The Vertical-Mill Stage

Design of a Vertical Movement Stage for Surface Machining

by

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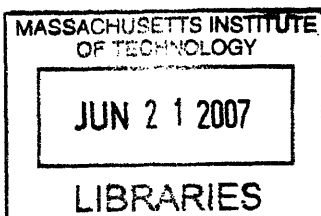
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ABSTRACT

To allow the widespread use of three dimensional patterns on walls, the manufacturing cost must be reduced. The goal of this project is to design a machine for vertical axis movement which can be measured and controlled. The key issues addressed in this thesis, are the motion past obstacles in the extrusions, a lightweight mechanism that is stiff with minimal horizontal deflections, can adjust to different heights and can support its own weight. The design favored consisted of a chain drive designed to travel 3 meters in 10 seconds at a weight of 30 Kilograms. The machine is kept in place by five points of contact, including a fixed sprocket and four sliding V-grooved rollers. The machine was tested for stability, stiffness and its ability to traverse the entire chain without problems.

Thesis Advisor: Alexander Slocum

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Acknowledgments

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1. Introduction

<u>Motivation</u>

Architects continuously seek to revolutionize the appearances of buildings. Using MIT as an example of evolving architecture, the prevalence of plain walls does not match the innovative appearance of the facades on the Stata Center or the McGovern Institute. Although current manufacturing technologies allow architects to create three-dimensional patterns in the internal walls of a building, the painstaking process limits these walls to a few number of luxury buildings. The limiting factors for the widespread use of adaptive walls are the manufacturing costs and the elaborate process required to transfer the finalized product from the manufacturing site to the building where it is being installed. The focus of this project is to further the development of a machine that will allow walls to be modified into three-dimensional shapes at the construction site, eliminating the cost and risk associated with transporting the panels. This project will focus in continuing the creation process of alternative walls spanning a three-dimensional space, increasing their penetration into buildings with smaller budgets

Tiffany Yang conducted the feasibility study for the overall design of the machine and provided a base to design the remaining components. Yang's design consists of a nested extrusion assembly driven by a chain drive on the top and bottom of the machine to control its horizontal displacement[1].

2. Design Requirements and Selection

The key design challenges for the vertical stage of the Wall Mill:

1. Ability to support the weight of the Router
2. Stiffness and Lightweight
3. Ability to drive over a 1/8" step
4. Mechanism can adjust to different heights

The mechanism was modeled as a cart in free ascent to determine the power requirements with estimates of the mass and desired ascent time, as shown in equation (1). The attributes used in this model are shown in Table 1.

$$P = \frac{4Mx^2}{t^3} \qquad (1)$$

Cart mass (M)	30 kilograms
Distance to Travel (x)	3 meters
Ascent time (t)	10 seconds
Power Required (P)	1.08 Watts

Table 1: Power Budget Estimation for Cart system

The maximum weight of the router and stage were estimated at 10 Kg, the common height of a building story was estimated to be 3 meters, and the desired ascent time was initially estimated at 10 seconds, these parameters are initial settings for operation and can be adjusted to improve the maneuverability of the wall mill.

The motion in the vertical axis of the wall mill requires motion that can be easily controlled, measured and defined at all points of the machine's track. Otherwise it could lead to inconsistencies during operation. Another functional requirement of the machine is to drive through a 1/8" step without problems, since the interface between the nested extrusions must be traversed during operation. Bidirectional stiffness is another crucial characteristic, since the stage must be able to cut material with the support of the vertical stage alone and it should have minimal displacements along the vertical axis.

Three different actuation mechanisms were considered, and the best match for the application was selected using the selection criteria aforementioned. The three mechanisms explored were a pulley system, a friction drive and a chain drive. The selection process can be summarized in Table 2.

Mechanism	Pulley System	Friction Drive	Chain Drive
Controllability	0	0	0
Bidirectional Stiffness	+	0	+
Traction	+	0	+
Position Sensing	+	0	+
Interface with other components	-	0	+

Table 2: Selection of Actuation Mechanism

Other concerns which are not addressed in this project are the location mechanism, or the correction of the offset that occurs when the stage moves out of the larger extrusion and into the inner extrusion.

All the components necessary for the actuation of the vertical axis must be supported by a stiff frame to reduce beam bending. A shape with a higher moment of inertia can reduce the bending caused by external loads, and a hollow shape can minimize the weight of this frame. The ideal shape for the frame materials will be a combination of these two attributes. Possibilities for this sub-unit are I-beams or a rectangular extrusion.

To determine the points of contact that the frame should make with the nested extrusions, a deterministic design was favored. The chosen configuration uses 3 axis of contact to restrict the motion of the stage to a single axis. Figure 1 demonstrates the placement of the contact points that restrict the motion of the stage to the vertical axis.
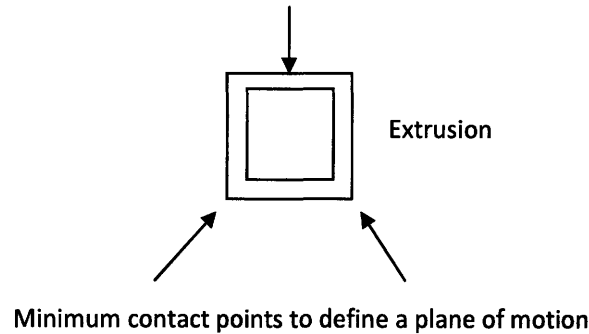
Minimum contact points to define a plane of motion

Figure 1: Interaction Between the Vertical Motion Stage and Nested Extrusions.

The nested extrusions presented a challenge to the design of the stage, the interface between the outer and inner extrusion forms a 1/8" step in the path of the vertical stage. After determining the desired points of contact, it is important to know that every element in contact with the extrusion remains in contact at all times. Pre-loading the elements that contact the extrusion is the most straightforward and feasible solution. The alignment of the components is crucial; an incorrect alignment could cause instability during operation and the machine could precipitate to the ground.

3. Detailed Design
3.1 Actuator System

The actuation of the vertical stage will use a chain drive as described in the previous section. The spacing between the holes in the extrusions will determine the pitch of the chain links used with snap-on fasteners to attach with the extending extrusion. The stage will also support the router stage, which will move the router into and out of the z-plane. The estimated weight of the vertical system is approximately 10 kg, so approximating the weight of the stage as 30 Kilograms builds into the design a safety factor of 3.

3.1.1 Chain and Sprocket Selection

The design chosen for the vertical actuation, requires chain to be easily added or removed from the supporting extrusions, ½" holes span the length of the supporting extrusions at a center to center distance of 1 inch. The existing holes can be connected to a single pitch chain adding link (McMaster: 7321K6) with a butterfly self ejecting fastener (Speedway Motors: 91007125). The chain that matches the distance between the holes is the ANSI chain number 40 (1/2" pitch); this chain will determine the sprocket used to transmit the actuation. The chain must support a load of 300 Newtons on a contact area of 0.008 square meters which equals 37 Kilopascals of stress, which is much lower than the yield stress of Steel.

3.1.2 Motor and Gearbox Analysis

To determine the optimal drive combination, equation (2) combines the inertia of the rotor J_m and gearbox J_{trans}, the radius of the sprocket R_{wheel} used in the chain drive and the mass of the stage M_{stage}.

$$n_{trans} = \frac{R_{wheel}}{\sqrt{\dfrac{J_m + J_{trans}}{M_{stage}}}} \tag{2}$$

Using these equation and a list of available motors and gearboxes from Maxon, the specifications for desired motor and gearbox were calculated, the ideal parameters are shown in table 3.

R_{wheel}	0.0643 m	n_{trans}	5.58
J_m	1330 g cm^2	Available Gear Ratio	5.2
J_{trans}	0.0124 g cm^2	Γ_{motor}	2.88 Nm
M_{stage}	30 Kg	$\Gamma_{gearbox}$	14.98 Nm
		F_{drive}	232.91 N

Table 3: Optimal Transmission Parameters for actuation system.

The specifications of the machine recommend that the travel speed of the stage be at least 0.6 meters/second, to allow traveling 3 meters in under 10 seconds starting from rest. It is important that the force produced by the optimal transmission is enough to fulfill the kinematic requirements. The minimum Torque required for the system to travel at 0.6 m/s and the corresponding angular velocity are given by equations 3.1 and 3.2.

$$\tau = m \cdot a \cdot r \tag{3.1}$$

$$\omega = \frac{v}{r} \tag{3.2}$$

For the transmission described in Table 3, the required torque and angular velocity outputs are 0.116 Nm and 560 rpm, which can be achieved with the selected motor and gearbox.

3.1.3 Bench Level Experiments

One of the foreseen issues in the design of the current mechanism was the problem that the transition between the larger and the smaller extrusion and could cause. To verify this would not represent a problem, the sprocket was driven over the transition manually to test whether the transition was a large resistance to motion. Since the size of the step was only 2% of the sprocket size, the sprocket transferred smoothly without any adjustments. Another key observation was that the alignment of the chain is crucial to a smooth uninterrupted motion, which will register better on the encoder attached to the motor.

3.2 Weight Bearing System

The vertical actuation stage must be in contact with the nested extrusions at all times, so the mechanism does not separate and fall straight to the ground. There are five points of contact, including the sprocket, and four rollers that can slide to adjust for both extrusions. The rollers used are McMaster's V-Grooved-Guided Track Rollers, which press the corner of the extrusions. Each roller can move independently to accommodate for the transition between the extrusions. To avoid interferences with the chain, four corner "V-rollers" were used across from the driving sprocket as shown in Figure 2. To make sure that the sprocket applied even loads on the rollers and to counterbalance torques, the sprocket was placed between the two sets of rollers. Several options were explored to spring load the V-rollers onto the extrusions.
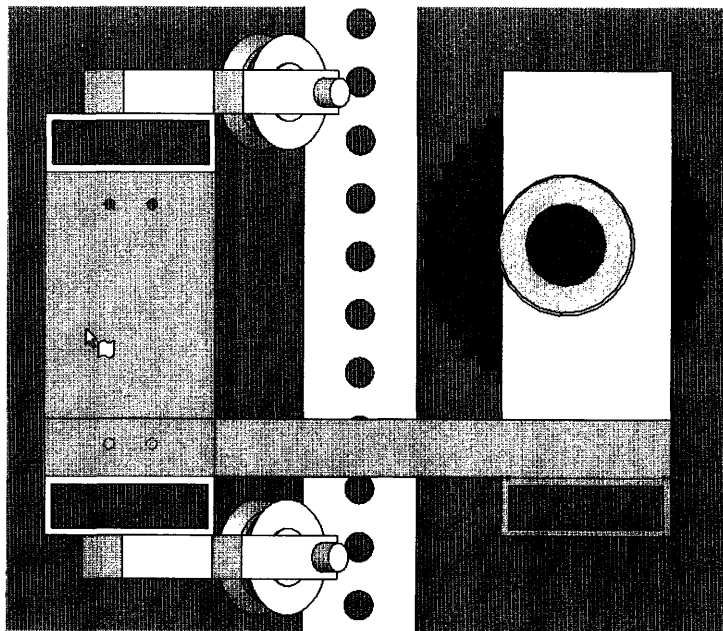

Figure 2: V-roller Bearing Positioning.

3.2.1   Flexure design

The first option considered was to mount the v-rollers on flexural bearings. The shaft of the v-rollers is ½" in diameter, so all of the flexures manufactured were based on a ¾" thickness. The minimum required thickness made the flexure design more difficult as the stiffness of ¾" Aluminum reduces its flexibility. A Matlab script was written to approximate each blade of the flexure as a cantilevered beam using equations 4.1 through 4.4, the program is shown in Appendix 1.

$$I = \frac{1}{12} bh^3 \qquad \text{(4.1)}$$

$$F = \frac{2\sigma_y I}{hL} \qquad \text{(4.2)}$$

$$\delta = \frac{FL^3}{3EI} \qquad \text{(4.3)}$$

$$n = \frac{D}{\delta} \qquad \text{(4.4)}$$

Using the script mentioned above with the parameters shown in Table 4, it was calculated that a load of 3.7 N was needed to displace the 13 blades of the flexure .09".

Blade thickness	h	0.05"
Flexure thickness	b	0.75"
Blade length	L	1.5"
Aluminum Yield Stress	σ_y	55×10^6 Pa (based on Solidworks value)
Young's Modulus	E	70 GPa

Table 4: Parameters used to model flexure blades as cantilevered beams.

3.2.1.1 Finite Element Analysis

After approximating the flexure's range of motion, a Finite Element Model of the component was drafted in SolidWorks and analyzed using CosmosXpress. The force F calculated with equation (4.2) was used as a load in the flexure's axis of motion to calculate the actual displacement of the flexure.

The maximum displacement of this flexure was half of the required displacement (see figure 3), so it would have to be positioned halfway between the two operating points and tensioned. The purpose of using a flexure was to eliminate the need of applying an external force on the bearing of the v-roller; however, this flexure still needed an external force thus reducing the advantages of using such a complex mechanism.
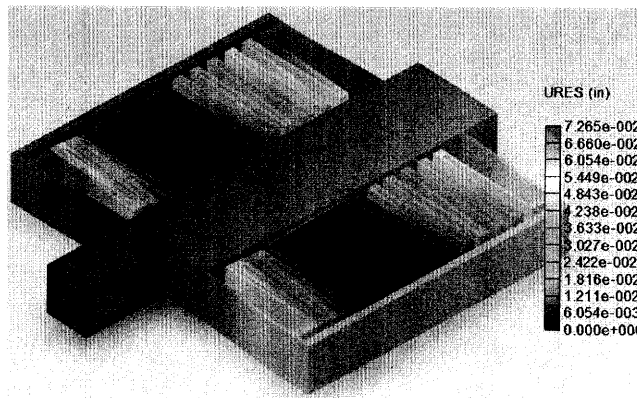
Figure 3: Displacement of Initial Flexure Design.

3.2.2 Spring-loaded Bearing

A spring loaded design to actuate the V-track rollers was also considered. This design used an aluminum block to support the rollers and fasten them to the main body frame as shown in Figure 4. Shoulder bolts were used to constrain motion to one dimension during operation and a nylon sheet was used between the sliding aluminum pieces to improve their mobility.
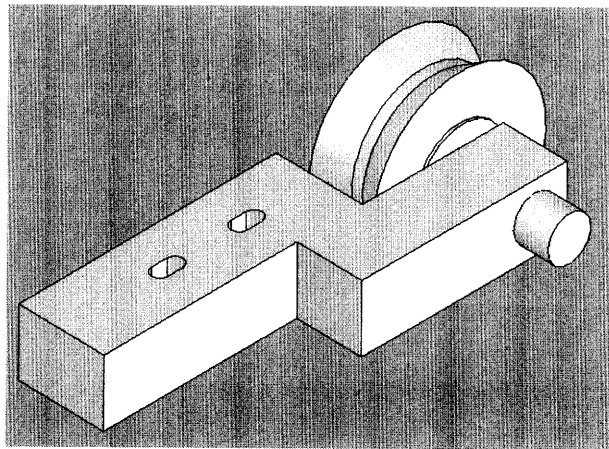

Figure 4: Bearing support for V-track rollers.

To ensure that the stage would be continuously in contact with the nested extrusions, it was decided that the four v-rollers should move to accommodate the change in size of the extrusions. Although moving four components is more complex than moving one, it was necessary to actuate the rollers, because actuating the sprocket itself would cause a poor bearing contact between the two rollers and the extrusions as shown in Figure 5.
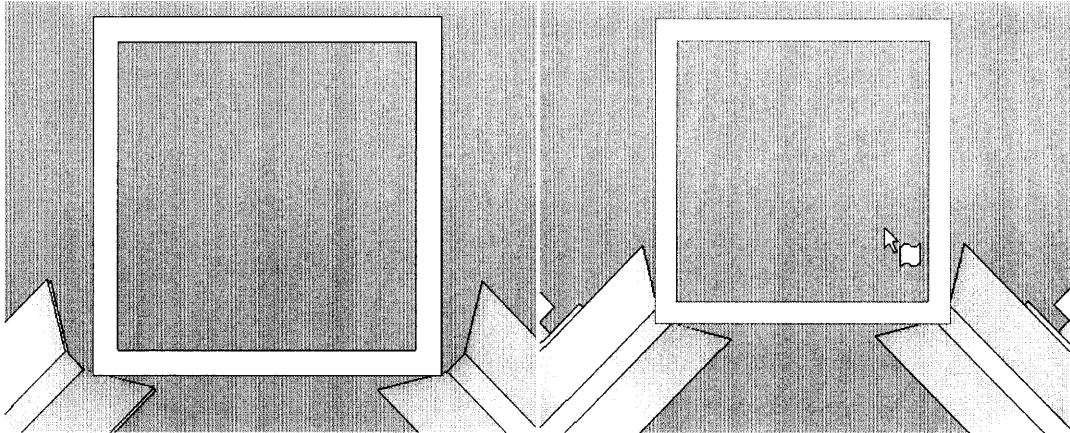
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Figure 5: Contact formed between fixed rollers and inner extrusion (left) compared with contact between sliding rollers and inner extrusion (right).

The frictional force formed between the rollers and the extrusion helped maintain the vertical stage at a fixed height. Estimating a mass *(M)* of 30 Kg, and using a coefficient of static friction *(μ)* between steel and steel of .7, it can be estimated with equation 5 that each of the four springs needs to exert a contact force of 107.1 Newtons (or 24.05 lbs) between the V-rollers and the extrusion.

$$F_{roller} = \frac{Mg}{4\mu}$$ (5)

To compress the bearing and track roller assembly onto the extrusion, compression springs will be used. The selected compression springs have a rate of 18.6 lbs / inch *(k)*. To support the weight of the stage they need to be compressed at least 1.3" *(x)* according to Hooke's Law (equation 6).

$$F_{spring} = kx$$ (6)

The springs will be mounted around a 0.5" shaft press-fit into a bearing angle to prevent buckling and misalignment as shown in Figure 6. It is important that the springs used have ground ends, otherwise they would not contact the bearings uniformly.
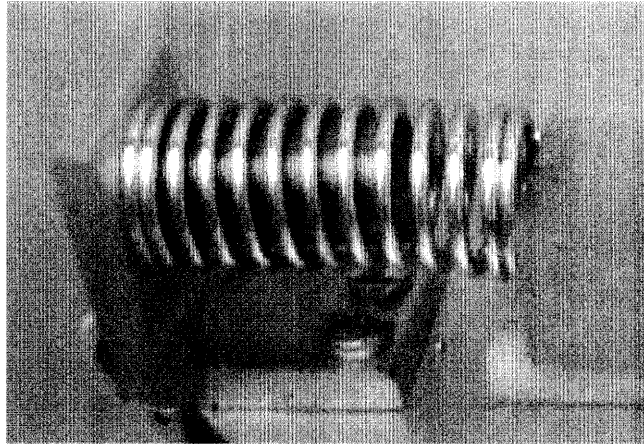
Figure 6: Bearing for mounting ground spring.

4. Results

4.1 <u>Integrated Design</u>

Integrating all the components yields (less fasteners and spring mechanisms) the design shown in Figure 7. The design is mostly composed of extrusions to facilitate manufacture and assembly, as well as providing mounting surfaces for other components. SolidWorks estimates the weight of the assembly as 6.3 lbs, which is well within the initial estimates of 30 Kg including the weight of the router and motor.
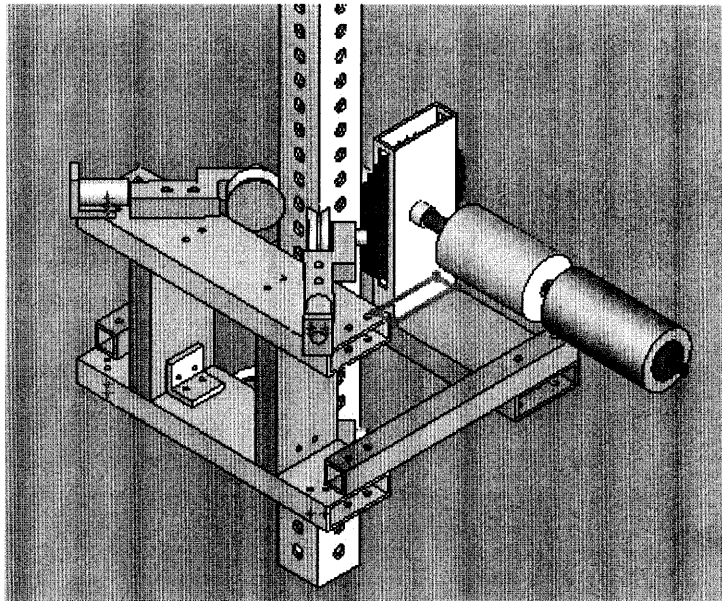

Figure 7: Final SolidWorks Assembly.

4.2 Experiment

The controllers for the machine have not been developed, so testing has been limited to sliding the machine manually to test its features. The features with the greatest importance for this test are

i)   The stability of the stage
ii)   Ability to move on both of the nested extrusions
iii)   Stiffness of the frame

The procedure followed to test the stage consisted of:

1. Mounting the machine onto an extrusion.
2. Checking the sideways torque that the machine could withstand without wobbling (Figure 8).
3. Rolling the machine on both of the nested extrusions.
4. Pushing the machine from different directions to make sure that the shape of the body does not change.
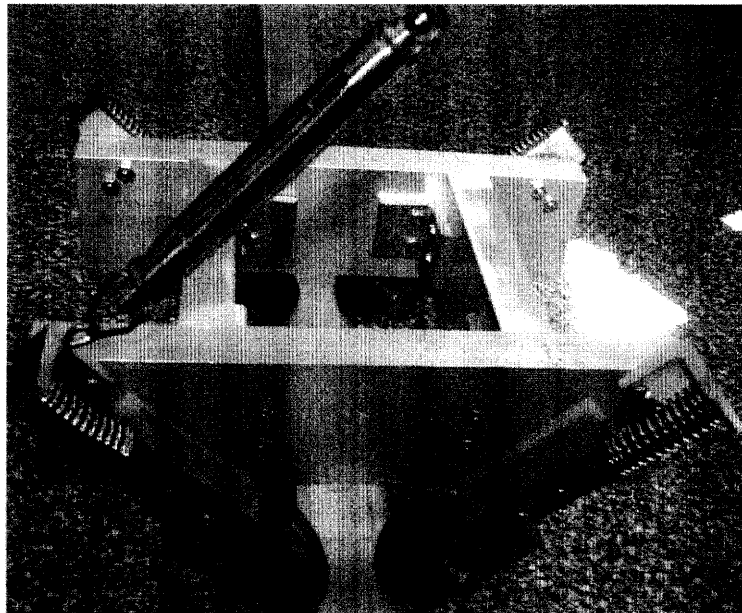

Figure 8: Torque applied perpendicular to the roller contact.

The machine resistant torsion forces very well, so twisting loads applied at opposite ends did not change its shape. The springs were able to adjust to both extrusion sizes (1.75" and 2"), however, adjusting the springs for the different sizes required a force greater than 100 Newtons. Further tests are needed to determine if during operation the machine would adjust to both sizes on its own. Using a force gauge to apply a load as shown in Figure 8 tests the resistance of the machine to a torque perpendicular to its axis of motion. The maximum torque applied without any

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deflection was 7 Newton-meters. Adding the chain and sprocket on the other side of the extrusion should increase this value and make the machine more stable.

5. Further Work

To make the vertical axis mill a useful addition to Tiffany's Wall Mill, the mechanisms must be integrated with a stage that will allow the wall mill to move in the z-axis as well. A linear stage could be used to control the motion of the cutting bit. The three axes of motion must be controlled by a single controller to allow the machine to move in each axis independently using one programmable interface.

The position of the stage can be determined using rotary encoders, which will calculate the displacement of the stage based on the revolutions the motors have moved. Since the motors driving the sprockets move along with the machine, feeding power and avoiding cable entanglement will be challenges to be considered in future work.

6. **Appendix 1**. Matlab program written to design flexures.

%This program uses the design parameters of each blade to determine the number of blades
%needed to move a set displacement

```
h=.05*.0254              % Thickness of each required blade (inches converted to meters)

b=.75*.0254              %Flexure thickness (inches converted to meters)

L=1.5*.0254              %Length of each blade (inches converted to meters)

I=1/12*b*h^3             %Moment of Inertia of each blade

E=70e9                   %Young's Modulus

Sy=55e6                  %Yield Stress of Aluminum based on SolidWorks value

F=Sy*I/2/(h/2)/L         %Force required to displace each blade

d=F*L^3/3/E/I            %Displacement of each individual blade

Totald=.125*2^.5*.0254/2 %Required Displacement

ratio=Totald/d           %Number of blades needed to achieve desired displacement
```