VOL. 11, NO. 13, JULY 2016 ISSN 1819-6608
ARPN Journal of Engineering and Applied Sciences

www.arpnjournals.com

DESIGN, DEVELOPMENT AND PRECISION SCANNING OF SINGLE DOF FLEXURAL MECHANISM USING DOUBLE FLEXURAL MANIPULATOR

Sharad S. Mulik[1], Suhas P. Deshmukh[2], Mahesh S. Shewale[3] and Hrishikesh Zambare[4]
[1]Department of Mechanical Engineering, Sathyabama University, Chennai, Tamilnadu, India
[2]Department of Mechanical Engineering, Sinhgad Academy of Engineering, Pune, Maharashtra, India
[3]Department of Mechanical Engineering, Trinity Academy of Engineering, Pune, Maharashtra, India
[4]Department of Mechanical Engineering, Purdue School of Engineering& Technology, IUPUI, Indianapolis, Indiana, United States of America
E-Mail: sharadmulik@gmail.com

ABSTRACT

Flexural mechanism generate motion based on the flexibility of the elements which offers advantages such as friction-free motion, zero backlash and high order of repeatability. Various attempts had been made to design, develop flexural mechanism for precision applications. Present article discuss about the design of such flexural mechanism using double flexural manipulator (DFM). Here, DFM is designed using classical as well as numerical approach to achieve straight line motion. DFM consists flexural manipulator, actuator (VCM i.e. Voice Coil Motor), optical encoder and high speed data acquisition microcontroller dSPACE DS1104 R and D Controller Board. Further, DFM is manufactured and integrated with dSPACE DS1104. Experimental investigation is conducted and experimental parameters are estimated which are having close match with theory as well as numerical FEA analysis. Frequency response system identification is conducted and experimental transfer function is identified and validated with due experimentations. Further, PID control strategy is implemented on DFM and numerous experiments are conducted to test its precision positioning at high speed of the scanning. It is observed a positioning accuracy of less than 2 microns at scanning speed of 2 mm per second (low speed scanning) and precision position of less than 5 microns at 60mm per second (high speed scanning).

Keywords: flexural mechanism, double flexural manipulator, system identification, PID control, precision positioning.

INTRODUCTION

The development in the sector of electronics, material science and advanced manufacturing has put up rising requirements for the ultra-fidelity techniques [1, 2]. Precision mechanisms with prominent cost-performance attributes will be required to raise the economic contingency of arising small-scale technologies. Micro and nano-positioning mechanisms play a very significant part in novel technology [3, 4]. It has utilization in many applications, like micromachining and scanning probe (such as scanning tunneling, atomic force, etc.) microscopy. Distinct XY scanning mechanisms are evolved which differ from screw type to high accuracy recirculation ball mechanisms. XY scanners established until now have numerous disadvantages like constrained scanning range, low performance attributes (coupled degrees of freedom, one stage motion subject to another), accuracy, backlash and many more. Moreover, it is intricate to generate suitable control system to attain required performance. Hence modern era of mechanisms called compliant or flexural mechanisms are developed for high speed precision applications [5, 6 and 7]. Flexures are versatile frameworks that depend on material elasticity for their operation. Motion is initiated because of anamorphosis at molecular level that proceeds in two primary attributes of flexures - uniform motion with accuracy and high speed application. There are many benefits of these types of mechanisms like smooth motion with zero friction, zero backlash, high speed scanning etc. Since the movement of flexures is an averaged consequence of molecular deformations, the adverse effects such as friction, stiction, and backlash are completely eradicated. If the excellence of displacement is certain orders of magnitude better, the quality of constraint may be non-ideal. Instead of this, there are minimum two causes that make flexures most suitable as constraint member in mechanisms where minute displacement is adequate. Flexures are classically lucid in composition and assembly and thus are dominant over air bearings and magnetic bearings. Second, in spite of non-ideal constraining effect of flexure, they are eminently repeatable and remarkably calculable [8, 9].

This work aims at design, development, experimental system identification and PID control implementation on planar double flexural manipulator for precision positioning at high speeds. Distinct operational specifications like stiffness, stress, frequency response and damping factor are investigated.

Paper is structured as follows - section 2 explains design of Double Flexural Manipulator and analytical procedure that comprises of determination of stiffness, parasitic motion and rotation of DFM. Section 3 describes system integration of DFM with PC via dSPACE DS1104 controller. Section 4 presents system identification in which static parameters such as stiffness and dynamic parameters such as ω_n, ξ and frequency response are determined. Section 5 presents experimental implementation of PID control strategy at high speeds of scanning.

VOL. 11, NO. 13, JULY 2016 ISSN 1819-6608
ARPN Journal of Engineering and Applied Sciences

www.arpnjournals.com

DOUBLE FLEXURAL MANIPULATOR

Figure-1 illustrates a double flexural mechanism that comprises of four flexural beams, secondary motion stage, and primary motion stage. Investigation describes that DFM yields appealingly no parasitic error displacement and results in perfect straight line motion.

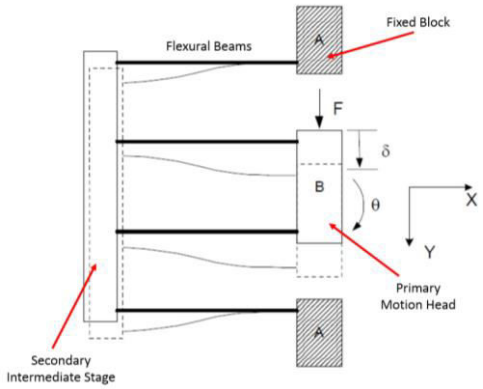


Figure-1. Layout of Double Flexural Mechanism (DFM).

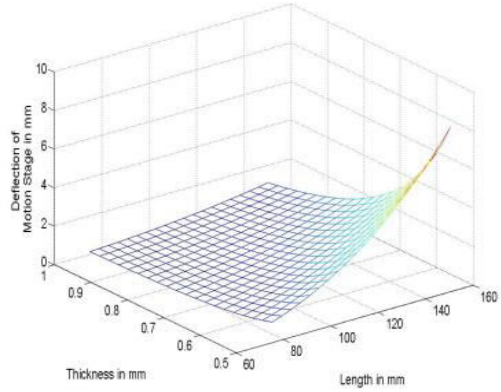
Displacement of a Double Flexural Mechanism is denoted by,

$$\delta = \frac{FL^3}{12EI}$$

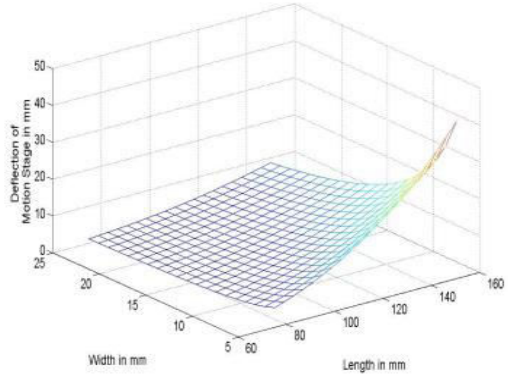
Where

F = applied force in N
L = length of flexural beam in mm
E = modulus of elasticity of flexural beam material in N/mm2
I = mass moment of inertia in mm4 = $wt^3/12$
w = width of beam in mm
t = thickness of beam in mm

Parametric modeling is carried out by determining and constructing important details of the design by using CAD software such as ProE 4.0. Finite element analysis is achieved with the help of ANSYS Parametric Module. Design co-ordinates are established by applying CCD (Central Composite Design) method. Input specifications: Length, Width and Length and Output specifications: Displacement of Motion Stage and Equivalent Stresses. Figure-2 describes a simulation results and denotes dependency of thickness, width and length of flexural beam on displacement of motion stage for DFM. Conforming to current prerequisites of application, a parametric investigation to determine the necessary dimensions for DFM has been carried out. The load in this effort was a direct force of 1 N which yielded a deformation of about 10mm (±5 mm) which is shown in Figure-2. Mechanism dimensions are finalized as below.
It can be seen that the desired displacement of ±10 mm is to be achieved. Length = 100mm, width = 20mm and thickness = 0.5mm gives a 9.968 mm displacements and stresses are not beyond the elastic limit (σ = 268 N/mm2, σ_{ut} = 500 N/mm2). Flexural stage is to be designed and established by employing DFM (Double Flexural Manipulator). This flexural stage has various benefits like it eliminates backlash, friction, high order of repeatability and possesses wide range for scanning etc. Figure-3 denotes experimental arrangement of DFM structure.



(a) Deflection vs length and thickness.



(b) Deflection vs length and width.

Figure-2. Parametric analysis of DFM.

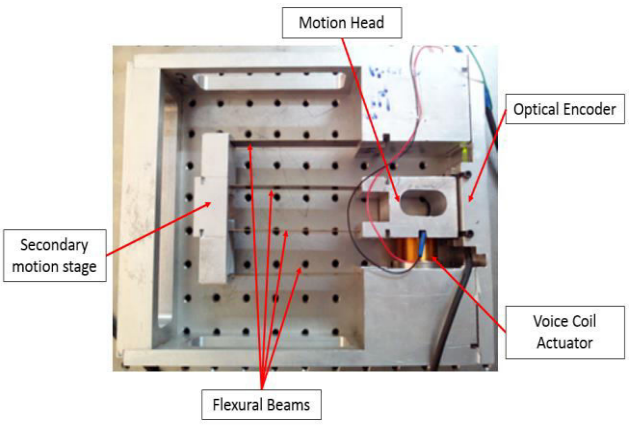


Figure-3. Manufactured single stage DFM manipulat.

The material for flexural beams is selected as beryllium copper and aluminum for their supporting structures. Flexural beams are manufactured by laser cutting method Supporting structures are fabricated by

milling, drilling, slotting and counter boring processes. All the components are assembled together by using alan screws.

MECHATRONIC INTEGRATION AND LAYOUT

System integration is necessary to associate the system to dSPACE DS1104 controller and finally actuate it with Control Desk (computer installed Graphical User Interface software) shown in Figure-4.

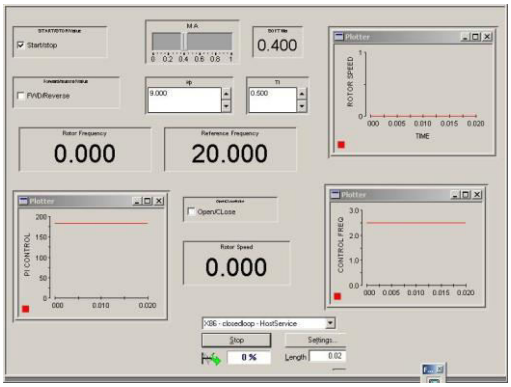


Figure-4. Control desk GUI.

The DAC port of dSPACE 1104 has an amplification ratio of 1:10 which means if we input 1V value in ControlDesk GUI, DAC port gives output of 10V. Operating force is applied to DFM subsequently transforming it to appropriate current-voltage signal. As current output of DAC port has extremely low magnitude consequently it is intensified through the Linear Current Amplifier (LCAM) and it has a current gain of 2A/V. The force causes the motion head to displace as the manipulated current signal is provided to voice coil motor which generates a force of 68.9 N/A. Proper care has been taken to add all these gains while preparing the mathematical model for the system.

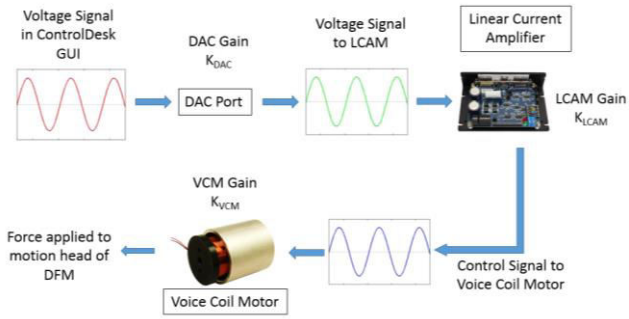


Figure-5. Control logic for application of force on motion head of DFM.

The displacement is measured with the help of Linear Encoder established in the system. The displacement signal through the encoder is forwarded to the dSPACE DS1104 controller. A control logic file is developed in MATLAB Simulink to correlate this encoder output signal with reference signal to determine the error signal that performs as the actuating force. Conclusively, all the outputs from different equipment are presented by using Control Desk that acts like GUI amidst the user and the integrated setup. Therefore, system integration is carried out and the setup is accessible for experimentation procedure. Electronics integration and experimental setup along with system integration is displayed in Figure-6.

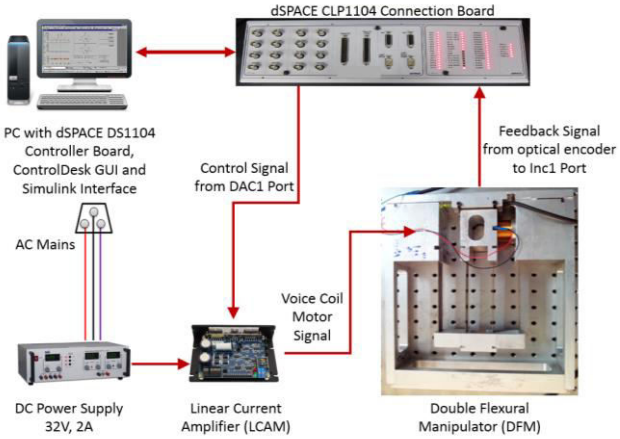


Figure-6. Mechatronic integration of DFM with dSPACE DS1104 R and D controller and experimental setup.

Different mechanical attributes such as stiffness, damping factor, etc. can be calculated precisely with the help of DFM in integration with dSPACE DS1104 controller. Natural frequency of the system is also determined for the setup.

STATIC AND DYNAMIC CHARACTERIZATION

Stiffness estimation: Stiffness i.e. slope of empirical force deflection plot is evaluated. MATLAB Simulink control logic was promoted to operate voice coil motor and displacement of motion head was sensed with the help of established optical encoder. Experimental evaluation of stiffness is carried out both in forward as well as reversed direction of motion. Figure-7 displays actual force - deflection plot of motion head of DFM. Table-1 illustrates discrimination of experimental and theoretical stiffness for DFM.

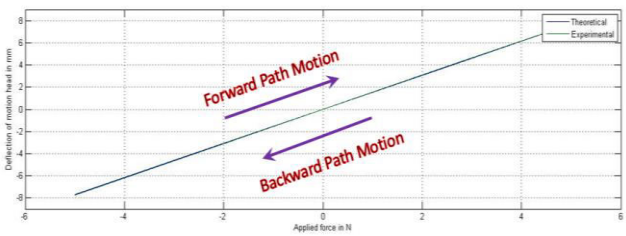


Figure-7. Stiffness attributes of DFM.

Table-1. Comparison of theoretical and experimental stiffness.

Force (N)	Experimental stiffness (N/mm)	Theoretical stiffness (N/mm)	Error (%)
-5	0.64683053	0.65	0.49
-2.5	0.64532783	0.65	0.724
2.5	0.64482848	0.65	0.802
5	0.64842433	0.65	0.243

Damping Factor Determination: Motion head is provided with an initial deflection by an ephemeral force and then the motion head is allowed to oscillate openly until it comes to rest. Experimental outcomes (see Figure-6) are retrieved and drafted as deflection vs time plot. Perception of Logarithmic decrement is applied next for determination of damping factor. Logarithmic decrement is denoted by,

$$\delta = \frac{1}{n}\left[\log\left(\frac{X_0}{X_n}\right)\right]$$

Where n is any integer number of successive positive peaks, X_0 is amplitude of the first peak in consideration and X_n is amplitude of peak at n periods away. Also damping factor is represented by,

$$\xi = \frac{\delta}{\sqrt{4\pi^2 - \delta^2}}$$

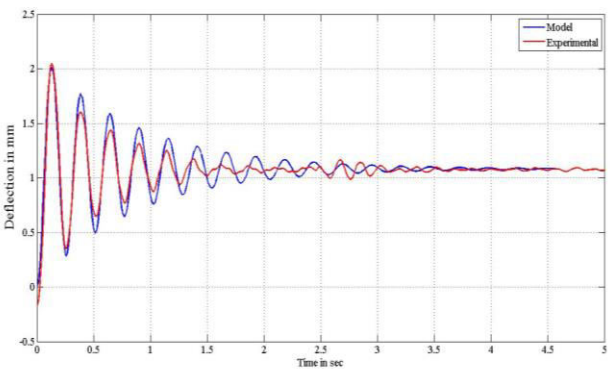


Figure-8. Experimental transient response of DFM mechanism.

From above experimental results (see Figure-8), logarithmic decrement and damping factor in DFM are found to be as below,

$\delta = 0.218308$ and $\xi = 0.034785$

Determination of natural frequency: Purpose of system identification is to establish empirical transfer function of DFM. Transfer function is a connection among input control signal to voice coil motor and real displacement of motion head (for uniform input amplitude and variable input frequency). We obtain frequency response by implementing sinusoidal input voltage and sensing appropriate displacement outputs. MATLAB Simulink control algorithm was established for problem solving time frequency response determination. When amplitude of 0.08 V was provided, a frequency response (1 Hz to 70 Hz) curve for system is brought up to interpret natural frequency and the nature of phase change. It is noted that the peak frequency for the system is 24.51 rad/s. This empirical statistics is further utilized to estimate experimental transfer function of DFM system.

Experimental and model results are correlated and displayed in Figure-9. It is ascertained that there is very good agreement among experimental and model results.

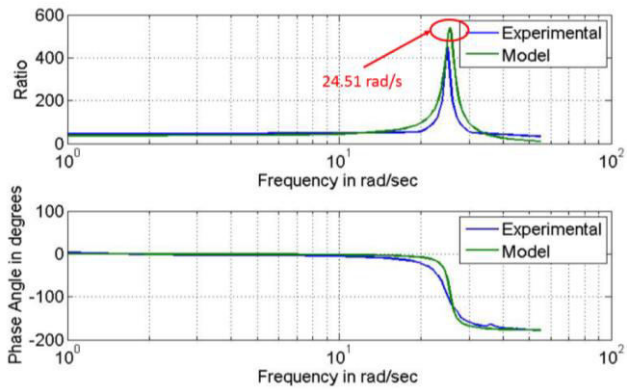


Figure-9. Experimental frequency responses for DFM.

Experimental statistical readings are then optimized with the help of MATLAB function `tf(num,den)` to evaluate empirical model. Experimental model is evaluated as,

$$G(s) = \frac{1}{s^2 + 2.346s + 600.7}$$

PID CONTROL IMPLEMENTATION

A proportional-integral-derivative type of controller (PID controller) is a control loop feedback mechanism extensively applied in industrial control systems (see Figure 10). A PID controller determines an error value by the variance amongst a measured process variable and a desired reference point called as set point. The controller endeavors to reduce the error with adjustment of the operation through implementation of a manipulated variable. The PID logic algorithm consists of three discrete constant specifications, and is consequently at times referred as three-term control: the proportional, the integral and derivative values, denoted by P, I and D.

VOL. 11, NO. 13, JULY 2016 ISSN 1819-6608
ARPN Journal of Engineering and Applied Sciences

www.arpnjournals.com

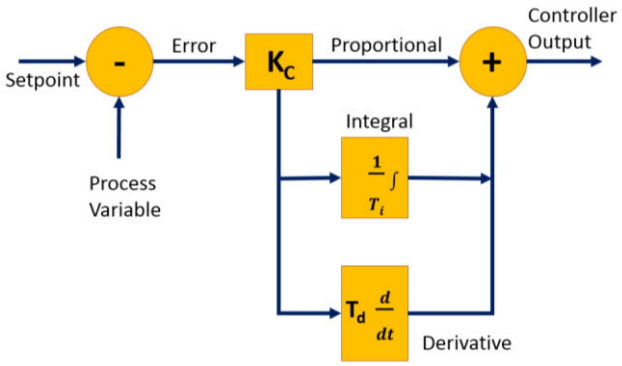
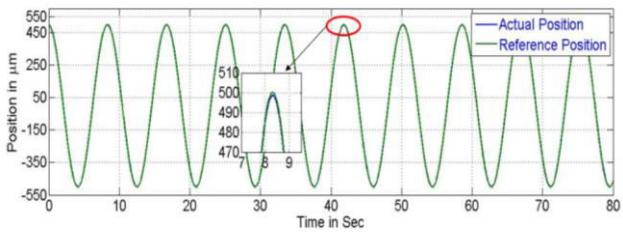
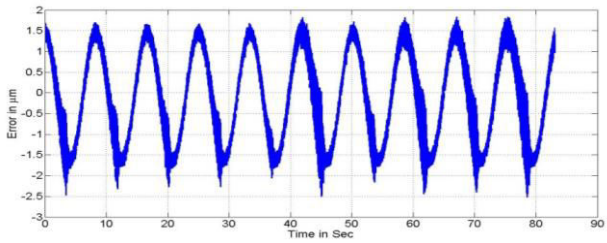


Figure-10. PID control algorithm.

The Ziegler–Nichols tuning approach is a speculative technique of tuning a PID controller. It was established by John G. Ziegler and Nathaniel B. Nichols. It is executed by applying integral (K_i) and derivative (K_d) gains as zero. The proportional gain (K_p) is further raised from zero till it approaches the ultimate gain K_u, at which the output of control loop oscillates with constant amplitude. K_u and the oscillation period T_u are used to set the P, I, and D gains based on type of controller applied.

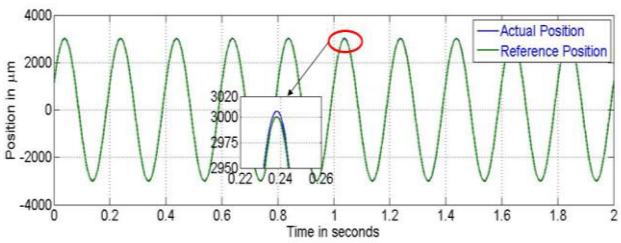


(a) Comparison of actual position and instructed reference position.

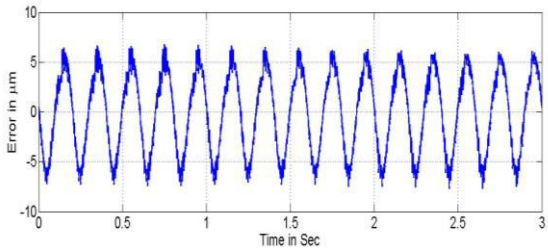


(b) Error in real time positioning

Figure-11. Real time PID implementation on DFM at 500 µm amplitude and 0.75 Hz frequency.



(a) Comparison of actual position and instructed reference position.



(b)Error in Real Time Positioning

Figure-12. Real time PID implementation on DFM at 3000 µm amplitude and 5 Hz frequency.

Figure-11 displays a Real time PID control results for DFM at lesser velocity (0.75 Hz frequency and Speed = 400 µm/s) and less scanning range (Amplitude = 500 µm) of motion head. Displacement accuracy of less than 2 µm is attained. Figure 12 illustrates a Real time PID control results for DFM at comparatively higher speed (5 Hz frequency and Speed = 2500 µm/s) and more scanning range (Amplitude = 3000 µm) of motion head. Displacement accuracy lower than 8 µm is achieved.

CONCLUSIONS

Double Flexural Mechanism (DFM) is developed and incorporated to dSPACE DS11004 R&D controller. The characterization of DFM is carried out on the basis of two distinct fields - (1) Static analysis is executed for finding out force deflection attributes for the range of entire displacement and (2) Dynamic analysis is accomplished using frequency response which gives characteristics of system with different frequency inputs. This frequency response is further utilized to accomplish experimental modelling of DFM. Empirical model of with the help of frequency response is found out using constrained minimization approach. Evaluated empirical model is further used for PID control algorithm implementation. PID control attributes (i.e. proportional gain, integral gain and derivative gain) are adjusted using Ziegler Nichols approach. Empirical model at the outset was examined and accuracy of lower than 1 microns was attained. PID algorithm was applied using dSPACE DS1104 R and D controller and Control Desk GUI environment. Actual positioning accuracy of lower than 2 microns is accomplished. PID control exhibited good disturbance rejection and the least amount of error when being used for tracking at different frequencies. LQR control could only be used for regulation purpose whereas LQI control attained good disturbance rejection during regulation. To improve tracking, we would use feed-forward control with LQI and with other control strategies and implement it on the setup.

ACKNOWLEDGEMENT

This work was financially supported by a grant from the Department of Science and Technology, HRD Ministry, Govt. of India. The authors thank to management of Sinhgad Academy of Engineering

Kondhwa, Pune for providing technical and managerial support to execute research work.

REFERENCES

[1] S. Awatar. 2004. Synthesis and analysis of parallel kinematic xy flexure mechanism, PhD Thesis, Massachusetts Institute of Technology, Cambridge, MA, see also URL http://web.mit.edu/~shorya/www/PHD/Thesis/Introduction.pdf.

[2] Shuichi Dejima, Wei Gao, Hiroki Shimizu, Satoshi Kiyono, Yoshiyuki Tomita. 2005. Precision positioning of a five degree-of-freedom planar motion stage, Mechatronics. 15: 969-987.

[3] Suhas Deshmukh, P.S. Gandhi. 2009. Opto-mechanical Scanning Systems for Microstereolithography (MSL): Analysis and Experimental Verification. Journal of Material Processing Technology. 209(3): 1275-1285.

[4] S. Awtar, A. H. Slocum. 2005. Design of parallel kinematic XY flexural mechanism, Proceedings of IDETC/CIE 2005 ASME 2005 International Design Engineering Technical Conferences and Computers and Information in Engineering Conference, Long Beach, California, USA.

[5] S. P. Deshmukh, Anand Tandel, A. R. Deshpande, K. R. Jagtap. 2014. Modeling, Analysis and PID Controller Implementation on Double Wishbone Suspension Using SimMechanics and Simulink, Elsevier Procedia Engineering. 97: 1274-1281

[6] Lobontiu N. 2003. Compliant Mechanisms: Design of Flexure Hinges, CRC Press.

[7] Suhas Deshmukh, P.S. Gandhi. 2007. A Novel Optomechatronic Focused Laser Spot Submicron Scanning System for Microstereolithography. Poster presentation at Banglore Nano conference 2007, International Conference on Nanotechnology, Banglore, INDIA.

[8] Qing Yao, J. Dong, P.M. Ferreira. 2007. Design, analysis, fabrication and testing of a parallel-kinematic micropositioning XY stage. International Journal of Machine Tools and Manufacture. 47: 946-961.

[9] Hwa Soo Kim, Young Man Cho. 2009. Design and modeling of a novel 3-DOF precision micro-stage, Mechatronics. 19: 598-608.

[10] Yangmin Li, Qingsong Xu. 2009. Modeling and performance evalutrion of a flexure-based XY parallel micromanipulator. Mechanism and machine Theory. 44: 2127-2152.

[11] Suhas P. Deshmukh, Hrishikesh Zambare, Kavidas Mate, Mahesh S. Shewale, Zeba Khan. 2015. System identification and PID implementation on Double Flexural Manipulator. International Conference on Nascent Technologies in the Engineering Field (ICNTE-2015), IEEE Xplore.

[12] Suhas Deshmukh, Rachel Patil, Y.P. Reddy, Kavidas Mate. 2015. FEA Analysis and Experimental Investigation of Building Blocks for Flexural Mechanism. 2015 International Conference on Nascent Technologies in the Engineering Field (ICNTE-2015), IEEE Xplore.

[13] Y. Tian, B. Shirinzadeh, D. Zhang, X. Liu, D. Chetwynd. 2009. Design and forward kinematics of the compliant micro-manipulator with leaver mechanisms, Precision Engineering. 33: 466-475.

[14] Ranier Clement, J. L. Huang, Z. H. Sun, J. Z. Wang, W. J. Zhang. 2013. Motion and stress analysis of direct-driven compliant mechanisms with general-purpose finite element software, The International Journal of Advanced Manufacturing Technology. 65(9-12): 1409-1421.

[15] Sergio Lescano, Micky Rakotondrabe, Nicolas Andreff. 2012. Micromechanisms for Laser Phonosurgery: A Review of Actuators and Compliants Parts. IEEE International Conference on Biomedical Robotics and Biomechatronics, BIOROB'12, Rome, Italy.

[16] L.F. Campanile, M Rose, E.J. Breitbach. Synthesis of flexible mechanisms for airfoil shape control: a modal procedure.

[17] Eric S. Buice, David Otten, Raymond H. Yang, Stuart T. Smith, Robert J. Hocken, David L Trumper. 2009. Design evaluation of a single-axis precision controlled positioning stage. Precision Engineering. 33: 418-424.

[18] Yung-Tien Liu, Bo-Jheng Li. 2010. Precision positioning device using the combined piezo-VCM actuator with frictional constraint. Precision Engineering. 34: 534-545.

[19] Won-jong Kim, Shobhit Verma, Huzefa Shakir. 2007. Design and precision construction of novel magnetic-levitation-based multi-axis nanoscale positioning systems. Precision Engineering. 31: 337-350.

[20] Chih-Liang Chu, Sheng-Hao Fan. 2006. A novel long-travel piezoelectric-driven linear nanopositioning stage. Precision Engineering. 30: 85-95.

[21] Deyuan Zhang, Chienliu Chang, Takahito Ono and Masayoshi Esashi. 2003. A piezodriven XY-microstage for multiprobe nanorecording Sensors and Actuators. A 108: 230-233.

[22] Byung-Ju Yi, Goo Bong Chung, Heung Yeol Na, Whee Kuk Kim, Il Hong Suh. 2003. Design and experiment of a 3-DOF Parallel Micromechanism Utilizing Flexure Hinges. IEEE Transactions on Robotics and Automation. 19(4).

[23] Chien-Hung Liu, Wen-Yuh Jywe, Yeau-Ren Jeng, Tung-Hui Hsu, Yi-tsung Li. 2010. Design and control of a long-traveling nano-positioning stage. Precision Engineering. 34: 497-506.