J. Appl. Comput. Mech., 7(1) (2021) 270-276
DOI: 10.22055/JACM.2020.32923.2102

ISSN: 2383-4536
jacm.scu.ac.ir

Shahid Chamran University of Ahvaz

Journal of Applied and Computational Mechanics

Research Paper

Dynamic Investigation of Non-linear Behavior of Hydraulic Cylinder in Mold Oscillator using PID Control Process

Yonghui Park[1], Changwoo Lee[2,3]

[1] Department of Mechanical Engineering, Yuhan University, 590, Gyeongin-ro, Bucheon-si, Gyeonggi-do, Republic of Korea, Email: yhpark@yuhan.ac.kr

[2] Steel-pipe Technology Team, Pohang Institute of Metal Industry Advancement
56, Jigok-ro, Nam-gu, Pohang-si, Gyeongsangbuk-do, Republic of Korea, Email: cw82.lee@pomia.or.kr

[3] Department of Mechanical Engineering, Pohang University of Science and Technology
77, Cheongam-ro, Nam-gu, Pohang-si, Gyeongsangbuk-do, Republic of Korea, Email: cw82.lee@postech.ac.kr

Received March 12 2020; Revised April 16 2020; Accepted for publication April 16 2020.
Corresponding author: Yonghui Park (yhpark@yuhan.ac.kr)

Abstract. To learn the dynamic characteristics of a mold oscillator, we establish a model that describes the relationship between force equilibrium of a hydraulic cylinder and mold under various oscillation conditions. The non-linearity caused by the servo-value and the operating fluid is considered as excitation, and is calculated as control error between an input signal and mold oscillation in real-time by a PID control process. Based on the non-linear property, we determine that the dynamic behavior is caused by mold oscillation displacement and hydraulic cylinder pressure. We define excitation frequency and harmonic terms, and determine that the sources of the harmonic peak frequency and high peak frequency; (50n ± exciting frequency ω_{exc}) are friction between the piston and hydraulic cylinder, and variable stiffness of the operating fluid. Finally, a mathematical model of the hydraulic chamber that can represent the unknown non-linear phenomenon is derived.

Keywords: Dynamics, Mold oscillator, Hydraulic cylinder, Beat phenomenon.

1. Introduction

Using information provided by a good simulation model, engineers can modify and improve an instrument by sensitivity analysis, optimization, and application of control methods. Despite the importance of a good model, most steel manufacturers do not have a design process to evaluate the mold oscillator that is used in the continuous casting process, so all instruments in the mold oscillator are imported [1-2]. To develop a comprehensive design and increase business profitability, these manufacturers should develop the required products or systems independently.

A mold oscillator with a hydraulic servo system has characteristics that differ according to system structure and the oscillation conditions including oscillating stoke and frequency. The hydraulic servo system is defined by governing equations that describe the operations of the orifice, the continuity, servo-valves' transfer functions, force equilibrium, and feedback control. [3-5] Due to the governing equation, the dynamic characteristic is correlated with complex design variables. Previous work suggested that the servo system has the greatest effect on the dynamic characteristic of the mold oscillator. Based on the fact that inverse dynamics gives a solution if numerical direct integration is possible, the dynamics characteristic from this previous work is only one of various candidates. [6-7] In other words, solution of the unknown nonlinear force is not effective under different oscillating conditions. To solve this problem, we identify the globally optimal solution of the unknown nonlinear force effect by applying feedback control. In this process, input signal of the unknown nonlinear force is primary interest.

First, we establish a control process that includes a dynamics model for the mold oscillator and PID controller, and define unknown non-linear force as in previous work. The non-linear force can change according to control status, but we suggest a non-linear force that is acquired by an optimization process to minimize control error. We observe the distribution of the control error at various control gains.

Second, we establish simple models to explain the dynamic behavior around specific peak frequencies in the displacement data. The excitation frequency ω_{exc} and its harmonic terms are defined by friction in the cylinder, and the high-frequency ($50n\pm\omega_{exc}$) Hz (n = 1 – 4) is simulated by a multi-degree-of-freedom (MDOF) mass-and-spring system with parallel connections.

2. System Modeling

2.1 Dynamics model of the mold oscillator with PID

The mold oscillator shakes the mold as specified by a user. The user should adjust oscillation configurations to minimize the error between the actual and desired oscillations. The hydraulic servo system maintains balance between force and displacement.

Published online: April 17 2020

However, the servo system cannot be modeled easily because it has non-linear responses to some inputs. [8-9] In the past, engineers checked dynamic behaviors by carefully operating a hardware model without a simulation model. This method can be dangerous, but a simulation model of the process does not always give the dynamic behaviors precisely, because mathematical techniques may not be sufficient. Therefore a reliable simulation model of the mold oscillator is required.

In this section, we establish an integrated model for the mold oscillation mechanism. We use a control diagram (Fig. 1) when calculating unknown non-linear forces. Generally, the PID controller is combined with the hydraulic pressure of the hydraulic cylinder to control the displacement of mold oscillator in Fig. 1. However, the mathematical model of mold oscillator does not reflect the unknown non-linear force of the hydraulic pressure, so we measured the hydraulic pressure and applied it as external force to the dynamic simulation as shown in Fig. 1 to find out the unknown non-linear force. The PID controller should be modified to minimize error between input and output. The PID controller has the advantage that it can estimate the error distribution easily, so the hydraulic servo system can be simplified by using a PID controller. The output of the PID controller is the unknown non-linear force. The mold oscillator is established by dynamics (Fig. 2). The mold is supported by two leaf springs and two hydraulic cylinders. These components are located symmetrically to the center of gravity of the mold. The hydraulic cylinder provides stiffness and pressure at the same time. We use experimental pressure data while applying sinusoidal oscillation at various frequencies and amplitudes ("sine sweeping"). The leaf spring is pre-compressed by force equilibrium with the weight of the mold. The mold has 6 DOF through a bushing joint with the ground. The integrated model gives non-linear force at the point of application of the hydraulic cylinder. Design variables and operating conditions were duplicated from previous work.

The hydraulic pressure is treated as disturbance in the integrated model. Although the disturbance changes according experiment conditions, our task is to find the non-linear force model that provides results that match experimental measurements. Eq. (1) is the non-linear force from the PID controller. [10-11]

$$F_\alpha = K_p\left(z(t)_{desire} - z(t)\right) + K_i\int_0^t \left(z(\tau)_{desire} - z(\tau)\right)d\tau + K_d\frac{d\left(z(t)_{desire} - z(t)\right)}{dt} \tag{1}$$

F_α is a compensated force [N], K_p & K_i & K_d are Proportional & Integral & Derivative control gains, z_{desire} is a target input signal and z is mold displacement.

Eq. (2) is the non-linear force by inverse dynamics:

$$\begin{aligned} &m\ddot{z} + c\dot{z} + (k_{hydraulic} + k_{leaf})z = F - mg + k_{leaf}z_{leaf} + \alpha = F_{total} \\ &\ddot{z}, \dot{z}, z \xrightarrow{\text{Inverse Dynamics}} = F \end{aligned} \tag{2}$$

m is mass of the mold, c is the damping coefficient of the system, $k_{hydraulic}$ is internal stiffness of the hydraulic cylinder, k_{leaf} is spring constant of the leaf spring, z_{leaf} is initial position of the leaf spring, g is the gravity acceleration, α is the unknown force and z is the displacement of the mold. The non-linear force is defined by the mold displacement, velocity, and acceleration. Dynamic behaviors can change if the assumptions used in numerical integration change. [12-14] However, the PID controller includes error status in detail, and can be changed by control gains to represent the operability of the servo system. To minimize the control error, we use the Ziegle Nichols method [15] to find the control gains K_p, K_i, and K_d. according to basic rules (Table 1).

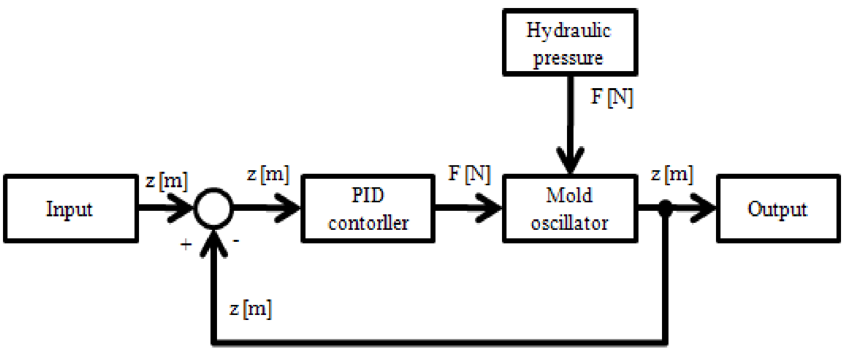

Fig. 1. Control diagram for calculating the unknown non-linear force

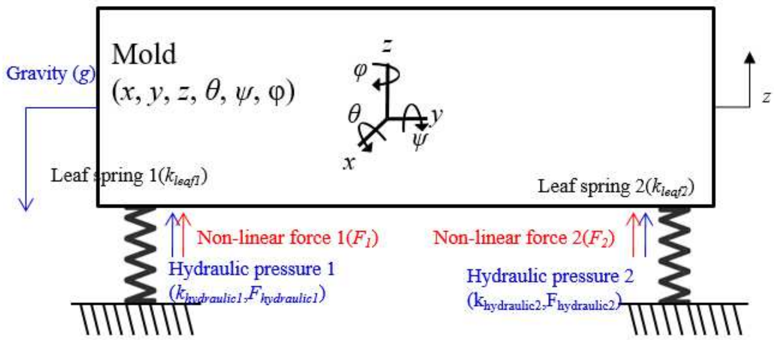

Fig. 2. Control diagram for calculating the unknown non-linear force

Table 1. Basic rule for selecting the control gain

Quantity	Variables
Objective function	Mean(z_{desire}-z)
Design variables	K_p, K_i, K_d
Constraints	K_p, K_i, K_d > 0

Table 2. Natural frequency [Hz] distribution

Excitation	Resonance frequencies			
Frequency ω_{exc} [Hz]	ω_1 / ω_5 [Hz]	ω_2 / ω_6 [Hz]	ω_3 / ω_7 [Hz]	ω_4 [Hz]
1	51 / 49	101 / 99	151 / 149	ω_4 (Stroke)
2	52 / 48	102 / 98	152 / 148	ω_4 (Stroke)
3	53 / 47	103 / 97	153 / 147	ω_4 (Stroke)
4	54 / 46	104 / 96	154 / 146	ω_4 (Stroke)
5	55 / 45	105 / 95	155 / 145	ω_4 (Stroke)

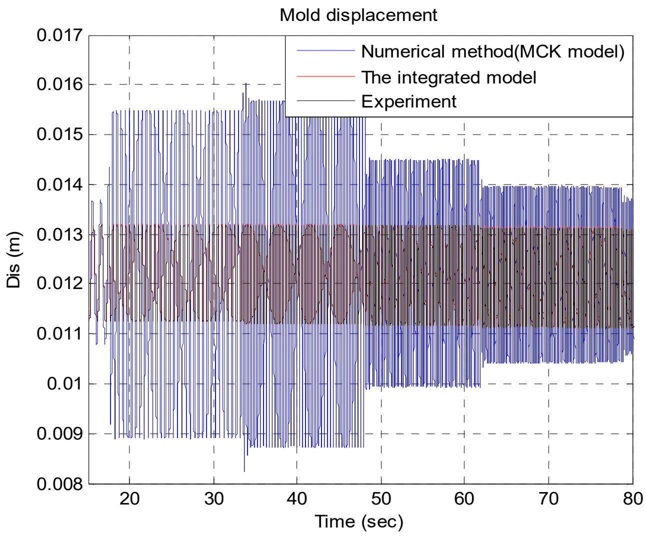

Fig. 3. Mold displacement from 3 models with sweeping excitation

2.2 Dynamic characteristic according controller gain

Unlike a normal PID controller, our PID controller should have high gain values instead of the hydraulic servo system. In other words, the PID controller compensates for the control error from displacement dimension [m] to force dimension [N] (Fig. 1). By trial and error, we determined that the control error decreased dramatically at $K_p > 3.0 \times 10^8$. We compared the displacement observed in the experiment, to that predicted by the integrated model including the PID controller, and the numerical solution (MCK model) by inverse dynamics and Newmark's beta method (Fig. 3). In Fig. 3, the red and the black line is being piled up one on another. The experiment and the integrated model show similar distributions, but the MCK model solution differs from them. However, when K_p is low, the integrated model and the numerical solution have the same distributions. These results indicate that inverse dynamics is acceptable within low acceleration variation, and that the non-linear force by the PID controller can be tracked in real time.

3. Mathematical model for the unknown non-linear force

3.1 Multi-degree of freedom mass-spring system

We establish an MDOF mass-and-spring system to represent the stiffness of the hydraulic cylinder. We focus on a high peak frequency ($50n\pm\omega_{exc}$) Hz that is not related to an excitation signal. We designed a 7-DOF mass-and-spring system for the piston, the operating fluid, and the cylinder (Fig. 4). We can expand and reduce degree of freedom to represent additional peak frequency on the piston by applying additional m_{hi} & k beside relative mass of the operating fluid. The mass and stiffness of the piston cause it to have a natural oscillation frequency. To represent the beat phenomenon, we adjust the hydraulic mass and stiffness corresponding resonant frequencies ($50n\pm\omega_{exc}$) Hz. We obtained the distribution of frequencies of the operating fluid and the piston's natural oscillation in the 7-DOF system (Table 2). The natural frequency ω_4 of the piston is much lower than the other frequencies, and is a function of stroke of the oscillating signal. We don't describe ω_4 in Table. 2, because space between the piston and operating fluid mass is not effective to determine ($50n\pm\omega_{exc}$) Hz. So, the stiffness values of k_4, k_5, k_6, k_7, k_8 and k_9 are much lower than k_1, k_2, k_3, k_{10}, k_{11} and k_{12}. However, we need to set appropriate mass and stiffness of the piston to get same the unknown nonlinear force as experiment, because they change according to excitation conditions.

3.2 Reaction force by the hydraulic oil

When we apply excitation displacement, velocity, and acceleration to the piston, the reaction force on it is exerted by springs:

$$F_{reaction} = k_4(z_4-(z_1+z_4))+k_5(z_4-(z_2+z_4))+k_6(z_4-(z_3+z_4))+k_7(z_4-(z_5+z_4)) \\ +k_8(z_4-(z_6+z_4))+k_9(z_4-(z_7+z_4)) \quad (3)$$

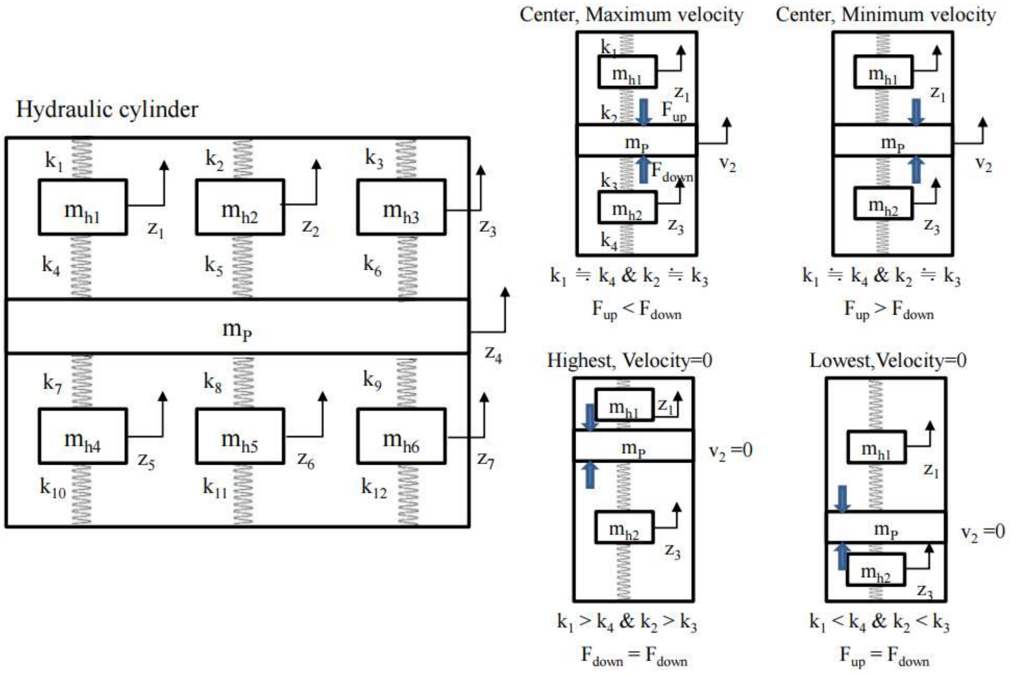

Fig. 4. Dynamics model for the hydraulic cylinder and mechanism according to position of the piston

The displacement of z_4 oscillates, but Eq. (3) does not include z_4. The general spring model gives a reaction force in the opposite direction. However, the non-linear force of the hydraulic cylinder always heads toward the piston; center, and the free position of the spring is renewed as current position in a moment. In other words, fixed end position of a spring is changed according its displacement of opposite end position. To identify the force balance in the hydraulic cylinder we define a 3-DOF mass-spring system (Fig. 4) instead of the multiple springs. F_{up} and F_{down} are non-linear forces in each chamber. These forces are always directed toward the piston. Depending on the velocity and position of the piston, the stiffness and force in the hydraulic chamber change. When the velocity of the piston is maximum or minimum, the amplitude of the beat phenomenon is strong because the up and down chambers have similar stiffnesses. In contrast, when the velocity of the hydraulic piston is low near the highest and the lowest position of the piston, the beat phenomenon is weak.

Eq. (3) is not sufficient to express stiffness variation and direction of the non-linear force. For instance, the reaction force of spring k_7 is not always positive, because the displacement of z_5 changes from negative to positive. But we can define variable stiffness of the hydraulic chamber by using the reaction force. When the piston is located at the highest or lowest position, the stiffness of the operating fluid k_1, k_2, k_3, k_{10}, k_{11} and k_{12} in Fig. 4 reverses because of the volume of the chamber. Generally, the stiffness of the operating fluid decreases as the chamber expands; i.e., the stiffness of the up chamber increases as the stiffness of the down chamber decreases, and vice versa. [5],[16] During the sine-sweeping experiment, high-frequency peaks were observed at 49 and 51, 48 and 52, …, and 43 and 57 Hz. By applying force difference between the hydraulic chambers as Eq. (3), we can easily define a conversion of the non-linear force that acts on the piston.

3.3 Numerical iteration and Simulation configuration

To solve the equation of motion, we apply Newmark's beta method, which is a form of numerical direct integration. This method calculates a solution that satisfies the equation of motion and a change, assuming that the dynamic response in each step changes gradually. The dynamic responses (Eqs. 4-6) include initial condition, time step, a_1, a_2 and a_3 coefficients from the linear acceleration assumption, and effective stiffness matrix in every step. We use $\beta = 0.25$, $\gamma = 0.5$ according to the average acceleration method. Newmark's beta method guarantees stability and convergence of solution regardless of time step and system. However, the computing efficiency of the method is low because it includes an inertia matrix without lumped method. However, this low efficiency is not important here, because our model has few degrees of freedom.

$$[M]\{\ddot{z}\}_1 + [C]\{\dot{z}\}_1 + [K]\{z\}_1 = F_1 \longrightarrow \{\ddot{z}\}_1 = [M]^{-1}\left(F_1 - [C]\{\dot{z}\}_1 - [K]\{z\}_1\right) \tag{4}$$

$$\vec{F}_{i+1} = F_{i+1} + a_1 z_i + a_2 \dot{z}_i + a_3 \ddot{z}_i \tag{5}$$

$$z_{i+1} = \frac{\vec{F}_{i+1}}{\vec{k}}, \dot{z}_{i+1} = \frac{\gamma}{\beta \Delta t}\left(z_{i+1} - z_i\right) + \left(1 - \frac{\gamma}{\beta}\right)\dot{z}_i + \Delta t\left(1 - \frac{\gamma}{2\beta}\right)\ddot{z}_i, \ddot{z}_{i+1} = \frac{1}{\beta(\Delta t)^2}\left(z_{i+1} - z_i\right) - \left(\frac{1}{\beta \Delta t}\right)\dot{z}_i - \left(\frac{1}{2\beta} - 1\right)\ddot{z}_1 \tag{6}$$

For the simulation, we pre-defined the dynamic response of the piston instead of applying force; i.e., displacement, velocity, and acceleration of the piston were given. In the model, the reaction force is determined by the relationship between the piston and each hydraulic mass. As in experiment, the mold displacement corresponds to the piston displacement, so we can define the reaction force as result from the pre-defined piston motion to calculate the unknown non-linear force.

Eq. (7) is the pre-defined response of the piston. The signal is edited by changing amplitude A and frequency ω_{exc}. We selected A = 1 mm, ω_{exc} = 1 Hz.

$$z_{piston} = A\sin(\omega_{exc} t), \dot{z}_{piston} = A\omega_{exc}\cos(\omega_{exc} t), \ddot{z}_{piston} = -A{\omega_{exc}}^2\sin(\omega_{exc} t) \tag{7}$$

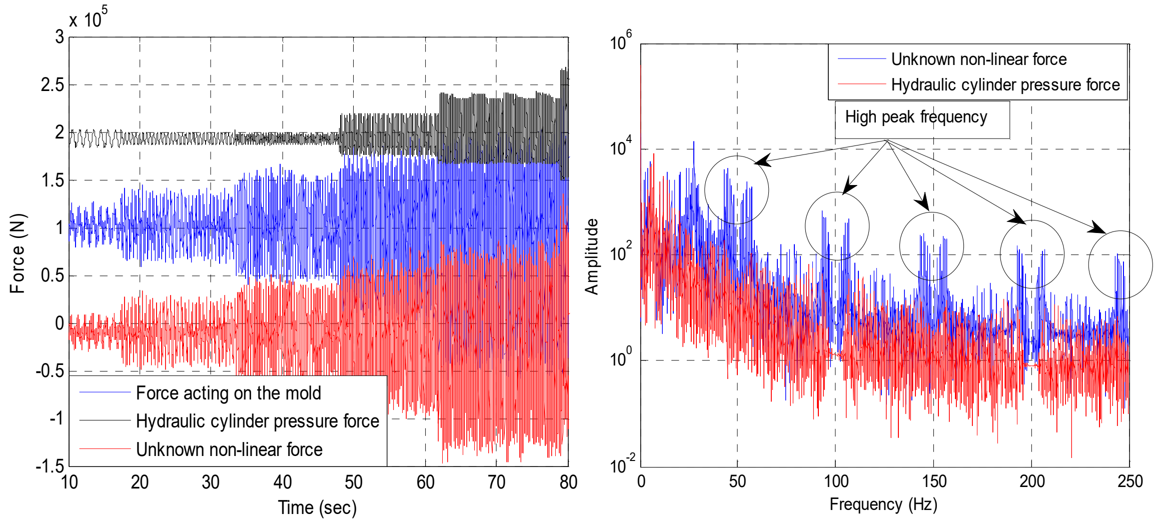

Fig. 5. Time history and frequency analysis of the unknown non-linear force

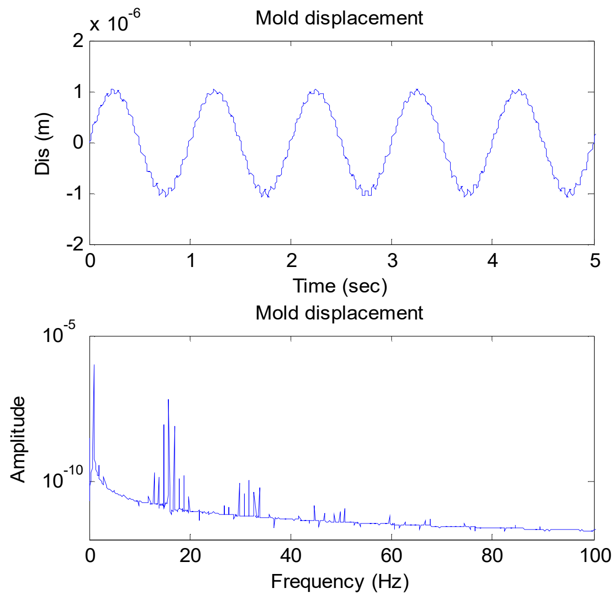

Fig. 6. Dynamic responses of the mass-spring under column friction

4. Result and Discussion

4.1 Dynamic characteristic of non-linear force and column friction model of piston

The unknown non-linear force from the integrated model was observed when a sine-sweeping input signal with A = 2 mm and $1 \le \omega_{exc} \le 6$ Hz was applied (Fig. 5). We adjusted PID gains to closely follow the input signal, which is the mold displacement. Generally, the PID gain is designed for as a transfer function to reduce overshoot, settling time, and other factors. However, we cannot define these behaviors and cannot modify oscillation conditions, because experimental data about hydraulic pressure are fixed. Although the criterion of PID gain is not clear, we try to minimize error between the input and output oscillating signal by optimization. The force distribution (Fig. 5) increased over time, as was observed in previous research. Because the distribution can be changed by step time and the PID gain, the dynamic response should be valid for the experiment data and for the input oscillating signal.

The unknown non-linear force distribution was translated to the frequency domain (Fig. 5). The plot shows $1 \le \omega_{exc} \le 6$ Hz and its harmonic terms, and also ($50n \pm \omega_{exc}$) Hz. The cause of these peak frequencies is related to oscillating conditions or hydraulic cylinder status. The experimental data including the displacement and the pressure can be regarded as steady-state responses, and the dynamics model includes damping ratio. Therefore, the non-linear force is an external force. If these frequencies are characteristics of the mold oscillator, the damper will gradually decrease the unknown non-linear force.

We define two models for representing the oscillation frequency and ($50n \pm \omega_{exc}$) Hz. In this section, the oscillation frequency and harmonic terms are investigated.

The displacement of the mass in response to the sine sweeping approximated a sine curve, contaminated by several harmonic components that were revealed after translation to the frequency domain (Fig. 6). However, this process only identifies the frequencies; it does not explain them. To find the causes of the harmonics, we model a 1-DOF mass-spring system with column friction; Eq. (8) is its equation of motion.

$$m\ddot{z} + \mu m g \times sign(\dot{z}) + c\dot{z} + kz = F, \ddot{z} + \mu g \times sign(\dot{z}) + 2\zeta\omega_n \dot{z} + \omega_n{}^2 z = F \quad (8)$$

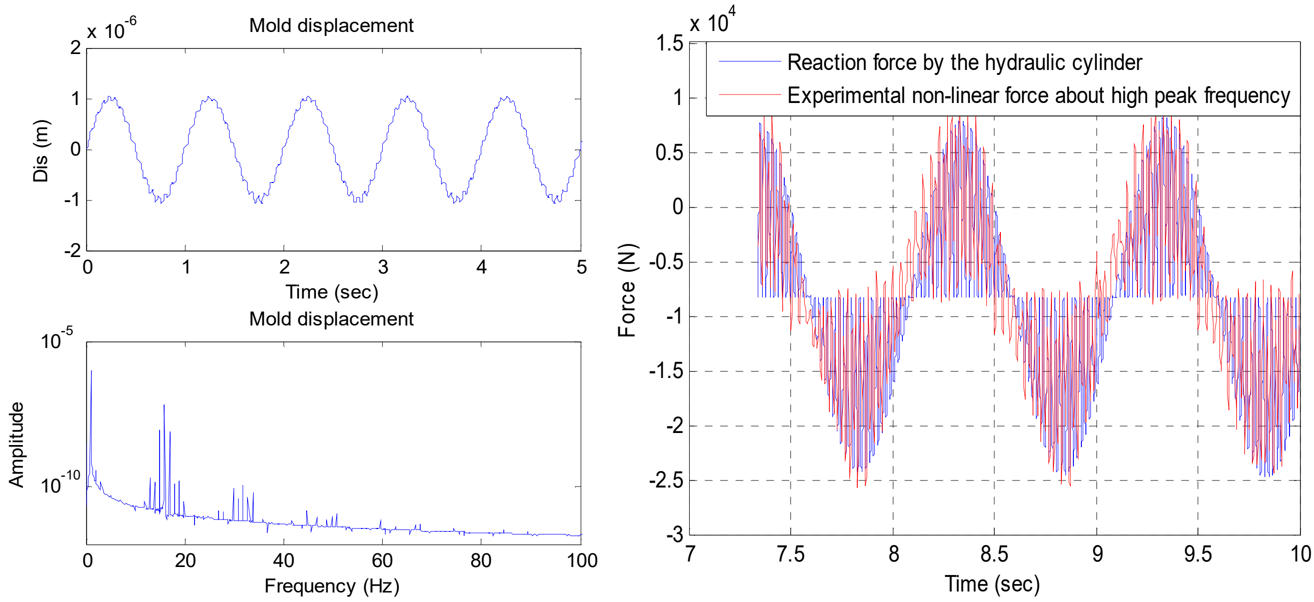

Fig. 7. Unknown non-linear forces in the 3DOF and 5DOF system

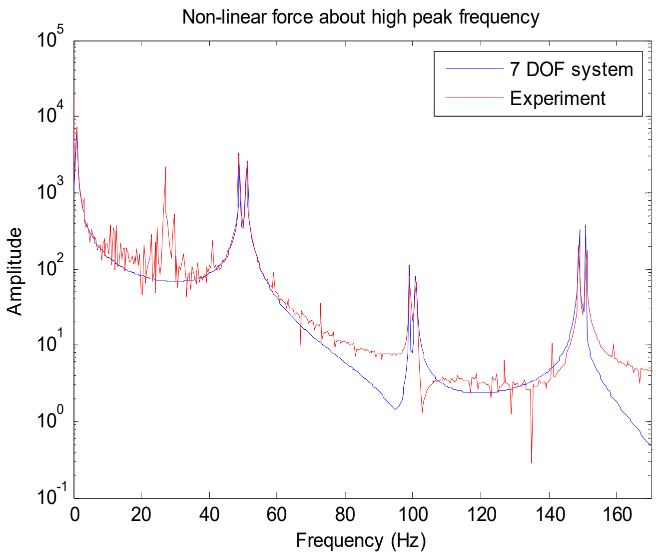

Fig. 8. Unknown non-linear forces in the 7 DOF system in Frequency domain

where m is mass μ is the friction coefficient between the piston and cylinder, c is the damping coefficient of the hydraulic cylinder and k is the internal stiffness of the hydraulic cylinder. The system is arbitrary to represent general phenomena. System variables were: natural frequency ω_n = 15.9 Hz, damping ratio ζ = 0.3, and k = 10^9 N/m. We calculated displacement of m when force F was applied at frequency 1 Hz (Fig. 6). The displacement includes ω_{exc} and its harmonic terms, and the natural frequency. ω_{exc} and the natural frequency had strong peaks, and the harmonic terms of ω_{exc} had weak peaks. By comparing this distribution to experimental mold displacement, we find that the harmonic terms of ω_{exc} are founded equally. By using the distribution of the peak frequency, we conclude that the hydraulic cylinder has a friction problem that corresponds to the harmonic terms of ω_{exc}, and can estimate system characteristic from experiment data regarding frequency.

4.2 Dynamic responses of the MDOF mass spring system for the hydraulic chamber

Whereas ω_{exc} and its harmonic terms are related to the excitation conditions, the ($50n\pm\omega_{exc}$)-Hz terms are related to relationship between the hydraulic chamber (Section 3). The frequency has paired peak frequencies that are symmetric around 50, 100, 150, and 200 Hz. Reaction forces were obtained for the 3-, 5-, and 7-DOF mass-spring systems (Fig. 7-8). As the number of DOF increased, the non-linear force for beat phenomenon became progressively more similar to the experiment data (Fig. 7). We define the piston's response as an excitation force (Section 3.3); i.e., the spring's reaction forces are not correlated. Depending on the number of DOF, the model can include frequency pairs at 199 and 201 Hz, 249 and 251 Hz, 299 and 301 Hz, and so on.

In the experiment result, the amplitude of the beats was proportional to the velocity of the mold oscillator, and was the lowest when the mold is not operated. According this fact, beating is amplified when the chamber has similar dynamics status; the chamber has same volume, and the piston locates at the center position.

We find that the hydraulic cylinder model has variable and various dynamic characteristics. Beats always occur during operation. Engineers must consider the peak frequencies of the hydraulic chamber. If engineers cannot identify these frequencies, the mechanical parts including the mold and additional coil spring will be on unstable area, resonance area by the operating fluid when the natural mode frequency coincides with the hydraulic chamber frequency.

5. Conclusion

We established an integrated model to calculate unknown non-linear forces that act on a mold oscillator. To extract the unknown non-linear force, we used a dynamics model of the mold oscillator, compensation, and experimental measurements of displacement and pressure. By using the model, we found that the unknown non-linear force consists of the excitation frequency ω_{exc} and harmonic terms and the hydraulic chamber frequency: $(50n\pm\omega_{exc})$ Hz. To investigate these signals, we established mathematical models of column friction and MDOF mass-spring system. First, ω_{exc} and harmonics term on the mold were caused by friction between the piston and cylinder. Displacement had high peak frequency at the natural frequency and at ω_{exc}, and local peaks at the harmonics of ω_{exc}. Second, we defined a MDOF mass-spring system to represent variable and various characteristics for the hydraulic chamber. The model represented the complicated unknown non-linear force easily.

Author Contributions

Author 1 planned the scheme, initiated the project and developed the mathematical modeling and examined the theory validation; Author 2 conducted the experiments and analyzed the empirical results. The manuscript was written through the contribution of all authors. All authors discussed the results, reviewed and approved the final version of the manuscript.

Acknowledgments

This work was supported by Yuhan University, POMIA and POSTECH with equipment including engineering software.

Conflict of Interest

The authors declared no potential conflicts of interest with respect to the research, authorship and publication of this article.

Funding

The authors received no financial support for the research, authorship and publication of this article.

References

[1] Jung, Y. J., Kang, G. P., Kim, Y. D., Kang, C. G., An Introduction to Design Technology of Slab Continuous Casting Machine, *Journal of Korea Foundry Society*, 21(3), 2001, 147-154.
[2] Park, J. C., Lee, S. J., HI, J., Development of diagnosis system of mold oscillation in a continuous slab casting machine, *Journal of the Korean Society for Precision Engineering*, 13(5), 1996, 84-94.
[3] Bonchis, A., Corke, P. I., Rye, D. C., Ha, Q. P., Variable structure methods in hydraulic servo systems control, *Automatica*, 37(4), 2001, 589-595.
[4] Milić, V., Šitum, Ž., Essert, M., Robust H∞ position control synthesis of an electro-hydraulic servo system, *ISA transactions*, 49(4), 2010, 535-542
[5] Park, Y. H., Park, H. C., Dynamic characteristic analysis of hydraulic servo system in a mold oscillating mechanism, *Advanced Materials Research*, 1025, 2014, 183-191.
[6] Ider, S. K., Inverse dynamics of parallel manipulators in the presence of drive singularities, *Mechanism and Machine Theory*, 40(1), 2005, 33-44.
[7] Tsai, M. S., Yuan, W. H., Inverse dynamics analysis for a 3-PRS parallel mechanism based on a special decomposition of the reaction forces, *Mechanism and Machine Theory*, 45(11), 2010 1491-1508.
[8] Montanari, M., Ronchi, F., Rossi, C., Tilli, A., Tonielli, A., Control and performance evaluation of a clutch servo system with hydraulic actuation, *Control Engineering Practice*, 12(11), 2004, 1369-1379.
[9] Seo, J., Venugopal, R., Kenné, J. P., Feedback linearization based control of a rotational hydraulic drive, *Control Engineering Practice*, 15(12), 2007, 1495-1507.
[10] El-Bardini, M., El-Nagar, A. M., Interval type-2 fuzzy PID controller for uncertain nonlinear inverted pendulum system, *ISA transactions*, 53(3), 2014, 732-743.
[11] Liu, L., Pan, F., Xue, D., Variable-order fuzzy fractional PID controller, *ISA transactions*, 55, 2015, 227-233.
[12] Craig, R. R., Kurdila, A. J., *Fundamentals of structural dynamics*, John Wiley & Sons, New York, 2006.
[13] Bathe, K. J., *Finite element procedures*, Cambridge, 2006
[14] Constantinides, A., Schowalter, W. P., Carberry, J. J., Fair, J. R., *Applied numerical methods with personal computers*, McGraw-Hill, Inc., New York, 1987.
[15] Kang, I. C., Choi, S. M., Choi, J. S., An Analytic Study on the Relations between the Ziegler-Nichols Tuning Methods for Controllers, *Journal of the Korean Society of Marine Engineering*, 26(2), 2002, 219-225.
[16] Park, Y. H., Lee, H. K., Park, K. T., Park, H. C., Practical behavior of advanced non-linear hydraulic servo system model for a mold oscillating mechanism depending on line volume, *Journal of Mechanical Science and Technology*, 30(3), 2016, 975-982.

ORCID iD

Yonghui Park https://orcid.org/0000-0001-6716-6935

How to cite this article: Park Y., Lee C. Dynamic Investigation of Non-linear Behavior of Hydraulic Cylinder in Mold Oscillator using PID Control Process, *J. Appl. Comput. Mech.*, 7(1), 2021, 270–276. https://doi.org/10.22055/JACM