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Design and Optimization for a Compliant Gripper with Force Regulation Mechanism

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Abstract**—**This paper presents a design and optimization for a compliant gripper. The gripper is constructed based on the concept of compliant mechanism with flexure hinge. A passive force regulation mechanism is presented to control the grasping force a micro-sized object instead of using a sensor force. The force regulation mechanism is designed using the planar springs. The gripper is expected to obtain a large range of displacement to handle various sized objects. First of all, the statics and dynamics of the gripper are investigated by using the finite element analysis in ANSYS software. And then, the design parameters of the gripper are optimized via Taguchi method. An orthogonal array L_9 is used to establish an experimental matrix. Subsequently, the signal to noise ratio is analyzed to find the optimal solution. Finally, the response surface methodology is employed to model the relationship between the design parameters and the output displacement of the gripper. The design of experiment method is then used to analyze the sensitivity so as to determine the effect of each parameter on the displacement. The results showed that the compliant gripper can move with a large displacement of 213.51 μm and the force regulation mechanism is expected to be used for high precision positioning systems.

Keywords**—**Flexure hinge, compliant mechanism, compliant gripper, force regulation mechanism, Taguchi method, response surface methodology, design of experiment.

I. INTRODUCTION

MICROGRIPPER is now widely used for handling fragile objects. It consists of a variety of various applications such as living cells, micromechanical pars, and so on. Nowadays, microgrippers have received considerable attention from both academia and industry. A mechanical gripper is designed with two or more hands, called jaws, so as to pick up and place a micro-sized object. Practical grippers are required a large gripping range in order to grasp different heavy and sized objects. In the past, the multi-degrees-of-freedom grippers were proposed to handle different object shapes [1]-[2]. Some were designed with underactuated mechanisms to increase the gripping capacity [3]-[4]. In addition, some grippers were developed by using the concept of compliant mechanism. In order to achieve a high precision, the compliant grippers were monolithically fabricated using some various techniques such as 3D printer, CNC, or wire electrical discharged machining process [5]-[8].

There were some technologies to control the grasping force to reduce the contact force on fragile objects [9], [10]. It required an extra closed or open loop force control algorithm. This was relatively expensive and costly. Moreover, because the hands are compliant, the position of the object relative to the hands was uncertain if the force sensors were not controlled. Hence, a force regulation mechanism is presented in this study to adjust the grasping force. The force regulation mechanism allows a constant force. It was a passive mechanism that was used for the vibration isolation [11]. This mechanism utilized the planar spring as flexure hinge to adjust the force.

In order to measure the displacement of the gripper's hands, laser displacement sensors or capacitive sensors were used; however, they were costly. To decrease the cost, strain gauges are attached on the surfaces of the flexure hinges to form a displacement sensor.

The aim of this paper is to develop a compliant gripper. The passive force regulation mechanism is presented to accurately adjust the grasping force for the compliant gripper. The gripper is expected to manipulate fragile objects. The Taguchi method is applied to optimize the design parameters of the force regulation mechanism. In this study, the displacement of the gripper is considered as an objective function. And then the response surface methodology is utilized to model the relationship between the design parameters and the displacement. Finally, the design of experiment is employed to analyze the sensitivity of the design parameters on the displacement.

II. DESCRIPTION OF COMPLIANT GRIPPER

Fig. 1 depicts an assemble model of the developed compliant gripper. The gripper was located on a vibration optical table by fixed holes to suppress any undesired outside disturbances. The gripper was driven by two piezoelectric actuators (PZT 1 and PZT 2, as seen in yellow color), separately. These two PZTs were actuated simultaneously. When the PZTs were activated in the y-axis, the two hands of gripper were opened and closed to grasp a micro-sized object. The two preload screws were utilized to exert a force to the structure of gripper, called as pre-stress, which was aimed to make a good contact between the PZTs and the structure. The input displacement of the PZTs are frequently measured via the laser displacement sensor, but this sensor is very expensive. Therefore, the strain gauges were attached on the flexure hinges with a suitable glue. The strain gauge-based flexure hinge was formulated into a displacement

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sensor, as shown in Fig. 2.

When the object was handed, the grasping force was almost controlled by a sensor force. Because the commercial sensor forces are relatively expensive, the force regulation mechanism (FRM) was presented in this study. The force of this mechanism was adjusted by using the two regulation screws. The regulation screws were aimed to provide an appropriate grasping force for the hands. Besides, the regulation mechanism could increase or decrease the range of motion of the hands so that the gripper could handle various micro-sized objects. To control a grasping force, the regulation screws would exert a suitable force to the planar springs. The planar springs were connected with the structure of the gripper through the flexure hinges. The planar springs were also made of flexure hinges in order to become more compliant. On the flexure hinges, the strain gauges were glued to make a force sensor, as indicated in Fig. 3. The size of the compliant gripper was described in Table I. All flexure hinges and planar springs had the same in-of plane thickness *t* of 0.5 mm the compliant gripper had the out-of-plane width *w* of 6 mm. The gripper was made of polypropylene. This material has mechanical properties as: Density, 950 kg/m^3, Young's modulus, 1100 MPa, Poisson's ratio, 0.42 and yield strength, 25 MPa. The gripper was desired to achieve a large range of motion over 50 µm. To obtain this specification, an optimization process was necessarily. Before the optimization process, the statics and dynamics were conducted in later section to describe the mechanical behaviors of the complaint gripper.

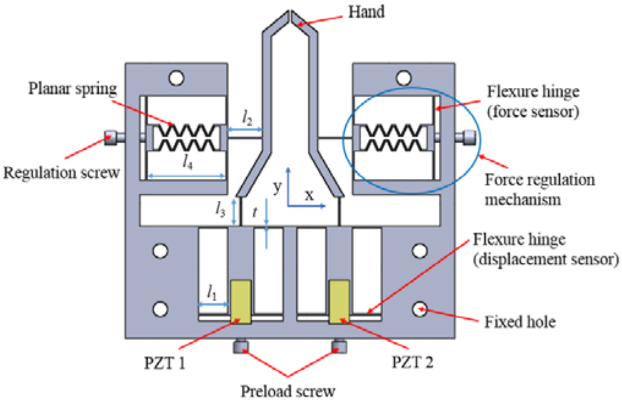

Fig. 1 Model of compliant gripper

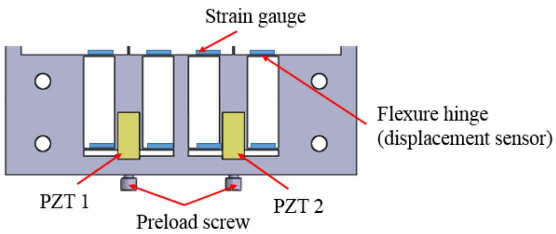

Fig. 2 Model of PZT and displacement sensor

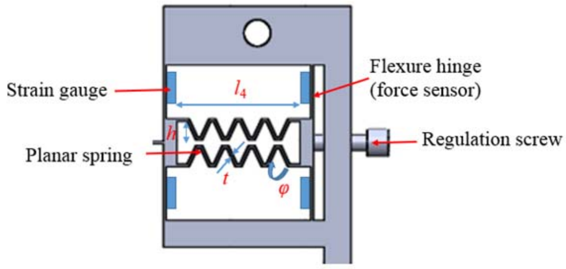

Fig. 3 Model of FRM and force sensor

TABLE I
INITIAL SIZE OF THE COMPLIANT GRIPPER

Symbol	Quantity	Value
l_1	Length of flexure hinge 1	10 mm
l_2	Length of flexure hinge 2	12 mm
l_3	Length of flexure hinge 3	10 mm
l_4	Length of planar spring	23 mm
t	Thickness of flexure hinge	0.5 mm
h	High of planar spring	4 mm
w	Width of gripper	6 mm
φ	Angle of planar spring	50 degrees
	Total: 113 mm x 108 mm x 6 mm	

III. STATIC AND DYNAMIC ANALYSIS

The finite element method-based computation analysis has been widely employed for design testing. In this section, the finite element analysis (FEA) in ANSYS 16 software was used to investigate the static and dynamic behaviors of the gripper. To perform this analysis, the deign parameters of the gripper were assumed as in Table I.

The automatically meshing method was utilized, and then each flexure hinge was refined to achieve analysis accuracy. A coarse meshing method was adopted in regions with smaller deformation, as given in Fig. 4. The boundary conditions were given as: The gripper was fixed at the holes and a force of 12 N was applied to both PZT1 and PZT2.

Under applying of the force from the PZT1 and PZT2 in the y-axis without the FRM, the gripper's hand was moved along the x-axis to grasp the object; however, the hand was also displaced along the z-axis. The z-axis movement was undesired motion or called as parasitic motion error. As seen in Table II, the results indicated that the parasitic errors are relatively small because the ratio of parasitic motion p_e to the displacement d_x is lower than 2%.

TABLE II
THE DISPLACEMENT AND PARASITIC MOTION ERROR WITHOUT FRM

Fore (N)	Displacement d_x (µm)	Parasitic motion (µm)	Raito p_e/d_x
4	20.38	0.35	1.71%
6	30.57	0.53	1.73%
8	40.76	0.71	1.74%
10	50.95	0.89	1.74%
12	61.14	1.07	1.75%

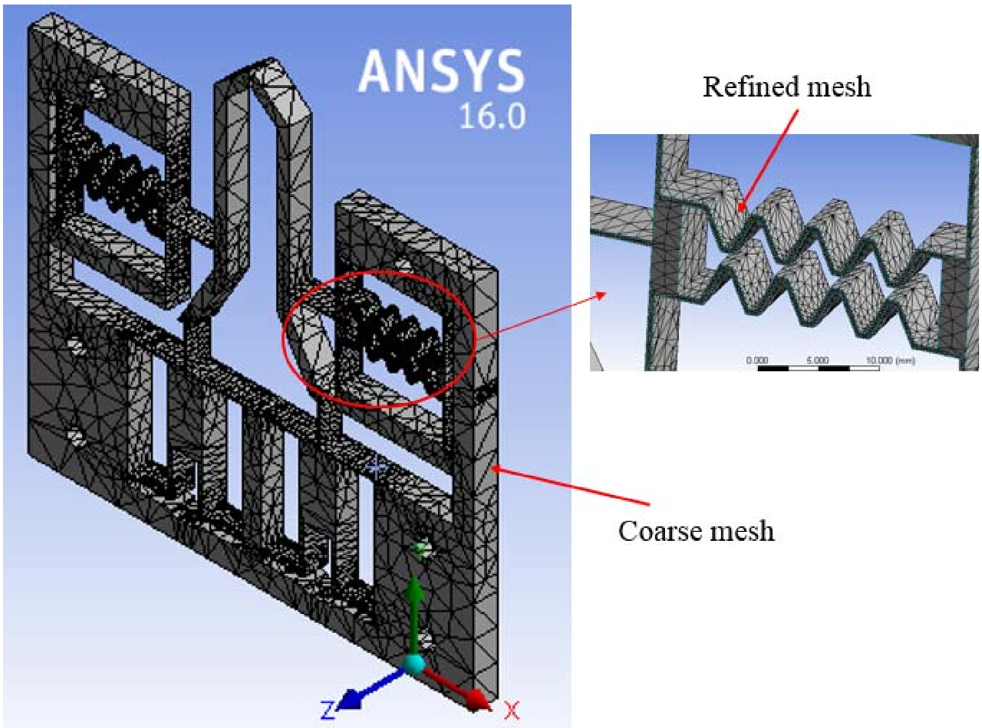

Fig. 4 Meshed model of the compliant gripper

When the PZTs applied a force of 12 N to the gripper; at the same time the regulation exerted a force the FRM, the displacement of the gripper would be changed as well as the parasitic error. As shown in Table III, the results revealed that the displacement of the gripper's hand with the FRM is larger than that of the gripper's hand without the FRM about 43 times. the reason is because the FRM increased the applied force to the gripper. So, it could conclude that it helps the gripper increase the range of motion at the open position and decrease the range of motion at close position to grasp the object. However, the parasitic error without the FRM was lower than that with the FRM.

TABLE III
THE DISPLACEMENT AND PARASITIC MOTION ERROR WITH FRM

Fore (N)	d_x (μm) without FRM	dx (μm) with FRM	Parasitic motion (μm) without FRM	Parasitic motion (μm) with FRM
1	5.09	218.87	0.08	0.71
1.2	6.11	251.37	0.10	0.78
1.6	8.15	328.37	0.14	0.92
2	10.19	450.37	0.17	1.06
3	15.28	598.86	0.26	1.42

The deformation of the gripper was depicted as in Fig. 5 and the stress distribution was indicated in Fig. 6. Through these figures, it could be seen that the maximum stress is at the end of flexure hinges.

The dynamic behavior of the gripper was investigated subsequently to understand the first resonance frequencies, called first natural frequencies and its modal shapes. This analysis was carried out by FEA. As shown in Fig. 7, the first mode had a frequency of 95.388 Hz and two hands rotated about the x-axis. The second mode had a frequency of 95.778 Hz and two hands also rotated about the x-axis. The third mode had a frequency of 96.843 Hz and the right hand rotated about the z-axis. The fourth mode had a frequency of 97.068 Hz and the left hand rotated about the z-axis. The fifth mode had a frequency of 204. 45 Hz and two hands rotated about the z-axis. The sixth mode had a frequency of 206.61 Hz and two hands also rotated about the z-axis.

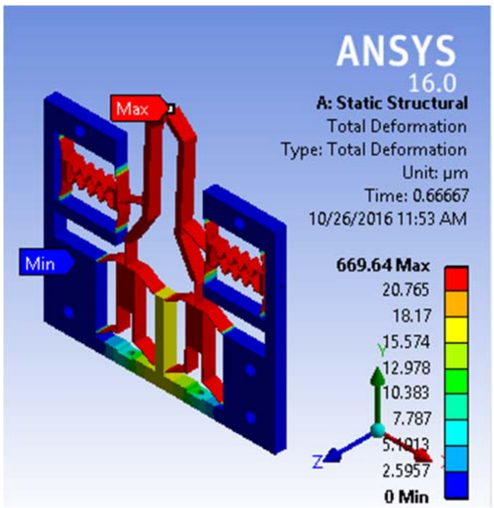

Fig. 5 Deformation of the compliant gripper

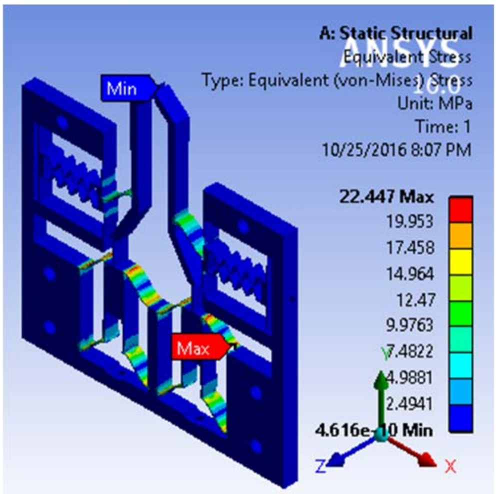

Fig. 6 Stress distribution of the compliant gripper

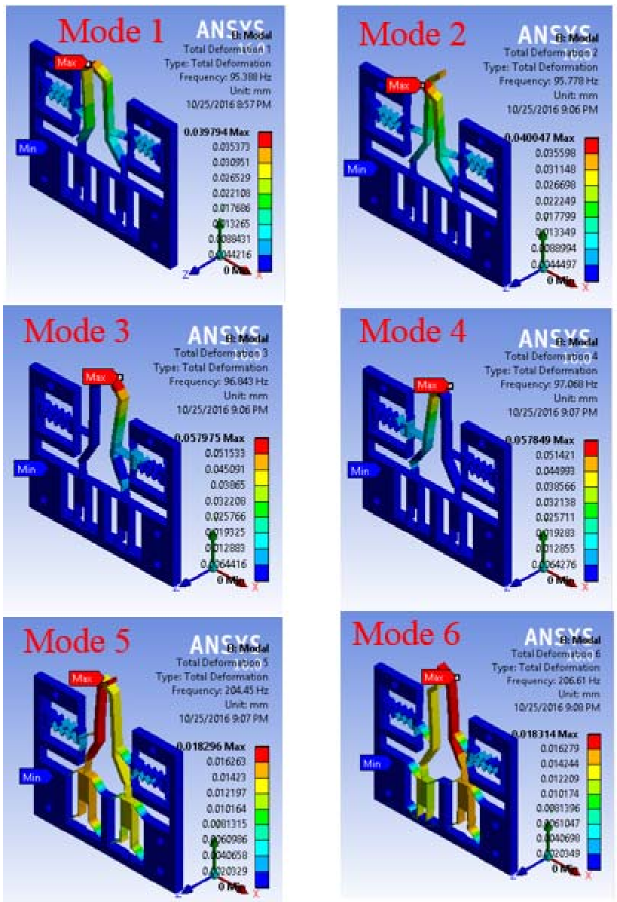

Fig. 7 Mode shapes of the compliant gripper

IV. Optimization

A. Taguchi Method

The Taguchi Method (TM) is widely used for robust parameter designs [12]-[15]. To create a robust design, the TM suggests a three-stage process to achieve the desired product quality: (1) Concept design; (2) parameter design; and (3) tolerance design. The concept design is the process of choosing the technology and design which will reduce production costs and result in high-quality products. The parameter design refers to the selection process of the control factors and the determination of the optimal levels for each factor. The purpose of the parameter design is to find the most suitable factor levels so as to make a robust system that is less sensitive to variations in uncontrollable noise factors. There are two types of factors that affect a product's functional characteristic: control factors and noise factors. The control factors can easily be controlled, whereas the noise factors are either too difficult or impossible or expensive to control.

The tolerance design process occurs after the parameter design and is used to reduce unwanted variations and improve quality. A better tolerance increases the product's cost or process because higher quality materials, components or machinery are needed.

Of the three design considerations, the TM primarily focuses on the parameter design because the use of this method can improve the quality and decrease costs while not requiring better materials, parts or production. The aim of the parameter design is to achieve minimum variations so the end product is consistently close to the desired target. The TM deals with the statistical and sensitivity analysis required in determining the optimum parameter settings and thereby achieving the robust quality response. The response to the parameter settings is considered as a measure that is not only the mean of the quality performance, but also its variance. The mean and variance are then integrated into a single performance measure, known as the signal-to-noise (*S/N*) ratio. The Taguchi robust parameter design categories depend on the desired performance response, which is described as follows:

Smaller the better: To make the system response as low as possible, the target value, quality variable, is zero. The *S/N* of this type is defined as:

$$\eta = -10\log\left(\frac{1}{q}\sum_{i=1}^{q} y_i^2\right), \tag{1}$$

Larger the better: To make the system response as high as possible, the target value, quality variable, is infinite. The *S/N* of this type is defined as:

$$\eta = -10\log\left(\frac{1}{q}\sum_{i=1}^{q}\frac{1}{y_i^2}\right), \tag{2}$$

Nominal the best: To reduce variability around a target, a specific target value is given. The *S/N* of this type is defined as:

$$\eta = -10\log\left(\frac{1}{q}\sum_{i=1}^{q}\left(y_i - m\right)^2\right), \tag{3}$$

where y denotes the quality response; subscript i the experiment number; q the number of replicates of experiment 'i'; and m is the target value of the response.

Regardless of the quality response category, a higher *S/N* ratio corresponds to a better quality characteristic. The TM uses a special orthogonal array to construct an experimental layout, and the mean of each level of factors is analyzed to determine the effects of the parameters and so find the optimum parameter settings based on the statistical results. In this study, the larger the better was used to find the optimal displacement for the gripper. The larger the better was used to find the optimal displacement for the gripper.

The optimization problem was stated as:

$$\text{Maximize } y(h,t,\varphi), \tag{4}$$

Subject to constraints as:

$$\begin{cases} 3\text{mm} \le h \le 4\text{mm} \\ 0.3\text{mm} \le t \le 0.7\text{mm}, \\ 40\text{mm} \le \varphi \le 60\text{mm} \end{cases} \tag{5}$$

where y is the displacement of gripper. h, t, and φ are the design variables.

TABLE IV
DESIGN PARAMETERS AND THEIR LEVELS

Symbol	Code	Level 1	Level 2	Level 3
h (mm)	A	3	3.5	4
t (mm)	B	0.5	0.5	07
φ (degree)	C	40	50	60

TABLE V
EXPERIMENTAL RESULTS AND *S/N* RATIOS

Trial. No	h (mm)	t (mm)	φ (degree)	d_x (µm)	*S/N* ratio (dB)
1	3	0.3	40	55.66	34.91
2	3	0.5	50	65.86	36.72
3	3	0.7	60	66.55	36.46
4	3.5	0.5	40	59.16	35.44
5	3.5	0.7	50	62.40	35.90
6	3.5	0.3	60	70.31	36.94
7	4	0.7	40	62.64	35.93
8	4	0.3	50	213.51	46.58
9	4	0.5	60	63.57	36.06

TABLE VI
RESPONSE TABLE FOR THE MEAN OF *S/N* RATIOS

Symbol	Level 1	Level 2	Level 3
h (mm)	35.92	36.10	39.53
t (mm)	39.48	35.96	36.10
φ (degree)	35.43	39.62	36.49

In this study, the design variables of FRM were taken into account in the optimization such as the high of the planar spring, the angle of the spring, and thickness of the spring. Three control parameters were divided into three levels, as shown in Table III. An orthogonal array L_9 of TM was applied to formed nine experiments. And then, the FEA results and *S/N* ratios of the displacement of gripper's hand were collected and computed, as indicated in Table IV. Finally, the response table for the mean *S/N* ratio at each level was calculated, as seen in Table V. Based on Table V, the response graph for the optimization process was drawn. The results revealed that the optimal parameters are at A3B1C2 at the 8th experiment in Table IV, i.e., the optimal value of parameter h was at 4 mm, the optimal value of the parameter t was at 0.3 mm and the angle φ was at 50 degrees. These optimal solutions will be verified in the future work by fabricating a prototype.

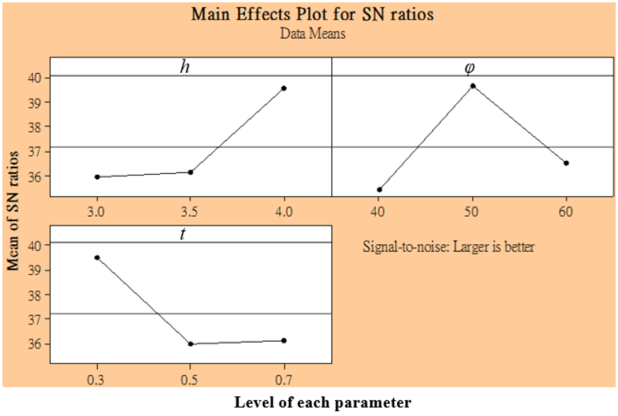

Fig. 8 Response graph of the displacement

B. Modeling and Sensitivity Analysis

The aim of sensitivity analysis is to investigate the effect of the design parameters on the displacement of gripper. Response surface methodology (RSM) is used to describe this relationship. This relationship is investigated by a quadratic polynomial regression model. A quadratic polynomial regression model is the most popular due to its flexibility to an approximate nonlinear response, and the mathematical model of the quadratic regression model is described below because first-order models often give lack-of-fit [16].

$$y = \beta_0 + \sum_{u=1}^{N} \beta_u x_u + \sum_{u=1}^{N} \beta_{uu} x_u^2 + \sum_u \sum_v \beta_{uv} x_u x_v + \varepsilon, \tag{6}$$

where y is the predicted output displacement, x is the design variable, N is the number of design variables, β_v (v = 0, 1, 2,…, N) are regression coefficients, β_{uu} and β_{uv} are quadratic coefficients. These regression coefficients can be determined by least square method. ε is the model error which is neglected because the computational optimization are carried out at constant temperature and ignored geometric errors. Based on Table IV, the relationship between design parameters and the displacement was modeled as:

$$\begin{aligned} y &= 9.53 - 621.615h + 51.322\varphi - 764.517t \\ &\quad + 96.023h^2 - 0.509\varphi^2 + 641.279t^2, \end{aligned} \tag{7}$$

To describe the effect of the design parameters on the displacement of the compliant gripper, the design of experiment (DOE) was used in Minitab software. As seen in

Fig. 9, the parameters *h*, *φ*, and *t* had the significant effects on the displacement.

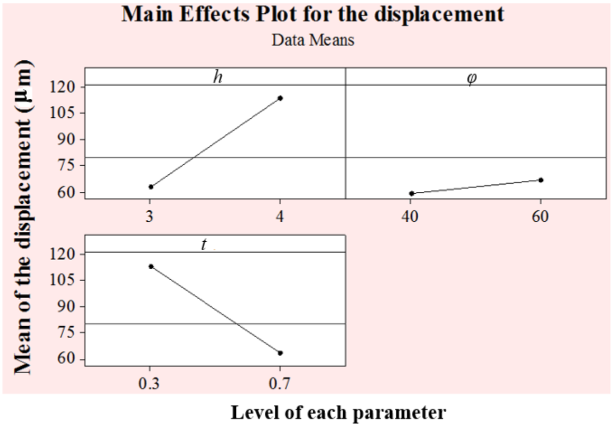

Fig. 9 Sensitivity analysis of parameters to the displacement

V. Conclusion

A compliant gripper with the FRM has been designed and optimized in this study. The mechanical structure of the gripper was constructed via using the compliant mechanism and flexure hinge. The FRM was designed using the planar spring. The gripper was driven via two PZTs simultaneously. The FRM was adjusted via the regulation screw while the preload screw was utilized to make a good contact between PZTs and the gripper.

The FEA was used to investigate the displacement and the parasitic motion error of the gripper without the FRM and with the FRM. It revealed that the displacement of gripper's hand with the FRM is higher than that of one without the FRM. The FRM is expected to adjust the control force for related grippers and high precision positioning systems. In addition, the strain gauges were tended to attach on the flexure hinges to measure the control force of the FRM as well as the displacement of the gripper. These will verify in the future work.

And then, the design parameters of the FRM were optimized via using the TM to find the best displacement of the gripper. The result revealed that the compliant gripper can provide a large displacement of 213.51 μm. The RSM was used to formulate the equation between the design parameters of the FRM and the output displacement of the gripper. The DOE method was used to analyze the sensitivity of the design parameters on the displacement. The results showed that three design variables of the springs have the significant effects to the displacement.

In future work, a prototype will be fabricated and its behaviors are tested to verify the FEA results.

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