INDIAN INSTITUTE OF TECHNOLOGY, KANPUR

ME 451A - 452A : B.Tech Project

PROJECT REPORT

Design and Development of Proof Mass Actuator

Supervisor

Dr. Mohit Law

<u>Group No. 22</u>

Abhishek Sahoo

Tarun Sharma

Vaibhav Munjole

ACKNOWLEDGEMENT

We take this opportunity to express our deepest gratitude to our revered guide Prof. Mohit Law without whose help and guidance, the project conceptualisation and manufacturing would not have seen the light of the day.

We would like to thank the PEC members Dr. Nachiketa Tiwari, Dr. P S Ghoshdastidar, Dr. Jishnu Bhattacharya, and Dr. S K Choudhury for their valuable feedback.
We would also like to thank Mr. Srijan Bharati for mentoring our project. We extend our gratitude to all the student members of Machine Tool Dynamics Lab for their continued support throughout the duration of the project.We also appreciate the efforts of the staff of Tinkering Lab, 4i Lab and Machine Tool Dynamic Lab who helped us immensely in the manufacturing process.

Above all we would like to thank the Department of Mechanical Engineering, Indian Institute of Technology Kanpur for giving us the golden chance of working on this project.

CONTENTS

1. THE PROBLEM AND THE SOLUTION

Reducing foreign dependence on technology and development of indigenous products is the trademark of every developed nation. India, being a developing country depends a lot on exports for technology and this needs to be changed.

So, we decided to build an electromagnetic actuator which are used in active damping devices.Currently, there are hardly any well known manufacturers of these actuators in India. So, mostly they are bought from Europe .Some of the active damping systems we currently have in our institute are as follows:

1. ADD45 by Micromega Dynamics gives a nominal force of 45 N . It comes in a size of 6.4Φ*18 cm and costs around INR 2.2 lakhs.
2. B&K4809 by Brüel & Kjær gives a nominal force of 45N. It comes in a size of 14.9Φ*14.3 cm and costs around INR 10 lakhs.
3. MICA300CM by Cedrat Technologies gives a nominal force of 300N . It comes in a size of 10Φ*12 cm and costs around INR 12.5 lakhs.

With a vision of building high-tech products indigenously, we aim to build an electromagnetic actuator. In this project, we try to build a first generation of such actuators .This would set the tone for in-house development of such devices in future .

2. CONCEPTUAL SOLUTION

We aim to build an electromagnetic actuator that would be compact and would be cost effective as compared to other products currently available worldwide.

The working principle is as follows:

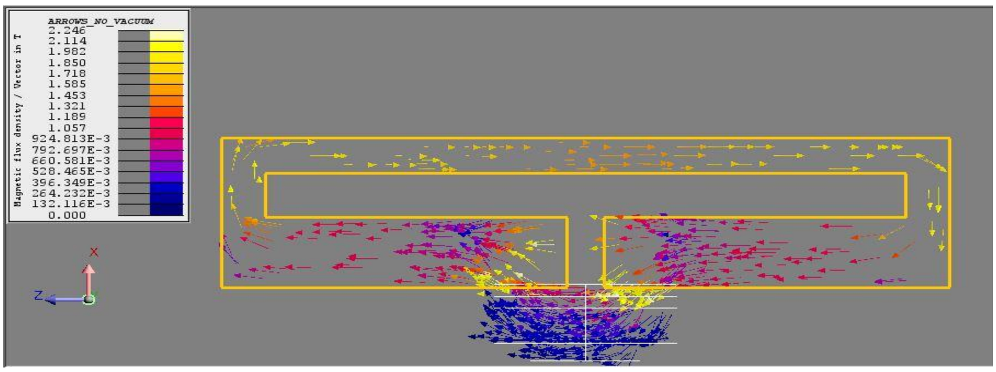


Fig 1: Field lines in the moving iron-stator assembly

As we excite the coil, a magnetic field gets generated around it. In the presence of the stator(core), all the field lines will pass through the stator since it has very high permeability and try to complete its path by passing through the stator itself.
Next, we cut a circular hole in the stator and allow our moving iron to pass through that hole. The moving iron is made in such a way that the air gap (which is the clearance between hole and the moving iron) is 0.5 mm.

Now, when the iron piece is placed just outside the hole and the coils are excited, all the field lines which were passing through the stator will now, in order to complete its path, try to pass through the moving iron and again enter the stator (least resistance path). We place the moving iron such that the field lines have to bend in order to achieve this. Hence , these bent field lines will apply a force on the stator which would have an axial component. The stator will experience a large amount of force because of the large number of field lines which are trying to pass through that very small region.

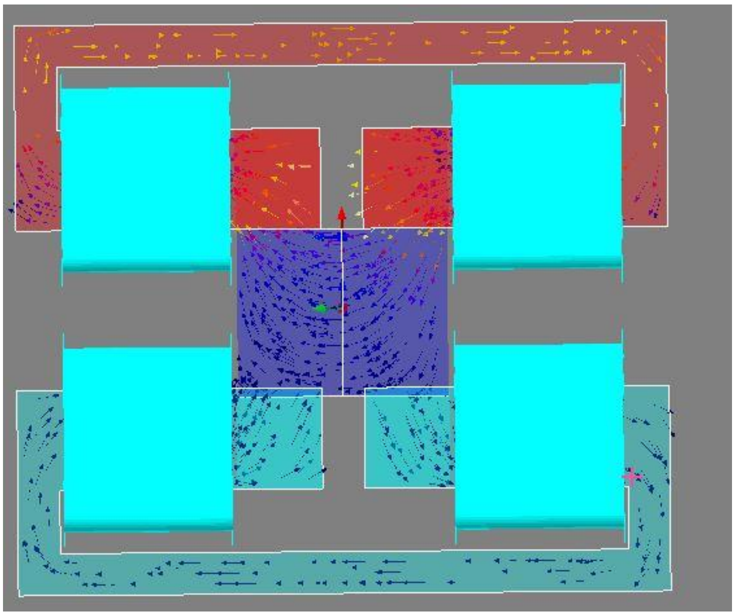


Fig 2: Oscillatory motion of the moving iron

The force pulls the moving iron towards the stator itself. Now we switch the current off in the upper stator and switch it on in the bottom stator. In the same way, the moving iron will now be pulled in the opposite direction.

In this way, we switch the currents across the two stators and hence control the frequency of the oscillatory motion of the moving iron. The actuator would be able to give a force proportional to current input in a specified bandwidth.

This motion would help in transferring the force on to the body of the actuator using springs , which in turn will be applied on the structure to which it is attached.

There are 3 subsystems of the overall design :

1. Actuation
   A proof mass is actuated by electromagnetic force. This actuation has to be proportional to current and should be substantial in a given working frequency bandwidth.

2. Force transfer
   The force on the proof mass has to be transferred on to the body to which the actuator is attached. This would be done by development of linear flexural bearings which would also help in restricting the motion of the proof mass in one direction.

3. Electronics and Control
   The force output would be controlled by a current input. It involves developing an input function and a power circuit to serve the purpose.

3. DELIVERABLES

Following are the target specifications of our actuator:

- Nominal Force : 50 N
- Peak force : 100 N
- Natural frequency of the system : 30 Hz
- Size : 15cm X 15cm X 15cm
- Total Weight : 10 kg

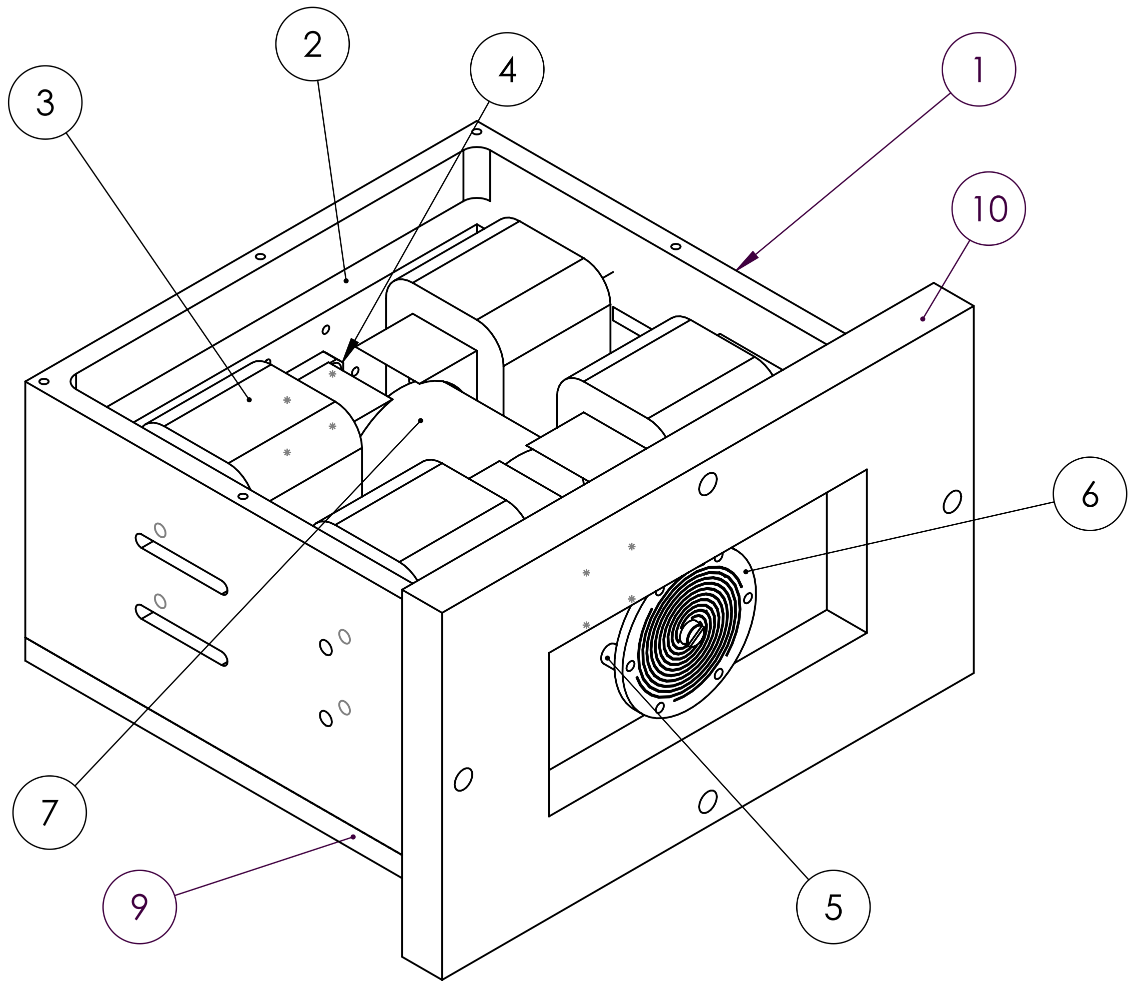
4. MANUFACTURING PARTS LIST

PARTS	QTY	MATERIAL	MACHINING
Cover	1	Aluminium 6061	Milling/Drilling
Stator	2	Cast Iron	Milling/Drilling
Spacer Nuts	12	SS 306	Turning
Linear flexural bearing	2	AISI 1045 Steel	CNC Wire EDM
Moving Iron	1	Cast Iron	Turning
Connector	1	SS 306	Turning
Cover Plate	2	Aluminium 6061	Cutting/Drilling

5. Isometric and Orthographic Drawings of assembly, sub-assemblies and parts

(All drawings are in first angle projection and dimensions are in mm unless otherwise specified)

ITEM NO.	PART NAME	PART NUMBER	QTY.
1	Cover	BTP_G22_01	1
2	Stator	BTP_G22_02	2
3	Solenoid Coil	BTP_G22_07	4
4	LINEAR BUSH BEARING	BTP_G22_08	2
5	Spacer Nuts	BTP_G22_05	12
6	Linear Flexural Bearing	BTP_G22_09	3
7	Moving Iron	BTP_G22_03	1
8	Connector	BTP_G22_04	1
9	Cover Plate	BTP_G22_06	2
10	Base Plate	BTP_G22_10	1



ONLY 1 COVER PLATE IS SHOWN

GROUP MEMBERS	GROUP	SCALE
Abhishek Sahoo Tarun Sharma Vaibhav Munjole	22	1 : 2

FACULTY ADVISOR

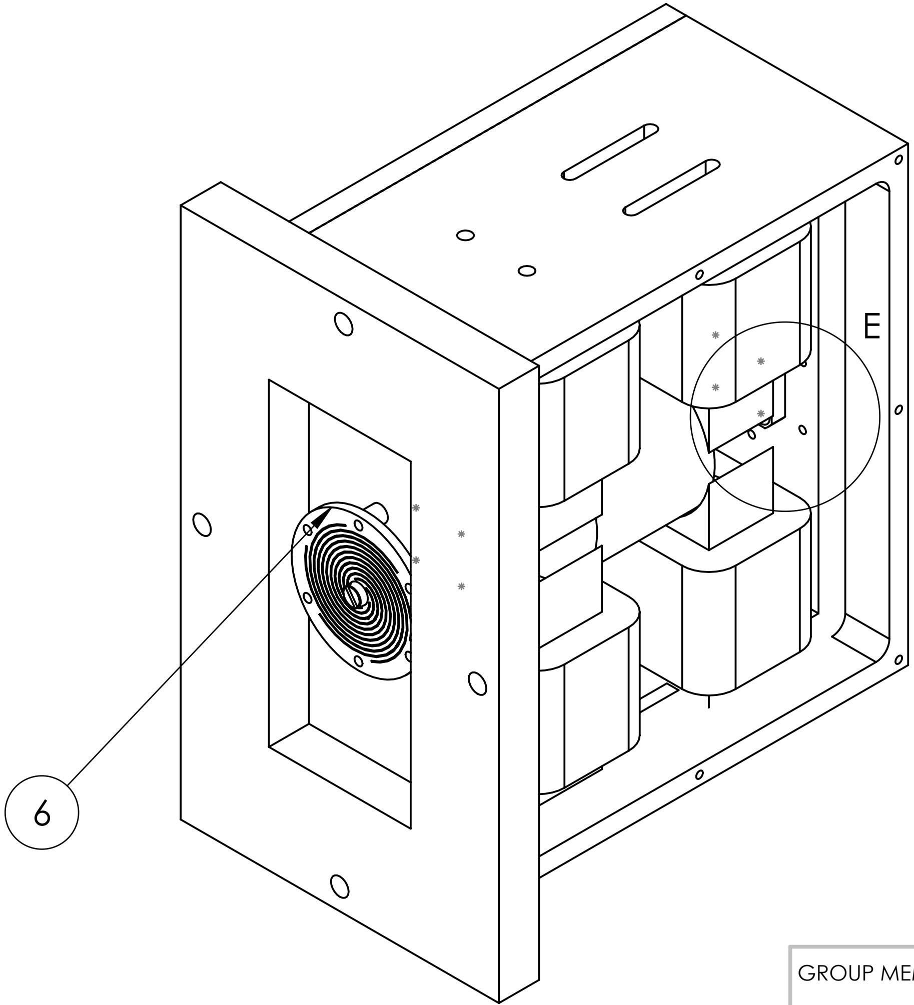
Mohit Law

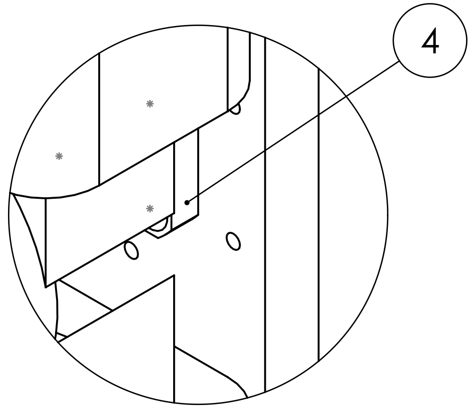
Design and Development of Proof Mass Actuator

ASSEMBLY DRAWING

IIT KANPUR
Indian Institute of Technology, Kanpur

SHEET 1/15 (1/3)

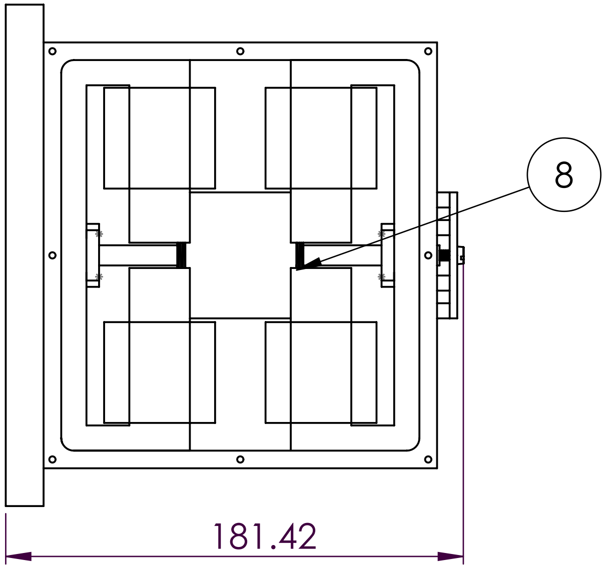


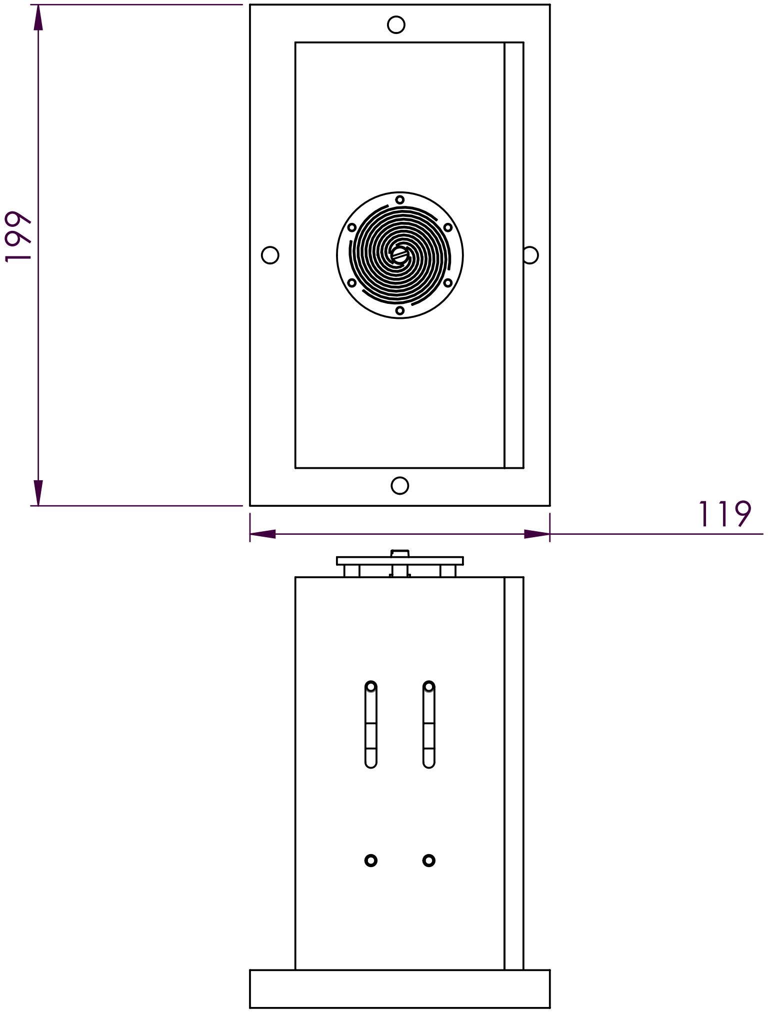


DETAIL E
SCALE 1 : 1

GROUP MEMBERS Abhishek Sahoo Tarun Sharma Vaibhav Munjole	GROUP 22		SCALE 1 : 2
FACULTY ADVISOR Mohit Law	Design and Development of Proof Mass Actuator		
IIT KANPUR Indian Institute of Technology, Kanpur	ASSEMBLY DRAWING		
			SHEET 2/15 (2/3)

D





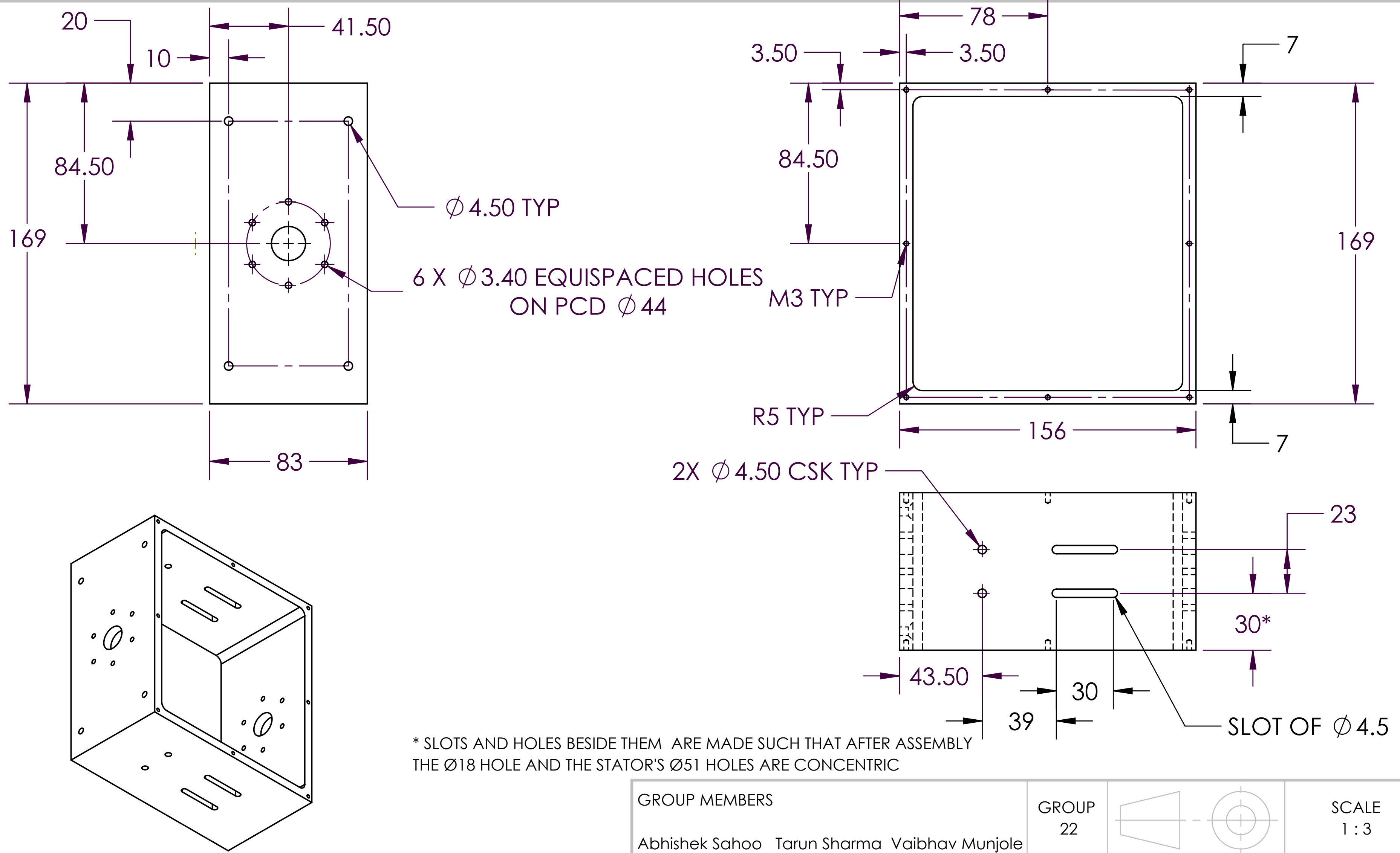
C

B

*All dimensions in mm

A

GROUP MEMBERS Abhishek Sahoo Tarun Sharma Vaibhav Munjole	GROUP 22		SCALE 2 : 7
FACULTY ADVISOR Mohit Law	Design and Development of Proof Mass Actuator		
IIT KANPUR Indian Institute of Technology, Kanpur	ASSEMBLY DRAWING		
		SHEET 3/15 (3/3)	



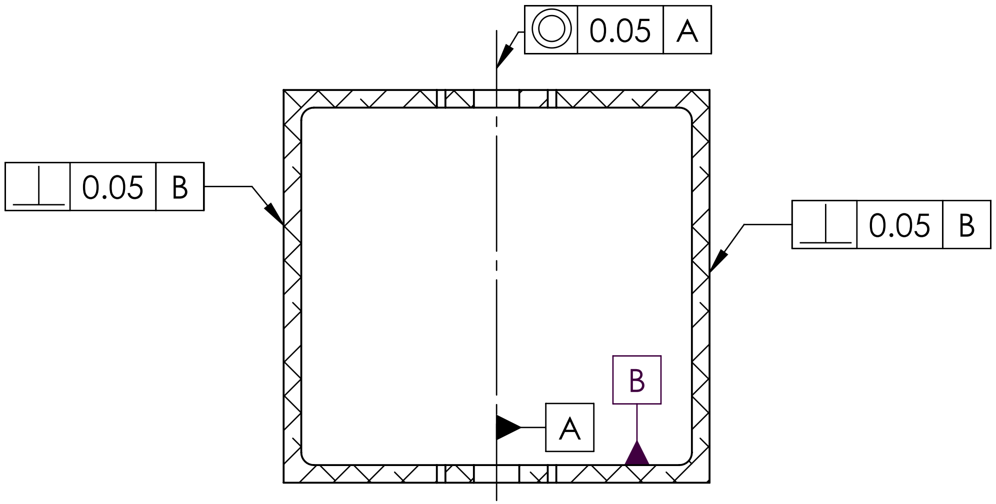
* SLOTS AND HOLES BESIDE THEM ARE MADE SUCH THAT AFTER ASSEMBLY
THE Ø18 HOLE AND THE STATOR'S Ø51 HOLES ARE CONCENTRIC

QTY : 1
MATERIAL : ALUMINUM 6061
MANUFACTURING PROCESS : MILLING ,DRILLING

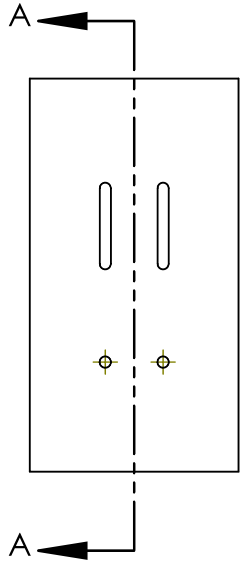
NOTE:
1. All Dimensions in mm
2. Remove all burrs and sharp edges
3. Round all edges max. radius 10mm
4. Unless stated otherwise,
   - Use tolerance for linear dimension = ± 0.1 mm

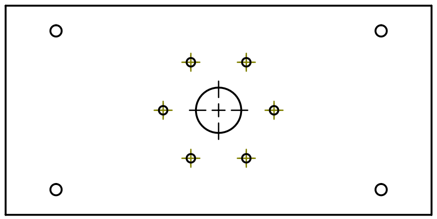
GROUP MEMBERS Abhishek Sahoo Tarun Sharma Vaibhav Munjole	GROUP 22		SCALE 1 : 3
FACULTY ADVISOR Mohit Law	Design and Development of Proof Mass Actuator		
IIT KANPUR Indian Institute of Technology, Kanpur	COVER		
	PART NO. BTP_G22_01	SHEET 4/15 (1/2)	

THE TOP AND BOTTOM Ø 18 HOLES SHOULD BE CONCENTRIC



SECTION A-A



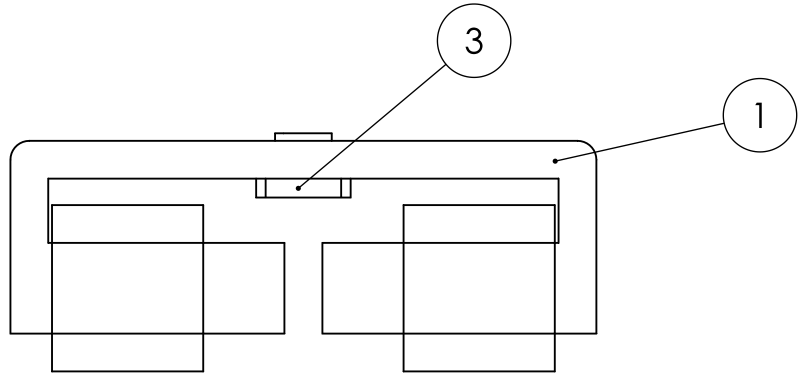


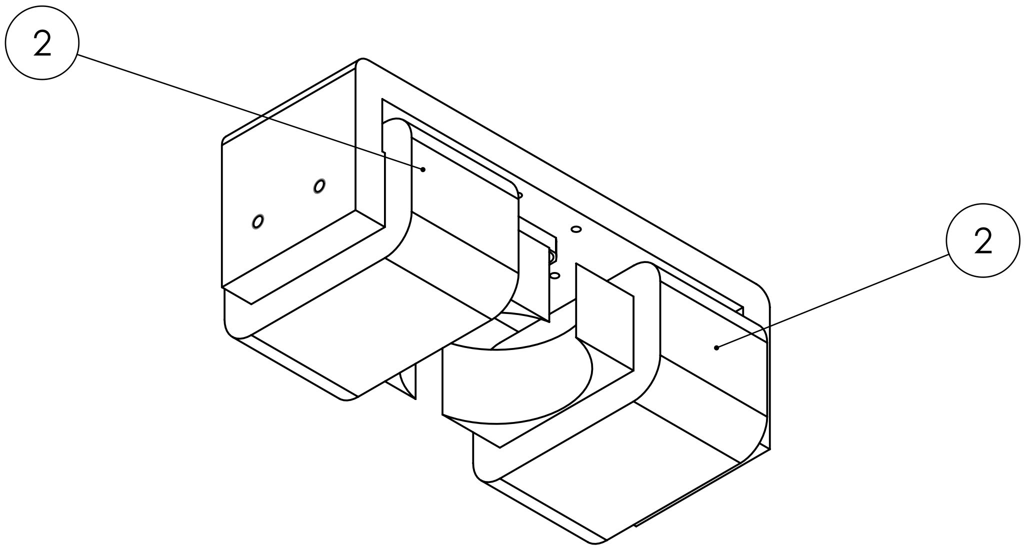
QTY : 1
MATERIAL : ALUMINUM 6061

NOTE:
1. All Dimensions in mm
2. Remove all burrs and sharp edges
3. Unless stated otherwise,
   - Use tolerance for linear dimension = ± 0.1 mm

GROUP MEMBERS Abhishek Sahoo Tarun Sharma Vaibhav Munjole	GROUP 22		SCALE 1 : 3
FACULTY ADVISOR Mohit Law	Design and Development of Proof Mass Actuator		
IIT KANPUR Indian Institute of Technology, Kanpur	COVER		
	PART NO. BTP_G22_01	SHEET 5/15 (2/2)	

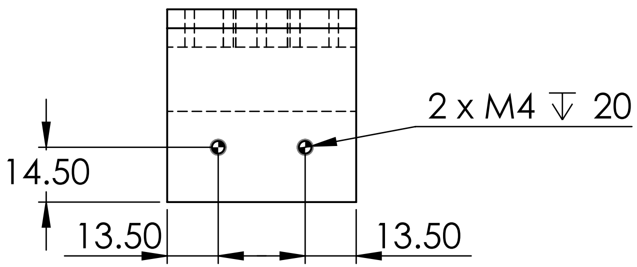
ITEM NO.	PART NAME	PART NUMBER	QTY.
1	STATOR	BTP_G22_02	1
2	SOLENOID COIL	BTP_G22_07	2
3	LINEAR BUSH BEARING	BTP_G22_08	1

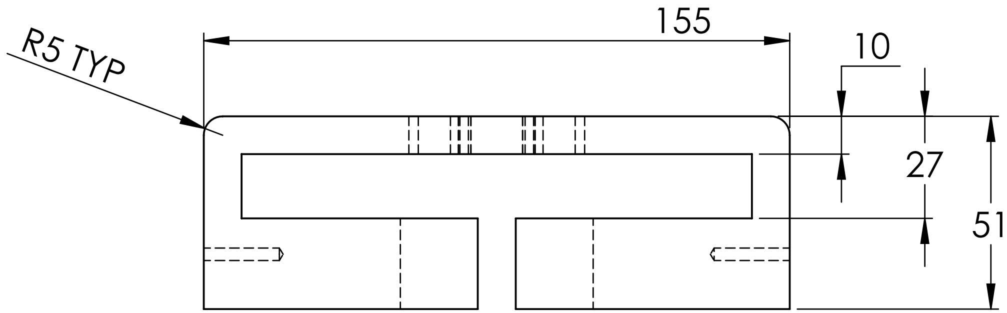


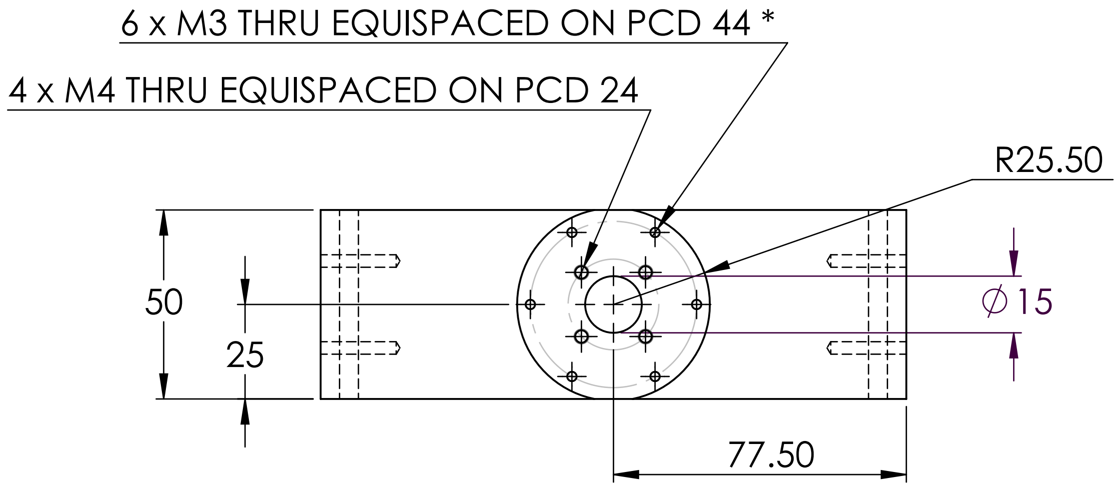


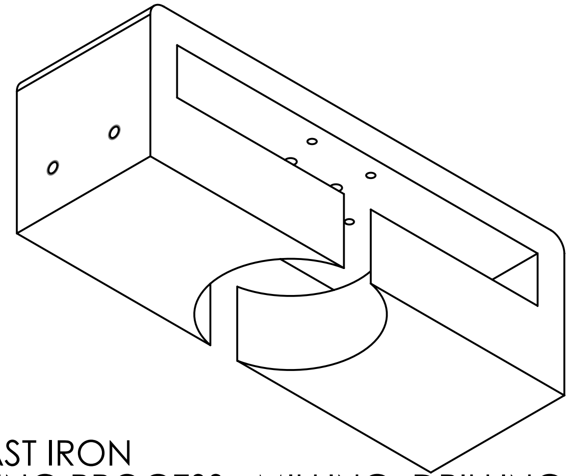
QTY : 2

GROUP MEMBERS Abhishek Sahoo Tarun Sharma Vaibhav Munjole	GROUP 22		SCALE 1 : 2
FACULTY ADVISOR Mohit Law	Design and Development of Proof Mass Actuator		
IIT KANPUR Indian Institute of Technology, Kanpur	COIL STATOR SUBASSEMBLY		
		SHEET 6/15 (1/1)	









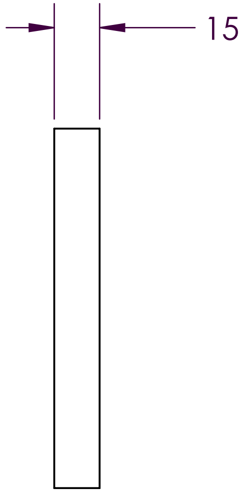
* HOLE MATCHING WITH PART PART NO. BTP_G22_01

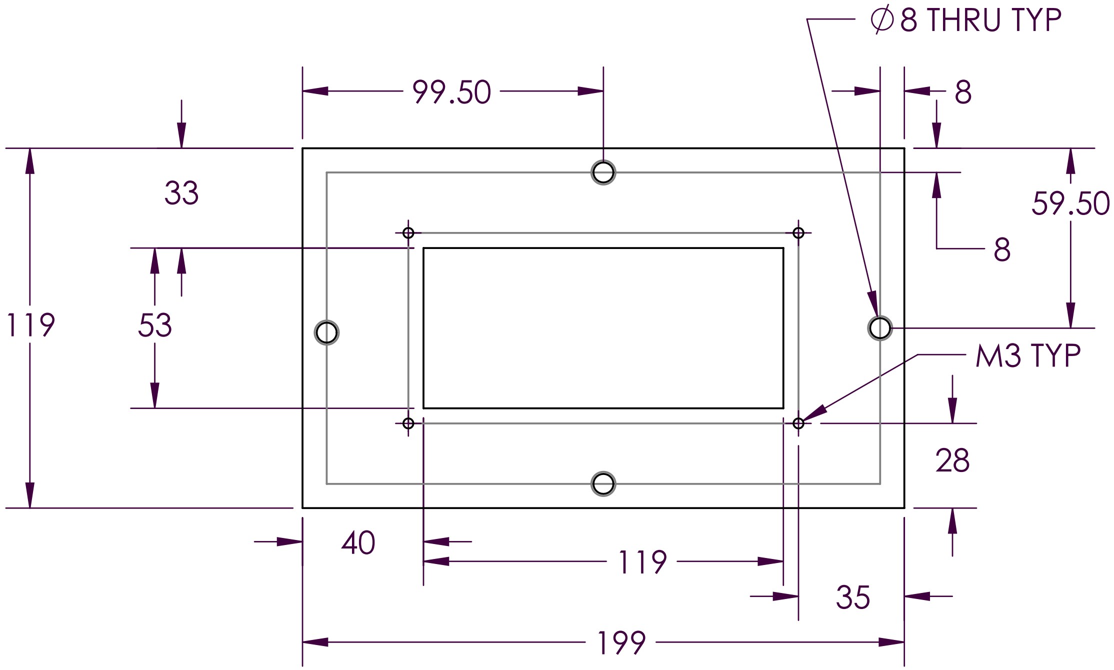
QTY : 2
MATERIAL : CAST IRON
MANUFACTURING PROCESS : MILLING, DRILLING

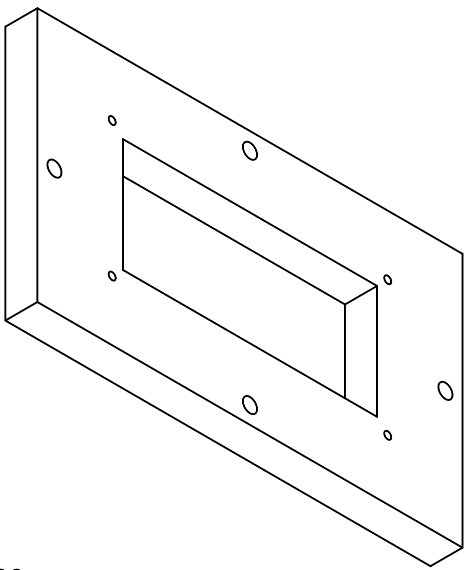
IMPORTANT : BTP_G22_09 AND BTP_G22_01 SHOULD HAVE CLEARANCE FIT SUCH THAT STATOR SHOULD BE ABLE TO SLIDE INSIDE THE COVER BUT THERE SHOULD BE NO LATERAL MOVEMENT

NOTE:
1. All Dimensions in mm
2. Remove all burrs and sharp edges
3. Unless stated otherwise,
   - Use tolerance for linear dimension = ± 0.1 mm

GROUP MEMBERS Abhishek Sahoo Tarun Sharma Vaibhav Munjole	GROUP 22		SCALE 1 : 2
FACULTY ADVISOR Mohit Law	Design and Development of Proof Mass Actuator		
IIT KANPUR Indian Institute of Technology, Kanpur	STATOR		
	PART NO. BTP_G22_02	SHEET 7/15 (1/1)	

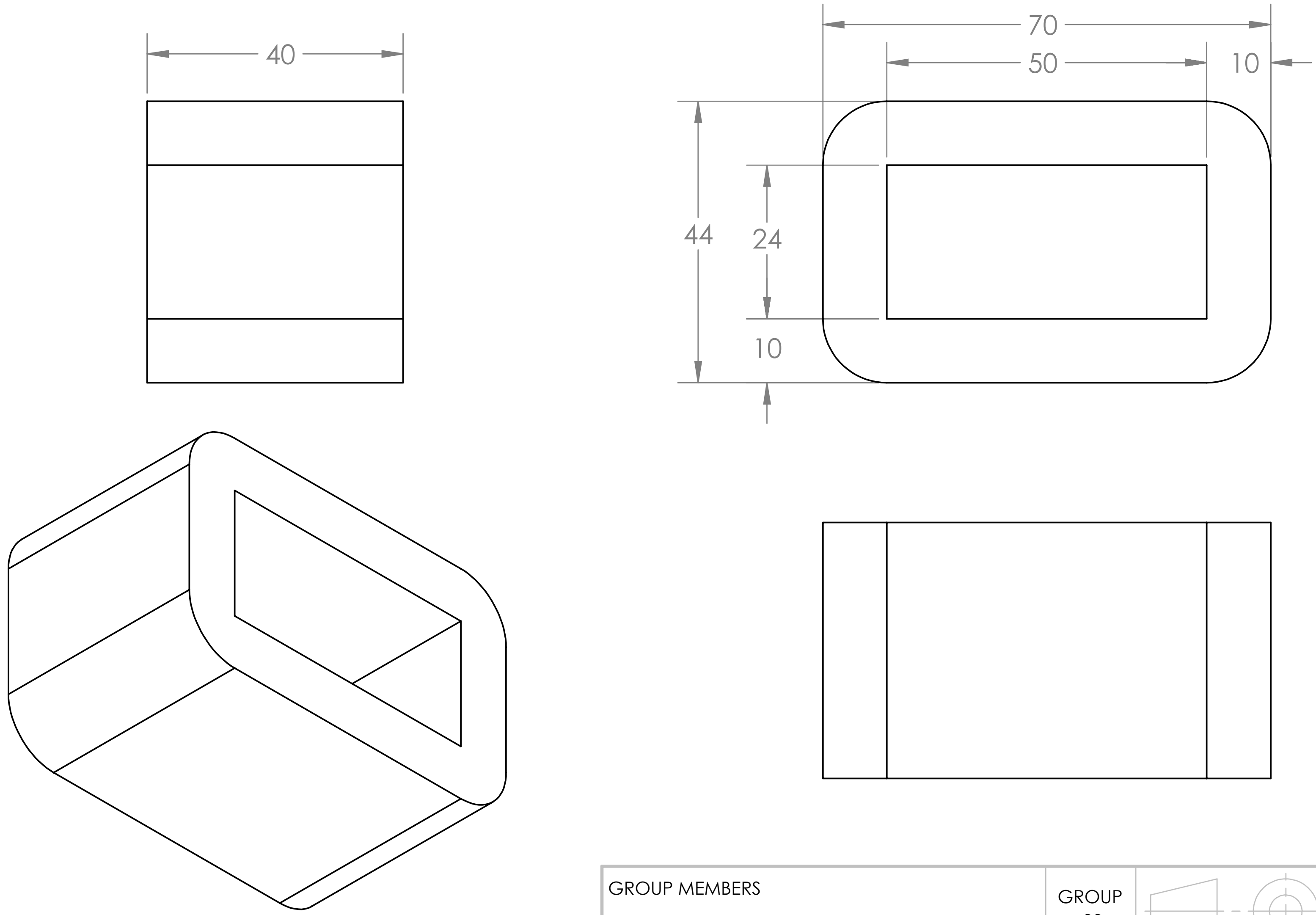






QTY : 1
MATERIAL : SS
Manufacturing Process: Milling, Drilling

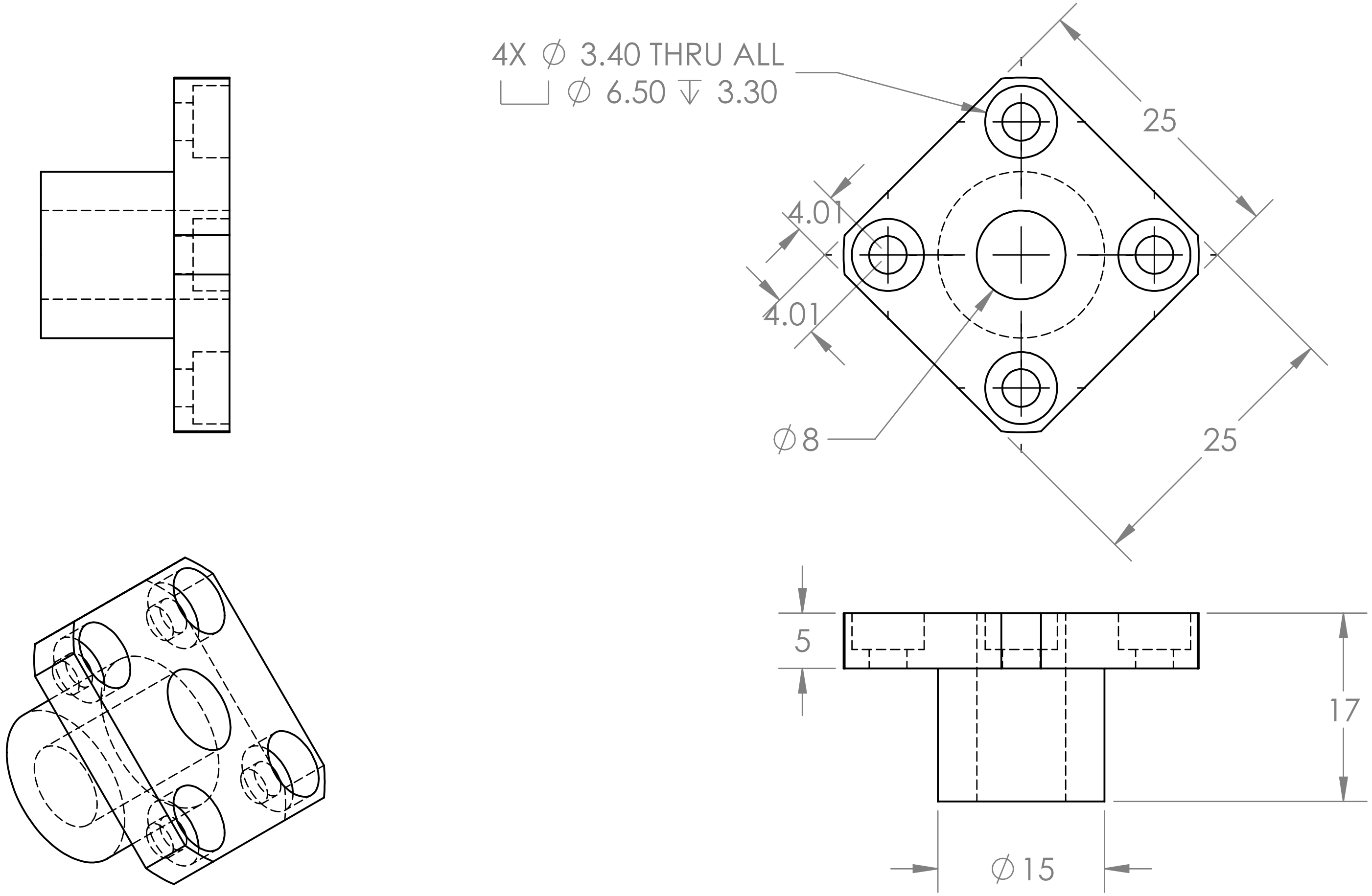
GROUP MEMBERS Abhishek Sahoo Tarun Sharma Vaibhav Munjole	GROUP 22		SCALE 2 : 5
FACULTY ADVISOR Mohit Law	Design and Development of Proof Mass Actuator		
IIT KANPUR Indian Institute of Technology, Kanpur	BASE PLATE		
	PART NO. BTP_G22_10	SHEET 8/15 (1/1)	



QTY : 4
MATERIAL : 1mm(AWG18) COPPER WIRE

No. of turns in 1 layer = 40
No. of layers = 10

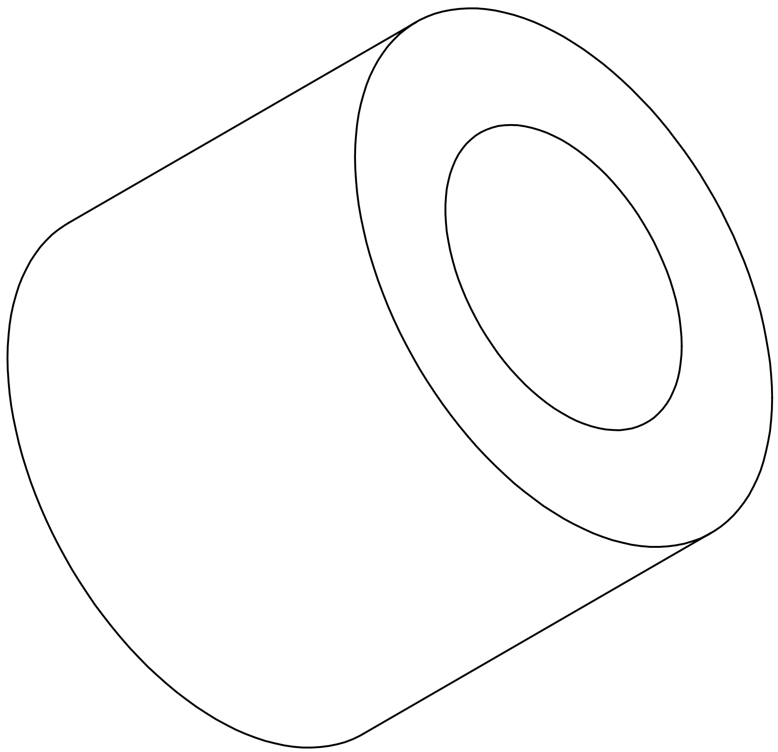
GROUP MEMBERS Abhishek Sahoo Tarun Sharma Vaibhav Munjole	GROUP 22		SCALE 1 : 1
FACULTY ADVISOR Mohit Law	Design and Development of Proof Mass Actuator		
IIT KANPUR Indian Institute of Technology, Kanpur	SOLENOID COIL		
	PART NO. BTP_G22_07	SHEET 9/15 (1/1)	

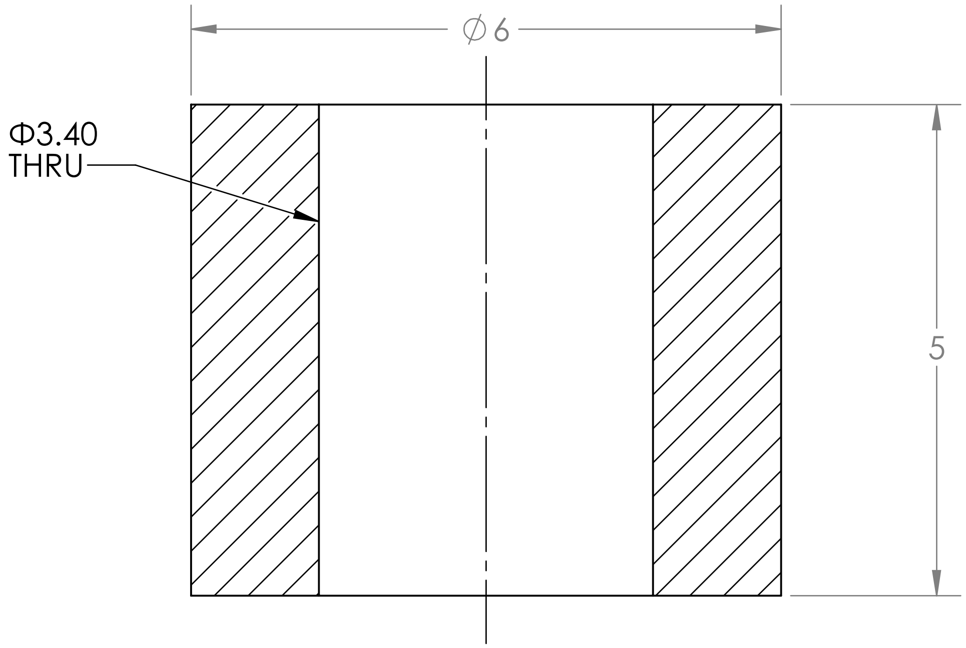


QTY : 2
TO BE PURCHASED DIRECTLY

NOTE:
1. All Dimensions in mm

GROUP MEMBERS Abhishek Sahoo Tarun Sharma Vaibhav Munjole	GROUP 22		SCALE 1 : 1
FACULTY ADVISOR Mohit Law	Design and Development of Proof Mass Actuator		
IIT KANPUR Indian Institute of Technology, Kanpur	LINEAR BUSH BEARING		
	PART NO. BTP_G22_08	SHEET 10/15 (1/1)	

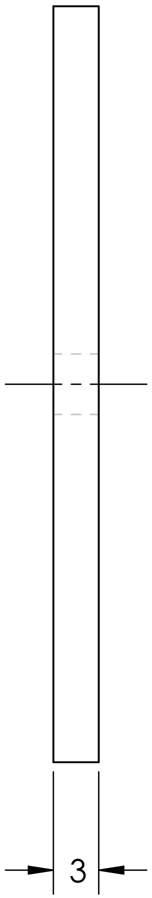


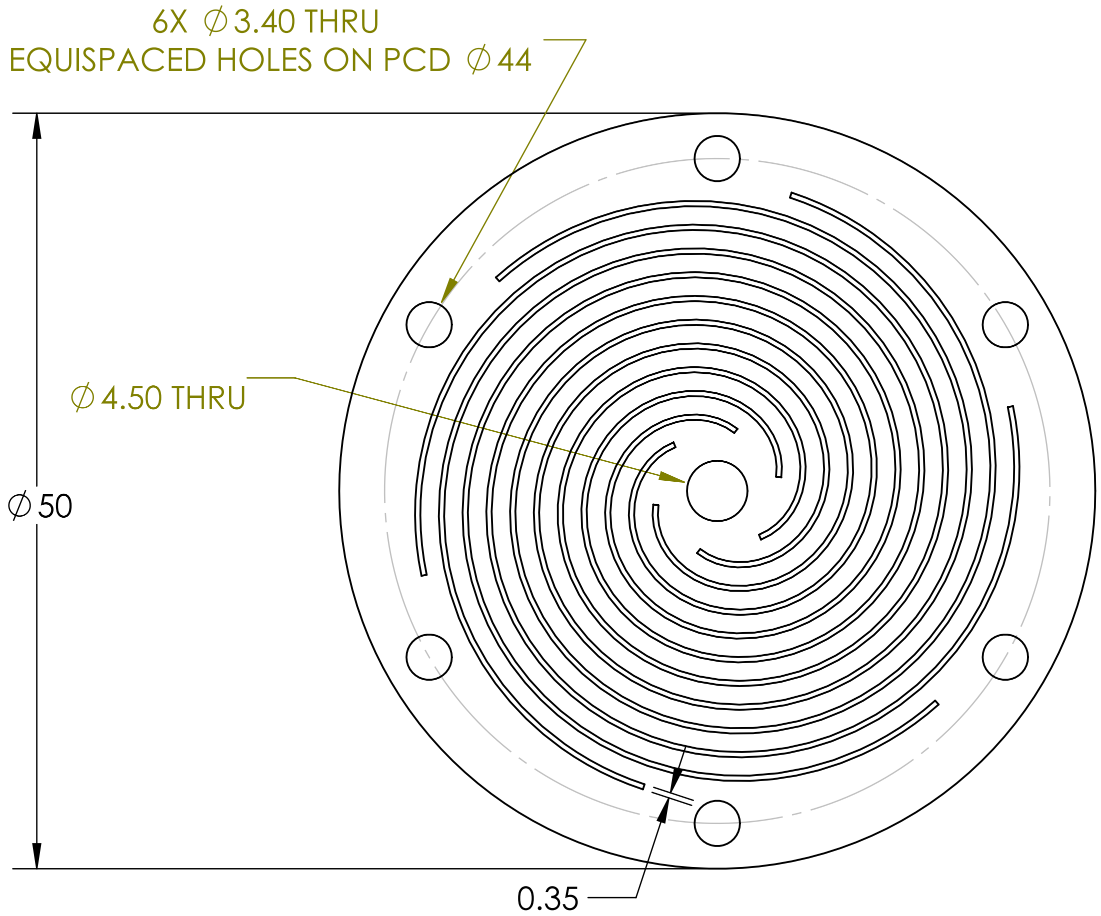


QTY : 12
MATERIAL : STAINLESS STEEL 306
MANUFACTURING PROCESS : TURNING

NOTE:
1. All Dimensions in mm
2. Remove all burrs and sharp edges
3. Unless stated otherwise,
   - Use tolerance for linear dimension = ± 0.1 mm

GROUP MEMBERS Abhishek Sahoo Tarun Sharma Vaibhav Munjole	GROUP 22		SCALE 13 : 1
FACULTY ADVISOR Mohit Law	Design and Development of Proof Mass Actuator		
IIT KANPUR Indian Institute of Technology, Kanpur	SPACER NUT		
	PART NO. BTP_G22_05		SHEET 11/15 (1/1)





-TO BE CNC WIRE CUT USING CAD MODEL

-THE CONCENTRICITY OF ∅4.50 HOLE HAS TO BE CAREFULLY CONTROLLED

QTY : 2
MATERIAL : AISI 1045 STEEL
MANUFACTURING PROCESS : CNC WIRE EDM

NOTE:
1. All Dimensions in mm
2. Remove all burrs and sharp edges
3. Unless stated otherwise,
   - Use tolerance for linear dimension = ± 0.1 mm

GROUP MEMBERS

Abhishek Sahoo Tarun Sharma Vaibhav Munjole

GROUP 22

SCALE 1 : 1

FACULTY ADVISOR

Mohit Law

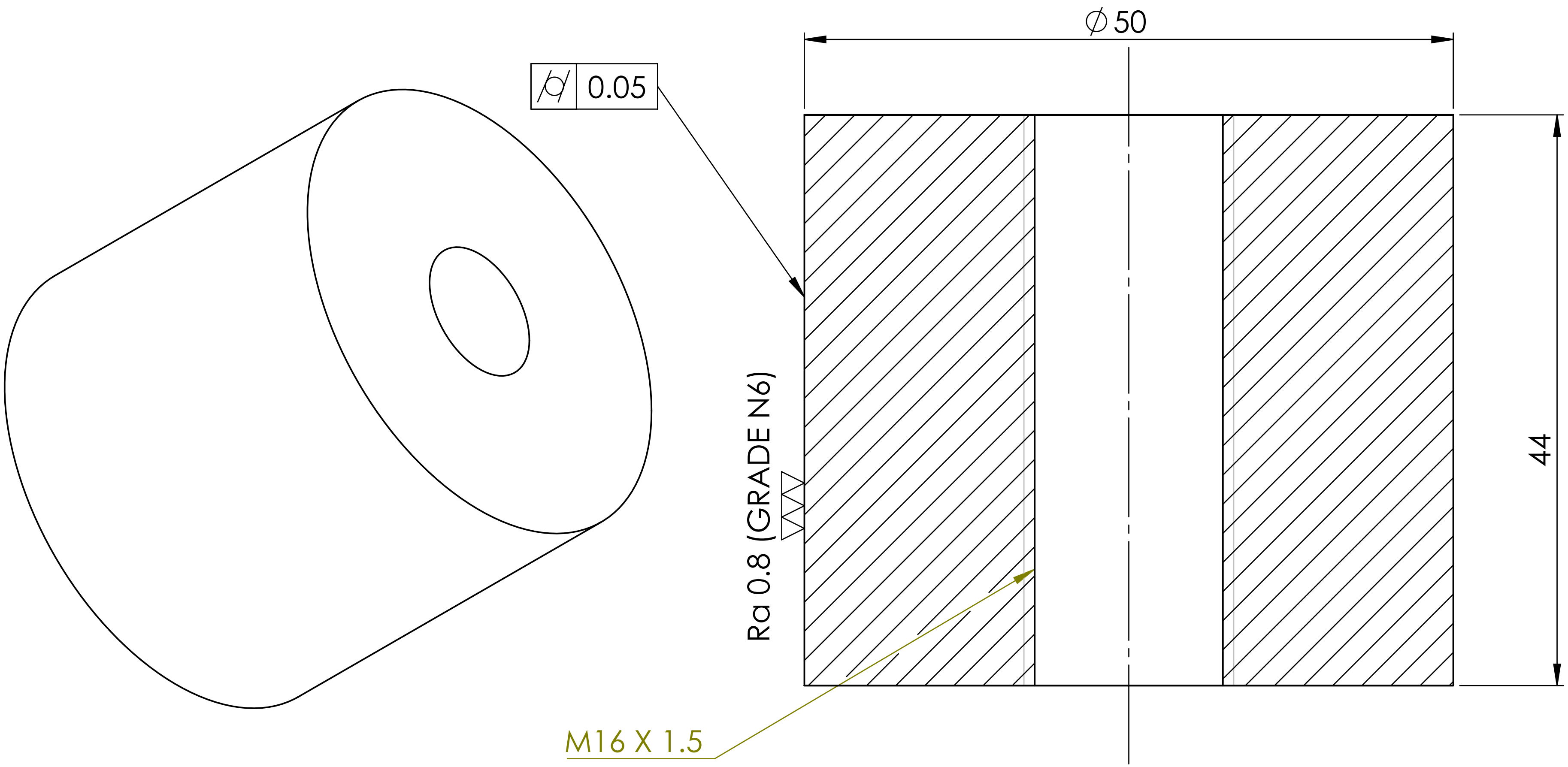
Design and Development of Proof Mass Actuator

IIT KANPUR
Indian Institute of Technology, Kanpur

LINEAR FLEXURAL BEARING

PART NO.
BTP_G22_09

SHEET 12/15
(1/1)

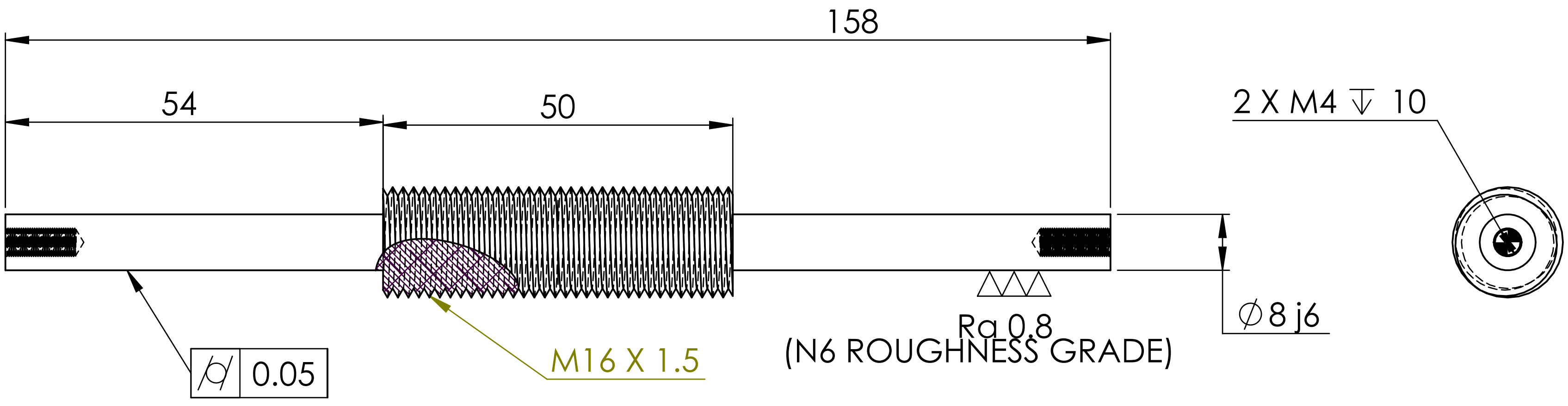


BTP_G22_03 IS SCREWED ON THE M16 EXTERNAL THREAD OF BTP_G22_04

QTY : 1
MATERIAL : CAST IRON

NOTE:
1. All Dimensions in mm
2. Remove all burrs and sharp edges
3. Unless stated otherwise,
   - Use tolerance for linear dimension = ± 0.1 mm

GROUP MEMBERS Abhishek Sahoo Tarun Sharma Vaibhav Munjole	GROUP 22		SCALE 1 : 1
FACULTY ADVISOR Mohit Law	Design and Development of Proof Mass Actuator		
IIT KANPUR Indian Institute of Technology, Kanpur	MOVING IRON		
	PART NO. BTP_G22_03	SHEET 13/15 (1/1)	



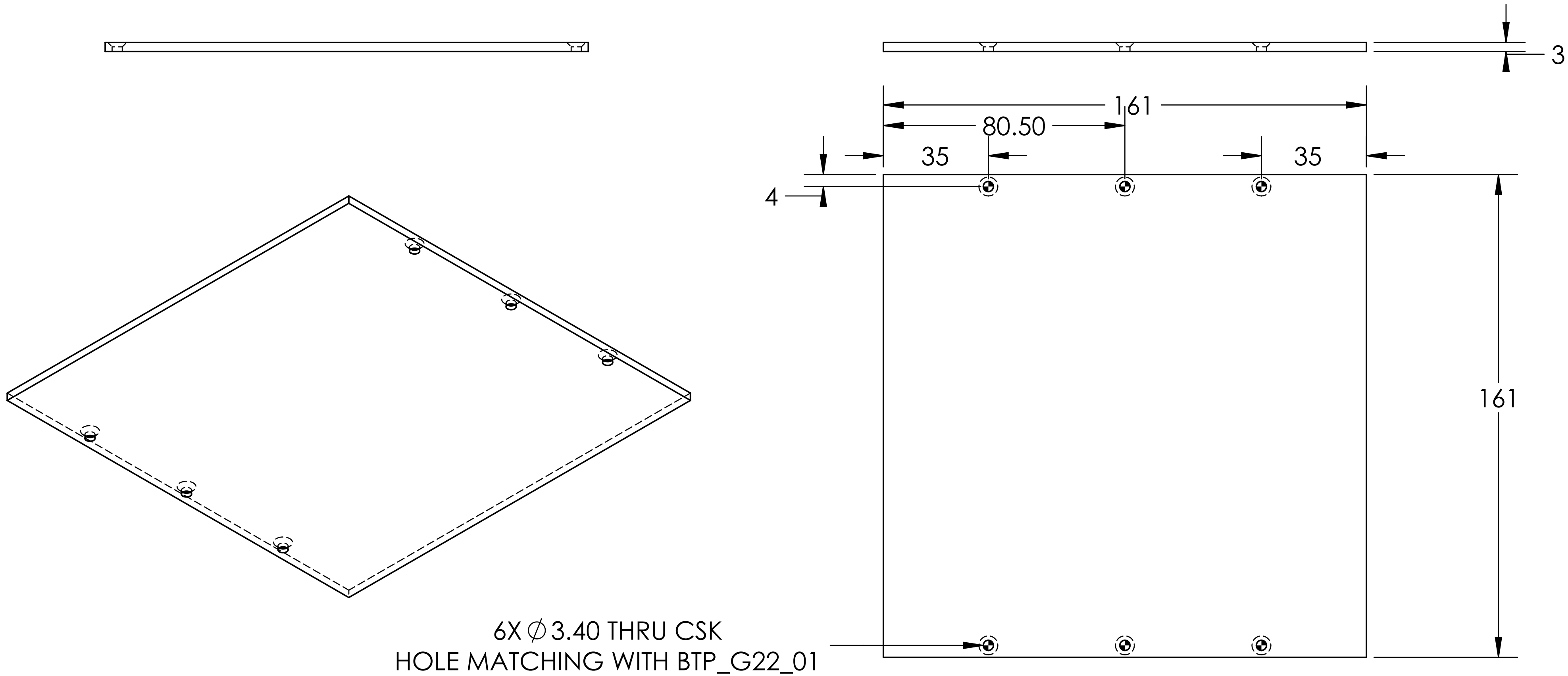
CONCENTRICITY OF M4 DRILLS , M16 EXT. THREAD AND AXIS OF THE PART TO BE CAREFULLY CONTROLLED

QTY : 1
MATERIAL : SS 306
MANUFACTURING PROCESS: TURNING

NOTE:

1. All Dimensions in mm
2. Remove all burrs and sharp edges
3. Unless stated otherwise,
   - Use tolerance for linear dimension = ± 0.1 mm

GROUP MEMBERS Abhishek Sahoo Tarun Sharma Vaibhav Munjole	GROUP 22	SCALE 1 : 1
FACULTY ADVISOR Mohit Law	Design and Development of Proof Mass Actuator	
IIT KANPUR Indian Institute of Technology, Kanpur	CONNECTOR	
	PART NO. BTP_G22_04	SHEET 14/15 (1/1)



QTY : 1
MATERIAL : ALUMINUM 6061
MANUFACTURING PROCESS : CUTTING, DRILLING

NOTE:
1. All Dimensions in mm
2. Remove all burrs and sharp edges
3. Unless stated otherwise,
   - Use tolerance for linear dimension = ± 0.1 mm

GROUP MEMBERS Abhishek Sahoo Tarun Sharma Vaibhav Munjole	GROUP 22	SCALE 1 : 2
FACULTY ADVISOR Mohit Law	Design and Development of Proof Mass Actuator	
IIT KANPUR Indian Institute of Technology, Kanpur	COVER PLATE	
	PART NO. BTP_G22_06	SHEET 15/15 (1/1)

6. DESIGN CALCULATIONS AND ANALYSIS

6.1.Design Calculations

1.Solenoid Coil

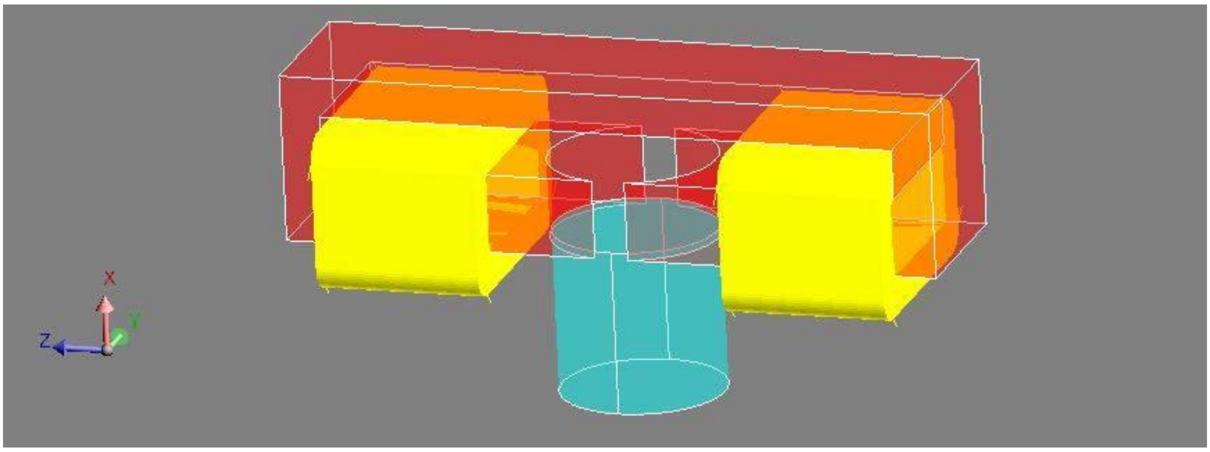


Fig 3: FLUX model for simulating the actuator

We require a unidirectional force. Thus,
$F_x = 100\ N$
$F_y = 0\ N$
$F_z = 0\ N$

Thus, symmetrical structure about $y = 0$ and $z = 0$

The following equations and theoretical procedure were carried out in COMSOL Multiphysics to get the geometry.

A = Cross Sectional Area of Coil
μ_0 = Permeability of Free Space
$= 4\pi \times 10^{-7}$
g = Air gap
N = Number of turns of coil per unit length
i = Current in the coil

Ampere’s law stipulates that the integration of magnetic field intensity, H, around a closed contour of length l is equal to the net current crossing the surface of the closed contour:

$$\oint \vec{H} \cdot \overrightarrow{dl} = Ni$$

where i is the applied current and N is the number of times the current encircles the contour.

Now,

$$H = \frac{\varphi}{\mu A}$$

$$\oint \frac{\varphi}{\mu A} \cdot \overrightarrow{dl}$$

Now, rewriting Ampere’s law in terms of magnetic flux and resistance through a material and using reluctance, R we get,

$$R\varphi = Ni$$

$$R = \frac{x_{gap} - x}{\mu A}$$

The flux linkage is defined as the number of turns in the coil multiplied by the magnetic flux passing through the cross-sectional area of the coil. The voltage may then be derived as:

$$V = \frac{d(N\varphi)}{dt} = \frac{\partial(N\varphi)}{\partial t}\frac{di}{dt} + \frac{\partial(N\varphi)}{\partial x}\frac{dx}{dt}$$

$$= L(x,i)\frac{di}{dt} + K_s(x,i)\frac{dx}{dt}$$

The speedance, Ks is the coefficient of motional EMF across the coil and can be derived as:

$$K_s(x,i) = N\frac{d\varphi(x,i)}{dx} = N\frac{d}{dx}\left(\frac{Ni}{R_g}\right)$$

Assuming no electrical power loss in the coil, the change in work done as the total flux changes is:

$$W = \int i.d(N\varphi) = \int R.\varphi.d\varphi = \frac{iN\varphi}{2}$$

The force acting on the second magnetic element is:

$$F = -\frac{dW}{dx} = -\frac{Ni}{2}\frac{d\varphi}{dx}$$

Taking the following values and optimising the variables N, I, A in COMSOL Multiphysics and(or) FLUX 2018 to achieve a maximum force of 100 N in x-direction , which is constant over at least 2mm of stroke length, we get

$N = 800\ turns$
$A = (24\ x\ 50)\ cm^2$ (Taking Rectangular Cross Section)
$i = 4A$

2.Stator and Cover

The dimensions of the stator is governed by dimension of the solenoid coil winding.
We are using AWG 19 wire which has a current rating of 10A. The wire diameter is 0.912mm.We need 800 turns.

We divide this winding symmetrically about the moving iron to cancel any lateral forces. So, in each coil, there are 400 turns.Length of 1 turn is equal to the wire diameter. So, total length required for 400 turns is 364.8 mm.

To make it compact, we make multilayer coil winding. If we apply 10 layers of winding , then length of solenoid will be 36.48mm.

Hence, total length of two solenoids is 72.96mm.

The moving iron will pass through the stator, hence it should have a gap of 51 mm.

Finally, we provide 10mm thickness for its strength. So, total length of stator is 72.96 + 51 + 20 = 143.96 mm. Considering some tolerance for coil winding, we decide length of the stator to be 155 mm.

The stator has to slide fit into the cover, hence the cover width is decided. Length of the cover is determined by the space required to fit the two stators and the moving iron.

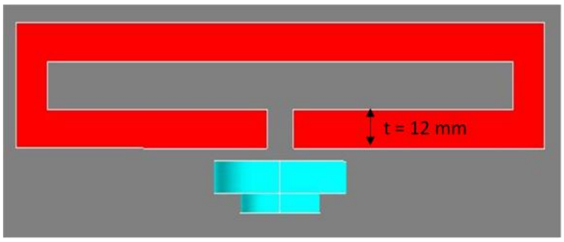
The cover has slots so as to adjust the distance between the two stators which will help us to try out different configurations during testing.

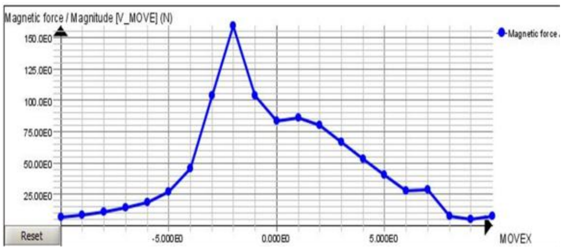
3.Moving Iron

The dimensions of moving iron was decided by analysis using FLUX.
We tried out different shapes and sizes to obtain a high electromagnetic force which should be constant over the length of the stroke.

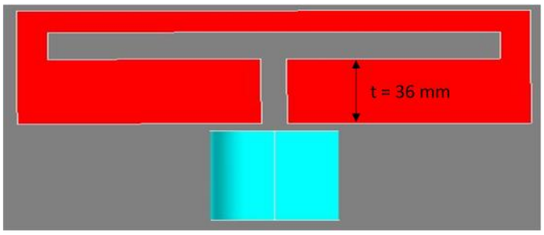
Below are a few major iterations and their results:

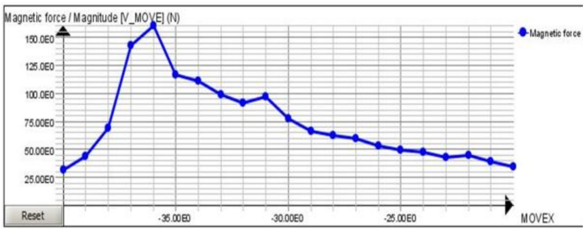
Iteration 1:



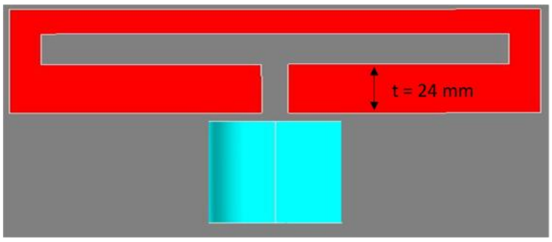


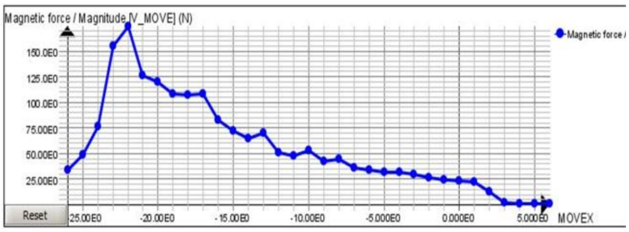
Iteration 2:



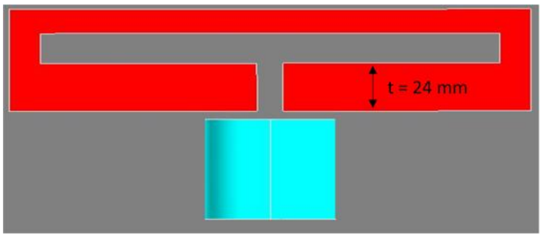


Iteration 3:





Iteration 4:



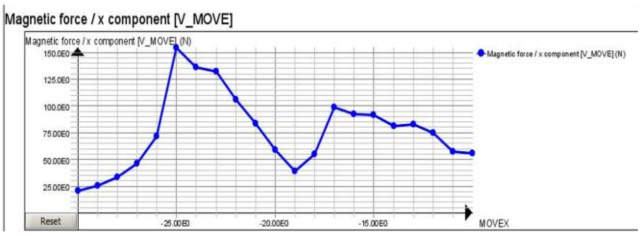


Fig 4: Force vs. position for different stator-moving iron geometry

In the third iteration , we see a constant force of 105 N over a range of 2mm.

Hence , diameter of the moving iron was found out to be 50 mm.

The graph shows electromagnetic force vs. the air gap. Notice that the force is high when the air gap is zero, but decreases sharply as the air gap is increased.

We need to keep in mind the machining tolerances and hence we decided to keep the air gap of 0.5 mm to get high electromagnetic force and make its manufacturing viable .

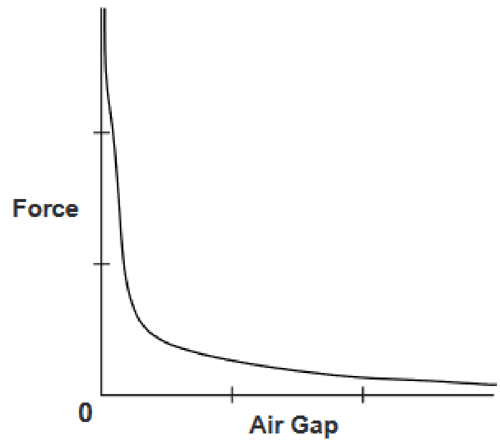


Fig 5: Electromagnetic force vs. Air gap

We provided internal threading in the moving iron so that we can adjust its initial position. This will give us flexibility while testing , to decide the initial position of the moving iron with respect to the stator.

4.Spring Design

We used FLUX to get the amplitude of oscillation, from which we keep it at 2 mm. It was done because the electromagnetic force was fell down drastically after a specific displacement. Thus in order for us to get a uniform forcing, we limit our amplitude of oscillation to 2 mm.

Hence, we find that the stiffness of the spring by the equation,

$$F = kx,$$

by assuming that the spring behaves linearly in x = +2mm to x = -2mm.

Since, we wanted our peak force to be of 100N, thus
$k = F/x = 100/(2 \times 10^{-3}) = 50000$ N/m

We take 2 springs in our model which are connected in parallel and hence the desired stiffness of each spring is **25000 N/m.**

We achieved a spring constant of **30000 N/m** by including design constraints and the mass of moving iron was about 1 kg

Thus, natural frequency of the system is

$$f = \frac{1}{2\pi} \sqrt{\frac{k}{m}} = 39\ Hz$$

Natural frequency of 39 Hz is desirable. This is because most machines have dominant modes above 100 Hz and so when our actuator works to damp oscillations above 100 Hz, it will be working above this resonant frequency and thus we will face no problems because of resonance.

Different spring designs were tried out along with their strength and fatigue analysis. This was done in Solidworks. The details of spring design can be found out in Appendix II.

** Design of all the parts are based on calculations.

6.2.Analysis

1.Static Analysis of Actuator

All the above mentioned dimensions were found out by doing static analysis of the model of our electromagnetic actuator on FLUX. We optimised the dimensions by carrying out simulations by changing the design parameters to meet our target specifications.

In addition to that we obtained the force vs. current graph for a given geometry and position. We see that there is almost linear dependence , so we can vary the electromagnetic force in a linear manner by varying the current in the coils of the stator. This improves controllability of the actuator.

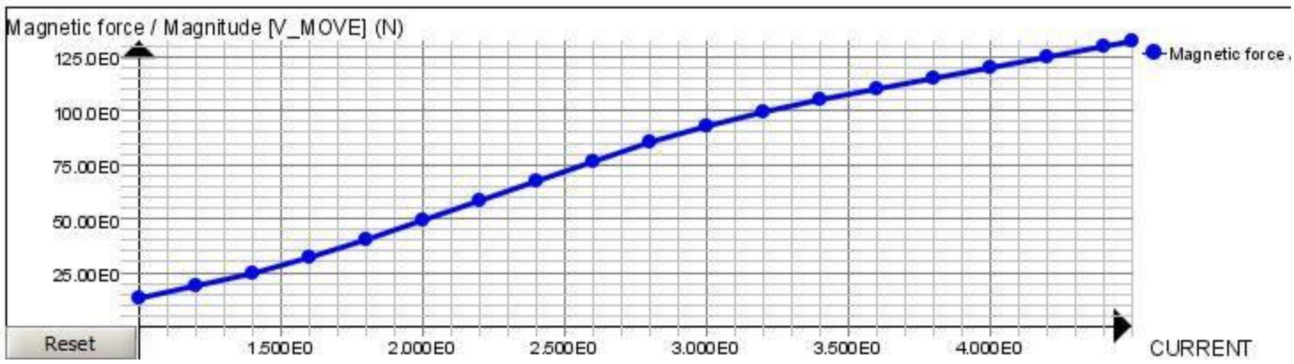


Fig 6: Force vs. current at a fixed position of moving iron

2.Transient Analysis of Actuator

Transient analysis of the actuator was done to find the bandwidth and nature of force output at resonant frequency of the moving iron.

We carry out transient analysis for 1 to 200 Hz and our natural frequency is around 40 Hz.

This is done using a sine sweep function which sweeps from 1 to 200 Hz in a specified amount of time.

The current function given in the top stator is

$$i(t) = A\sin\left(2\pi * \left(f_1 t + \frac{f_2 - f_1}{t_0} t^2\right)\right)$$

where,

A = amplitude of the current

f_1= frequency at time = 0

f_2= frequency at time = t_0

The same function with a phase difference of π/2 is sent into the bottom stator.

The time step of the transient analysis was kept as $\Delta t = \frac{1}{10 * f_{max}}$. This ensures that we get 10 readings / cycle at the maximum frequency and avoid aliasing.

Then the time domain response is converted into frequency domain response by FFT(Appendix I).

Here are the results:

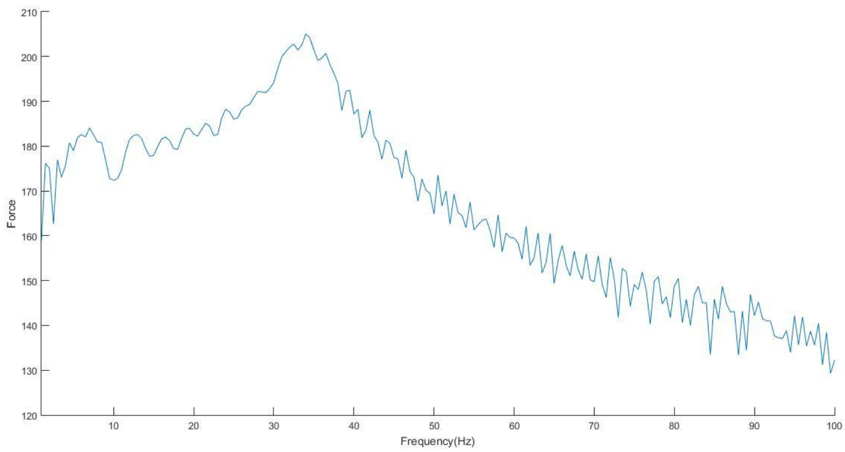


Fig 7: Force vs. Frequency (1 -100 Hz)

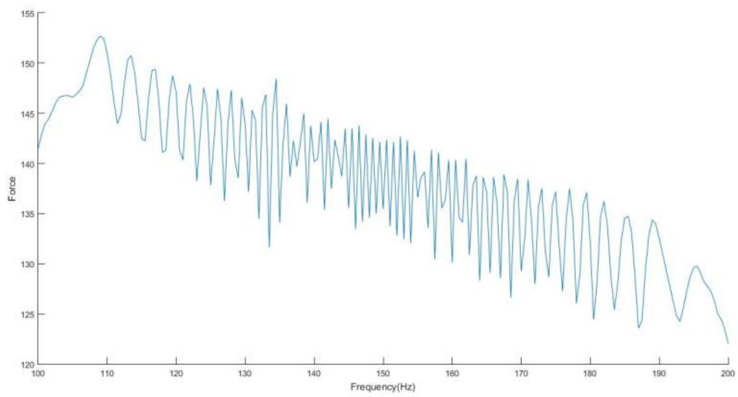


Fig 8: Force vs. Frequency (100 -200 Hz)

(i) We see a peak at 39 Hz, which is the natural frequency of the moving iron.

(ii) The force drops off with increase in frequency beyond 150 Hz and is no more useful.

3.Finding input current function

By applying a sine and a cosine wave across the two stators , the motion of the moving iron was not smooth . This result in poor force output curve.

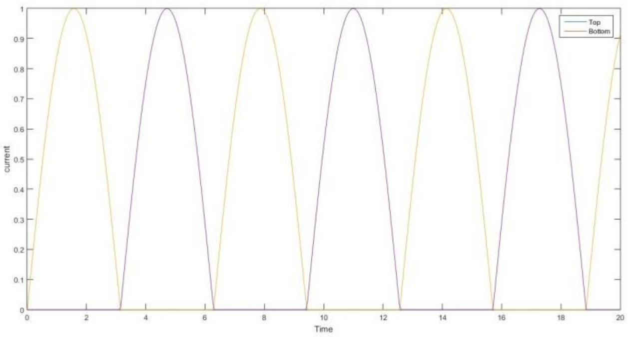


Fig 9: Old Current function

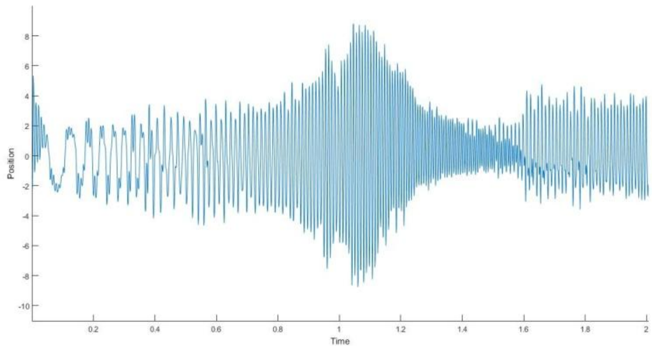


Fig 10 : Position vs. Time for sweep of old function

We found a new input function which dramatically smoothened the motion of the moving iron.

For the top stator, $i(t) = \frac{A}{2}(|\sin(2\pi f t\ )| + \sin(2\pi f t\ ))$ and

For the bottom stator, $i(t) = \frac{A}{2}(|\sin(2\pi f t\ )| - \sin(2\pi f t\ ))$.

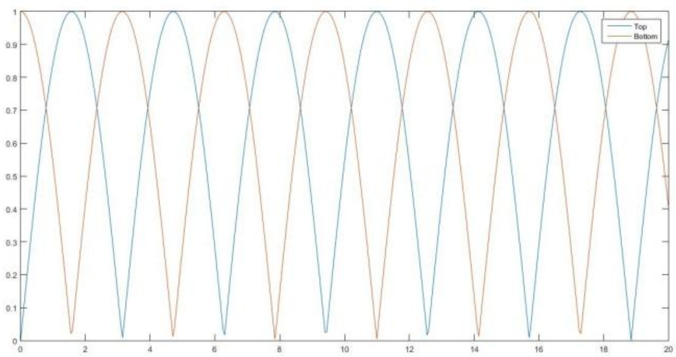


Fig 11: New Current function

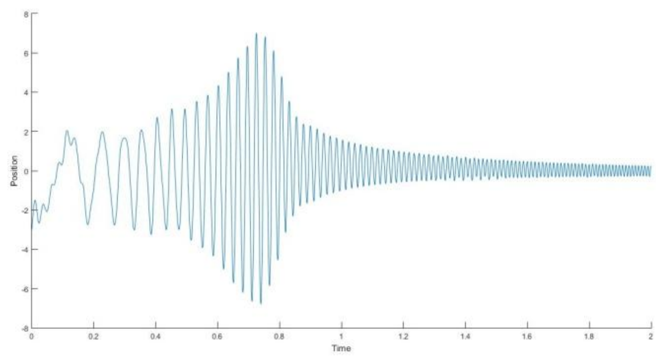


Fig 12 : Position vs. Time for sweep of new function

4.Electrical Circuit

We need an electrical circuit to send the current signal into the actuator. As of now, it is an open loop function as shown in previous section, and the circuit design is still in preliminary stage.

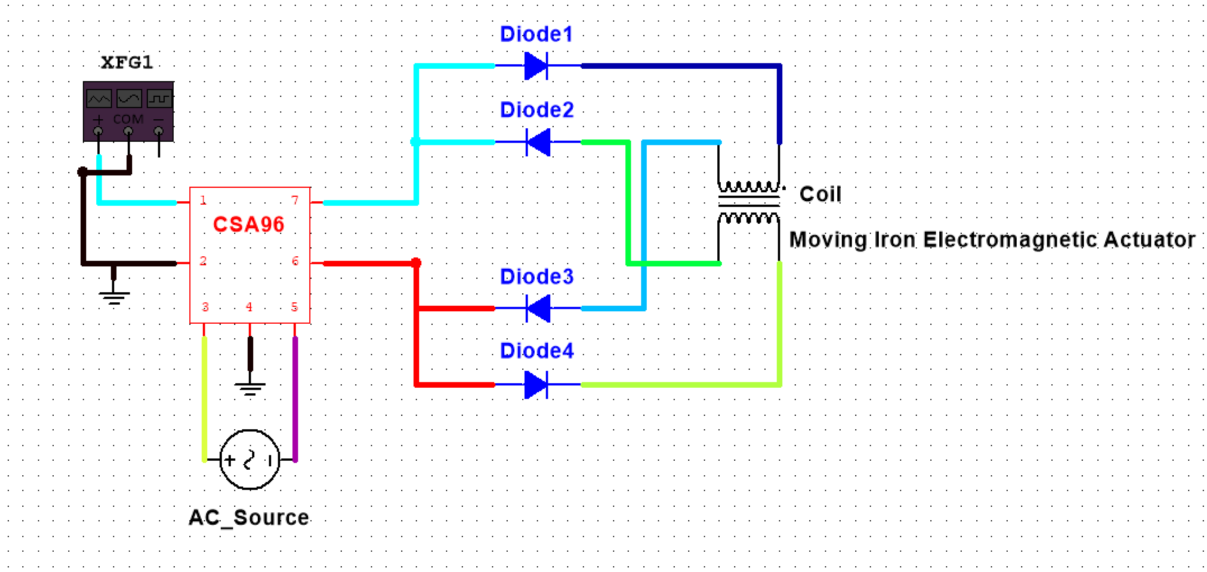


Fig 13:Schematic diagram of the electrical circuit

We are using a function generator XFG1 to generate a sinusoidal signal. This is fed into the amplifier to amplify the current. This amplified signal is then sent into the upper and lower coil by an arrangement of diodes that ensure that in one cycle of a sinusoid, upper half of the cycle goes into one stator and the lower half of the sinusoid goes into the bottom stator. In this way we push the desired function into the coils of the stator.

7. PRODUCT STRUCTURE

Part	Qty	Specification/Material	Machining/Vendor	Cost(INR)
Cover	1	Aluminium 6061	Milling/Drilling	NA
Stator	2	Cast Iron	Milling/Drilling	NA
Spacer Nuts	12	SS 306	Turning	NA
Linear Flexural Bearings	2	AISI 405 Steel	CNC Wire EDM	NA
Moving Iron	1	Cast Iron	Turning	NA
Connector	1	SS 306	Turning	NA
Cover Plate	2	Aluminium 6061	Cutting/Drilling	NA
Linear bush bearing	2	LMK8SUU	Hardware store	400
Solenoid wire	500 m	AWG19 wire	Sun Electronics,Gumti	1500
M4 bolts	20	Allen, countersunk	Hardware store	200
M4 bolts	20	Allen	Hardware store	200
M4 nuts	20	Allen	Hardware store	200
M3 bolts	30	Allen	Hardware store	300
M3 nuts	20	Allen	Hardware store	200
M6 bolts	10	Allen, countersunk	Hardware store	100
Current Amplifier			Machine Tool Dynamics Laboratory	
Diodes	10	MUR860	Sun Electronics,Gumti	100
Resistors	5	1KΩ	Sun Electronics,Gumti	100
Heat Sink	4	NA	Sun Electronics,Gumti	20
Fuse	4	5A	Sun Electronics,Gumti	50
GP Board	1	15 cm X 15 cm	Sun Electronics,Gumti	100

*All manufacturing will be done outside institute.

8. RESULTS AND ANALYSIS

The designed and the manufactured actuator vary in a few aspects:

1. Due to manufacturing constraints, the number of coils in each solenoid is 210 turns instead of 400. Also, the coils are not identical to each other. Hence the force generated by the actuator is reduced since the electromagnetic force is a function of the number of turns.
2. We could not maintain the intended air gap of 0.5 mm between the stator and the moving iron. So, we had to increase the gap to 1.5 mm. This leads to drop in the force generated by the actuator since the electromagnetic force is a function of the air gap.
3. The two springs are not aligned in a completely concentric way. So, the air gap is not constant at all points on the moving iron. This generates force in transverse directions which is undesirable.
4. The stiffness of the designed springs and the manufactured ones vary a lot. Due to this, the natural frequency has shifted from intended 30Hz to around 100 Hz for 3mm springs.
5. Finally, due to power constraints of the amplifier, the actuator could not be tested beyond 200Hz for 2.5 A current and 150 Hz for 3.5 A current.

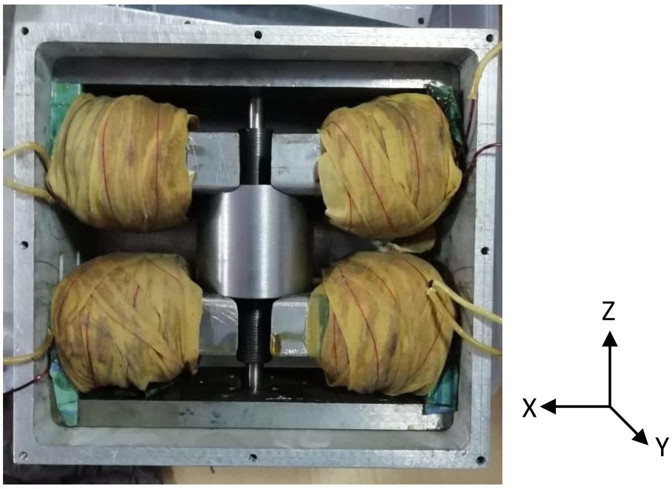


Fig 14: Axes of the actuator

As shown in fig. 14 , the z- direction is along the axis of motion of the moving iron. This is the direction in which we intend to extract the force. The x- direction is along length of the stator and y – direction is perpendicular to the other two axes. This convention for direction is used henceforth.

1. Force - Frequency characterisation of the actuator

Using an intermediate block , the actuator is mounted on the dynamometer. An input signal of given frequency and amplitude is generated either by a function generator or using LabView and fed into the amplifier. The amplifier amplifies the current to required current or voltage rating and then is sent into the actuator.

Starting from frequency of 15 Hz, we provide input to the actuator at 5 Hz intervals upto 120 Hz at a constant current rating of 2.5 A and the experiment is repeated for 3.5 A. This is chosen against direct sweep because it is observed that the actuator required a certain time interval to settle at any specified frequency.

In the fig. 15 and 16, the force-frequency characteristics is shown for the actuator using the springs of 1.5 mm thickness. Along z- axis, we can see that in both cases the force first increases upto a certain frequency , and then it is relatively constant and then it shows a peak at its natural frequency(≈75 Hz). Beyond that , the force falls again and then slowly becomes negligible. Similar trend is observed along y- direction.

However , the force along x- axis is almost constant for all the frequencies.

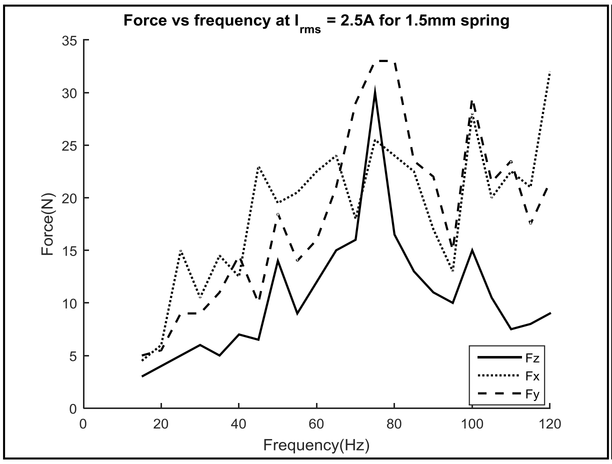


Fig 15: F vs. f at 2.5A for 1.5mm spring

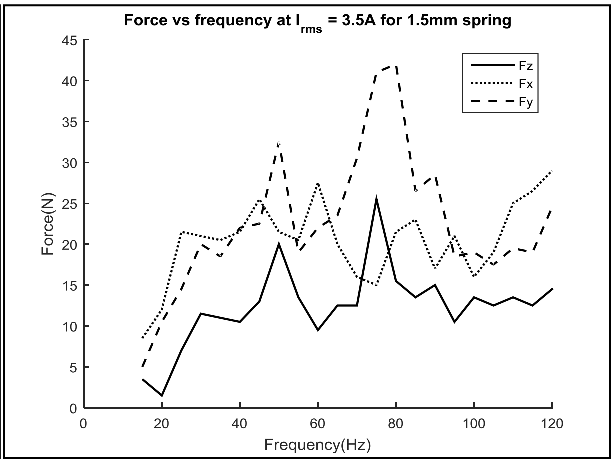
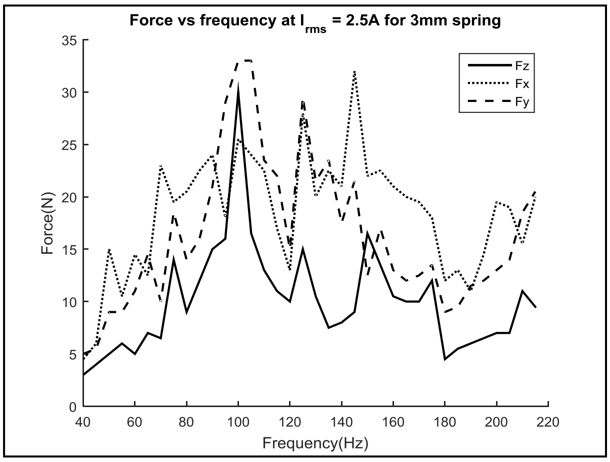


Fig 16: F vs. f at 3.5A for 1.5mm spring

We have similar trends when the 1.5 mm spring is replaced by the 3 mm spring. Now, we have the peak due to natural frequency at 100 Hz.



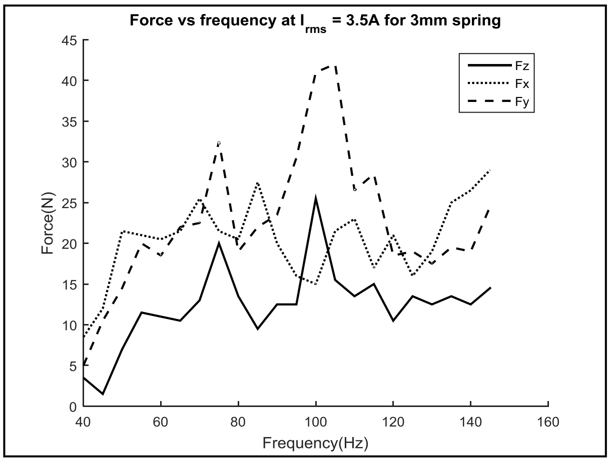
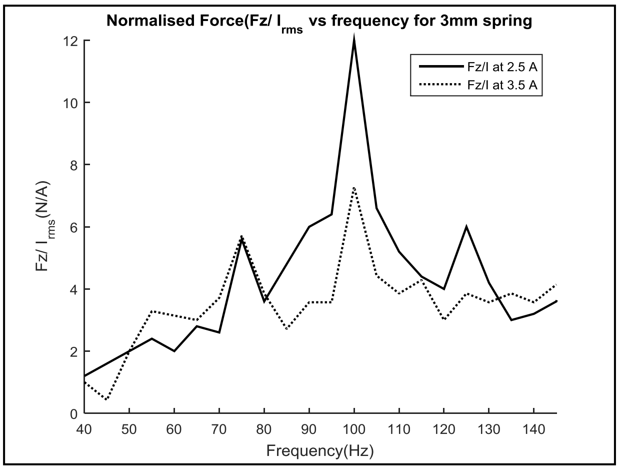


Fig 17: F vs. f at 2.5A for 3mm spring

Fig 18: F vs. f at 3.5A for 3mm spring

Fig. 19 and 20 shows the current-normalised force - frequency plot for both spring arrangements along z- axis. When we normalize the force by dividing the current applied , we see for both the cases that the two plots almost overlap each other. This shows that the system is slightly non-linear, as predicted by the model. Another thing that can be seen is the force is independent of the stiffness of the spring.



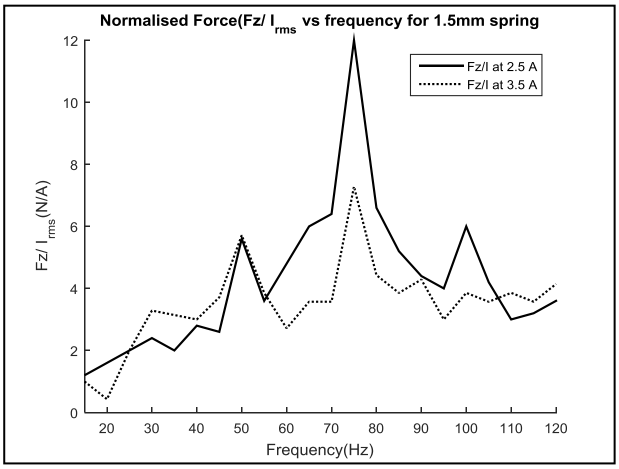


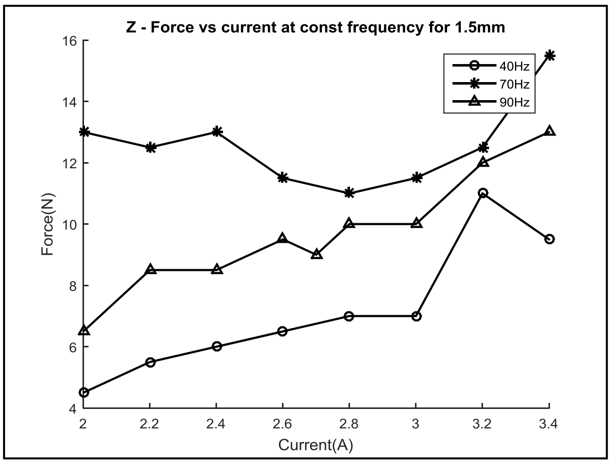
Fig 19: F_z /I vs. f for 3mm spring

Fig 20: F_z /I vs. f for 3mm spring

2. Force - Current dependence of the actuator

The frequency is now kept constant and current is increased from 2 A to 3.4 A in steps of 0.2 A and the force is measured in all three directions.

We see that force along z-direction increases with current. However, in case of 70Hz & 1.5mm spring, no such trend is observed. This maybe because 70Hz is the natural frequency of the system .However this explanation does not hold for 3mm spring at 60 Hz. Even though natural frequency is at 100 Hz, there is no clear relation between force and current at this frequency.



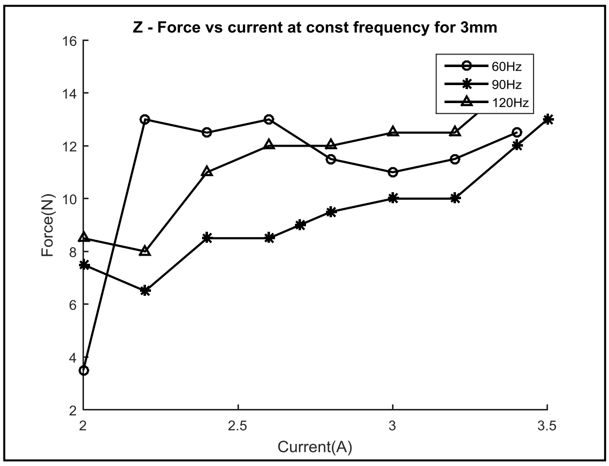
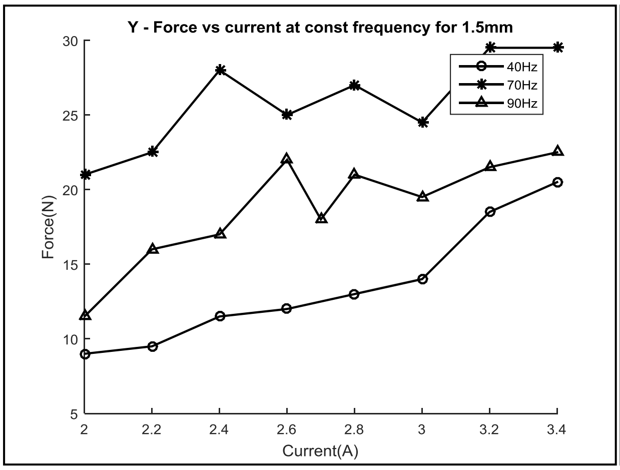


Fig 19: F_z vs. I for 3mm spring

Fig 20: F_z vs. I for 3mm spring

A similar trend is observed along y-direction. At 40 Hz and 60 Hz for 1.5mm and 3mm springs respectively, there is no clear correlation between force and the current. Also, we see that force is almost constant at 120 Hz for the 3mm spring arrangement.



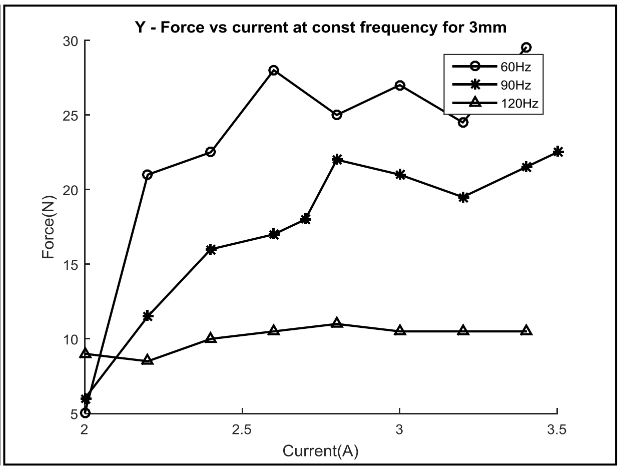
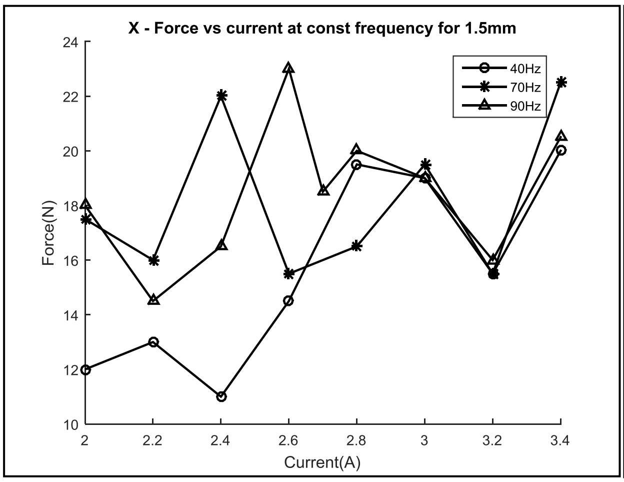


Fig 19: F_y vs. I for 1.5mm spring

Fig 20: F_y vs. I for 3mm spring

However, along x-direction, there is no clear correlation between the force and the current at any frequency . This suggests that the force component along x-direction is indeed from the misalignments,which leads to differing air gaps between moving iron and the stator at different points and hence is random.



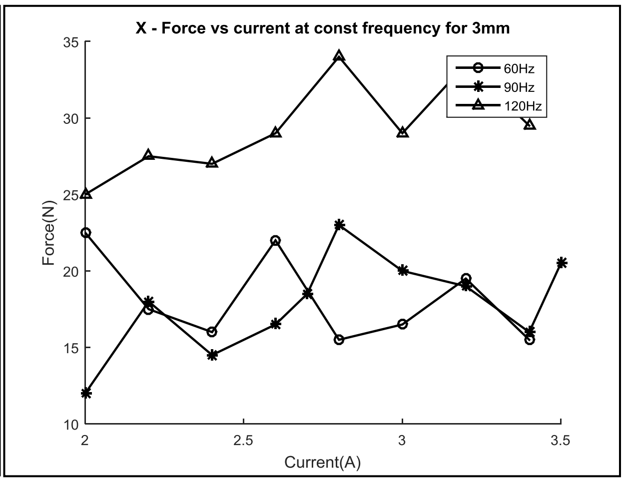


Fig 19: F_x vs. I for 1.5mm spring

Fig 20: F_x vs. I for 3mm spring

9. CONCLUSIONS

After conducting the tests on our actuator and comparing it with expectations from the model , the following conclusions are drawn:

1. As mentioned in the previous section, there were a few compromise in manufacturing of the actuator due to manufacturing constraints.Due to decrease in number of turns and increase in air gap , the force output of actuator has reduced substantially. The actuator now delivers about 13 N force in the frequency range of 30 – 140 Hz.
2. The force along z-axis and current show an approximately linear relationship. Also, the normalised force vs. frequency plots show that the system is slightly non-linear.
3. The force frequency characteristics suggests that there is an almost constant force output in the bandwidth of the actuator, except at the natural frequency.
4. There are unwanted force components along x and y directions. These are most probably caused due to misalignment between the moving iron and the stator,resulting in different air gaps at different points.
5. There is no heating as such when the current is limited to 3.5A (rms).After testing the actuator, it is observed that the body heats up after 90 minutes of constant use.
6. When excited at lower frequencies, the actuator vibrates at twice the frequency of the current. For e.g. when current input is at 30Hz, the actuator gives a force at 60Hz. However, this can be avoided by fixing the moving iron firmly onto the connecting rod using glue or arresting nuts.

10. FURTHER IMPROVEMENTS

In this project , we developed a first generation of the actuator. So, with a very simple design we got to know about some basic characteristics of an electromagnetic actuator with a moving iron. However, there are a lot of questions left to be answered. On the basis of what we learnt, we suggest the following should be implemented in the future iterations to better understand the working and get work output closer to expected values:

1. More care has to be taken in the manufacturing, regarding concentricity and maintaining the air gap. Since force output falls drastically with increase in air gap, it is very important to keep the air gap as small as possible and at the same time avoid the moving iron from interfering with the stator. Maintaining concentricity would help in reducing the transverse force components.
2. The design of the stator needs to be improved. It was very difficult to wind the coils around the stator. The new design should ensure that coils can be wound using the machine and not by hand. Also all the four windings need to be identical to reduce forces along the transverse direction.
3. Electric steel can be used to make the stator. It has high permeability which would give more force and has small hysteresis area which would reduce power loss. Also, the core can be made of laminates instead of being a single solid entity. This would reduce eddy current losses.
4. The connector rod has to be hardened. It was observed that the balls in the linear bush bearing indented the surface of the rod further compounding the problem of concentricity.
5. The stiffness of the spring can be reduced. Force output is independent of the spring stiffness. Hence a reduced stiffness would lower the natural frequency below 30 Hz.
6. More powerful amplifier can be used to increase current beyond 3.5A (rms). Increasing current would increase force output. Care has to be taken to use wire rated for higher current capacity.Also, cooling system needs to be incorporated if more power is being provided to the amplifier.

11. APPENDIX I

Code for processing transient analysis results from FLUX in Matlab.

```
%% Construct FRF between output - input given time domain signals
 clc; clear all; close all;
% Load the time domain input-output files provided to you
load actual1to100.txt ;
% The file given to you is organized as:
% 1st column - time;
% 2nd column - position (output);
% 3rd column - force (output);
% 4th column - current(input)
% Assign them in above order
k = 50;

time = actual1to100(:,1);
position = actual1to100(:,2);
 force = k*position;
%force = actual100to200(:,3);
current = actual1to100(:,4);

% Lenght of signal
L = length(time);

% Sampling Freq
dt = time(2) - time(1);
Fs = 1/dt;

% Define a freq. vector
f = Fs/2*linspace(0,1,(L/2+1)); % in Hz
w = (f*2*pi)'; % in rad/sec


% Take the Fourier transform using fft
fftpos = fft(position,L);
fftpos = fftpos(1:length(position)/2+1); % retain only the single sided
spectrum
fftforce = fft(force,L);
fftforce = fftforce(1:length(force)/2+1); % retain only the single sided
spectrum
fftcurrent = fft(current,L);
fftcurrent = fftcurrent(1:length(current)/2+1); % retain only the single
sided spectrum

FRFxf = fftpos./fftforce;


% plot the results
% mag
figure(1);
hold on;
plot(time,force);
xlabel('Time');
ylabel('Force');
```

```
figure(3);
hold on;
plot(time,position);
xlabel('Time');
ylabel('Position');

figure(5);
hold on;
plot(time,current);
xlabel('Time');
ylabel('Current');

% fftforce = smooth(fftforce);
figure(2);
hold on;
plot(f,20*log(abs(fftforce)));
xlabel('Frequency(Hz)');
ylabel('Force');
 xlim([1 100]);


% fftpos = smooth(fftpos);
figure(4);
hold on;
plot(f,20*log10(abs(fftpos)));
xlabel('Frequency(Hz)');
ylabel('Position');

% fftcurrent = smooth(fftcurrent);
figure(6);
hold on;
plot(f,(abs(fftcurrent)));
xlabel('Frequency(Hz)');
ylabel('Current');
```

12. APPENDIX II

Detailed spring design

Flexure bearing is an integrated concept for combination of functions of spring and bearing into one single entity. They are mostly used in the systems of piston and motor assemblies to support and oscillate with the piston. These flexure bearings are inexpensive and compact in nature. The radial stiffness of the spring in comparison to axial stiffness is much higher. These bearings have the shape like that of disk with different designs and patterns cut onto the disk. These distinct patterns result in variations between fatigue limits, yield strength and force displacement characteristics. We chose the model similar to that used in Oxford cryocooler, the three arm spiral profile with certain modifications on design part, not so unalike from the original one, to meet our target requirements.

In our theoretical analysis we first built different models of flexure bearing with the specified parameters in Solidworks 2016. Then using Finite Element simulation tool we meshed the models and obtained the desired force displacement characteristic, stiffness and fatigue limits.

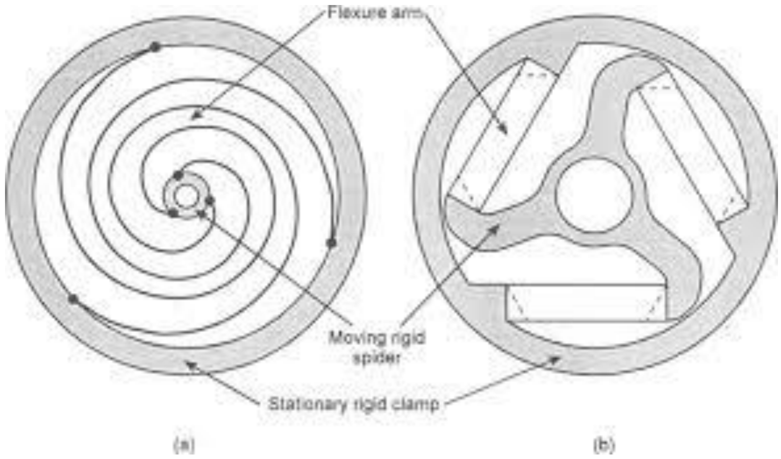


Fig 21 – (a) Oxford flexure bearing (b) Triangular Flexure bearing

1) Target Specification

We need to build a spring with an axial stiffness between the range 25000-30000 N/m, Outer diameter should not be greater than 50mm and the thickness should be less than 3mm. The bearing will be working at a force in range close to 100 N therefore we also have to analyse yield strength and fatigue limits for this range to ensure that spring will not yield or get fatigue during the course of functioning.

2) Fixed and Variable Parameters

In figure 22 there is a detailed description of different features of flexure bearing

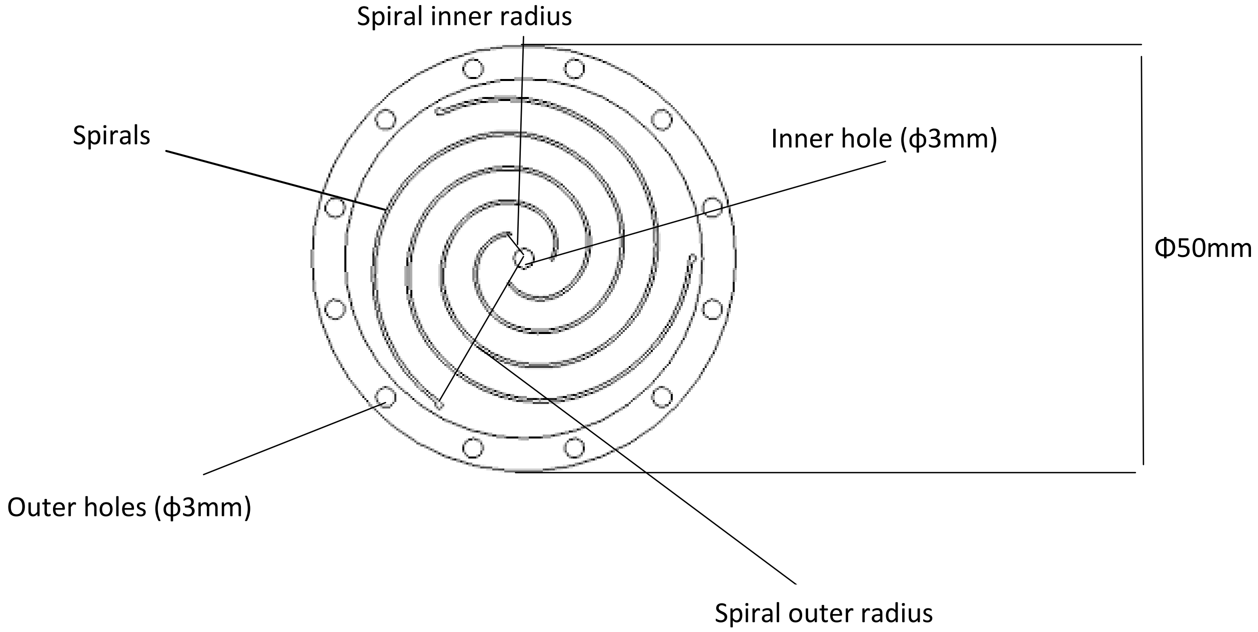


Fig 22: Features of a flexural bearing

I. Fixed Parameters

Some of the parameters that are fixed in all the models of bearing these are:

- Outer Diameter(D) - 50mm
- Inner and outer holes(d) - There is one inner hole and 6 outer hole with same diameter of 3mm. Inner hole is for mounting of connecting rod and outer holes will be fixed using nut and bolts.

II. Variable parameters

- Thickness of Disk(w)- The simulation is done for three different values of 1mm,2mm and 3mm.
- Slot thickness(t) - spiral slot thickness is varied for three different values 0.3mm,0.5mm and 0.7mm.

- Spiral pitch(p) and no. of spirals(N) – The equation for spiral curve is shown below

$$x = p * [\cos\left(t - n * \frac{\theta}{N}\right) + t * \sin\left(t - n * \frac{\theta}{N}\right)]$$

$$y = p * [\sin\left(t - n * \frac{\theta}{N}\right) - t * \cos\left(t - n * \frac{\theta}{N}\right)]$$

Here θ is constant and equal to 2π

N - number of spirals (varied for three values, 3, 6 and 9)

n is varied from 0 to N-1 and unique for different spiral curves

p - pitch (varied with different values)

3) Material Properties

The spring material is AISI 1045 Steel, cold drawn. Some of the properties of material are listed in table below:

Property	Value
Yield Strength	5.3e08 N/m^2
Tensile Strength	6.25e08 N/m^2
Elastic Modulus	2.05e11 N/m^2
Poisson’s Ratio	0.29
Mass Density	7850 kg/m^3
Shear Modulus	8e10 N/m^2
Thermal Expansion	1.15e-005 /Kelvin

4) Simulation Studies

I. ***How to model flexure bearing in solid works***

First build a solid disk and then extrude the inner and outer holes. Then using the curve tool put the equation of spirals according to desire no. of spirals. Using thin extrude cut the disk along the spiral curves, you will get a simple model of flexure bearing with specified parameters.

Fig 23: CAD of flexural bearing

II. ***Static study***

By using FEA simulation tool ,select new static study. Then select the material 1045 steel cold drawn, force 100 N along the axis of bearing at the central hole, set fixtures at the outer holes and run the study.

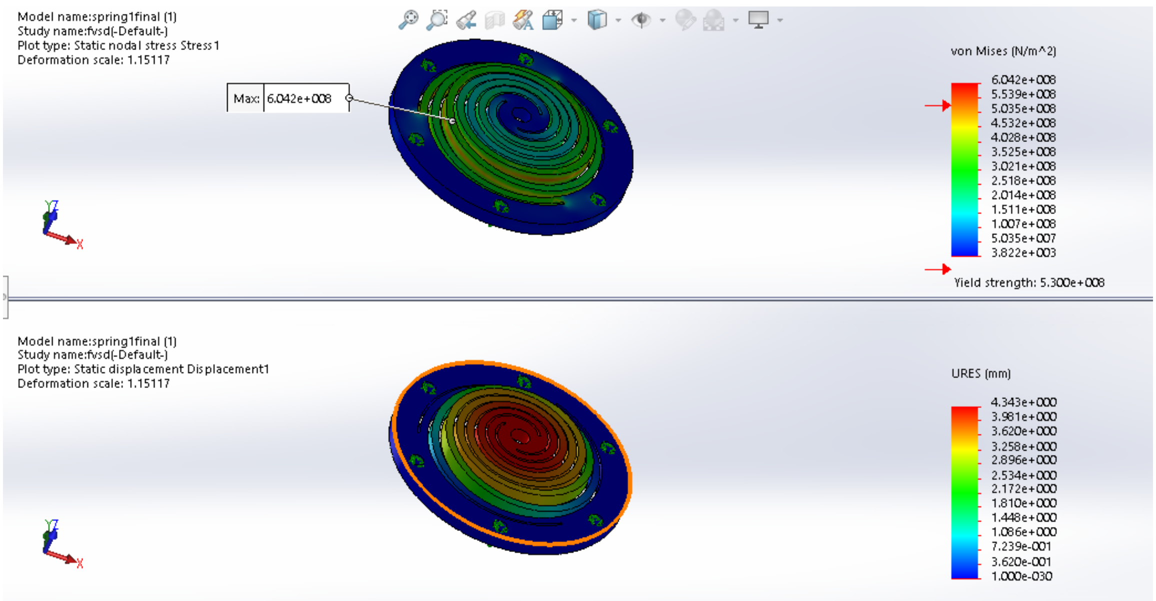


Fig 24: force displacement characteristics for parameters T=0.5, N=3, W=3 and P =1 in given units

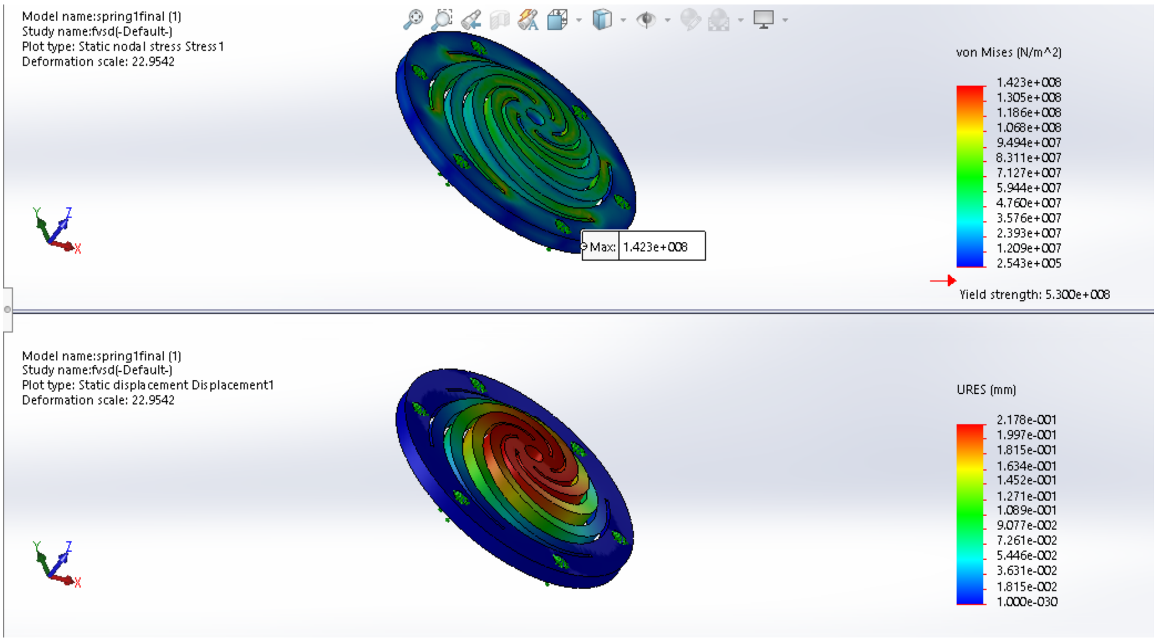


Fig 25: Force displacement characteristics for parameters T=0.7, N=6, W=3 and P =1.5 in given units

III. ***Fatigue study***

Repeat the process of selecting new study. Then select fatigue study, choose S-N curve for material, feed no. of cycles, put reversible force 100N and run the study.

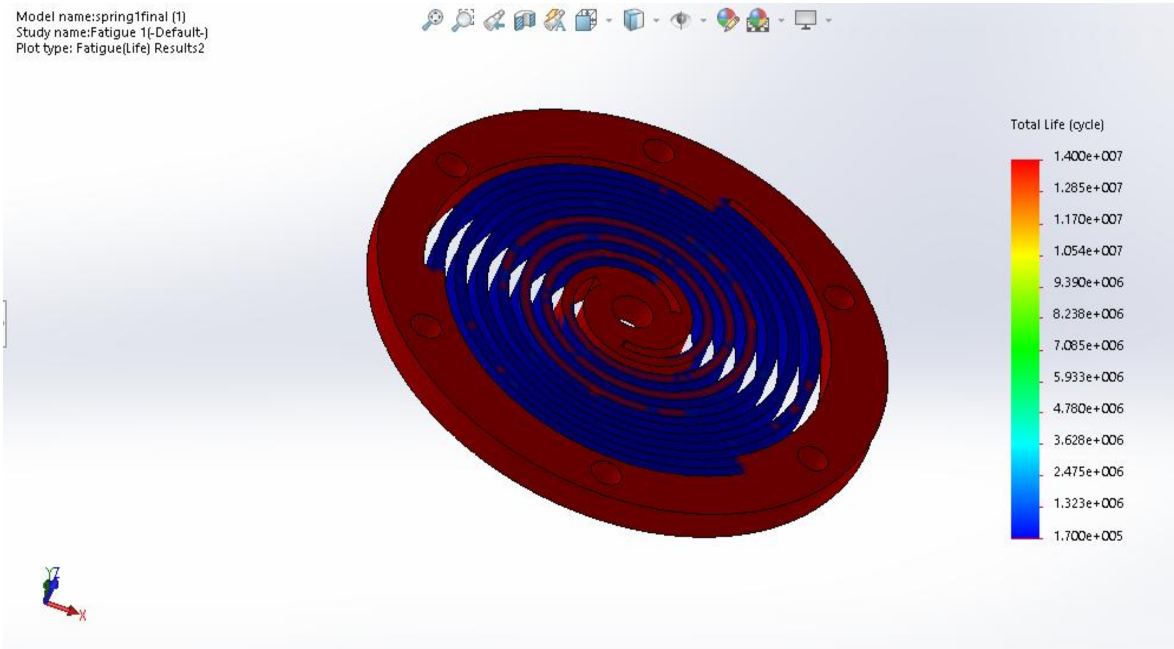


Fig 26: Fatigue characteristic for parameters T=1, N=3, W=3 and P =1 in given units

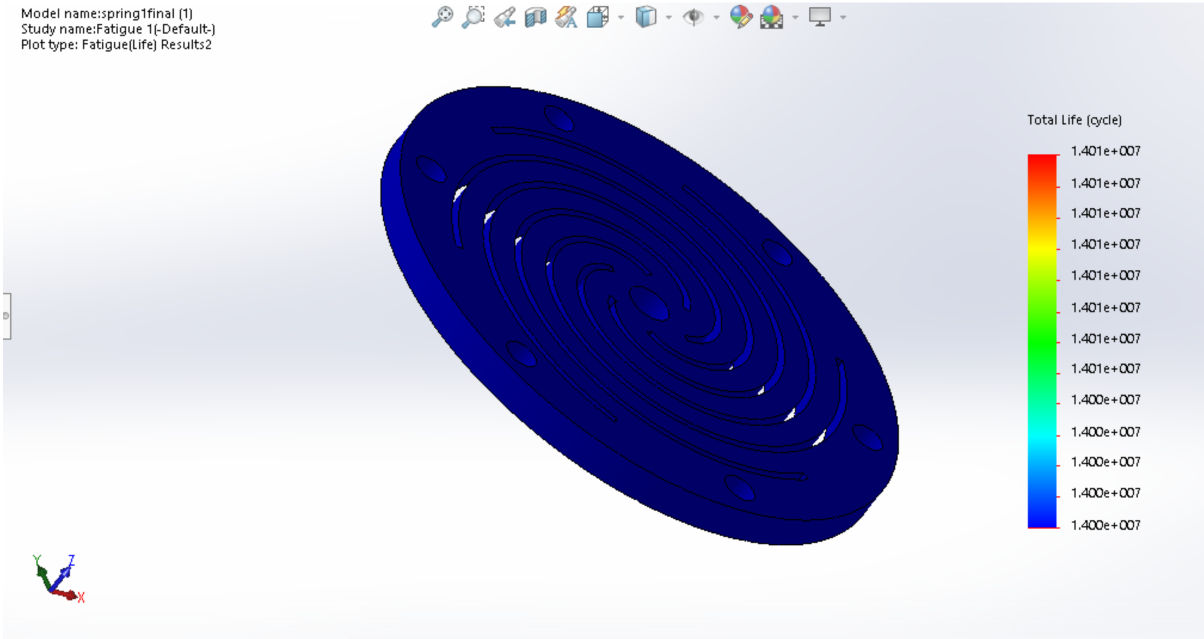


Fig 27: Fatigue characteristic for parameters T=0.7, N=6, W=3 and P =3 in given units

5) Results

No of Spirals	Spiral Pitch(mm)	Slot thickness(mm)	Disk Thickness(mm)	Max stress Concentration (N/mm^2)	Max Displacement(mm)
6	1.5	0.3	1	2.07E+09	18.06
6	1.5	0.7	1	3.00E+09	38.7
6	1.5	0.3	3	4.34E+08	2.709
6	1.5	0.7	3	9.08E+08	7.87
6	3	0.3	1	1.23E+09	2.47
6	3	0.7	1	9.52E+08	3.064
6	3	0.3	3	1.93E+08	0.16
6	3	0.7	3	1.42E+08	0.2178
3	1	0.5	3	6.04E+08	4.343
3	1	1	3	1.14E+09	11.84
3	3	0.5	1	1.11E+09	1.648
3	3	1	1	8.603+08	1.931
3	3	0.5	3	1.15E+08	0.08049
3	3	1	3	1.16E+08	0.095

6) Conclusion

From the above results the most suitable result are coming out to be in the third case. The stiffness in this case is coming out to be 33000 N/m which is very close to our target requirements. Further the max strength is below the point of yield strength and fatigue cycles limit are slightly above 20000.

No of Spirals	Spiral Pitch(mm)	Slot thickness(mm)	Disk Thickness(mm)	Max stress Concentration(N/mm^2)	Max Displacement(mm)
6	1.5	0.3	3	4.34E+08	2.709

7) References

1. Saurabh Malpani, Yogesh Yenarkar, Dr. Suhas Deshmukh, S P Tak, D.V. Bhope —Design OF Flexure Bearing For Linear Compressor by Optimization Procedure Using FEA‖, International Journal of Engineering Science and Technology (IJEST) Vol. 4 No.05 May 2012 pp.1991-1999
2. Fayaz H. Kharadi, Mayur S. Jadhav, Sachin D. Kanhurkar, Penelope A. Pereira, Dr.Virendra K. Bhojwani, Suneeta Phadkule- Selection Of High Performing Geometry In Flexure bearings For Linear Compressor Applications Using FEA, INTERNATIONAL JOURNAL OF SCIENTIFIC & TECHNOLOGY RESEARCH VOLUME 4, ISSUE 01, JANUARY 2015