이력특성을 고려한 초정밀 SIDM 회전기구의 모델링
Modeling of an Ultra Precision SIDM Rotational Device Considering Hysteresis

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1. Introduction

Nawadays, impact microactuators have attracted a lot of attention due to ease of fabrication, capability of batch processing, robustness to environmental perturbations, high accuracy and high power output, high response speed of actuation elements.

Recently SIDM (Smooth Impact Drive Mechanism) based ultra precision rotational device was developed in the Micro-Electro-Mechanical Systems laboratory of University of Ulsan [1]. This paper presents the above device modeling using accurate frictional model and considering hysteresis.

2. Ultra Precision Rotational Device

The investigated ultra precision rotational device has central symmetrical structure (see Fig. 1). It consists of preload mechanism, rotational disk and driving unit. The last includes the piezoelectric element that joined to the flexible hinge and friction part through its holder. The friction part is kept in touch with rotational disk. Rotational disk is covered by alumina (Al_2O_3) cap and uses 3.5 inch hard disk rotation bearing. Rotational axis of the disk is fixed. Preload mechanism is installed onto slide rails and composed of base, rolling bearing, spring and loadcell. Spring force defines preload force value. Preload force's direction is perpendicular to the vibration direction of the piezoelectric element and is applied to the friction part.

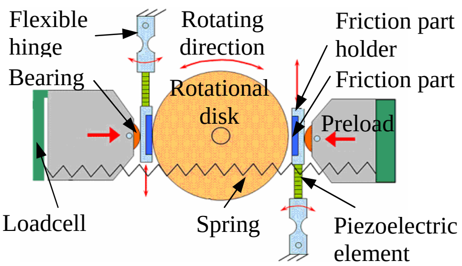

Fig. 1 Schematic diagram [1]

The principle of rotation is based on the Smooth Impact Drive Mechanism (SIDM) [4]. This mechanism uses slow extension and rapid contraction of the friction part. At low driving frequency the friction part rotates the disk during slow motion and slips during rapid motion. However, at high driving frequency the sliding always occurs even during a slow extension. In this case, rotation of the disk results from the difference between slow extension time and rapid contraction time.

3. Dynamic Model

Multilayer piezoelectric element that used in the system can be considered as a mass-damper-spring system. In such a way, the free-body diagram of the investigated rotational device is become as shown in Fig. 2. And the dynamic equations [7] is given as

$$\begin{aligned}&(m_p + m_f)\ddot{x}_p + c_p\dot{x}_p + k_p x_p + F_d = F_p \\ &F_d \cdot r = I\varepsilon,\end{aligned} \tag{1}$$

where m_p and m_f are masses of the piezoelectric element and friction part (including friction part holder), respectively; x_p, c_p, k_p are displacement, damping coefficient and stiffness coefficient of the piezoelectric element, respectively; and F_p, F_d are the piezoelectric force and the driving force (friction force).

The piezoelectric force F_p in consideration of hysteresis effect that described by Bouc-Wen model [5] is defined as

$$F_p = k_p(d_e V_p - z), \tag{2}$$

where d_e is piezoelectric constant; z is a state variable α_p is parameter that controls the restoring force amplitude; β_p and γ_p are parameters that control the shape of hysteresis loop. The state variable [5] is given as

$$\dot{z} = \alpha_p d_e \dot{V}_p - \beta_p |\dot{V}_p| z - \gamma_p \dot{V}_p |z|. \tag{3}$$

The friction force is realized by the Leuven model [6] as well as the hysteresis effect of the friction force is described by Bouc-Wen model [5]. In detail the Leuven model consists of a nonlinear state and a friction force equations [6] as given respectively

$$\begin{aligned}&\frac{dh}{dt} = v\left(1 - \operatorname{sgn}\left(\frac{F_h(h)}{s(v)}\right)\left|\frac{F_h(h)}{s(v)}\right|^n\right) \\ &F_f = F_h(h) + \sigma_1 \frac{dz}{dt} + \sigma_2 v,\end{aligned} \tag{4}$$

where n is a coefficient used to shape the transition curve; v is velocity; h is a state variable; $F_h(h)$ is the hysteresis frictional force; and function $s(v)$ describing a constant–velocity behavior [6] is given as

$$s(v) = \operatorname{sgn}(v)\left(F_c + (F_s - F_c)e^{-\left(|v|/v_s\right)^{\delta}}\right), \tag{5}$$

where δ is an arbitrary exponent; F_c, F_s are Coulomb frictional force and static frictional force respectively; v_s is a Stribeck velocity.

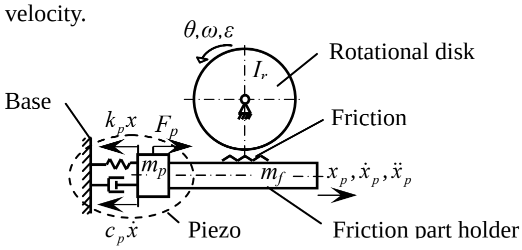

Fig. 2 Free-body diagram

4. Simulation

Simulation was performed using MATLAB package. To solve the system of equations (1), (3)-(5) Runge-Kutta method was applied.

The simulation of the established nonlinear dynamic model with taking into consideration the hysteresis effect was carried out and compared with the experimental results. The saw-tooth waveform driving voltage $V(t)$ was used to excite the piezoelectric element (Fig.3). In the Fig.4 the experimental and simulated angular displacement of the rotational device are presented. A macroscopic outlook exhibits well agreement of the simulation and the experimental results.

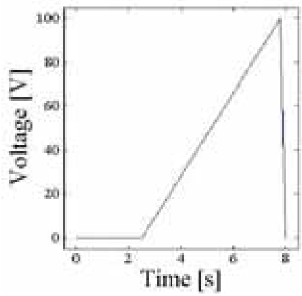

Fig.3 The single-wave saw-tooth waveform driving voltage

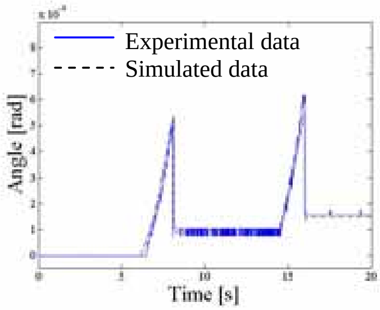

Fig.4 IDM dynamic response on two single-waves

In the following Fig. 5 the effect of the piezoelectric element hysteresis can be observed. It is obvious that the simulation with piezoelectric hysteresis of the final angular positioning is much more accurate then the one neglecting it. This can be explained by the fact that the piezoelectric force forms a hysteresis loop with respect to the input voltage and the residual hysteresis still occurs in the piezoelectric element even if there is no any input voltage (Fig. 6).

In spite of the time and computational complexity of frictional force modeling it needed to be considered for the system requiring a high precision output such as the rotational device been investigated in this work.

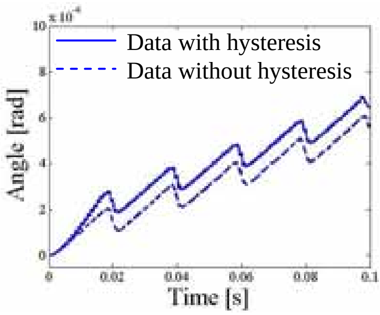

Fig. 5 IDM dynamic response

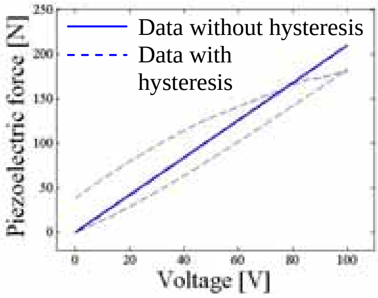

Fig. 6 Hysteresis effect of the piezoelectric element due to the single-wave input

5. Conclusion

In this paper modeling and simulation of the recently developed ultra precision rotational device was carried out. Under this investigation, the accurate dynamic model has been obtained. It became clear that the frictional force and the hysteresis effect of the piezoelectric element should be considered at the same time to obtain accurate simulation results. The Leuven model [6] has been applied to describe the frictional force. The hysteresis effect of both the piezoelectric and the frictional force were described by the Bouc-Wen model [5]. The angular displacement output has been tested. The calculated results closely matched with the experimental results.

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Reference

1. Lee, S., and Jeon, J. U., "A Study on Ultra Precision Rotational Device Using Smooth Impact Drive Mechanism", J. KSPE, **25**, 140-147, 2008.
2. Fung, R. F., Han, C. F., Ha, J. L., and Chang, J. R., "Effects of Frictional Models on the Dynamic Responses of the Impact Drive Mechanism", J. Vibration and Acoustics, **128**, 88-96, 2006.
3. Low, T. S. and Guo, W., "Modeling of a Three-Layer Piezoelectric Bimorph Beam with Hysteresis", J. MEMS, **4**, 230-237, 1995.
4. Yoshida R., Okamoto Y., Higuchi T., and Hamamatsu A., "Development of Smooth Impact Drive Mechanism (SIDM)—Proposal of Driving Mechanism and Basic Performance", J. JSPE, **65**, 111–115, 1999.
5. Ikhouane, F., Manosa, V., and Rodellar, J., "Dynamic Properties of the Hysteretic Bouc–Wen Model", Systems and Control Lett., **56**, 197–205, 2006.
6. Swevers, J., Al-Bender, F., Ganseman, G., and Prajogo, T., "An Integrated Friction Model Structure with Improved Presliding Behaviour for Accurate Frictional Compensation", IEEE Trans. Automatic Control, **45**, 675–686, 2000.
7. Canudas de Wit, C., and Tsiotras, P., "Dynamic Tire Friction Models for Vehicle Traction Control", Proc. 38th IEEE Conf. Decision and Control, 3746-3751, 1999.