6th International & 27th All India Manufacturing Technology, Design and Research Conference (AIMTDR-2016), December 16-18, 2016, at College of Engineering., Pune, Maharashtra, INDIA.

Design Guidelines for an Electro-Hydraulic Actuator to Isolate Machines from Vibrations

Mohit Law

Department of Mechanical Engineering

Indian Institute of Technology, Kanpur, India

mlaw@iitk.ac.in

Abstract

This paper provides instructive guidelines to design electro-hydraulic isolators that maintain dynamic excitations transmitted from the ground to the machine tool to below permissible limits. An electro-hydraulic isolator, akin to a double-acting hydraulic cylinder, is a classic example of a valve-controlled piston. Flow to the actuator is regulated to move the actuator and the machine that the actuator supports to compensate for ground motion experienced by the machine. The dynamic performance of these electro-hydraulic actuators is governed by a combination of its mechanical, hydraulic, servo, and control elements. Actuator performance is also a function of the moving mass of the actuator, total volume of fluid in the cylinder chambers, active area of the piston annulus, the designed static stiffness, a hydraulic stiffness, hydraulic fluid properties, servo valve parameters and control architecture. Each of these parameters influences the dynamic response of the actuator differently.

This paper presents a systematic methodology based on a complete electro-hydro-mechanical dynamical model of the actuator to evaluate sensitivity of actuator's response to changes in its components/elements and their operating parameters. Models that were experimentally validated on a first generation prototype of the actuator are used to establish instructive guidelines for the construction of a second generation electro-hydraulic actuator that will guarantee its fast response across high bandwidths without compromising its ability to support large machine tool inertial loads.

Keywords: Machine tools, Active vibration isolation, Electro-hydraulic actuator, Design guidelines, Sensitivity analysis

1. INTRODUCTION

Vibration isolators maintain dynamic excitations transmitted to the machine from ground motion to levels below the allowable machining and measurement deviations. Isolation is paramount for high-precision and high-performance machine tool applications, and much research and development efforts have resulted in effective isolation solutions through passive and/or active means [1-3].

Passive isolators though effective, often result in rocking and other related instabilities [2-4]. To overcome these issues, active isolators that include a sensor, a controller and an actuator are used. Amongst the family of active isolators, electro-hydraulic actuators, if/when designed right, are preferred in machine tools over other pneumatic, electromagnetic and piezoelectric actuators because of their superior dynamic characteristics and their ability to respond to spectral and spatial characteristics of the floor vibrations [5-6].

A schematic representation of an electro-hydraulic isolator is shown in Fig. 1(a). The machine is mounted on four such actuators. Basic elements of the electro-hydraulic system are shown schematically in Fig. 1(b) and constructional details of the actuator are shown in Fig. 1(c). The electro-hydraulic actuator is akin to a double-acting hydraulic cylinder. Flow to the actuator is regulated through a proportional valve to move the actuator and the machine that the actuator supports to compensate for ground motion experienced by the machine.

The dynamic performance of electro-hydraulic actuators is governed by a combination of its mechanical, hydraulic, servo, and control elements. Actuator performance is also a function of the moving mass of the actuator, total volume of fluid in the cylinder chambers, active area of the piston annulus, the designed static stiffness, a hydraulic stiffness, hydraulic fluid properties, servo valve parameters and control architecture. Each of these parameters influences the dynamic response of the actuator differently. Even though the performance of a first generation prototype of such an electro-hydraulic actuator was reported as satisfactory in [5-6], there is still a need to establish guidelines for optimal actuator design to guarantee fast response across high bandwidths without compromising its ability to support large machine tool inertial loads.

This paper presents a methodology to establish instructive guidelines for the construction of a second generation electro-hydraulic actuator. An overview of the dynamical model is first presented in Section 2. Section 3 discusses performance of the first generation prototype. Sensitivity analysis of actuator performance to variations in design parameters is carried out in Section 4. This is followed by design recommendations for an optimal actuator in Section 5, followed by the main conclusions in Section 6.

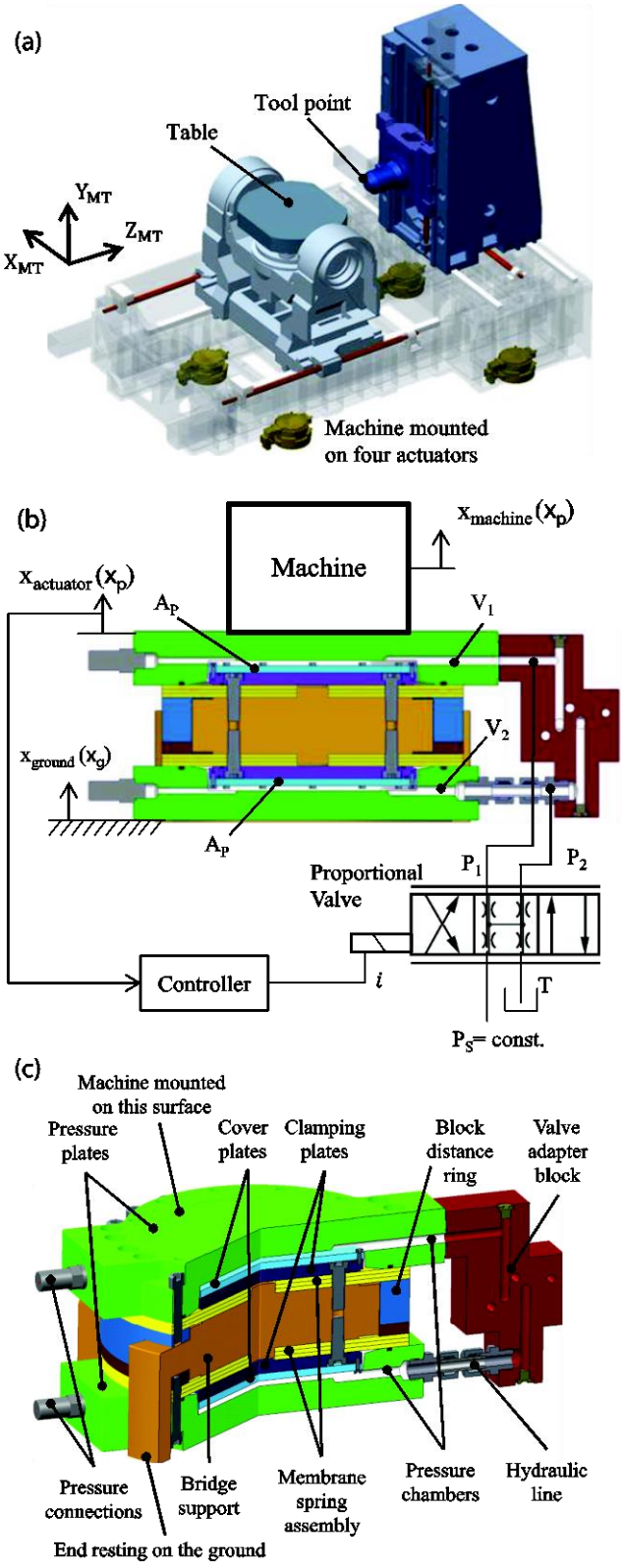

Fig. 1. (a) Schematic representation of a machine mounted on four actuators; (b) Schematic of valve controlled electro-hydraulic system; (c) Sectional view of the construction of the actuator. Figure adapted and modified from [5] Nomenclature in Fig. 1(b) is described in Section 2.

2. MODEL OF THE ELECTRO-HYDRAULIC ACTUATOR

The electro-hydraulic system shown in Fig. 1(b) is a classic example of a valve-controlled piston [6-8]. The system response is governed by the valve and its dynamics, the mechanical system and its dynamics, the hydraulics and the control system.

Ground motion detected by a sensor at the machine mounting location(s) acts as an input to the controller that drives a proportional valve that controls the fluid flow to the actuator. Flow to the actuator results in movement of the pressure plates and the machine to counter the motion caused by the ground vibrations. In the absence of any external disturbance, the actuator behaves like a passive system, with the membrane spring assembly (see Fig. 1(c)) and the hydraulics providing a very high passive stiffness. First generation prototype was designed to support a static load of up to 4 tonnes [5].

A detailed dynamic model of the electro-hydraulic actuator that has been experimentally validated is presented elsewhere in earlier work [5]. The reader is directed there for more details. Discussions here are limited to the dynamics of the main elements to obtain the transfer function between input to the valve and actuator position output. Machine is assumed to be rigid.

2.1 Valve dynamics

The valve's electronic driver converts the commanded voltage to current. The valve's solenoid converts this current into mechanical force acting on the spool against a return spring, which in turn results in spool movement and regulation of valve flow. Although valve dynamics are nonlinear [7], valves are often linearized in the Laplace domain as:

$$G_{valve} = \frac{x_v}{v}(s) = K_v \left[\frac{1}{1 + \left(\frac{2\zeta}{\omega_{n_v}}\right) s + \left(\frac{s}{\omega_{n_v}}\right)^2} \right] \qquad (1)$$

wherein x_v is the valve spool displacement, v is the supplied voltage, K_v is a gain, ω_{n_v} is the apparent natural frequency of the valve and ζ is the apparent damping ratio of the valve.

2.2 Dynamics of the hydro-mechanical system

Valve flow supplied to the pressure chambers of the actuator results in movement of the pressure plate of the actuators and the machine mounted on the actuator. Assuming valve orifices to be matched and symmetrical under constant supply pressure (P_S), the linearized flow equations are:

$$Q_1 = K_Q x_v - 2K_C P_1 \qquad (2)$$
$$Q_2 = K_Q x_v + 2K_C P_2$$

wherein Q_1 and Q_2 are the forward and return flows in m^3/s, and P_1 and P_2 are the forward and return

pressures. K_Q corresponds to a valve flow coefficient represented in $m^3/s/mA$, and K_C is the valve flow-pressure coefficient in $m^3/s/Pa$. Adding the valve flow equations gives:

$$Q_L = K_Q x_v - K_C P_L \tag{3}$$

wherein $Q_L = (Q_1 + Q_2)/2$ is the load flow, and $P_L = P_1 - P_2$ is the load pressure difference. Assuming that the pressure in each chamber is uniformly distributed, and that fluid leakage is negligible, application of the continuity equation to each piston chamber yields:

$$\begin{aligned} Q_1 &= \frac{dV_1}{dt} + \frac{V_1}{\beta}\frac{dP_1}{dt} \\ -Q_2 &= \frac{dV_2}{dt} + \frac{V_2}{\beta}\frac{dP_2}{dt} \end{aligned} \tag{4}$$

wherein β is the effective bulk modulus of the hydraulic system. V_1 and V_2 are the volumes of the forward and return chambers, which may be expressed as:

$$\begin{aligned} V_1 &= V_{01} + A_1 x_p \\ V_2 &= V_{02} - A_2 x_p \end{aligned} \tag{5}$$

wherein V_{01} and V_{02} are the initial volumes in each of the chambers and x_p is the displacement of the piston/pressure plates. A_1 and A_2 are the areas of piston, which are identical in the present case, i.e. $A_1 = A_2 = A_P$. Assuming the piston is centred such that: $V_{01} = V_{02} = V_0$, the total volume of fluid under compression becomes: $V_t = V_1 + V_2 = 2V_0$.

The volume and continuity equations can be combined, linearized and Laplace transformed by substituting Eq. (5) into Eq. (4) to yield:

$$Q_L = A_p s x_p + \frac{V_t}{4\beta} s P_L. \tag{6}$$

The dynamic equilibrium equation of the actuator piston under an external load can be expressed as:

$$F_p = P_L A_p = M s^2 x_p + C s x_p + K_L x_p + F_L \tag{7}$$

wherein F_p is the force generated by the actuator and F_L is the external load on the actuator. M is the translating mass (actuator mass plus machine mass), C is the viscous damping coefficient and K_L corresponds to the static stiffness of the actuator. Solving Eq. (3), (6), and (7) simultaneously under the assumptions of negligible membrane damping and no external load, the transfer function, G_{act} between valve and actuator position can be shown to be [5]:

$$G_{act} = \frac{x_p}{x_v} = \frac{K_Q/A_p}{\left(s + \frac{K_C K_L}{{A_p}^2}\right)\left(\frac{s^2}{{\omega_h}^2} + \frac{2\delta_h}{\omega_h} + 1\right)} \tag{8}$$

wherein $\omega_h = \sqrt{\frac{4\beta {A_p}^2}{V_t M}}$ is the hydro-mechanical natural frequency and $\delta_h = \frac{K_C}{A_p}\sqrt{\frac{\beta M}{V_t}}$ is the dimensionless hydraulic damping ratio, both of which play a deciding role in the behaviour of the electro-hydraulic actuator. Eq. (1) and Eq. (9) are combined to obtain the complete open-loop electro-hydraulic transfer function from voltage input to the valve to actuator/machine position output, G_{EHA}:

$$G_{EHA} = \frac{x_p}{v} = G_{valve} G_{act}. \tag{9}$$

This electro-hydro-mechanical dynamical model was validated with experiments conducted on the first-generation prototype of the actuator [5], results of which are discussed in Section 3.

3. Performance of First-Generation Prototype

The first-generation prototype is shown in Fig. 2. Actuator design parameters such as its moving mass (M), total volume of fluid in chambers (V_t), active area of the piston annulus (A_p), and its static stiffness (K_L) were obtained from the actuator's CAD model. Operational parameters such as the flow gain (K_Q) and flow-pressure (K_C) coefficients were identified from experiments [5]. Apparent valve natural frequency (ω_{n_v}) and damping ratio (ζ) were obtained from valve manufacturers catalogue [9]. These parameters are listed in Table 1.

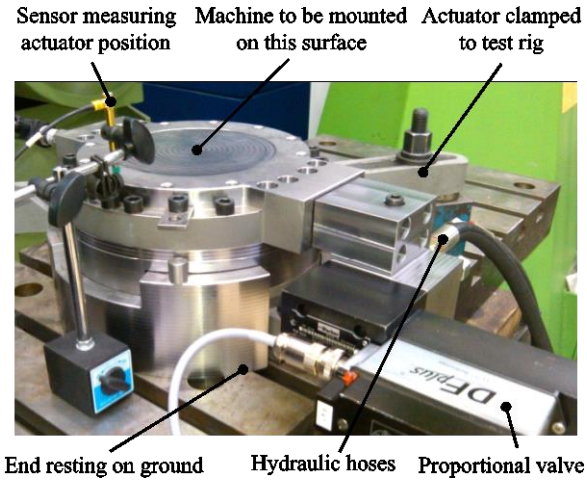

Fig. 2 First-generation prototype of the isolator [5]

The open-loop transfer function from voltage input to the valve to actuator position output using the initial design parameters and experimentally identified operational parameters as listed in Table 1 is shown in Fig. 3 for the actuator supporting a machine load of 4 tonnes. As evident from Fig. 3, the hydro-mechanical natural frequency is estimated to be 257 Hz. This is very close to the valve natural frequency of 250 Hz when the valve operates at around ±5% of its commanded signal [10].

TABLE I

Parameter	Symbol	Initial value	Parameter variation
Moving mass	M [kg]	4056	-
Active area of piston	A_p $[m^2]$	3.18×10^{-2}	±100%
Viscous coefficient	C [Ns/m]	3900	-
Total chamber volume	V_t $[m^3]$	5.39×10^{-4}	±100%
Actuator static stiffness	K_L [N/m]	6.8×10^8	-
Flow gain coefficient	K_Q $[m^3/s/mA]$	1.94×10^{-6}	±100%
Flow pressure coefficient	K_C $[m^3/s/Pa]$	6.6×10^{-11}	±100%
Natural frequency at ±5% of signal	ω_{n_v} [rad/s]	1571	-
Damping ratio	ζ	0.65	-
Gain	K_v	2	-
Bulk modulus of fluid	β [N/m^2]	1.4×10^9	-

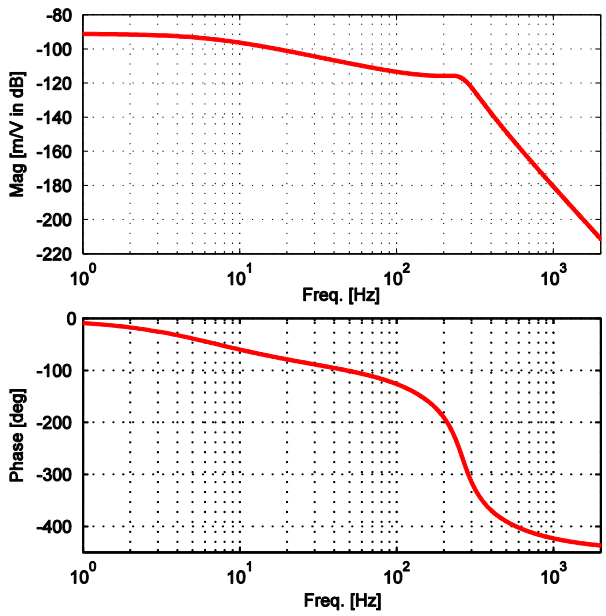

Fig. 3 Open-loop transfer function of the electro-hydraulic actuator with its initial design parameters.

The closed-loop bandwidth of the actuator is limited by the lesser of the two natural frequencies of the valve and the hydro-mechanical frequency. However, since there is a 90˚ phase lag around the valve natural frequency, and because the hydro-mechanical natural frequency is close to it, controlling the electro-hydraulic actuator will be difficult up to 250 Hz. For better control performance, these frequencies need to be well separated, as they were in the case of the first-generation prototype [5]. However, in that case the valve was assumed to be to be operated at ±90% of its commanded signal, and its natural frequency was found to be 100 Hz [10]. Though well separated, the valve frequency limited the closed-loop controllability of the actuator to be below 100 Hz. Since this is thought to be too low for a high-performance machine tool isolator that is to respond to broadband spectral and spatial floor vibrations, a next-generation actuator is sought with significantly improved dynamics.

Since dynamics of the electro-hydraulic actuator are limited by both the valve frequency and the hydro-mechanical frequency, and their being very well separated, systematic performance investigations are carried out in Section 4 under design and operational parameter variations to characterize sensitivity of the actuator to parameter changes. A high-bandwidth performance is sought with the valve and hydro-mechanical frequencies being very well separated.

4. Sensitivity analysis of the isolator to design and operational parameter changes

Since it is very difficult to achieve snappy, responsive control of the electro-hydraulic system with a low hydro-mechanical frequency, designs which push this frequency higher are targeted. The hydro-mechanical resonant frequency is related to the volume of fluid as well as to a change in the piston area; see Eq. (8) for these relationships. These parameters along with valve operational parameters are varied to check for their influence on system dynamics.

4.1 Sensitivity to changes in chamber volume

The open-loop transfer function of the isolator was evaluated by keeping all parameters the same as in Table 1 and varying the chamber volume to within ±100% of its initially designed value. These results are shown in Fig. 4. As evident, an increase in chamber volume, i.e. an increase in the volume of fluid under compression lowers the hydro-mechanical frequency to 182 Hz, whereas a decrease in the chamber volume increases in hydro-mechanical frequency to 363 Hz. In both cases, the hydro-mechanical frequency is very well separated from the valve frequency, which is more favourable than the initial design. Since the hydro-mechanical frequency becomes the system bottleneck when it is less than the valve's frequency [11], a design with a larger chamber volume should be avoided. Moreover, the design with a larger chamber will also lower the closed-loop bandwidth of the system as compared to the initial design. A smaller chamber also results in a higher hydraulic damping (0.3) as compared to a

design with a large chamber which has a hydraulic damping ratio of 0.15. A design with a smaller chamber is hence the preferred option.

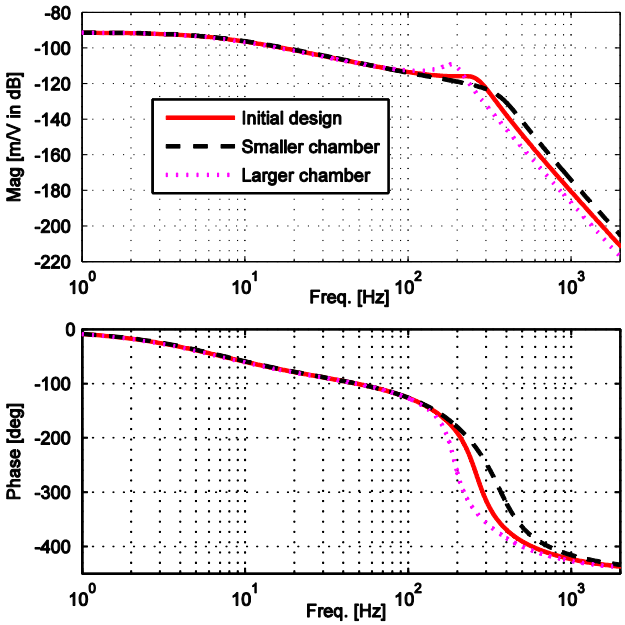

Fig. 4 Sensitivity of open-loop transfer function of the electro-hydraulic actuator to changes in chamber volume.

4.2 Sensitivity to changes in area of piston

Keeping all parameters the same as in Table 1 and varying only the area of the piston by ±100% of its initially designed value, we get the response of the system as shown in Fig. 5.

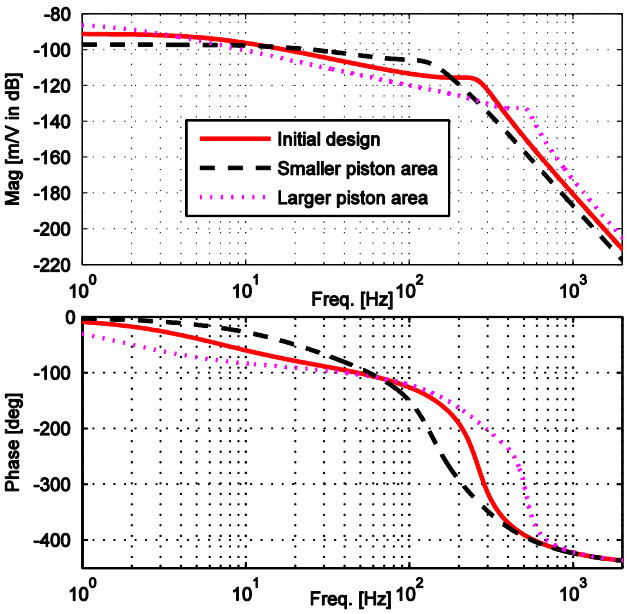

Fig. 5 Sensitivity of open-loop transfer function of the electro-hydraulic actuator to changes in piston area.

The hydro-mechanical frequency is directly proportional to a change in the area of the piston, with a large piston area resulting in a hydro-mechanical frequency of 513 Hz and a design with a small piston area resulting in a hydro-mechanical frequency of 128 Hz. Even though an increase in area of the piston reduces the hydraulic damping, which is not ideal, a larger piston area is still preferable since it results in well separated valve and hydro-mechanical frequencies as well as the bandwidth of the closed-loop system being limited by the valve and not the hydro-mechanical system.

4.3 Sensitivity to changes in flow coefficients

Keeping all other parameters the same as in Table 1 and varying only the flow gain and flow pressure coefficients in turn allows investigations of sensitivity of the isolator's performance to these coefficients. These results are shown in Fig. 6. The flow gain coefficient simply acts as a gain, and hence has no major influence on the dynamics of the isolator – as is evident in Fig. 6(a). The flow pressure coefficient on the other hand influences damping. The relationship between the flow pressure coefficient and hydraulic damping being linear (see Eq. (8)), a higher coefficient naturally increases the hydraulic damping, and hence when/if possible, the valve should be operated with a higher flow pressure coefficient.

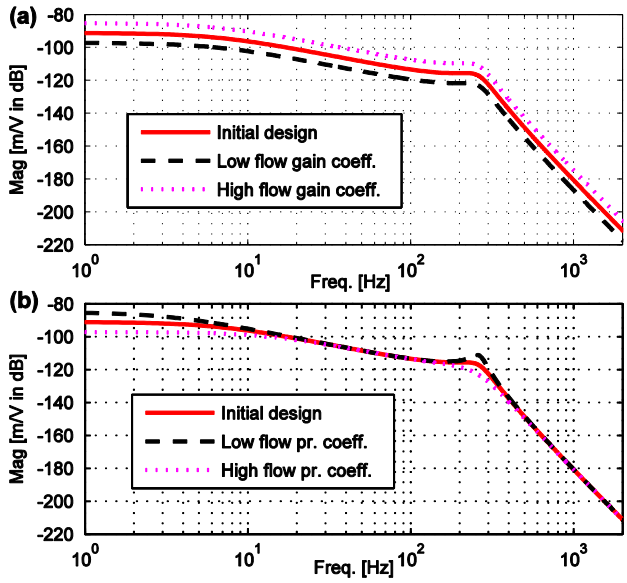

Fig. 6 Sensitivity of open-loop transfer function of the electro-hydraulic actuator to changes in flow gain coefficient (a), and to changes in flow pressure coefficient (b).

5. Design Recommendations for Next-Generation Actuator

Based on the sensitivity analysis in Section 4, it is amply clear that an actuator that has a small chamber, high piston area and one that operates a valve with a high flow pressure coefficient will result in a sufficiently high hydro-mechanical frequency with superior hydraulic damping. Hence it is

recommended that the next-generation isolator be designed with these properties. Isolator performance with recommended parameters is compared with the initial design in Fig. 7 to demonstrate potentially improved capabilities with a new design.

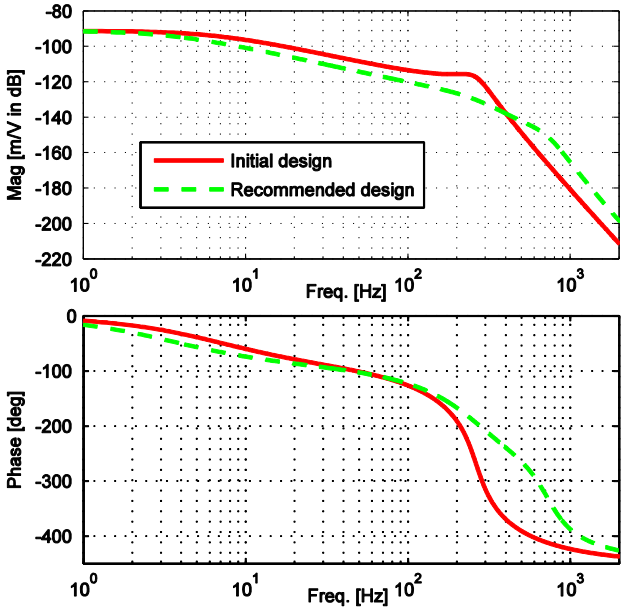

Fig. 5 Comparison of the open-loop transfer functions of the actuator with initial and recommended design parameters.

The recommended design would result in a system with the hydro-mechanical frequency being 726 Hz and a hydraulic damping of 0.3. The hydro-mechanical frequency would be very well separated from the valve frequency of 250 Hz. This is far better than the initial design which had issues of the two frequencies interacting with each other to make for difficult controller design.

Moreover, simulation based investigations into the gain margin for closed-loop controller stability using the open-loop transfer function shows that the recommended design would result in a gain margin of 128 dB at 230 Hz, which is far superior to the gain margin of 116 dB at 187 Hz for the initial design. In the case of the initial design, because the hydro-mechanical frequency and the valve frequency were so closely spaced, it is difficult to identify which of the two limit the closed-loop stability. However, in the case of the new design, it is very evident that the valve's dynamics limit the closed-loop stability.

Furthermore, since the hydro-mechanical frequency in the new recommended design is so well separated from the valve's frequency, there is room yet for using a valve with yet higher performance, for example using a valve with a frequency of 400 Hz or more. This would result in a very high-performance isolator that can respond quickly and robustly to any kind of broadband spectral and spatial floor vibrations, thus further increasing the utility of such electro-hydraulic actuators. Physically realizing a design that has a small chamber and a high piston area may pose challenges, but that is a different problem, and is outside the scope of this paper.

6. Conclusions

Electro-hydraulic actuators if/when designed right have the potential to isolate machine tools from broadband spectral and spatial floor excitations. Based on a complete electro-hydro-mechanical dynamical model of the actuator, sensitivity based investigations were carried out to evaluate sensitivity of actuator's response to changes in its design and operating parameters. It was found that an isolator with a small chamber, large piston area, and one operating a valve with a high flow pressure coefficient has significantly improved dynamic response as compared to the first-generation prototype. These results are instructive to guide the construction of a second generation electro-hydraulic actuator that guarantees its fast response across high bandwidths without compromising its ability to support large machine tool inertial loads.

Acknowledgment

The Author acknowledges the Fraunhofer IWU in Germany, where the first-generation of this actuator was prototyped. Ideas for this paper resulted from the Author's previous work on that prototype.

References

[1] DeBra D. B., Vibration Isolation of Precision Machine Tools and Instruments, Annals of the CIRP, 41 (1992), 711-718.
[2] Rivin E., Vibration isolation of precision equipment, Precision Engineering, 17 (1995) 41-56.
[3] Rivin E., Vibration isolation of precision objects. Sound and Vibration, 40 (2006), 12–20.
[4] Okwudire C. and Lee J., Minimization of the residual vibrations of ultra-precision manufacturing machines via optimal placement of vibration isolators, Precision Engineering 37 (2013) 425–432.
[5] Law, M., Wabner, M., Colditz, A., Kolouch, M, Noack, S. and Ihlenfeldt, S., "Active Vibration Isolation of Machine Tools Using an Electro-Hydraulic Actuator", CIRP Journal of Manufacturing Science and Technology, Vol. 10, 2015
[6] Wabner M., Law M. and Ihlenfeldt S., Dynamic modelling of an electro-hydraulic actuator to isolate machine tools from ground vibrations, Proc. of the 11th Int. Conference on High Speed Machining, HSM 2014, Czech Republic, 2014
[7] Merritt H. E., Hydraulic Control Systems, Wiley, 1967.
[8] The Mechatronics Handbook, CRC Press, 2007.
[9] Transfer Functions for Moog Servovalves, Technical Bulletin, Moog Controls Inc., 1965.
[10] Direct Operated Proportional DC Valve, Series D3FP, Technical Catalogue, Parker, 2011.
[11] Jhonson, J, The frequency response function – Parts 1-3, Hydraulics and Pneumatics, 1996, accessed via: www.hydraulicspneumatics.com on 20th June 2016.