euspen's 20th International Conference & Exhibition, Geneva, CH, June 2020
www.euspen.eu

Active hydraulic actuators for vibration reduction in machine tool applications

Harkaitz Urreta[1], Lander Rodriguez[1]

[1]*IDEKO Technological Centre, Arriaga Industrialdea 2, Elgoibar (Basque Country) Spain*

hurreta@ideko.es

Abstract
The research work presented in this paper is focused on the development of a new hydraulic actuator for machine tool applications. Vibration isolation has been one of the main focuses of precision engineering in recent years. For this purpose, electromagnetic actuators have been widely used. However, in cases where high forces are required, the size, the heat on the actuator and the cooling systems could be limiting constraints. Therefore, it is considered that vibration reduction in heavy duty machine tools, with forces higher than 10 kN and frequencies close to 50 Hz, cannot be covered with electromagnetic actuators. To meet these requirements hydraulic actuators are analysed in this paper. Hydraulic solutions have been historically viewed as low dynamics or even quasi-static solutions for industry, where their biggest advantages are the large applying force and power density. The present study covers the mathematical modelling together with the experimental analysis of a preliminary hydraulic actuator design, emphasizing the facets which enhance its frequency response. Experimental and simulation results exhibit good agreement for the different tests performed and thus, the model has been validated for the tested range of operation. The mathematical model is later used for a parametrical analysis, whose results are the preliminary design guidelines for a new hydraulic actuator that will be defined in further steps of this research project.

Keywords: Hydraulic actuator, active damping, machine tool

1. Introduction

Isolation of precision machines against external disturbances is essential to achieve the required machining tolerances. Vibrations are considered to be one of the main disturbing agents as they can widely affect the machining accuracy [1]. To satisfy the increasing requiring demands, passive and active isolators have been developed [2].

Passive vibration isolation is usually accomplished by a high mass on a low stiffness spring [1]. However, theses systems are usually tuned to a fixed frequency, which is not sufficient for highly position-dependent machines [3]. Moreover, passive isolators usually introduce dynamic instabilities which lead to rocking of the machine [3],[4]. Active isolators are composed of a sensor, a controller and an actuator which can overcome the dynamic issues associated to passive isolators providing an improved frequency performance.

Among active isolators, inertial drives have been recommended for heavy duty machine applications [5]. Electromagnetic inertial drives are most often used in industry [6], whose diverse applications can be found in the literature [5],[6],[7]. Nevertheless, if forces of several kN were to be counteracted, electrogmanetic actuators would lead to space and heating issues.

Hydraulic actuators are considered to be an alternative for these applications due to their high power to weight ratio and stiffness characteristics. However, their frequency response is limited due to the inherent non-linearities and delays found in hydraulic systems [5],[8].

An electrohydraulic active damping system was developed with short pipes and close gaps between piston and cylinder to reduce pressures losses and friction respectively [9]. For the presented design, the depth of cut could almost be tripled.

An electrohydraulic actuator was designed to isolate the machine from ground vibrations and with a high passive stiffness to support inertial loads [10]. Simulation results suggested the actuator could work up to 100 Hz with displacements in the µm range. Moreover, in [11] design guidelines were proposed for the development of new electrohydraulic actuators.

In the present study, applications which require forces of several kN, displacements in the mm range and a frequency bandwidth of 50 Hz are considered. In the search of these specifications and in order to gain some insight in hydraulic systems, the main objectives of the present document are the mathematical modelling of a hydraulic actuator along with test bench validation. The gained knowledge in conjuction with the design variables analysis will be the initial step for the desired new electrohydraulic actuator design.

With that aim, this document has been structured in 3 different sections. In section 2, the adopted methodology for the experiments and the mathematical modelling is explained. In section 3, the experimental results and model validation are presented followed by section 4, in which the main clonclusions are drawn.

2. Methodology

In this section the experimental set-up together with the different experiments to be conducted will be firstly presented. Afterwards, the mathematical modelling of the main components followed by the optimisation procedure to validate the model will be explained. Finally, a design variable analysis will be presented for the new generation hydraulic actuator.

2.1. Experimental set-up

The test bench is schematically shown in Figure 1. The backbone of the actuator consists of a hydraulic pump that feeds the system with fluid from a tank; an accumulator to feed the system with additional power in case abrupt need of energy is required; a D1FP Parker proportional valve which directs the flow; and a GLUAL KI 40/18x250 hydraulic asymmetric cylinder to drive the load. The pump is included in the hydraulic unit of Ideko ultraprecision laboratory capable of delivering up to 100 bar pressure.

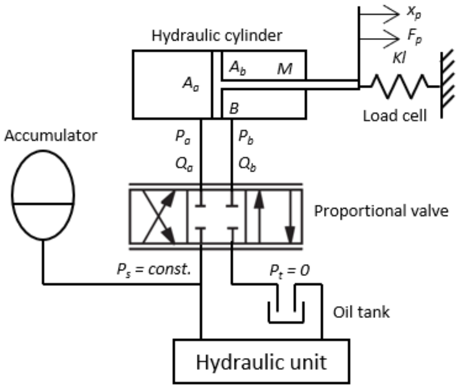

Figure 1. Actuator test bench scheme

The actuator was tested in open-loop configuration. Voltage to the valve was provided in the form of a sinestream signal with 3 repetitions of 120 frequencies ranging from 1 Hz to 70 Hz. In order to analyse the influence of both input signal and supply pressure, various experiments were proposed. In Table 1, the proposed test conditions are presented.

Table 1 Test conditions for the hydraulic actuator

Test	P_s [bar]	V_{in} [V]
1	30	2
2	30	2
3	45	4
4	45	4

An Interface SSM-AJ-20kN load cell was used to measure the actuator force and pressures were measured in both cylinder chambers and in the supply port of the proportional valve with HYDAC 3445-A-060-000 pressure sensors. In addition, valve spool position and input signal to the valve were measured. Signal control and acquisition was performed with Ingesys IC3 PAC controller.

2.2. Mathematical modelling

The actuator dynamics are mainly influenced by the proportional valve and the hydraulic cylinder dynamics. Despite the fact that the valve response is non-linear, it is often linearized and characterized as a second order transfer function as [12]

$$\frac{x_{sp}}{x_{in}} = \frac{K_v}{s^2 + 2\delta_v\omega_v + \omega_v{}^2} \qquad (1)$$

where x_{sp}, x_{in}, K_v, δ_v and ω_v are the spool position, input signal, valve proportional gain, valve apparent damping ratio and valve apparent natural frequency, respectively.

The supply pressure, P_s, is assumed to be constant at the input port of the proportional valve due to the accumulator influence. Besides, the tank pressure, P_t, is assumed to be null. Assuming the valve ports are matched and symmetrical, pressures will rise above and below $P_s/2$ by equal amounts. Thus, flow equations can be linearized as [13]

$$Q_a = K_q x_{sp} - 2K_c P_a \qquad (2)$$

$$Q_b = K_q x_{sp} + 2K_c P_b \qquad (3)$$

where Q_a, K_q, K_c, P_a, Q_b and P_b are line A flow, valve flow gain, valve flow-pressure coefficient, line A pressure, line B flow and line B pressure respectively. K_q and K_c are known as the valve coefficients and determine the frequency response and stability of the system. On the one hand, the open loop gain of the system is directly proportional to K_q and thus, it directly influences the system stability. On the other hand, K_c directly influences the damping ratio of the valve-piston system. Both of them are extremely dependent on the operating conditions [13].

Assuming constant temperature and density, no saturation nor cavitacion inside the piston chambers and no leakage between valve ports, the continuity equation in each chamber yields [13]

$$Q_a = \frac{dV_a}{dt} + \frac{V_a}{\beta_e}\frac{dP_a}{dt} \qquad (4)$$

$$-Q_b = \frac{dV_b}{dt} + \frac{V_b}{\beta_e}\frac{dP_b}{dt} \qquad (5)$$

where V_a, β_e and V_b are chamber A volume, effective Bulk modulus and chamber B volume respectively. Assuming identical initial volumes

$$V_a = V_0 + A_a x_p \qquad (6)$$

$$V_b = V_0 - A_b x_p \qquad (7)$$

where V_0, A_a, x_p and A_b are initial volume in each chamber, piston area in chamber A, piston position and piston area in chamber B respectively. The pressure load, P_l, is given by

$$P_l = P_a - P_b \qquad (8)$$

and due to the cylinder asymmetry the piston force, F_p, is given by

$$F_p = \begin{cases} P_l\, A_a, & \text{if } P_l > 0 \\ P_l\, A_b, & \text{if } P_l < 0 \end{cases} \qquad (9)$$

Assuming there are no external forces, the last equation is given by Newton's second law applied to the mass spring system of the piston and load cell. Applying the Laplace transformation yields [13]

$$F_p = Ms^2\, x_p + Bs\, x_p + K_l\, x_p \qquad (10)$$

where M, B and K_l are piston mass, viscous damping coefficient and load cell stiffness respectively.

Dynamics of the system are greatly influenced by the valve natural frequency and the hydrodynamic natural frequency, ω_h, which can be written for a symmetric cylinder as [13]:

$$\omega_h = \sqrt{\frac{4\beta_e A^2}{2V_0 M}} \qquad (11)$$

These two frequencies should be as separated as possible in order to enhance the actuator controllability [11]. These equations were implemented in Simulink, Matlab 2018. The parameters were determined according to the operating conditions expected in the experiments stated in Table 1. Finally, simulation results were compared with experimental ones for the optimisation that will be explained hereunder.

2.3. Optimisation

As previously stated, various parameters are highly dependent on the operating conditions. Therefore, similarly to [10], system identification was performed solving the following optimisation problem

$$\min f_{obj}(x_d) = \sum_{i=1}^{n_p} r_i{}^2 \ with\ l_b \le x_d \le u_b \qquad (12)$$

where f_{obj}, x_d, n_p, r_i, l_b and u_b are the objective function, set of model parameters, number of frequency points, the residual and lower and upper bounds of the set of model parameters respectively. The residual is calculated as:

$$r_i = abs(P_{a\,exp} - P_a) + abs(P_{b\,exp} - P_b) \qquad (13)$$

The pressures transfer functions have been considered for the residual calculation due to the load cell and piston relative displacement, which hindered clear force signal acquisition. Pressures provide an understanding of the system response as their difference is the image and likeness of the force the actuator can generate.

The set of parameters is composed of *ωv, δv, Kq, Kc, B and Kl.* The upper and lower bounds of the valve parameters are determined according to the charts provided by the manufacturer [12]. *B* and K_l are not a priori known. The model will be considered valid in case the experimental results match well with the simulated ones.

2.4. Parameter analysis

Once the model is validated, a parameter analysis will be performed in order to understand their influence in the frequency response of the actuator. The results from this analysis will be the first seed for the development of the new hydraulic actuator.

3. Results and discussion

In the present section the experimental results will be firstly analysed. Afterwards, the simulation results will be compared with the experimental ones followed by an analysis of the parameter influence simulations.

3.1. Experimental results

Chamber A pressure variation Bode diagrams are shown in Figure 2 for the different tests specified in Table 1. For the test bench arrangement shown in Figure 1, it is apparent that the pressure variations reach the maximum value possible, $P_s/2$, in all the studied cases for low frequency values. This suggests that the maximum force can be definitely reached in the low frequency range.

As it was expected, an increase in the supply pressure induces higher pressures in both chambers, increasing the actuator force capacity. However, the amplitude decay for equal input signal amplitudes occurs at similar frequency values for the different supply pressures. In opposition, an increase in the input signal amplitude involves the pressure to be kept at its maximum value for higher frequencies, and so will force.

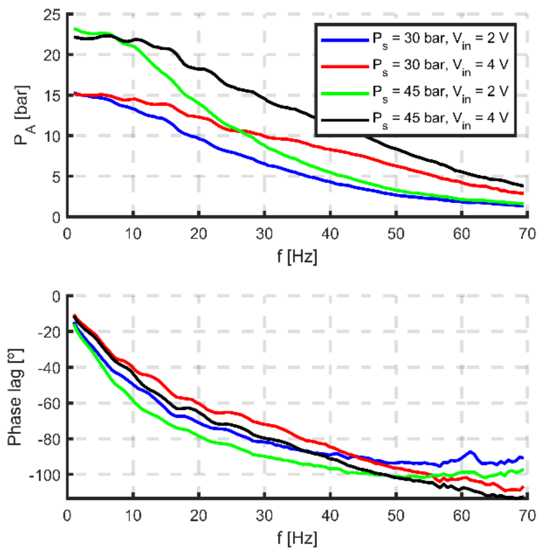

Figure 2. Chamber A pressure Bode diagram for the tests specified in Table 1

However, the phase lag response of the system is similar for all of the cases. There exists the common low frequency lag in pistons attached to spring loads [13] and similar phase lag trends were obtained somewhere in [10]. It is true that higher input signal amplitudes provide slightly lower time delays, which is in accordance with the amplitude response. The small differences could be attributed to the limited incrementss imposed in the supply pressures and input signal amplitudes studied.

3.2. Simulation and experimental results comparison

The optimisation problem specified in *Eq. (12)* was solved for the different test conditions. The initial valve parameter values were selected according to the expected operating conditions while the lower and upper bounds were equal for all of the cases. In order to gain a complete picture of the differences between model and reality, Figure 3 shows the Bode diagram of the pressure variation in A chamber for Test 3. Although there exist slight differences, the mathematical model reesembles reality to a great extent.

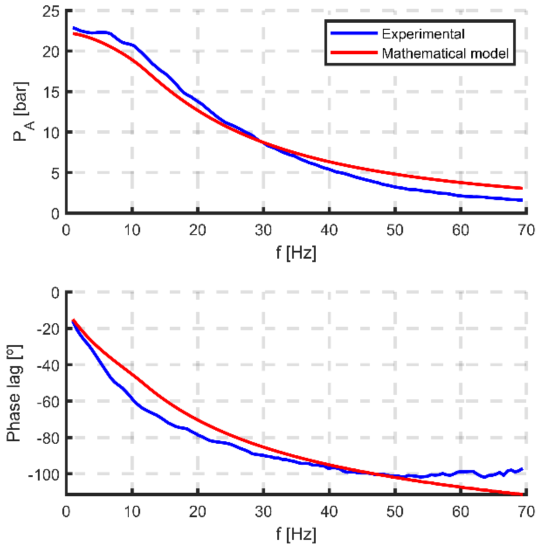

Figure 3. Experimental and simulation pressure response for P_s = 45 bar, V_{in} = 2 V

The differences can be attributed to the non-linearities that have not been modelled like the valve dead zone, pipes and orifice losses, flow forces, leakages… Moreover, valve coefficients have been considered constant in each specific case and are highly dependent on the operating conditions, hindering a perfect match. Specifically, K_c is remarkably susceptible to changes in the excitation frequency [10], accentuating these differences at higher frequencies. This effect can be observed at the highest frequencies of the pressure amplitude plot. Based on the good match between the simulation and the experiments, the model is considered valid.

3.3. Parameter analysis

From the experimental results, it can be concluded that the maximum force is actually achievable while considerable difficulties are encountered with maintaining high pressures with increasing frequency. Therefore, the rapid decrease in phase and amplitude should be enhanced. All things considered, the parameter analysis has been focused in the determination of their influence for enhanced frequency response.

In Table 2 the influence in the frequency response of the different parameters analysed is displayed together with the refitting easiness in future designs. It was concluded that valve coefficients and piston area are the most influential parameters. On the one hand, valve parameters are difficult to be modified in terms of design and vary with different operating conditions. On the other hand, lower areas provided better response in terms of pressure. This occurs because the low frequency lag introduced by pistons with spring loads at K_cK_l/A^2, *15 Hz*, influences the frequency response in the tested range of operation to a larger extent than the hydrodynamic, *350 Hz*, and the valve, *300 Hz*, natural frequencies. However, if the actuator is intended to be used as an active isolator, it will not act against a spring load and this effect will be removed. In this instance, greater areas will provide better piston force and displacement frequency response [11], as it can be concluded from *Eq. (11)*.

Table 2 Influence and refitting easiness of actuator parameters

Parameter	Influence	Refitting easiness
V	Moderate	High
A	High	High
K_q	High	Moderate
K_c	High	Moderate
β	Moderate	Low
M	Moderate	High

Among the other parameters, the volume and the mass are rather easily modifiable with the piston chamber length. On the contrary, the Bulk modulus should remain the same as oil is the common fluid in machine tools.

Based on this analysis, it was decided to make an analytical exercise of how the cylinder of the test bench could be improved for an enhanced frequency response. Its main constraits for the intended applications were the high piston chamber length, *250 mm*, as well as its asymmetry. Besides, the valve and the hydrodynamic natural frequencies are similar, hindering the actuator controllability

Therefore, a cylinder volume reduction was applied, decreasing its length to *50 mm*. In addition, symmetry was introduced in the piston for better controllability. The obtained chamber A pressure variation Bode diagram is shown in Figure 4.

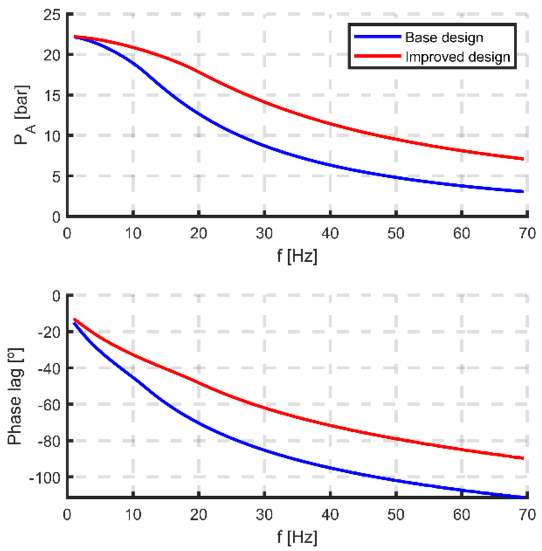

Figure 4. Improved actuator design compared to base design pressure response for P_s = 45 bar, V_{in} = 2 V

It is obvious that a reduction in volume improves the frequency response of the actuator. The phase lag is decreased in a good manner and the pressure is doubled between both designs for the highest frequencies, providing in consequence higher forces and displacements. Similar conclusions were obtained somewhere in [11]. Besides, the hydrodynamic frequency is increased from 350 Hz to 782 Hz. This widens the separation between the valve and the hydrodynamic natural frequency, enhancing the actuator controllability. The results suggest that focusing the attention on the cylinder design and applying enhanced operating conditions, a new actuator design that meets the application requirements can be obtained.

4. Conclusions and future work

The current research has presented the experiments and mathematical modelling of a hydraulic actuator. The mathematical model has been validated and the parametic results and the experience acquired will be applied in the development of the new generation hydraulic actuator. The following design guidelines will be considered in the new actuator:

1. Symmetry will be used when designing the hydraulic cylinder for better controllability.
2. The piston length will be kept as small as possible.
3. For active damping, greater areas will provide better piston force and displacement frequency response.
4. Pipes between valve and cylinder will be kept as small as possible to reduce pressure losses.
5. Higher supply pressures increase the force capacity of the actuator.
6. Higher input voltages enhance the frequency response of the actuator.

Finally, one of the main focuses of the future work should be the analysis of the influence of the viscous damping coefficient in the frequency response. The unawareness of the usual values of this parameter made it impossible to perform a proper analysis of its influence. This issue will be addressed with CFD simulations of a new cylinder design based on the design guidelines stated above.

References

[1] Schellekens P, Rosielle N, Vermeulen H, Vermeulen M, Wetzels S and Pril W S and Morkoc H 1998 Design for Precision: Current Status and Trends *Annals of the CIRP* **47** 557-578

[2] Rivin E I 1995 *Precision Engineering Journal* **17** 41-56

[3] Neugebauer R, Denkena B and Wegener K 2007 Mechatronic Systems for Machine Tools *Annals of the CIRP* **56** 657-686

[4] Wabner M, Law M and Ihlenfeldt S 2014 Dynamic Modelling of an Electro-Hydraulic Actuator to Isolate Machine Tools from Ground Vibrations *11th International Conderence on High Speed Machining* 1-14

[5] Ehmann C and Nordmann 2002 Low Cost Actuator for Active Damping of Large Machines *IFAC Conference on Mechatronic Systems* 179-184

[6] Mancisidor I, De La Fuente J, Barrios A, Barcena R and Munoa J 2013 Diseño de un amortiguador activo para la eliminacion de chatter en maquina herramienta *19 Congreso de Máquinas-Herramienta y Tecnologías de Fabricación* 1-11

[7] Munoa J , Mancisidor I, Loix N, Uriarte L G, Barcena R and Zatarain M 2013 Chatter suppression in ram type travelling column milling machines using a biaxial inertial actuator *CIRP Annals – Manufacturing Technology* **62** 407-410

[8] Fazil Ahamed M and Chauhan S 2016 Hydraulic Actuator Systems with Non-Newtonian Working Fluid *Bonfring International Journal of Industrial Engineering and Management Science* **6** 135-139

[9] Brecher C and Schulz A 2005 Electrohydraulic active damping system *Annals of the CIRP* **54** 389-392

[10] Law M, Wabner M, Colditz A, Kolouch M, Noack S and Ihlenfeldt S 2015 Active vibration isolation of machine tools using an electro-hydraulic actuator *CIRP Journal of Manufacturing Science and Technology*

[11] Law M 2016 Design Guidelines for an Electro-Hydraulic Actuator to Isolate Machines from Vibrations *6th International & 27th All India Manufacturing Technology, Design and Research Conference*

[12] Parker 2011 Direct Operated Proportional DC Valve, Series D1FP, Technical Catalogue

[13] Merritt H E 1967 Hydraulic Control Systems Tools New York John Wiley & Sons 132-173