Design OF Flexure Bearing For Linear Compressor By Optimization Procedure Using FEA

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Abstract:

Bearing are used to allow the relative motion between two surfaces. A shaft has to rotate about its casing or a piston has to slide about the cylinder. Both requires relative motion to happened least frictional losses. The flexural bearing however ,offers a different approach in supporting the bearing surfaces. The elements of bearing surfaces are deformed on application of load to one of the surfaces, allowing the relative motion between the two surfaces on removal of the load ,the surfaces go back to their original position subjected to condition that caused deformation of the bearing element due to applied load is within the limit of elasticity. This eliminates the wear ,vibration and frictional losses. However ,the deformation has to be limited. The precision and micro machining applications and some medical applications very low relative motion. Hence, flexural bearing in this kind of application is a better.

The present work is specific to the typical flexure bearing used in linear compressor. Since the flexural bearing designed procedure is not available, this paper proposes to the FEM as a tool to find the axial stiffness that would be offered by a typical flexure used in the linear compressor application for cryocooler. The cryocooler has a linear compressor use for compressing a gas with a typical displacement of 5 mm. The typical design has the flexural bearing with spiral cuts in the flexures. These spiral cuts allows each of the flexure to move itself axially on application of load in the axial direction. There are two states of flexures called stacks on either side of linear motor supporting a piston rod which moves the piston either side i.e. back and forth in gas displacer. As mentioned typical displacement of the piston is 5mm causing each of the flexure to get deformed by same amount. Since there is no any standard method available for calculating axial stiffness of bearing ,we have consider an example of disc.

Keywords: Cryocooler Flexure Bearing, FEA by Ansys 11, Testing

1. Description

A typical unit of a flexure suspension system used in linear compressor is shown in Figure 1. Each unit is in the form of a thin flat metal disc having three spiral slots, yielding three spiral arms which bear the radial and the axial loads. Each spiral sweeps an angle of 480o.The outer diameter of disc is 69 mm and P.C.D. of outer clamped holes is 61 mm. The central hole is having 3 mm diameter while outer clamped 12 holes having diameter 6 mm and thickness of disc is 0.3 mm. The spirals are situated 10 mm apart from centre of disc. The spiral arm is having equal to thickness of 0.5 mm.

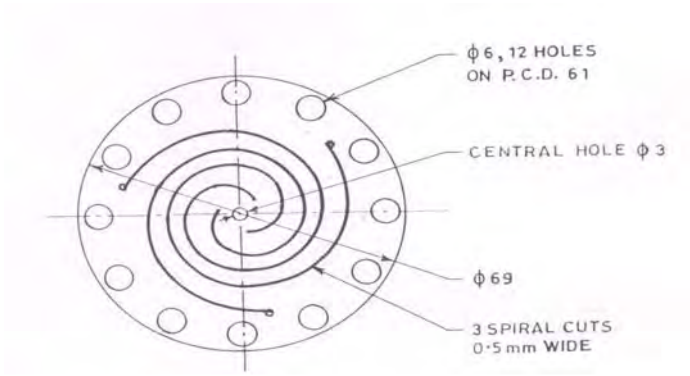


Fig.1 Typical Flexural Unit For Linear Compressor bearing

The present work is specific to the typical flexure bearing used in linear compressor. Since the flexural bearing designed procedure is not available, this project proposes to the FEM as a tool to find the axial stiffness that would be offered by a typical flexure used in the linear compressor application for cryocooler. The cryocooler has a linear compressor use for compressing a gas with a typical displacement of 5 mm. The typical design has the flexural bearing with spiral cuts in the flexures. These spiral cuts allow each of the flexure to move itself axially on application of load in the axial direction. There are two states of flexures called stacks on either side of linear motor supporting a piston rod which moves the piston either side i.e. back and forth in gas displacer. As mentioned typical displacement of the piston is 5mm causing each of the flexure to get deformed by same amount. A good flexure shall allow this with least possible resistance that is flexures which have least axial stiffness would be best flexure. Therefore the different flexure design by varying thicknesses and angle of spiral analyzed using FEM. The axial stiffness for each case is calculated and the best choice is chosen best on the axial stiffness criteria and equivalent stress criteria. Further experimentation has been carried out to validate the results. To summarize the typical scope involves

- Geometric modeling of disc having hole at centre.
- FEA Of Disc having hole at centre.
- Geomentric modeling of flexure.
- FEA Analysis of flexures by applying different loads and boundary conditions.
- The load is chosen best on the deformation criteria

2. A Sample Problem OF Deflection OF Disc

Flexural bearing under investigation in the project is made up of stack of flexures each of which is a disc having hole and the spirals hence ,it is apt to analyze a sample disc using the typical software to be used for further analysis. It should be noted that Timoshenko [8] has given the empirical formulas for finding the axial deflection of disc with hole if the axial load is applied on the periphery of hole with the outer boundary of the disc is simply supported.Fig 4.1 shows a typical disc with hole for which parameter ‘a’ and ‘b’ have been defined.

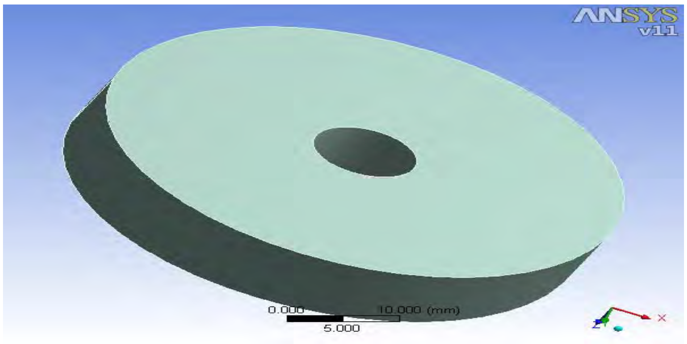


Fig 2 Disc Having Hole At Centre

The Timoshenko have deduced the formula for finding the deflection of disc which is as follows

Wmax=k1*q*a4/E*h3

It was decided to analyze the disc with hole of the following specifications using analytical as well as FEA approach.

Outer diameter a=25 mm

Inner diameter b = 5 mm

Material Used = Copper

Loading Point = At center peripheral surface

Force Applied = 5 N

Analytical Solution

As par Timoshenko deflection above referred disc can be found as

Wmax=k1*q*a4/E*h3

1) By using book theory of plates and shale's by TIMOSHENKO second edition and from page no.-62,63, we are calculating following results.

2) k=a/b=25/5=5

3) P=F/A

F=P*A

5=P*2 ∏ R t

5=P*2*3.14*5*5

P=0.0318 N/mm2 -----------(Answer)

4) Wmax=k1*q*a4/E*h3

Wmax=0.238*0.0318*254/1.1*105*53

Wmax=0.00021501 mm ----------(Answer)

Solution Using FEA

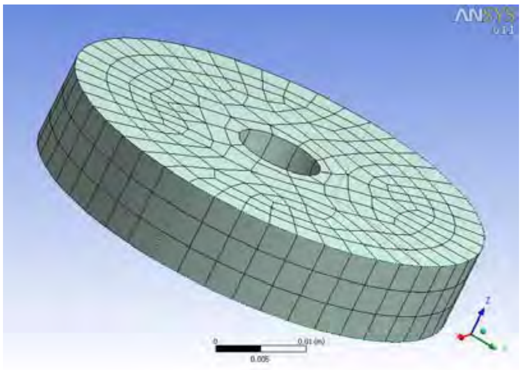
The geomentric modeling created in the CATIA and IGES file was imported to ANSYS 11.Further file is loaded in ANSYS.

FEA Modelling

The disc model is descretized into so many number of elements by using 20 node solid 186 element shown in fig 3.Then same was constrained by fixing the periphery loads and the pressure along the axial direction was applied.Pressure to be applied is calculated

P=F/A=F/2πRt

Fig 4 shows disc model with load and boundary condition applied.The force was varied from 1 N to 5 N



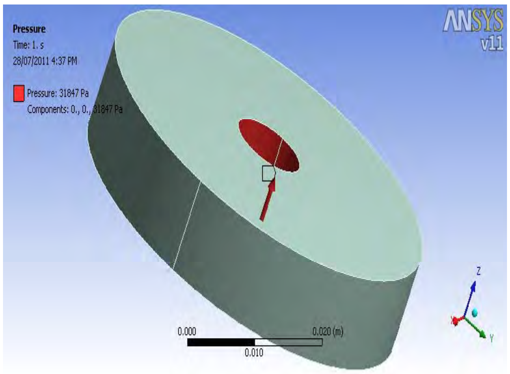


Fig 3 Disc Model With Meshing

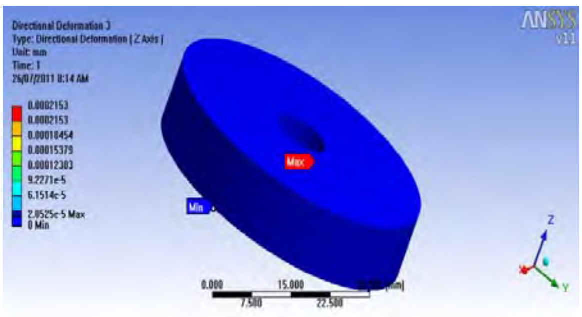
Fig 4 Disc Model With Load

Table 1: Pressure Calculation

Sr.No.	Force(N)	Pressure(F/2 πRt)
1	1	0.00636
2	2	0.0127
3	3	0.0019
4	4	0.0254
5	5	0.0318

Analysis And Results

The FEM model was analyzed by ANSYS 11 workbench and following are the results of axial deflection obtained from various load condition



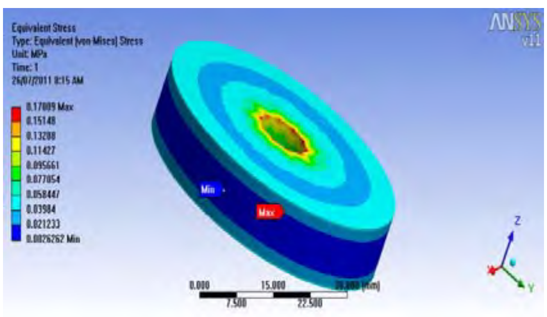


Fig 5 Z-Directional Deformation

Fig 6 Equivalent Stress

Table 2 : For Axial Theoretical And FEA Deformation Results

F(N)	Pressure (F/2 πRt)	Axial Deformation Analytical Results	FEA Deformation Results
1	0.0063	0.0000430	0.000044
2	0.0127	0.000085	0.000086
3	0.0019	0.000120	0.000135
4	0.0254	0.0001717	0.000187
5	0.0318	0.00021501	0.0002153

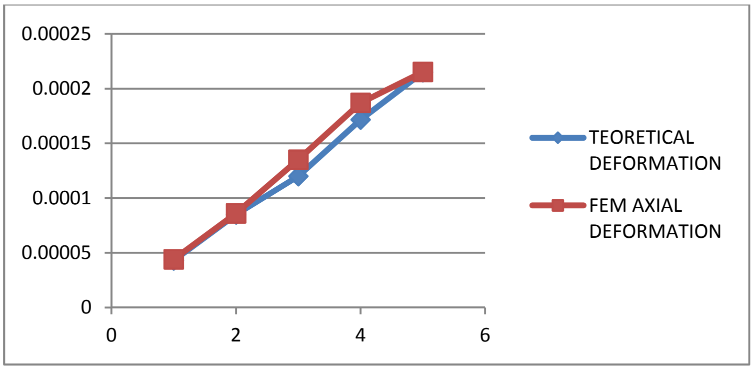


Fig 7 Force Verses Axial Deformation Theoretical And FEM

Discuss from the Fig. 7 it is evident that of disc under investigation has FEA deformation same as theoretical calculation as mentioned in formula. Hence, it can be concluded that it is possible to analyze a disc with hole and hence a flexure using 20 node solid 186 element used.

3. Modelling And Analysis Of Flexure Bearing

The very first step of modeling for FEM is a creation of geometric model. For the advantages of working and competence and handling, it was decided to use CATIA as geometric modeling software. The flexure disc was created using CATIA V-5 R-10 .

To create flexure, first go to part design. Then select plane. Click sketcher WB (workbench) icon. By using circle command, draw circle of required diameter that is 69 mm. Then constraint the circle. Also draw twelve holes along the periphery of circle having required diameter equal to 6mm.Also draw a hole at centre having diameter equal to 3 mm. Exit from WB .Pad circle by 0.15 mm mirror extend i.e. on both side. So, that it will get required thickness of 0.3 mm. For making 12 hole along periphery use pattern command in which give instances and mentioned angle. Similarly, create one hole at centre, use hole command and hole should positioned and set the option up to next. It will get required hole. The typical task while making model of flexure bearing was to make spiral. These spirals have situated 120 degree apart from each other and 10 mm apart from center of disc. To make spiral select shape. Then go inside GSD Workbench (Generative Shape Design).Use spiral command from a icon at lower side. Then use translate command to rotate spiral along particular axis (suppose y axis) such that each spiral will be at 120 degree apart from each other. Lastly, by using pocket command, select thick and up to next such that pocket of 0.5 mm create .

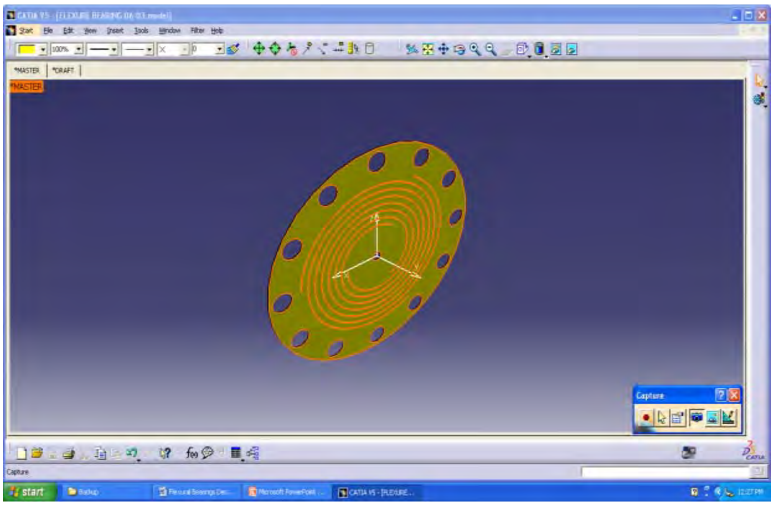


Fig 8 CATIA Model For Flexural Bearing

Material And Element Selection

For the required case of ANSYS copper was chosen and the element chosen was 20 node solid 186 as justified in the previous chapter. Following table shows the material properties. The ISO of copper and copper alloy is ISO 1190-1:1982

Meshing

The geometric model is meshed in ANSYS 11 workbench. Hence, it was decided to create maximum possible elements as offered by the default assistant ANSYS. Table shows the no. of nodes of element while Fig 9 shows meshed flexure.

Elements 6584

Nodes652

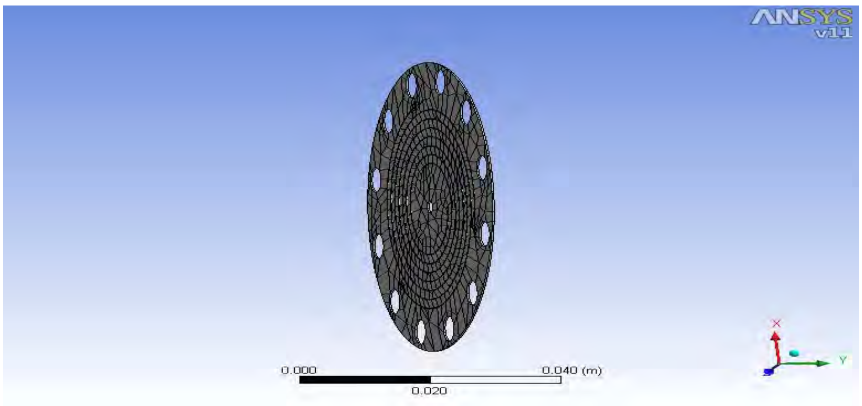


Fig .9 Meshing of Flexure Bearing

Table 3: Force and Deformation for Spiral Angle 480° and Thickness 0.30 mm

Pressure(MPa)	Force(P*2πRt)	Axial Deformation(mm)
0.004800	0.01356	0.44669
0.009600	0.0271	0.983
0.01500	0.04239	1.537

0.030	0.08478	2.561
0.03840	0.1085	3.93
0.045	0.12717	4.611
0.049	0.138474	5.028

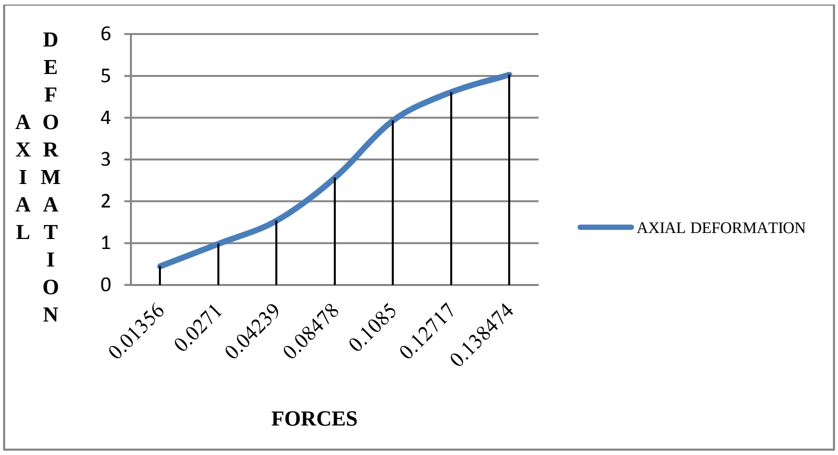


Fig 10 Axial Deformation

The axial deformation for applied force has been plotted in fig. 10 which indicates nonlinear variation of the deflection for the applied force. Initially the rate of deflection is high which further increases. When the flexure opens more. However ,ones it has opened ,the rate of deformation falls almost exponentially. Dropping the axial deformation par unit force applied for the higher loads as shown in fig 10.Beyond 0.1085 N ,there is significant drop in the axial deformation for the increment in applied load.ion procedure

4. Optimization Flexure Bearing

The similar study was done for the various cases which included varying thicknesses from 0.15mm ,0.30 mm,0.45 mm,0.60mm for the distinct spiral angles of 480°,600°,720°.The analysis of all this cases shows the similar pattern of deflection ,equivalent stresses and axial stiffness for applied forces. Table shows the forces obtained for the axial displacement of 5 mm for flexure disc. These cases have shown in table by varying angles and thickness. By using table a graph is plotted in between cases and axial stiffness and equivalent stress to conclude optimization results. It has been observed that flexure having more spiral angle and less thickness have least axial stiffness as well as less equivalent stress.

Table 4 :Different Cases For Varying Spiral Angle And Spiral Thickness

CASES(spiral angle*thickness)	PRESSURE(MPA)	FORCE (N)	DISPLACEMENT (mm)	EQU.STRESS MPA	AXIAL STIFFNESS
1-480*0.15	0.0125	0.01766	5	30.53	0.003532
2-480*0.30	0.0490	0.01738	5	58.15	0.0034
3-480*0.45	0.105	0.445	5	80.85	0.089
4-480*0.60	0.182	1.02	5	103.72	0.204
5-600*0.15	0.0092	0.01299	5	26.548	0.002598
6-600*0.30	0.134	0.04741	5	81.166	0.094
7-600*0.45	0.078	0.33064	5	67.158	0.06661284
8-600*0.60	0.13	0.73476	5	84.541	0.146952
9-720*0.15	0.007	0.00989	5	19.689	0.00197
10-720*0.30	0.099	0.03538	5	80.858	0.007077
11-720*0.45	0.060	0.25010	5	63.711	0.050202
12-720*0.60	0.100	0.55954	5	81.675	0.11190

Observation And Conclusion

From above table it has been observed that bearing having angle 720° and thickness 0.15 mm have less stresses and least axial stiffness. So, from case 1 and case 2 it has been concluded that bearing having less thickness have less axial stiffness. But when case 3 studied it has been observed bearing having more spiral angle should have less axial stiffness. By combining all three cases we conclude that we have to choose a bearing for design having less thickness and more spiral angle. To conclude this ,it has been drawn a graph below cases in x-axes verses equivalent stress(MPa) and axial stiffnesses(N/m) in y-axes. From graph, it has been again clear that case no 9 have least axial stiffness and less equivalent stress.

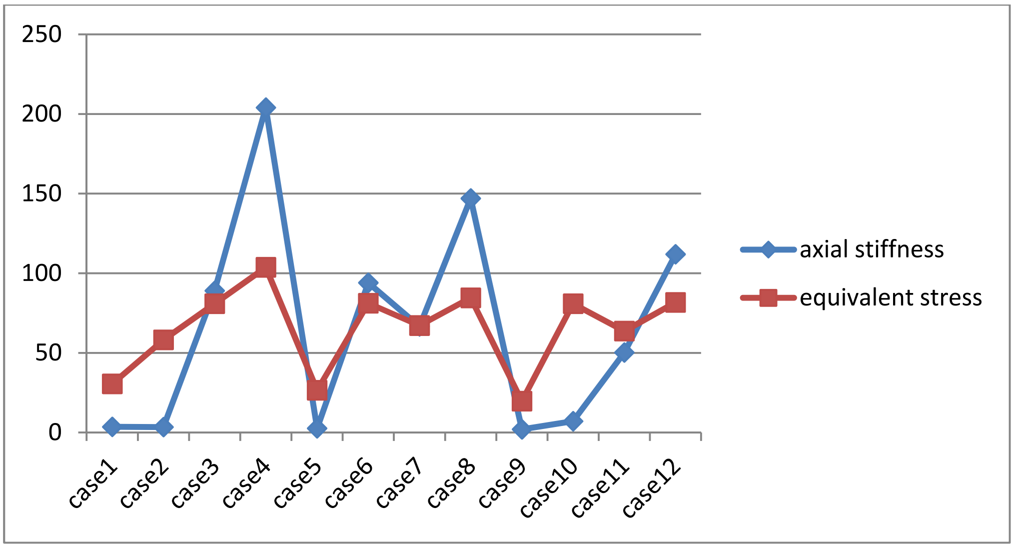


Fig 11 Cases Verses Axial Stiffness And Equivalent stress

4. **Testing OF Flexure Bearing**

Fig 11 shows the pictorial view of set up prepared to test the deformation of flexures on application of load. The flexure is fixed at the periphery boundary by clamping in the fixed plates .As depicted in the picture, a thread run over the pulley with the pan suspended at the another end was used to apply the varying loads on the flexure.

A height gauge with least count of 0.02 mm was used to measure the deformation of flexure . Graph in fig 12 shows the variation of observed deflection against the applied force. The applied load varying from 5 gram to 18 gram as the weight of pan is 4 gram

Fig 11 Experimental Set – Up With Height Gauge

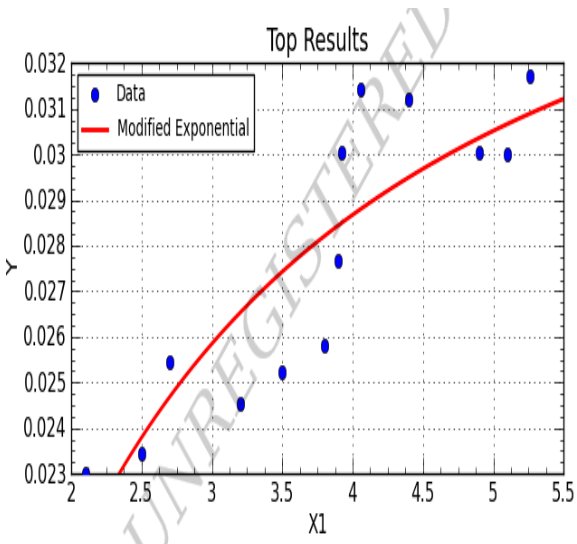


Fig 12 Graph Between Axial Deflection And Axial Stiffness

The Graph in fig 12 shows the variation of axial stiffness against the deflection. This graph obtained experimentally is also non linear as obtained in FEM. However the variation of pattern of two graph is inherent limitations of experimentation like

1)It is not possible to apply exact force used in FEM.

2)Weight of pan indispensible hence least possible load given is 5 gram

3)There are imperfection in the measurement even though height gauge has been used.

4)The hysteresis loss is not accounted as it is not possible to a stepwise increment and decrement of load.

However despite of the inherent limitation of the experimentation the overall stiffness variation pattern similar to that of FEA analysis.

References

[1] A.S. Gaunekar, T. Giiddenhenrich and C. Heiden , "Finite element analysis and testing of flexure bearing elements ",cryogenics 1996 volume 36, Number 5,page 359-364
[2] N. Chen a, X. Chen b, Y.N. Wu b, C.G. Yang a, L. Xu a, " Spiral profile design and parameter analysis of flexure spring", cryogenics 46(2006),page 409-419
[3] A.S.Gaunekar, G.P.Widdowson,N.Srikanth, Wang Guangneng, "Design and development of a high precision lens focusing mechanism using flexure bearing"
[4] H .P. Luo a, B. Zhang a,b,*, Z.X. Zhou, "A rotary flexural bearing for micromanufacturing",CIRP Annals- Manufacturing Technology 57(2008),page 179-182.
[5] B.J.Haung , Y.C. Chen," System dynamics and control of a linear compressor for stroke and frequency adjustment" ,transactions of the ASME,Vol.124,MARCH 2002.
[6] Finite elements in Engineering,Pearsons Education,By T.R.Chandrupatla,A.D.Belagundu
[7] Practical finite element analysis ,finite to infinite publications,pune,By Nitin Gokhale,Sanjay Deshpande,
[8] Theory Of Plates And Shells by Timoshenko,second edition, Mc Graw Hill publication ,page no.62,63.
[9] A.S.Gaunekar, K.G.Narayankhedkar and S.L.Bapat, " Dynamics and thermodynamic analysis of doubly motorized miniture stirling cryocooler using double coil linear motor" , cryogenics 1994,volume 34,Number 1, page 43-50