Out of Plane Transfer of Motion Using Orthogonally Coupled Micro Gears

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Abstract

The possibility of utilizing two orthogonal gears as a means of transferring out of plane motion in the micro scale is explored in this paper. A generalized method of constructing such a mechanism is also developed. These design procedures were created with the purpose of providing an entry point to the development process and not a method of optimization. A variety of design parameters and their effects on the quality of motion have also been investigated. The quality of the motion was rated using the accuracy of the mechanism when used as a point-to-point motion provider.

1. Introduction

In the MEMS community, a significant amount of work has been done [1,2] in the investigation of various mechanisms that transfer in plane motion.

This project focuses on the examination of the behavior of a gear set, in which the rotational axes of the gear and pinion have been placed perpendicular to each other. This mechanism has been designed specifically to allow motion to be directly transferred out the plane parallel to the Silicon substrate. The pinion is placed on the horizontal plane to ease motion generation using any of the many currently available sources. The gear is mounted on a freestanding base plate. The goal of this project is to discover the change in the movement quality, as some of the gear-set's pertinent design parameters are altered.

With reproducibility kept well in mind, the MCNC Multi User MEMS Process (MUMPS) has been chosen as the means to manufacture all of the proposed test structures.

2. Design and Layout

Figure 1 shows the particular design that has been pursued in this project. The design was constructed using the idea of a set of bevel gears. Due to the current constraints of the MEMS fabrication methods, which prevented the creation of true 3-D features, a pair of spur gears was used. In order to meet the design's need of three structural polysilicon layers, POLY 0 that is normally used as a ground layer has been utilized as the first structural layer. The feasibility of this particular fabrication technique is not covered in this paper. [3].

The particular structures that are placed on POLY 0 are the base of all the hinges, the base of the horizontal gear's pin, and the base plate for the vertical gear structure. Aside from the vertical base plate these structures were designed to have relatively large footprints in order to guarantee the anchorage of the entire test structure. The next structural layer, POLY 1, contains the two spur gears, the gear's pins, additional enforcement layers of the previous bases, a pair of cantilever springs, and two security flaps. POLY 2, finally, completes the test structure by forming the end caps of the two gear pins, and bridges for all of four hinges.

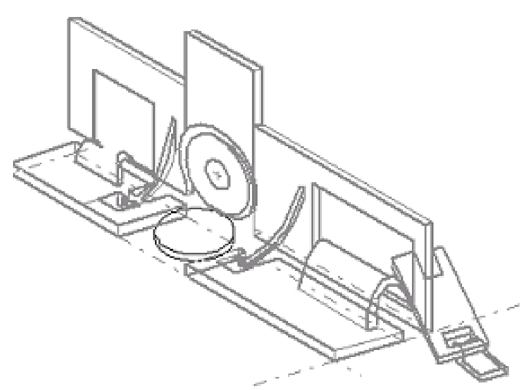

Figure 1: General Test Structure Design

Figure 2 shows the layout of the test structure. As it can be seen from the layout, etch holes were introduced to the vertical base plate in order to insure the completion of the Nitrite etching process. As a safety measure, the area between these holes was design to be about 10% of the area of the smallest anchor. It is recommended for the concentrated HF (49%) to be used as the etching chemical agent. This technique will prevent the polysilicon structure from getting etched away while the agent consumes all of the sacrificial Nitrite-layer [3].

Frictional forces have also been considered during the development of the test structure. After the final release-etch of the structure, the horizontal gear, also known as the pinion, will fall towards the substrate. In order to reduce the area of contact between the rotating gear and the stationary surface, a support structure was introduced utilizing the POLY 0 layer. This structure again can be observed in figure 2.

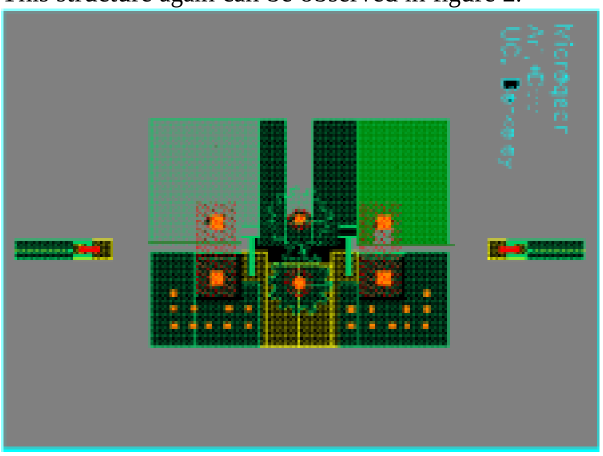

Figure 2. *Layout of the Test Structure Components on the Silicon Wafer*

To hold the vertical base plate and the gear fixed in an orthogonal direction with respect to the substrate two security flaps are used. Also to further increase the stability of this freestanding structure, two flexible cantilever beams were introduced as torsion springs. This torque application will help prevent any external forces, which can be introduced to the structure due to vibration, from unlatching the security flaps. Similar application of this type of torsional springs can be observed in [4].

Another important issue that has been tackled during the development of this structure is the fact that during assembly both the gear and the pinion will fall towards the substrate. In order to accommodate for this movement, the gear set has been designed so the final position of the set will reflect the correct position for proper meshing. The pinion will approximately travel 1.5 micron to its steady running position and the gear will move about 0.5 micron. The presence of one-micron net displacement was corrected by placing the gear and pinion at a distance of 1 micron closer to each other than their intended final positions.

Due to the relatively thin POLY 0 layer, it was discovered the need to reinforce the vertical base plate, which supports the gear. This reinforcement will help reduce the effects of any warping from preventing proper meshing of the gear set.

3. Theory

A generalized approach to designing mechanisms similar to the previous test structure has also been developed during the course of this project. The following steps [5] will allow designers to start with preliminary design parameters that can be used in the development of the first prototype of the mechanism. Although these steps can be used as guidelines during the development, they should not be used to optimize the design.

Table 1 lists all of the variables used in the equations used throughout the design procedure.

Table 1. *Variables Definitions Used in Design Procedure*

θ_V	gear's angular position from vertical [rad/s]
θ_H	pinion's angular position from vertical [rad/s]
d	distance from gear's or pinion's center to the contact line [m]
t	thickness of the gear tooth [m]
$\theta_{H,in}$	angle of entry point in relation to the vertical line [rad]
r_{ch}	pinion's radial distance to the contact point [m]
θ_{BTW}	angle between two consecutive front surfaces or back surfaces of the pinion [rad]
J_V	length of gear's projection on the contact line [m]
P_C	position of the contact point along the contact line [m]

The first step that needs to be taken is to determine the sizes of both the pinion and the gear, and their respective locations. These parameters are more than likely will be the results of the geometrical constraints introduced by other parts of the overall system. Then the amount of overlap that is desired between the members of the gear set needs to be determined. Following this step, the height of the individual gear-tooth should be decided. The physical meaning of tooth height as used in this paper can be observed in figure 4.

Once the previous design parameters have been determined, the first calculation step can be taken. This step calculates the range of angular position where a particular tooth of the pinion is in contact with its mate. This contact range can be found by drawing two lines from the center of the pinion to the contact line of the gear set with a magnitude of the radius of the pinion's base plus the height of the tooth. The angle formed by these two lines regulates the range of contact. In order to have complete control of the position of the gear in a particular direction, at least one tooth of the pinion has to be located within this range at all times. Figure 3 displays the previously described contact range and how it is defined in this design procedure.

Therefore, by determining the contact range, the minimum number of teeth that a particular pinion needs to have has been found. As long as the angular distance between the contact-faces of two consecutive teeth is smaller than the contact, than the minimum control requirement can be satisfied. The following set of equations summarizes the previous calculation step.

$$\theta_{in,h} = \cos^{-1}\left(\frac{d}{r_{ch}}\right) \tag{1}$$

$$\#teeth \geq \frac{360^{\circ}}{\theta_{contact}} = \frac{360^{\circ}}{2\theta_{in,h}} \tag{2}$$

Once the number of teeth has been determined, the thickness of these teeth should then be set. The procedure for finding this thickness is started by finding the relationship between the angular position of the pinion and the gear. Figure 4 shows that the position of the contact point between the two meshing teeth varies as the teeth traverse across the contact range. It shows the need to separate the motion across the contact range into two separate portions. The motion of the contact point can be represented by the following set of equations:

$$r_V \sin(\theta_V) = d \tan(\theta_h) \tag{3}$$

$$r_h \sin(\theta_h) = d \tan(\theta_V) \tag{4}$$

Now, the contact point between the two teeth has been defined. Using the previous set of equations, the angular position of the gear at any point along the contact range can be determined from the angular position of pinion. The next step is to model the behavior of the mate-tooth as both teeth move along the contact region. The amount of area along the contact region that is occupied by this tooth can be calculated through the estimation of the projection of this tooth as seen from the pinion.

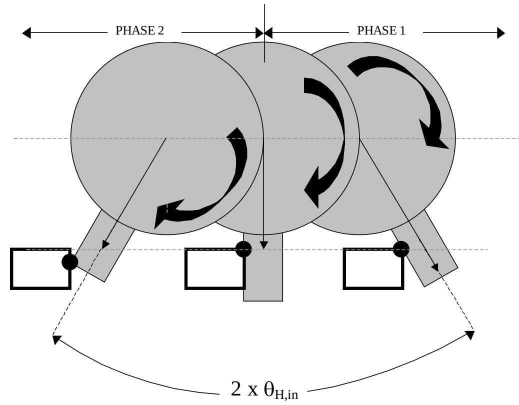

Figure 3. *Position of Contact Point Across Contact Range*

Figure 4 shows the projections of both the driving and mating teeth seen from the gear and the pinion respectively. Equation 5 shows how the end point of this projection area can be traced with respect to the contact point. The angular position of the gear can be obtained from equation 3.

$$J_{V,C} = d \tan\theta_h - \left(\frac{t}{\cos\theta_V}\right) \tag{5}$$

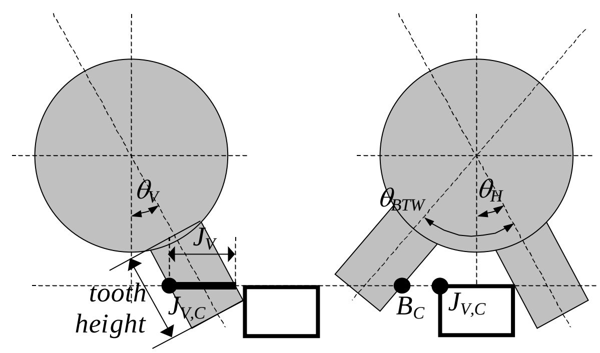

Figure 4. *Teeth Projection Seen From Both the Gear and the Pinion*

Since the position of the mating tooth is now known throughout the contact region, the thickness of the teeth of the pinion can be designed so no overlapping occurs in this region. If overlapping does occur, the tooth of the gear should be redesign until no overlap is present. Equation 6 provides a summary of this relationship. As long as the value generated using this equation is greater than zero, than no overlap occurs. Equation 6 can also be used to find the amount of slack that is present between the meshing gears. Utilizing this equation will allow the determination of the accuracy level of this structure when used in a point-to-point motion.

$$\overline{B_C J_{VC}} = d\tan(\theta_{btw} - \theta_h) - t\cos(\theta_{btw} - \theta_h) + d\tan(\theta_h) - \frac{t}{\cos(\theta_V)} \quad (6)$$

4. Test Structures

The different test structures that are proposed in this project have the same generic structure as the previously described mechanism. The differences between them are the number of teeth, thickness of the teeth, and the amount of overlapping. The purpose of this experiment is to observe the change in the gear behavior as the previous design parameters are changed. The experiment is also going to be used to verify the validity of the previously proposed design procedures.

5. Expected Results

Figure 6 shows what is expected to happen as the number of teeth on the pinion is increased. This particular test structure has the gear diameter of 50 micron, gear thickness of 5 micron, and the gear and pinion are 48 micron apart from each other. Equation 1 predicted that a minimum of 16 gear teeth is required on the pinion in order to have any control at all times. As the number of the teeth increases, the distance between the teeth of the header, pinion teeth directly in front of the currently pushing teeth, and the mating teeth decreases.

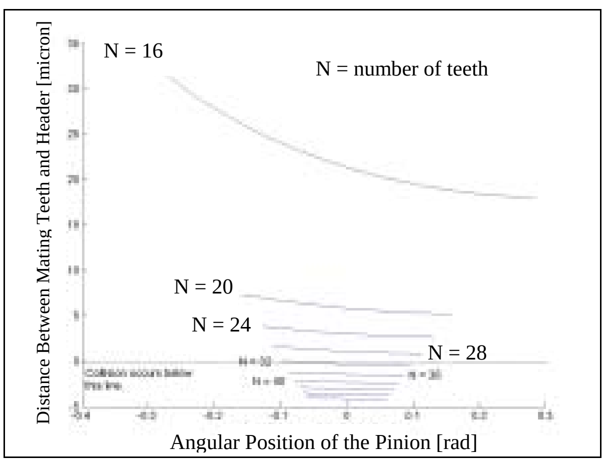

Figure 5. *Simulation Results of the Addition of the Number of Gear Teeth*

This method also allows the measurement of the mechanism's accuracy. The lesser the distance between the two mating teeth and the header, the less likely wobbling is going to occur. Therefore, the accuracy of point-to-point is going to be increased.

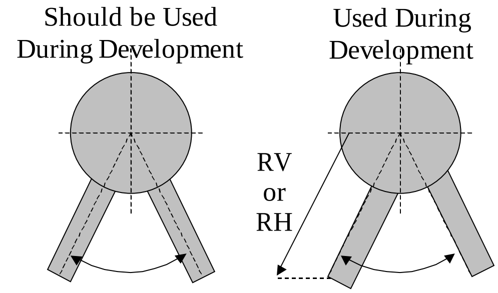

Figure 6. *Estimation Used in the Design Process*

The previous design procedures can be followed during the preliminary state of the design process. Assumptions, such as the one depicted in figure 5, prevented these procedures to be used in the more advance state of the design process. Here, it is clear that during the development process it is assumed that there is a radial line that goes from the center of the gear to the end of the teeth. It was also assumed that this line is perpendicular to the width of the teeth.

6. Conclusion

Utilizing the previous design procedures will allow MEMS designer to start constructing orthogonal gear structures in a very efficient manner.

The test structures that are proposed can provide new information about the importance of specific design parameters in determining the accuracy of a pair of micro gear structures. This new insight will allow the creation of new generation structures that can move out of the substrate's plane effectively.

Although preliminary equations show promising results, further inquiries into this particular area of MEMS design is highly encouraged.

7. References

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