http://www.thomsonbsa.com/pdf/ThomsonBSA-Lead-and-Ball-Screw-Catalog.pdf

[10] CNC Linear Controller, P/N 8800. Available: http://www.sherline.com/8800inst.pdf

[11] Renold Lifting Chain Catalogue. Available: http://www.renold.com/nmsruntime/saveasdialog.asp?lID=463&sID=1860