MODIFICATION OF SUSPENSION SYTEM OF EXISTING LINEAR COMPRESSOR USING FINITE ELEMENT ANALYSIS

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Abstract: Linear compressor is extensively used in cryo cooler to cool some application to cryogenic temperature because of its high electrical efficiency. Linear compressor employs specialized bearings called flexure bearing to support reciprocating mass. Compressor under study has 28 flexure discs to satisfy design constraints. These many flexure discs not only increase cost of manufacturing but also make compressor difficult to assemble. Main focus of present study is to reduce number of disc used in compressor keeping design constraints constant. Present study is carried out using finite element analysis. Effects of various geometrical parameters of spiral geometry on axial stiffness are also studied in this study.

***Index terms* – Flexure disc, stiffness, FEA, start and end radius.**

I. INTRODUCTION

Cryo cooler is used to cool application to cryogenic temperature. It employs linear compressor because of its enhanced electrical efficiency. Schematic diagram of linear compressor is shown in figure 1. Linear compressor as shown in figure has strong cylindrical magnet. Magnet has annular air gap. Coil is placed inside annular gap of magnet when coil exited produces axial force according to Fleming's Left hand rule. This force is used to compress gas inside compression chamber with help of piston. Piston is supported by Specialized bearing called Flexure bearing. When piston reciprocates these flexure bearings deform in direction of piston movement but they will limit self-deflection of moving mass attached to piston. Due to application of flexure bearing very small radial clearance is maintained between piston and cylinder. Due to this wear free clearance, reliability of compressor increases. Linear compressor does not require oil for its lubrication. As oil is not used in in compressor different types of refrigerant can be used with various temperature range. Schematic diagram of opposed piston linear compressor is given figure 1.

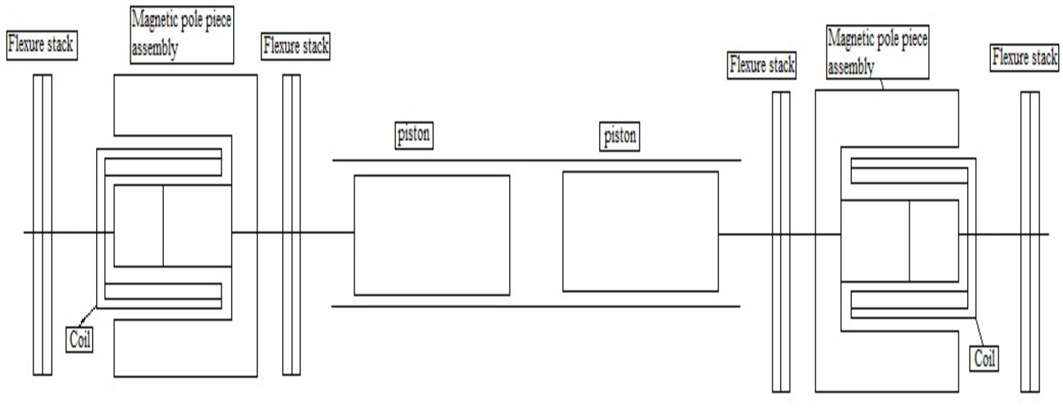

Figure 1: Schematic diagram of linear compressor

Flexure bearing has to draw minimum power input for their efficient use. They should have low axial stiffness so linear motor of compressor has to exert less force for piston to reciprocate. They should have high radial stiffness to offer minimum self-deflection. Existing compressor present at Walchand College of Engineering Sangli has power requirement of 100 watts. It has stroke of 5mm. It radial clearance maintained in cylinder bore is of 15 micron. Suspension system required for this compressor is designed using equations given in reference [1].

Design axial stiffness of suspension system is 7.291 N/mm. Design radial stiffness of Suspension system is 181.82 N/mm. To satisfy above design requirements present compressor employs 28 flexure disc.it has two stacks of 7 flexure bearing placed on either side of compressor. These many flexure discs not only increases cost of manufacturing of compressor but also they make compressor difficult to assemble. Main focus of present modification is to reduce number of flexure discs keeping design parameters constant.

II. MODIFICATIONS USING FINITE ELEMENT METHOD

2.1 Scope of modification

The design parameters for flexure bearings are geometry of profile, thickness of flexure bearing, diameter of flexure bearing, material and the total number of discs in stack. These parameters affect the flexure bearing characteristics. Out of this parameter, geometry of profile plays very important role in deciding axial and radial stiffness as well as fatigue strength. The profile of the bearing is shown in figure 2.

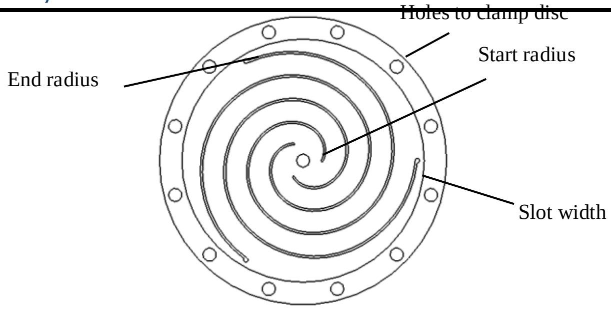

Figure 2: Spiral profile

To achieve total axial stiffness the number of flexure disc required 'N' is given by,

$$N = \frac{\text{total Axial stiffness}}{\text{axial stiffness of each disc}}$$

Effective axial stiffness of present flexure disc is 0.521 N/mm based on previous design calculations. Scope for modification is tabulated in following table.1

Table 1: Scope for modification

Sr. no.	Number of flexure discs on each side of compressor	Required axial stiffness in N/mm
1	14	0.521
2	12	0.608
3	10	0.729
4	8	0.911
5	6	1.215

For present modification number of flexure disc to be used in the compressor are selected as 10. Required axial stiffness of each flexure disc is 0.729 N/mm. Any further reduction of flexure discs would increase axial stiffness of each disc considerably.

2.2 Procedure of finite element analysis

The finite element analysis procedure consists of three steps which are as follows:

1. Pre-processing **-** This is first step of finite element analysis. This consists of development of models which can be carried out in any modelling software. In our case it has been carried out in CATIA v5. Then material selection needs to be carried out. Next process contains descretization of model. This can also be done in any meshing software. In our case ANSYS workbench itself is used to descretize the models.
2. Solution **-** This is second step of finite element analysis. This consists of applying loads and boundary conditions at appropriate location.
3. Post-processing **-** This is third and last step of finite element analysis. In this step results are found out. These results are then reviewed using different contour plots of stress and strains. Results can also be plotted in tabular and graphical form.

ANSYS workbench (Static structural analysis) is used to carry out whole analysis procedure. The reason behind using this software for analysis is that it is very adaptable.

Material chosen for the analysis was Beryllium copper which has young's modulus of 1.3×10^5 MPa. Poisson's ratio was taken as 0.3. Tensile strength of Beryllium copper is 1080 N/mm^2. Meshing has been carried out in workbench. Workbench has used Hex 20 element for the descretization. The next step of analysis is solution. In this step boundary conditions and loading conditions needs to be applied. In our case clamping holes need to fixed and we have to give displacement in X direction and Y direction. As we are giving displacement in both directions it is called as biaxial loading. This is shown in Figure 3. As we can see in Figure 3 fixed support is shown by blue colour around the clamping holes. Displacement of 5 mm was given in X direction and -0.015 mm was given in Y direction. This displacement was applied at inner circle face. Element size 0.6 is selected for analysis as this element size will not create much deviation.

In last step of analysis result was find out. Model was checked for von-mises strains and von-mises stress. It is also checked for force in X direction and force in Y direction which will help us to find out axial stiffness and radial stiffness respectively. Model after deformation and von-mises stresses developed. Loading conditions are shown in figure 3.

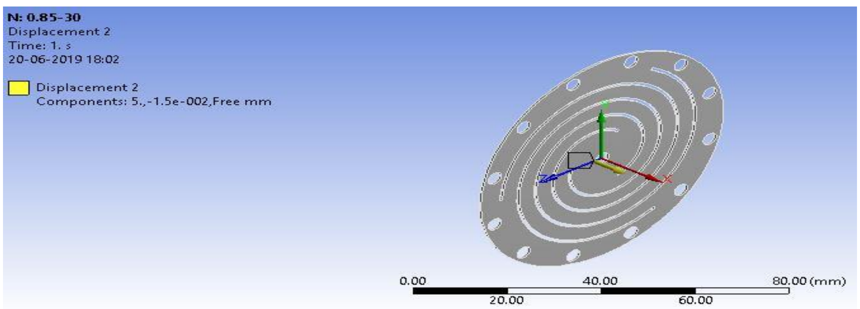

Figure 3: Loading conditions

III. RESULTS AND DISCUSSION

3.1 Thickness modification

Study is carried out check effects of variation of thickness of spiral profile on axial stiffness. Results are tabulated in table 2.

Table 2: Thickness analysis

Serial no.	Slot width in mm	Thickness in mm	Von-mises stress in MPa	Axial stiffness N/mm	Radial stiffness N/mm
1	0.69	0.4	152.6	0.264	60.966
2	0.69	0.5	362.9	0.548	79.213
3	0.69	0.6	411.7	0.936	95.060

Result shows for 0.6 mm thickness axial stiffness is 0.936 N/mm so this value is justifies thickness of profile selection.

3.2 Start radius modification

Study is carried out check effects of start radius of spiral profile on axial stiffness. Results are tabulated in table 3.

Table 3: Start radius analysis

Start radius	Von-mises strain	Von-mises stress in MPa	Axial stiffness N/mm	Radial stiffness N/mm
5mm	0.00296	383.6	1.198	147.953
6mm	0.00230	298.5	1.075	122.300
7mm	0.00191	251.9	0.970	100.287
8mm	0.00162	208.3	0.877	81.600
9mm	0.00147	187.4	0.784	65.802

As we can see from above table spiral profile having start radius of 9mm yields axial stiffness near to required value so start radius of 9mm is selected for spiral profile

3.3 End radius modification

Study is carried to check effects of variation of end radius of spiral profile on axial stiffness of flexure disc. Spiral end radius analysis is shown in table 4.

Table 4: End radius analysis

End radius	Von-mises strain	Von-mises stress in MPa	Axial stiffness N/mm	Radial stiffness N/mm
29mm	0.00131	167.20	0.656	46.494
28mm	0.00128	164.92	0.579	36.442
27mm	0.001148	147.66	0.509	28.270

As we can see from above table spiral profile having end radius of 29 mm to 27mm yields axial stiffness less than required value so we can conclude that by varying end radius will not satisfy design criteria.

3.4 Slot width modification

Study is carried to check effects of variation of slot with of spiral profile on axial stiffness of flexure disc. Spiral slot width analysis is shown in table 5.

Table 5: Slot width analysis

Slot width	Von-mises strain	Von-mises stress in MPa	Axial stiffness N/mm	Radial stiffness N/mm
0.85mm	0.001836	233.95	0.772	63.937
0.80mm	0.001477	187.46	0.784	65.802
0.75mm	0.001942	247.50	0.785	67.613
0.7mm	0.002010	256.26	0.791	69.473

As we can see from above table spiral profile having slot width yields axial stiffness near to required value so slot width of 0.85 mm is selected for spiral profile.

3.5 Optimum parameters of profile

From above analysis we found following optimum parameters for modified spiral flexure profile. Optimum values of spiral profile are tabulated in table 6.

Table 6: optimum parameters of spiral profile

Serial no.	Parameter	Value
1	Material	Beryllium copper
2	Disc thickness	0.6mm
3	Spiral angle	480^0
4	Spiral slot width	0.85mm
5	Spiral start radius	9mm
6	Spiral end radius	30mm
7	Diameter	73mm

After incorporating these optimum parameters in present spiral profile we get following results shown in table 7.

Table 6: Results of optimum values of spiral profile

Von-mises strain	Von-mises stress in MPa	Axial stiffness N/mm	Radial stiffness N/mm
0.001558	202.48	0.728	57.032

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Total deformation is given in figure 4.

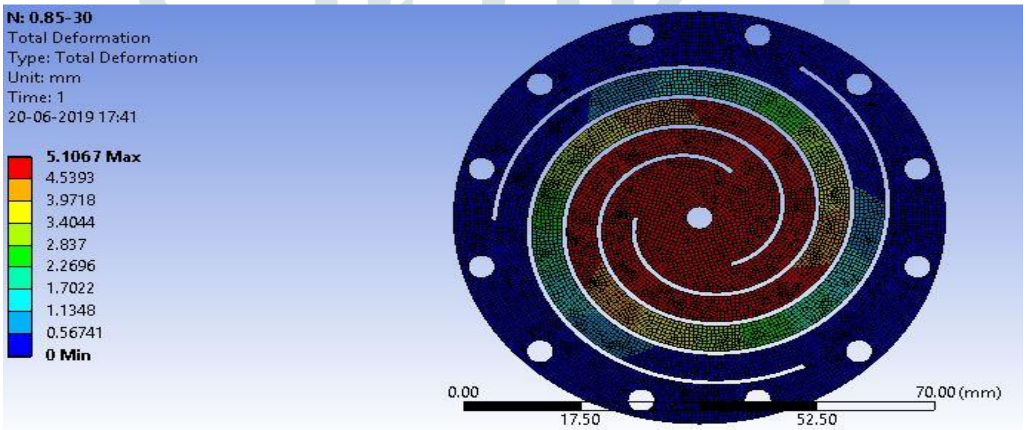

Figure 4: total deformation of flexure disc

Total Equivalent strain is given in figure 5.

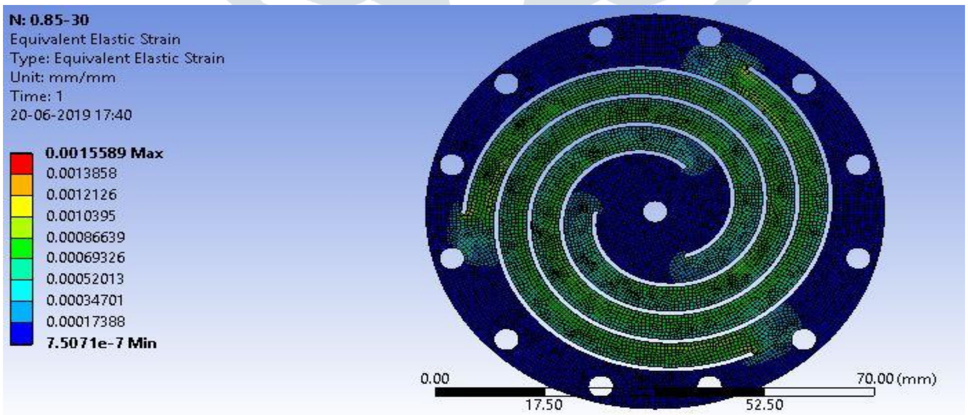

Figure 5: Equivalent strain of flexure disc

Total Equivalent stress is given in figure 6.

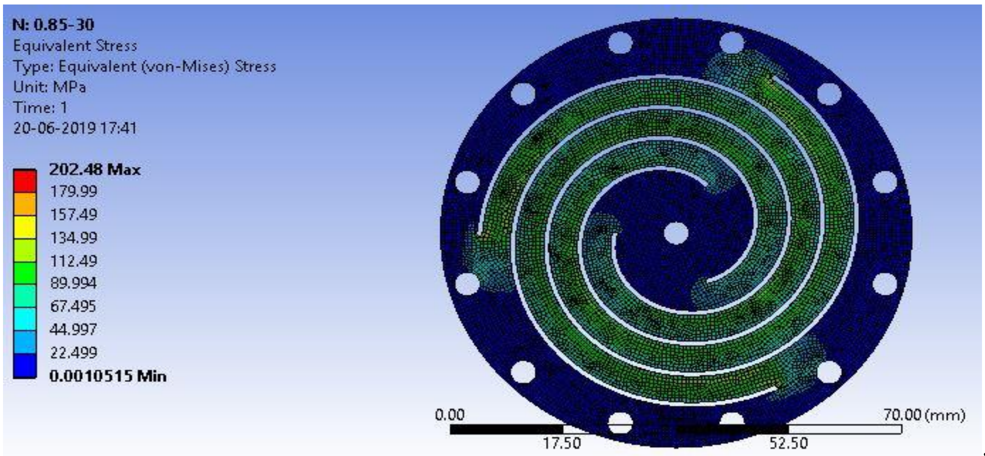

Figure 6: Equivalent stress of flexure disc

The maximum strain in this case is 155.89×10^{-5} corresponding to a stress 202.48 MPa which is well below the recommended design stress limit of 345 MPa. Total radial stiffness of flexure stack is 570.32 N/mm which well above design radial stiffness is 181.82 N/mm. This new modification in geometry of spiral profile satisfies design constraints.

IV. Conclusion

Previously compressor employed 28 flexure discs but due to this modification now number flexure disc required by compressor is 20.So total number of flexure disc required to compressor is reduced by 8 discs. This study elaborates effects of various geometrical parameters of spiral profile on axial stiffness of flexure disc. This modification not only reduces cost of the compressor but also it will reduce complication in assembly of compressor.

V. References

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