MM Science Journal | www.mmscience.eu
ISSN 1803-1269 (Print) | ISSN 1805-0476 (On-line)

Special Issue | HSM 2014
11th International Conference on High Speed Machining

September 11-12, 2014, Prague, Czech Republic

HSM2014-14038

DYNAMIC MODELLING OF AN ELECTRO–HYDRAULIC ACTUATOR TO ISOLATE MACHINE TOOLS FROM GROUND VIBRATIONS

M. Wabner*, M. Law, S. Ihlenfeldt

Fraunhofer Institute for Machine Tools and Forming Technology IWU, Chemnitz, Germany

*Corresponding author; e-mail: markus.wabner@iwu.fraunhofer.de

Abstract

This paper presents the dynamic modelling of a novel electro-hydraulic actuator that decouples, attenuates and isolates ground motion to keep dynamic excitations transmitted to machine tools below permissible levels. Closed-loop dynamics are formulated by considering the hydraulic system dynamics as well as the dynamics of the actuator and machine tool. Controlled closed-loop transmissibility shows the actuator to have excellent attenuation of ground vibrations for excitation frequencies of up to 30 Hz. Numerical investigations representative of typical floor vibration levels experienced by machine tools demonstrate that the device isolates ground motion by up to 80%.

Keywords:

Machine Tools; Active Vibration Isolation; Electro-hydraulic Actuator

1 INTRODUCTION

In typical production environments, machine tools are often subjected to high levels of inertial accelerations from ground motions that strain and distort the machine; the distortion depending on machine construction, machine mounting and the amount and direction of acceleration that the machine experiences [DeBra 1992]. Since it is difficult to insulate the machine from its production environment or from vibration producing machines and devices, it becomes necessary to isolate the machine from such disturbances. Machine isolation is critical for assuring their adequate performance, especially in situations where the amplitude of external vibrations significantly exceeds the magnitude of allowable machining and measurement deviations. Additionally, since vibration transmission can also result in machine instability and even lead to failure of some critical high value parts, it becomes ever more important to isolate the machine from ground vibrations.

The goal of vibration isolation hence is to protect the machine from severe ground vibrations and to reduce the intensity of dynamic excitations transmitted to the machine such that relative vibration amplitudes in the work zone are kept below permissible levels. This is achieved by decoupling or attenuating the ground motion such that the relative vibrations are acceptable [DeBra 1992].

In most cases the isolation requirements can be satisfied by judiciously designed passive isolators that have high damping and relatively small support stiffness [Rivin 1995]. Though passive solutions offer design simplicity and cost-effectiveness, they can lead to static and dynamic instabilities that result in rocking of the isolated machine on the soft passive mount; and they also suffer from poor high-frequency isolation. Moreover, in situations when the ground vibrates with an unpredictable waveform which has a broadband spectrum, it becomes difficult to design and implement effective passive isolation solutions and active solutions are often necessary.

Active isolation solutions include a feedback circuit that consists of a sensor, a controller and an actuator. Such solutions have shown to be very effective in automotive suspensions, telescopes, satellites and even in seismic applications. Actuating mechanisms in these applications include isolators that use hydraulic, pneumatic, electromagnetic, piezoelectric, and magnetorheological (MR) fluid-elastomer solutions. However, because of the small deflection capacity of piezoelectric actuators [Huang 2003] and the nonlinear force hysteresis characteristics of MR mounts [Choi 2005], these have found limited use in machine tool applications.

Competing requirements of high static stiffness that can support the large inertial load of the machine combined with low dynamic stiffness requirements for effective isolation, makes the design and use of active isolators difficult in machine tool applications. Active isolators for machine tools must be sensitive to the dynamic characteristics of the machine being isolated and to the spectral characteristics of the floor vibrations. Additionally, active isolators must not directly alter the undamped dynamics of the system, nor should they adversely affect the static stiffness of the system.

To address the above issues, a novel electro-hydraulic actuator has been developed at the Fraunhofer IWU; see [Bischoff 2009] and [Wabner 2012] for details. The first generation of such electro-hydraulic actuators were reported in [Abicht 2002], and were used for effectively isolating vibrations of Stewart-platforms. The proposed solution advances earlier prototypes by developing a double-acting isolation system that works on the principle of displacement compensation. Proposed solution also has a high passive stiffness. This electro-hydraulic actuator is designed to mitigate ground vibrations only in vertical directions and is not effective against lateral ground vibrations, even though these may also degrade performance of precision machine tools [Rivin 1995].

To verify the effectiveness of the proposed solution, this paper presents the dynamic modelling of this electro-hydraulic system to demonstrate how ground vibrations may be efficiently isolated. At first, the working principle of the actuator is discussed in Section 2. This is followed by formulating the closed-loop dynamics of the electro-hydraulic system in Section 3. Though the machine is flexible and its flexibility influences the closed-loop dynamics of the electro-hydraulic system, for preliminary investigations in this study, the machine is modelled as a lumped mass system. Actuator dynamics are evaluated virtually in Section 4 and simulation driven investigations are used to demonstrate effective isolation of ground vibrations in Section 5 followed by the main conclusion in Section 6.

2 ACTUATOR WORKING PRINCIPLE

The basic elements of the electro-hydraulic system are schematically shown in Fig. 1 and constructional details of the actuator are shown in Fig 2. The machine is rigidly mounted onto the actuator, as shown in Fig. 1. The ground motion sensor picks up acceleration signals from the bridge support of the actuator that is resting on the ground. Acceleration detected by the sensor provides the command signal (u_c) to the controller, which in turn produces a signal to drive the proportional valve. The proportional valve controls the fluid flow to the actuator in proportion to its drive current. The electro-hydraulic actuator is akin to a double-acting hydraulic cylinder with two different pressure chambers, as can be seen in Fig. 2. Flow to the actuator results in movement of the pressure plates and the machine against reaction from the membrane spring assembly. Actuator output is measured with a transducer that converts motion to an electric signal. This feedback signal (u_f) is compared with the command signal resulting in a finite error signal. A well designed controller will help to counter the disturbance by moving the machine in an attempt to zero the error signal.

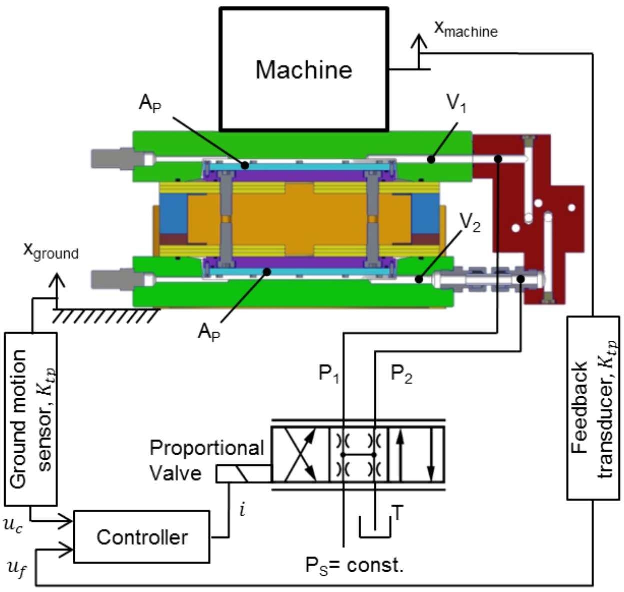

Fig. 1: Schematic of position controlled electro-hydraulic system with ground motion as an external disturbance. Nomenclature used above is explained in Section 3.

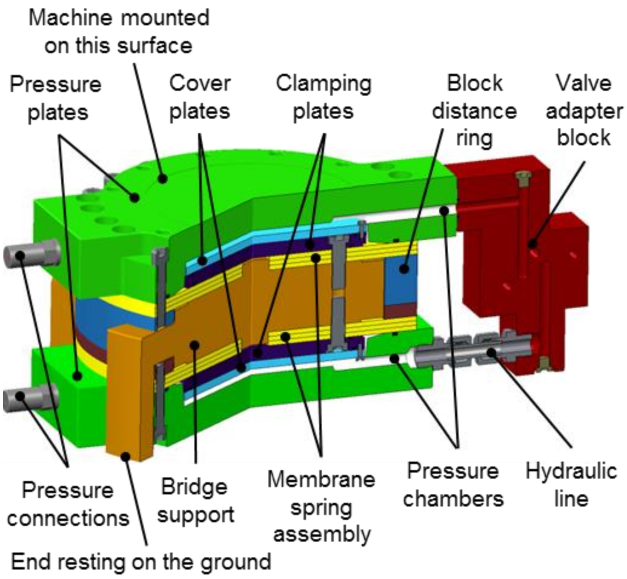

Fig. 2: Sectional view of the construction of the actuator.

The actuator acts only in the presence of a command signal received from the ground motion sensor. In the absence of any external disturbance, the actuator behaves like a passive system, with the membrane spring assembly providing a very high passive stiffness. The device shown in Fig. 2 is designed and dimensioned for a static load of up to 4 tonnes [Bischoff 2009].

3 MODELLING THE CLOSED-LOOP DYNAMICS

Since the actuator has to act against ground motion which may have broadband spectrum, dynamics of each of the subsystems shown in Fig. 1 that include the valve, piston and controller dynamics need to be modelled to formulate the closed–loop dynamics of the electro-hydraulic system subject to ground excitations, as discussed below.

3.1 Valve dynamics

An electro-hydraulic proportional valve acts as a high gain electrical to hydraulic transducer. The valve's electronic driver converts the commanded voltage to current. The valve's solenoid converts this current into a mechanical force acting the spool against a return spring; which in turn results in spool movement and regulation of valve flow. Proportional valves are highly complex devices that exhibit high-order nonlinear responses. Valve dynamics are influenced by nozzle and orifice sizes, spring rates, spool diameter, spool displacements, supply pressure, input signal levels, hydraulic fluid temperature, valve loadings, etc., making it very difficult to formulate an accurate mathematical model. Since the proportional valve selected approximates real servo valve dynamics [Parker 2011], it is convenient to represent the valve dynamics by a second order transfer function for preliminary level of analysis, which may later be updated by empirical approximations of the measured servo valve response at an advanced stage of investigations. Hence, the proportional valve's dynamics may be represented as [Moog 1965]:

$$\frac{Q}{i}(s) = K_Q\left[\frac{1}{1+\left(\frac{2\zeta}{\omega_n}\right)s+\left(\frac{s}{\omega_n}\right)^2}\right] \tag{1}$$

wherein Q is the valve flow in m^3/s; i is the current; ω_n is the apparent natural frequency of the valve; and ζ is the apparent damping ratio of the valve. K_Q corresponds to a valve flow coefficient represented in $m^3/s/mA$.

3.2 Valve flow and chamber pressure

Valve flow is supplied to the pressure chambers of the actuator that results in movement of the pressure plates of the actuator and the machine mounted on the actuator. Valve flow rate can be calculated based on the classical orifice equation given as [Meritt 1967]:

$$Q = A(x_v)C_d\sqrt{\frac{2}{\rho}(P_1 - P_2)} \tag{2}$$

wherein $A(x_v)$ is the cross-sectional area of the orifice which is a function of the valve spool position x_v, and the valve parameters (spool radial clearance and width of ports); C_d is an empirically obtained discharge coefficient; ρ is the density and P_1, P_2 may correspond to either the supply pressure (P_S), return pressure or chamber pressures in the cylinder, as shown in Fig. 1.

Furthermore, for the dynamic equilibrium condition of the actuator, the flow continuity equation in the chambers of the actuator may be represented as [Bishop 2002]:

$$\rho\left(\sum Q_{IN} - \sum Q_{OUT}\right) = \frac{d(\rho V)}{dt} = \rho\frac{dV}{dt} + V\frac{d\rho}{dt} \tag{3}$$

wherein Q_{IN} and Q_{OUT} represent the flows entering and leaving the chambers and V is the chamber volume, i.e. for either of the chambers, V_1 or V_2.

From the definition of the compressibility modulus of oil β, we can replace the density term in Eq. (3) as:

$$\frac{dV}{V} = -\frac{d\rho}{\rho} = \frac{dP}{\beta}. \tag{4}$$

Substituting the above Eq. (4) into Eq. (3) we get:

$$\sum Q_{IN} - \sum Q_{OUT} = \frac{dV}{dt} + \frac{V}{\beta}\frac{dP}{dt} \tag{5}$$

wherein the left hand side of the equation corresponds to the net flow delivered to the chamber by the valve. The first term on the right hand side in Eq. (5) is the flow consumed by the changing volume caused by motion of the piston, and the second term accounts for any compliance (compressibility) in the system. Eq. (5), when rearranged as:

$$P_{1,2} = \frac{\beta}{V}\int\left(Q_{1,2} - \frac{dV_{1,2}}{dt}\right)dt \tag{6}$$

can be used to determine the instantaneous pressure in any of the chambers.

3.3 Piston dynamics

Once the two chamber pressures are known, the net force acting on the piston (F_P) can be calculated as a function of the piston area (A_p) and load pressure (P_L):

$$F_P = (P_1 - P_2)A_p = P_L A_p. \tag{7}$$

Additionally, since the valve drives the actuator which is connected to a translating mass and is akin to a single degree of freedom system (under a lumped mass model assumption for the machine), the dynamic equilibrium equation of the piston can be expressed as:

$$F_P = M\frac{d^2x_{machine}}{dt^2} + F_f + K_L x_{machine} \tag{8}$$

wherein M is the translating mass which includes the mass of the piston and the load, i.e. the machine; F_f is the viscous friction which is a function of the piston velocity and K_L is the load stiffness corresponding to the static stiffness of the actuator.

3.4 Simplified analysis

Clearly, from the above formulations, i.e. Eq. (2-8), it is evident that the valve flow, pressure and piston forces are a function of multiple parameters that results in the flow and pressure to vary dynamically. However, as a first level of analysis, for the special case of a centred piston, the above system of equations, namely Eq. (2) and Eq. (5-6) can be linearized to obtain the following Laplace transformed flow equations [Rydberg 2008]:

$$K_Q x_v = A_p s x_{machine} + \left(K_{ce} + \frac{V_t}{4\beta}s\right)P_L \tag{9}$$

$$P_L A_p = Ms^2 x_{machine} + Csx_{machine} + K_L x_{machine} \tag{10}$$

wherein K_Q corresponds to a valve flow coefficient as in Eq. (1); x_v is the valve spool position as in Eq. (2); A_p is the area of the piston; K_{ce} is a flow pressure coefficient which has units of $m^3/s/Pa$; V_t ($V_t = 2V_1, or = 2V_2$) is the total oil volume in the chamber, i.e. on both sides of the piston and P_L is the load pressure, i.e. the difference in pressures between both sides of the chambers. The leakage flow (if any) is assumed negligible in this analysis.

These equations, i.e. Eq. (9-10) result in a block diagram level description of the system shown in Fig. 3.

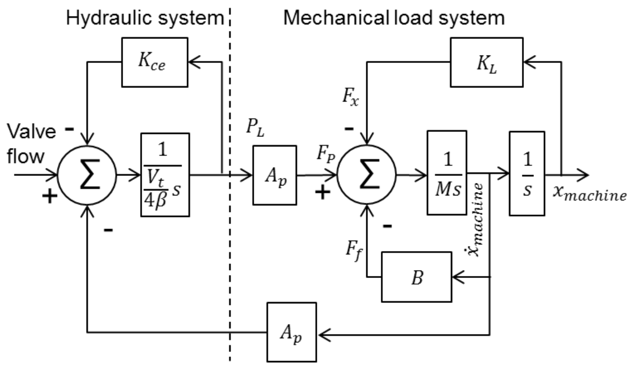

Fig. 3: Simplified block-diagram of the valve-controlled hydraulic actuator supporting a load.

B within Fig. 3 corresponds to the friction coefficient. For an assumed constant supply pressure and no load pressure, the valve flow coefficient K_Q represents the slope of the control flow and the input current curve at a specific operating region, and may be represented as:

$$K_Q = \left[\frac{\partial Q}{\partial i}\right]_{P_L = constant}. \tag{11}$$

This coefficient may be directly read off the charts provided by the valve manufacturer [Parker 2011], and in the present case is calculated as for an operating range for the command signal between 40-100% of the maximum current output possible:

$$K_Q = \frac{(100-30)\ lpm}{\frac{(100-40)}{100}\times 20\ mA} = 5.833\left[\frac{lpm}{mA}\right] \\ = 9.74\times 10^{-5}[m^3/s/mA]. \tag{12}$$

The flow pressure coefficient K_{ce} may be similarly represented as:

$$K_{ce} = \left[\frac{\partial Q}{\partial P_L}\right]_{i = constant}. \tag{13}$$

This coefficient can also be directly read off the charts provided by the valve manufacturer [Parker 2011], and in the present case is calculated for a 210 bar supply pressure and a pressure drop of 70 bar as:

$$K_{ce} == \frac{(160-100)\ lpm}{(210-70)\ bar} = 0.428\ \left[\frac{lpm}{bar}\right] = 7.15 \times 10^{-11} [m^3/s/Pa]. \quad (14)$$

Having modelled the electro-hydraulic dynamics, ground motion is now modelled as discussed in Section 3.5.

3.5 Modelling ground motion

Ground motion will result in machine motion, whether or not the actuator counters the ground motion. Hence, neglecting the electro-hydraulic dynamics, the equation of motion for the machine-actuator combination is:

$$M\ddot{x}_{machine} + C(\dot{x}_{machine} - \dot{x}_{ground}) + K_L(x_{machine} - x_{ground}) = 0 \quad (15)$$

wherein the damping coefficient C is a function of the damping ratio ζ_m of the mounting system, and can be represented as: $C = 2\zeta_m \omega_m M$; wherein $\omega_m = \sqrt{K_L/M}$ is the natural frequency of the machine-actuator system. ζ_m is assumed to be 0.02. Since medium sized machine tools typically have a mass of ~2500 kg and will be supported on three such actuators in the case of a three point support, each actuator will support ~800 kg of the machine and its own mass of 56 kg; hence M in Eq. (15) is 856 kg.

These ground vibrations against which the electro-hydraulic actuator will act are isolated by designing a controller such that it will help counter the disturbance by moving the machine in an attempt to zero the error signal.

3.6 Controller

The error signal between the commanded signal and the feedback from the transducer is manipulated by a controller that has the structure of a PID controller. The controller output u_v can be represented as:

$$u_v(t) = K_p u_e(t) + K_i \int u_e dt + K_d \frac{du_e}{dt} \quad (16)$$

wherein K_p, K_i and K_d are the PID constants and u_e is the error signal which is the difference of the commanded signal (u_c) and the feedback signal (u_f).

4 ACTUATOR DYNAMICS

Based on the above discussions and to ease analysis, a MATLAB Simulink model is formulated as shown in Fig. 4 to virtually test the closed-loop performance of the electro-hydraulic actuator. Each of the parameters pertinent for the model is listed in Tab. 1. At first, based on the CAD model of the developed actuator, the moving mass of the actuator, the volume of the chambers and the active area of the piston annulus are estimated, as listed in Table 1. The maximum stroke for the actuator is limited at ±100 µm, through introducing a saturation block in the Simulink model. Tab. 1 also lists other parameters pertinent to the proportional valve, the hydraulic supply and the motion sensors/transducers.

Tab. 1 Actuator, valve and hydraulic system data

	Description	Symbol	Value
Actuator data – estimated from CAD model [Bischoff 2009]	Moving mass	M	856 kg
	Max. stroke possible	$x_{p_{max}}$	±100 µm
	Active area of piston	A_p	3.18e-4 m^2
	Friction coeff. (assumed)	B	25000 N-s/m
	Total chamber volume	V_t	3.18e-6 m^3
	Actuator static stiffness	K_L	6.8e8 N/m
Prop. Valve D3FP Series [Parker 2011]	Flow gain coefficient	K_Q	9.74×10^{-5} $m^3/s/mA$
	Flow pressure coefficient	K_{ce}	7.15×10^{-11} $m^3/s/Pa$
	Max. input voltage	V_{max}	±10 V
	Natural freq.	ω_n	1256 rad/s
	Damping ratio (assumed)	ζ	0.48
Hydraulic prop.	Bulk modulus of fluid (assumed)	β	1.4×10^9 N/m^2
Transd.	Sensitivity	K_{tp}	8 mV/µm

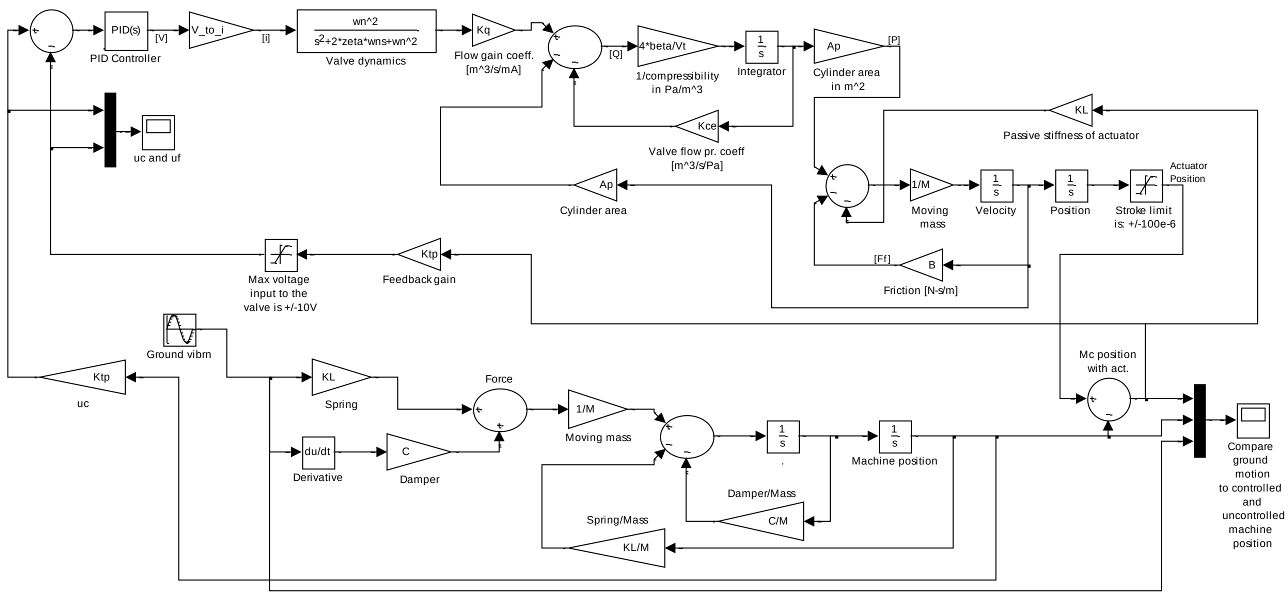

Fig. 4 Simulink model for the electro-hydraulic system subjected to ground motion as external disturbance.

Control parameters are tuned to obtain the tuned closed-loop system response between the disturbance input (ground motion) and controlled (isolated) machine response. This response, otherwise also referred to as the transmissibility is shown in Fig. 5. Response comparisons are limited to 200 Hz which is thought to be far greater than typically observed ground excitation frequencies in machine tool applications. The corresponding tuned control performance parameters are listed in Tab. 2.

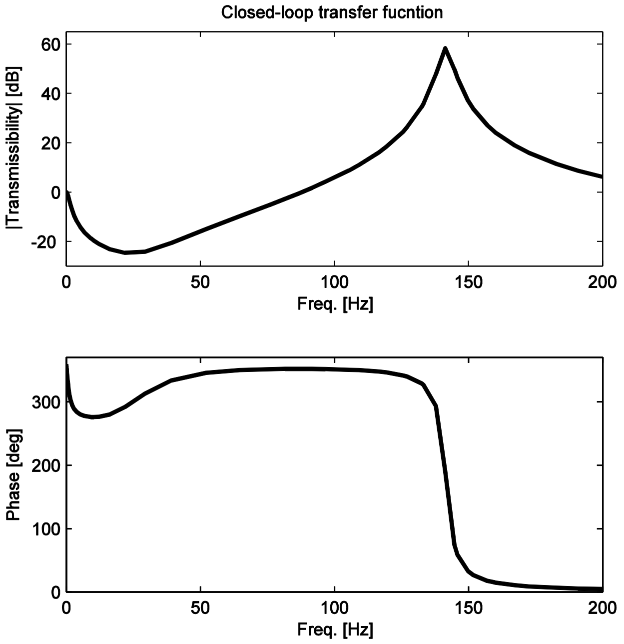

Fig. 5 Closed-loop transfer function between ground motion and isolated machine response.

Tab. 2 Tuned closed-loop performance parameters

Performance parameter	Tuned value
Rise time [s]	0.337
Settling time [s]	0.602
Overshoot [%]	~0
Peak	0.998
Proportional gain (K_p)	0.981
Integral gain (K_i)	64776
Derivative gain (K_d)	0
Closed-loop stability	Stable

As evident from Fig. 5, the negatively sloped relationship between the ground motion and the controlled machine position suggests that the electro-hydraulic actuator may indeed isolate ground vibrations up to ~30 Hz of ground excitation frequencies. The dominant mode at ~142 Hz corresponds to the machine tool system. The phase curve suggests that though machine may be isolated from the ground motion, it will be out of phase. This however is not thought to be of great significance, since as long as the machine is decoupled from ground vibration, it matters little what the phase of vibrations is at the tool point.

Although the closed-loop system response is stable, from parameters listed in Tab. 2, it does appear that the system has slow transients. Though these transients may limit performance under real practical implementation scenarios, the control parameters and strategy could provide useful guidelines when testing the real system. Though this model needs to be validated on the experimental prototype being developed at the Fraunhofer IWU, which forms part of the planned future work, it is assumed for now that the models developed are representative of the physical system. Hence, having evaluated the model based closed-loop stability and performance of the electro-hydraulic system, the actuator's ability to isolate ground motion is investigated in the next Section using two numerical case studies.

5 ACTIVE ISOLATION OF GROUND VIBRATION

Typical floor vibrations levels range from 1-50 µm within frequency bands of 1-50 Hz [Rivin 1995]. Higher amplitudes of floor vibrations are typically observed at lower frequencies and lower amplitudes at higher frequencies. These are the conditions under which the system is tested in the simulation environment. Performance of the device is tested under the two different conditions of: (i) ground motion of 20 µm at 5 Hz and (ii) ground motion of 10 µm at 20 Hz overlaid with random vibrations that have maximum amplitude of 1 µm.

Commanded and feedback signals for each of the above cases are compared along with the corresponding actuator position. Additionally, for each case the input ground motion is also compared with uncontrolled and controlled (isolated) machine response.

5.1 Case 1: Ground motion of 20 µm at 5 Hz

As evident in Fig. 6, despite the poor transient response of the actuator, the actuator is able to significantly isolate ground motion in the steady-state; isolating up to ~80% of the motion experienced by the machine. The phase difference between the commanded signal and the actuator feedback signal results in the isolated response not being in phase with the ground motion which, though important, is of little significance due to reasons outlined earlier. Though the machine mode at ~142 Hz gets excited during the transient state, steady-state spectrum of the controlled machine response contains only the excitation/response frequency of 5 Hz. The actuator operates well within its stroke limit of ±100 µm.

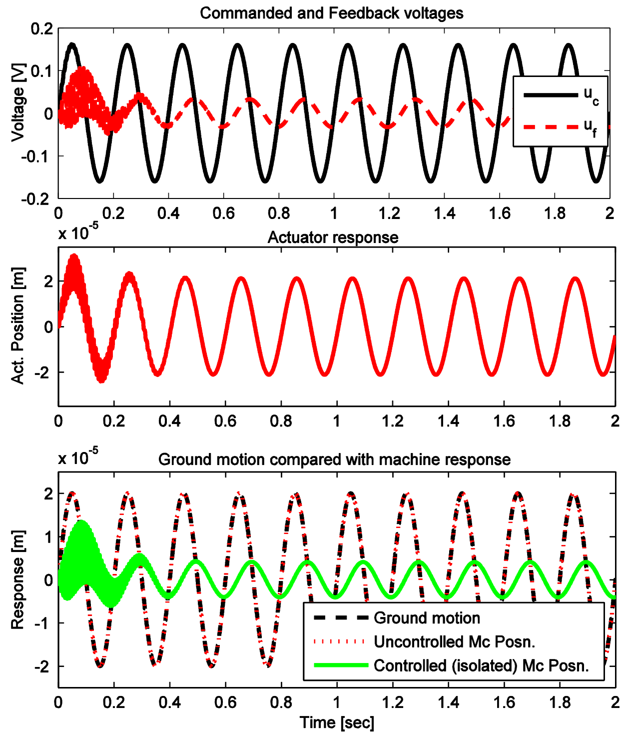

Fig. 6 Response for Case 1: Comparison of command and feedback signals (top); actuator position (middle); and comparison of ground motion to uncontrolled and controlled/isolated machine position (bottom).

5.2 Case 2: Ground motion of 10 µm at 20 Hz overlaid with random vibrations that have maximum amplitude of 1 µm

Even for the situation when ground motion of 10 µm at 20 Hz is overlaid with random vibrations that have maximum amplitude of 1 µm, the actuator is able to sufficiently isolate ground motion, as evident from comparisons in Fig. 7. The inset in Fig. 7 clearly shows the controlled machine response to be significantly less than the level of uncontrolled machine response. The mean level of isolation in this case is ~65%. As before, the transients excite the machine mode at ~142 Hz, and, though the steady-state response spectrum is dominated by the excitation frequency of 20 Hz, it also contains frequency content that corresponds to the random excitation.

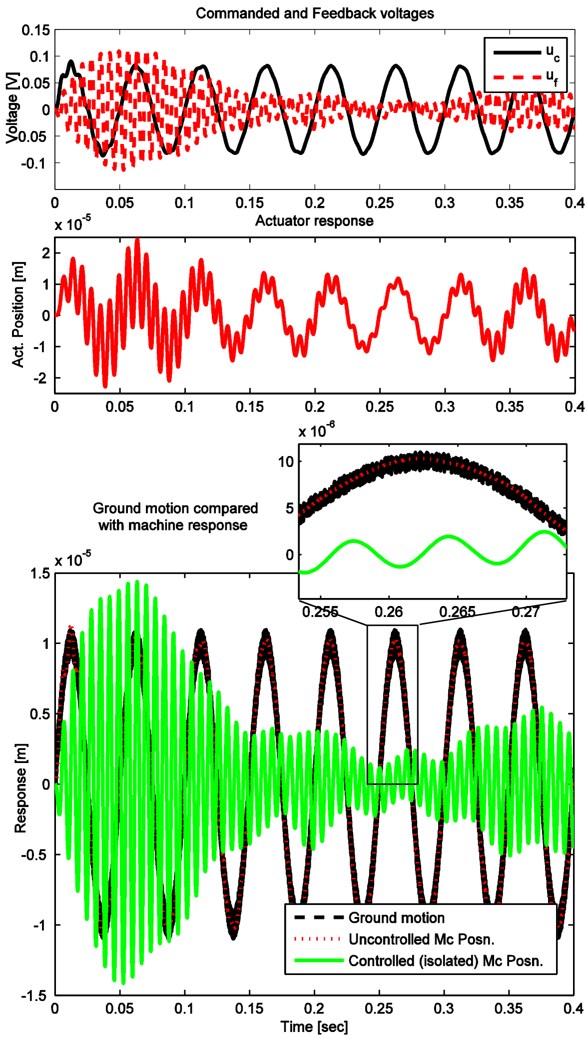

Fig. 7 Response for Case 2: Comparison of command and feedback signals (top); actuator position (middle); and comparison of ground motion to uncontrolled and controlled/isolated machine position (bottom).

6 CONCLUSIONS AND OUTLOOK

A comprehensive virtual model is formulated to investigate the closed-loop dynamics of the electro-hydraulic actuator to be used for effective isolation of ground vibrations in machine tools applications. Closed-loop transfer function between ground motion and isolated machine response suggests that the actuator may effectively isolate ground vibrations for excitation frequencies of up to 30 Hz. Numerical investigations representative of typical floor vibration levels experienced by machine tools demonstrate that ground motion may be isolated by up to 80%.

Preliminary investigations are promising and provide useful guidelines for testing the real system that is presently being prototyped at the Fraunhofer IWU. Poor transient response of the electro-hydraulic system may be overcome by implementing other advanced control strategies, which forms part of the planned future work.

The machine was presently modelled as rigid; however it is necessary to test the system (virtually) with a flexible description of the machine. Since in the case of a flexible description of the machine, the desired point of interest where the vibrations are to be isolated is the tool point, which is different than the machine mounting location, the model may need adjustments to achieve effective isolation. Furthermore, the possibility of spatial distribution of ground vibrations that may potentially result in different levels of excitations at different mounting locations, each with a different phase, pose interesting and challenging questions for effective isolation strategies. These investigations are also planned as part of the future work.

Because of their high passive stiffness, in the absence of external disturbances, the active machine mounts can also be used in the passive condition. Since the device can be used on demand, i.e. only when high disturbances are observed or when high precision is required, it offers an energy efficient and economical solution.

7 ACKNOWLEDGMENTS

This research was supported by the Fraunhofer Gesellschaft's ICON Project for Strategic Research Co-Operation on Sustainable Energy Technologies.

8 REFERENCES

[Abicht 2002] Abicht, C., Ulbrich, H. and Riebe, S., Active vibration isolation of a stewart-platform using high response hydraulic actuators, Proceedings of the 6th Int. Conference on Motion and Vibration Control, 2002.

[Bischoff 2009] Bischoff, Martin, Entwicklung eines aktiven Aufstellelementes zur Schwingungsisolierung von Werkzeugmaschinen, Masterarbeit, Hochschule für Technik, Wirtschaft und Kultur Leipzig, 2009 (in German).

[Bishop 2002] The Mechatronics Handbook (Editor-in-Chief: Robert Bishop), CRC Press, 2002, 602-607.

[Choi 2005] Choi, Y.T., Wereley, N.M. and Jeon, Y.S. Semi-Active Vibration Isolation Using Magnetorheological Isolators, Journal of Aircraft, 2005, 42, 1244-1251.

[DeBra 1992] D. B. DeBra, Vibration Isolation of Precision Machine Tools and Instruments, Annals of the CIRP, 1992, 41,2, 711-718.

[Huang 2003] X. Huang, S.J. Elliott, M.J. Brennan, Active isolation of a flexible structure from base vibration, Journal of Sound and Vibration, 2003, 263, 357–376.

[Meritt 1967] H. E. Merritt, Hydraulic Control Systems, Wiley, 1967.

[Moog 1965] Transfer Functions for Moog Servovalves, Technical Bulliten, Moog Controls Inc., 1965.

[Parker 2011] Direct Operated Proportional DC Valve, Series D3FP, Technical Catalogue, Parker, 2011.

[Rivin 1995] Rivin, Eugene, Vibration isolation of precision equipment, Precision Engineering, 1995, 17, 41-56.

[Rydberg 2008] Rydberg, Karl-Erik, Hydraulic servo systems – Lecture Notes, IEI/ Fluid and Mechanical Engineering Systems, Linköpings Universitet, 2008.

[Wabner 2012] M. Wabner, S. Ihlenfeldt, Aufstellelement, Patent No. DE102012010196 (A1), 2012 (in German).