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Experimental and FEA Approach for Optimization of Spiral Flexure Bearing

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Abstract- In present situation problem to run duel piston compressor is lubrication. Since contaminant can often migrate to the cold end of the cry coolers. So the lubricants may contaminate the system and dry bearing lead to wear out the system. To overcome this problem we need oil free compressor, so that spiral bearings are developed Spiral flexure bearings moves concentrically without touching the piston. Experimentation was done for different bearing considering thickness, external diameter, spiral angle and Slot Width as a important parameter. Optimization of design parameters using experimental and FEA has been done by selecting axial and radial stress as the dominating parameter. Experimental and FEA results are found in good agreement. Optimized flexure bearing found strong in radial stiffness and weak in axial stiffness.

Keywords- FEA, Flexure bearing, cry cooler, radial stiffness, axial stiffness.

I. INTRODUCTION

Cryocooler is the equipment required to reach low temperature. Two main parts of any cryocooler are compressor and expander. Earlier, the expander used to have moving parts, but after the development of pulse tube coolers, compressor is the only part having moving components in it. These are low input power, high reliability, long life time, maintenance-free operation, low weight, minimum vibrations and noise, compactness. These diverse requirements affect the type and design of compressor to be used (as the compressor is the only moving part in cryocooler) and hence it determines the reliability of the cryocooler.

The linear motor compressors developed earlier have been developed by using materials such as magnets, bearings etc. available at the time of development but advancing technology demands for more robust, light weight and compact compressors for present applications such as in satellites. So the present work is aimed at designing, fabricating the opposed piston, moving coil compressor which will use new materials such as rare earth magnets and different types of flexure bearings and will be compact compared to its predecessors.

The first cryocooler using flexure bearings was used by the Oxford University in 1980s. Davey [1] reviews the development of these oxford style cryocooler. Since then, flexure bearings have proved to be the best option for small capacity cryocooler used for satellite based cooling applications requiring very high reliability in performance and a high operation life. Marqaurdt and Radebaugh [2] [3] have reported scaling laws and approximate equations for the design of linear arm flexures analysis of the linear arm flexure using FEM. Narayan khedkar et al. [4] established the link between thermodynamic performance and dynamics of two reciprocating components in doubly motorized free piston free displacer (FPFD) Stirling cryocooler, thus presenting design tool for such cryocooler. Gaunekar [5] has developed moving double coil linear motor compressor with 27 W output power for small capacity Sterling cryocoolers of cooling capacity 1 W at 80 K. He has also developed displacer motor with 0.2 W power output for above purpose. Haung. et al [6] have reported that flexure bearing is a key technology for long life of space borne cryogenic refrigerators (cryocoolers), TonnyBenschop et al. [8] have used FEM simulation package Algor for spiral flexure design. Gaunekar et al have analyzed spiral flexure bearing using finite element method and reported an optical method for accurate measurement of associated strains on the flexure disc so as to validate the analytical results the literature review addresses the trend in the development of different geometry and flexure material. The advancement in computational analysis will add to the improvement in design of flexure bearing.

Current study mainly focused on optimization of spiral flexure bearing by using experimentation and finite element method. Axial and radial stress is taken as the important optimizing parameter for the present study.

II. FLEXURAL BEARING

Each disc is in the form of a flat metal disc having three spiral slots, yielding three spiral arms, which bear the radial and axial loads. Each spiral sweeps an angle of 480°. Twelve peripheral holes are used to clamp the disc rigidly onto a support structure. These holes are intentionally sufficiently oversized with respect to bolt to provide freedom in radial positioning. The central hole in the disc allows the shaft to fit in snugly. To relieve stress concentration small holes have been provided at the end of the spiral cuts. Beryllium copper is used as the material because of its good spring properties. The flexure discs are manufactured by photo-etching method

Dimensional view of flexure disc with spiral section is as shown figure

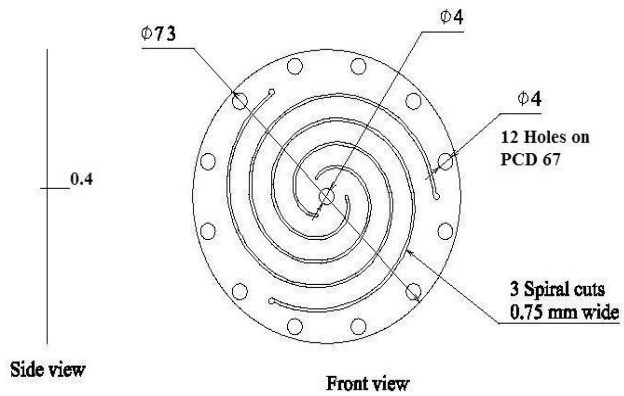

Fig. 1 Cad Model of Spiral Flexure

III. DESIGN OF EXPERIMENTS

Each flexure unit is in the form of a flat metal disc having three spiral slots yielding three separate spiral arms which bear the radial and axial loads. Twelve peripheral holes are used to clamp the disc rigidly onto a support structure.

The design parameters of flexure bearing are:

Effective diameter

Thickness of the disc

Spiral swept angle

Slot width

Different models of flexure bearings are to be analyzed by varying different parameters using FEA software like ANSYS. The parameters and their levels identified for this analysis are as follows:

Table 1 Design Parameter for Spiral Bearing

Sr. No	Parameter	Variation	No. of Levels
1	Effective diameter	40-60 mm	3
2	Thickness	0.2-0.6 mm	3
3	Swept angle	3600-6000	3
4	Slot Width	0.5-1 mm	3

No. of possible combinations for spiral flexure bearing considering parameters and levels are 81. But to study the effect of geometric parameters on the response variables to arise at the most suitable geometry, we need not analyze all 81 geometries. Hence Design of Experiments (DOE) theory needs to be used. From various methods available, Taguchi method is selected for present application. This is fractional factorial method. From this method it is decided to use L27 array . Thus this way no. of combinations reduces to 27.

Typical geometries of spiral flexure bearings under investigation have been shown these flexures are flat metal discs with three spiral slots. These spiral slots bear the axial and radial loads. The spiral in swept angles 420^0, 480^0 and 600^0 respectively. The hole at center is of 3 mm diameter to insert piston rod of compressor. The effective diameter of flexure element is selected from 40 to 80 mm. The outer edge radius of each slot is parallel to inner edge shifted by 0.25 to 1 mm which defines the slot width. At the inner end, semicircle arc provides the stress relief.

3.1 FE model

Finite element model is prepared by optimization of axial stiffness, radial stiffness and fatigue strength.

Table 2 FEA Parameters

Element Type	Shell-63
Real constant(thickness)	ranges from 0.1mm to 0.5mm with step of 0.1mm
Material	Beryllium copper
Young's modulus (E)	130 ×103 N/mm2
Poisson's ratio	0.3

Element type Shell 63 having six degrees of freedom at each node, suitable for elastic member and having both bending and membrane capabilities. Stress stiffening and large deflection capabilities are included. In practice all the structures are three dimensional. However, geometrical approximations are made to facilitate simple stress analysis of a part. The approach is valid and useful in FE modeling. If the geometry and loads of a problem can be completely described in one plane, then the problem can be modeled in one plane as two dimensional. The real constants give the third dimension. For the same we have used shell 63 element.

3.2 Boundary conditions

The model with boundary condition. The nodes on the outer clamped rim of the model were fully constrained by putting all six degrees of freedom of each node to zero. All the nodes in the region of the central spacer were constrained so that no rotation about in-plane axes (X and Y) was possible. They were then displaced perpendicular to the disc plane by an amount(1 to 5 mm) representing the axial displacement of the moving component (piston) and the solution obtained. For every value of axial displacement, the program was run with the addition of a fixed radial displacement of 15 µm so as to obtain the solution for such a combined (axial + radial) loading. (In general, a radial clearance of about 15 µm between the piston and the cylinder serves well as a clearance seal, and hence it is expected that any disc in the flexure bearing assembly would not deviate radially by more than the clearance).

IV. EXPERIMENTAL SET UP

4.1 Strain measurement

Generally strain gauges are used for the strain measurement. A strain gauge is a device used to measure the strain of an object. The most common type of strain gauge consists of an insulating flexible backing which supports a metallic foil pattern. As the object is deformed, the foil is deformed, causing its electrical resistance to change. This resistance change, usually measured using a Wheatstone bridge, is related to the strain by the quantity known as the gauge factor. Since it is required to measure the strain in three directions at the same time to calculate von Misses strain, strain gauge rosette needs to be used.

Strain gauges are positioned at 450 from each other. One gauge is mounted on top of another so that it covers a smaller area than that having entire gauge in one plane and so would give more accurate results.

To carry out strain measurement KAPTL KDM-P-6800-S-8 strain indicator is used. This indicator is connected to computer. This has 8 channels and has sampling rate of 100 samples per second. It gives micro strain value which is stored in the Excel file format.

LVDT is connected to flexure bearing and then through LVDT indicator, it is connected to computer. Displacements are shown on monitor as well as can be stored in excel format. Strain gauge rosette is pasted in flexure bearing. Then it is connected to the 120 Ω modules. There are 120 Ω and 350 Ω modules area available with the strain indicator to connect strain gauges of 120 Ω and 350 Ω respectively. On the module, there is provision for selecting full bridge, half bridge and quarter bridge circuits. In present case quarter bridge circuit is used. Module is the connected to the strain indicator and to the computer where micro-strain value is displayed.

5 RESULTS AND DISCUSSION

All the experiments designed after Taguchi method is analyzed by using Finite element analysis and experimental techniques. Static analysis of the all the models gives results, for this particular research we have taken into account of von misses stress, axial and radial force. The contour result of von misses stress for model no 3 is as shown in figure no 2.The von misses' stress contours shows maximum stress at the end of flexural ribs.

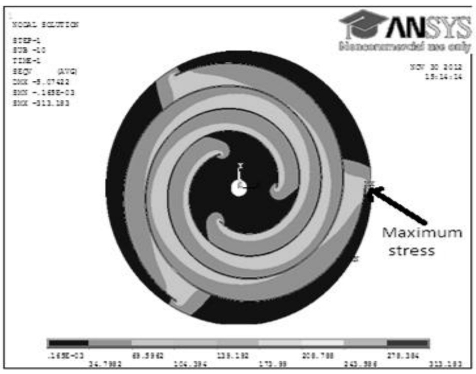

Fig 2 ANSYS Result

(1) Force Vs. Displacement

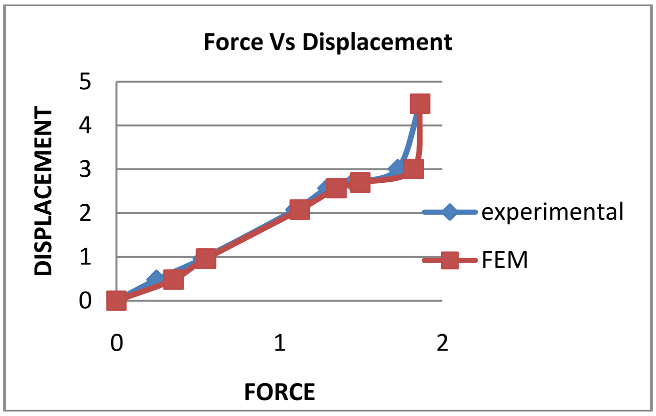

Fig. 3 Compliance Graph for Experimental and FEA

Fig. 3 shows very good agreement between FEM and experimental. Coefficient of x in trend line equation gives the value of axial force which is 0.53 N. This value is in good agreement with FEM value of 0.56 N.

(2) Axial stiffness VS Displacement

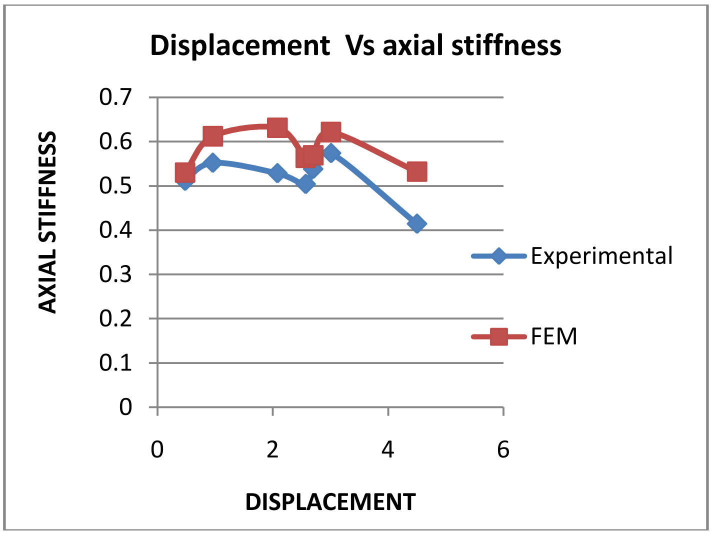

Fig. 4 Experimental Vs. FEM for Displacement and Axial Stiffness

Fig 4 shows that, values got from experimental setup and FEM are having good linearity among them. Coefficient of x in trend line equation gives the value of axial stiffness which is 0.52 N/mm. This value is in good agreement with FEM value of 0.54N/mm.

(3) Stress Vs. Displacement

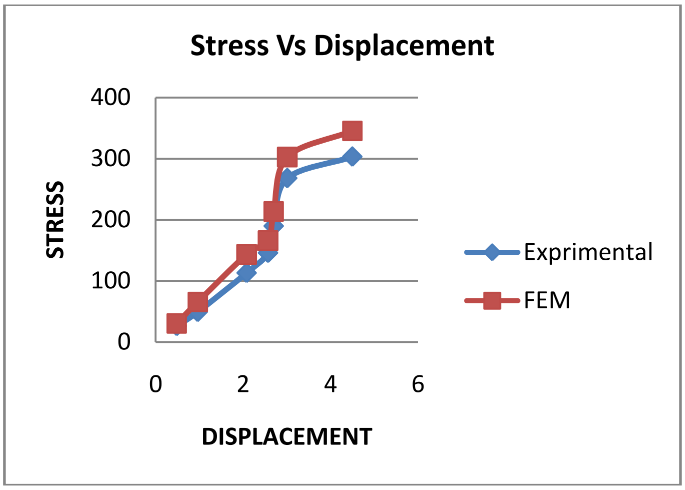

Fig 5 Experimental Vs. FEM for Displacement and Stress

Fig 5 shows that, values got from experimental setup and FEM are having good linearity among them. In experimental setup for 4.5 mm displacement stress having 330 N/mm2 value, on other hand it gives 300 N/mm2 for same displacement in FEM.

6 CONCLUSION

A set of design graphs has been developed using finite element modeling to provide the maximum radial stiffness and optimized geometry for spiral flexure bearings. The design curves can quickly provide stiffness values and reduce modeling time.

Validation of FEA results with experimental results nearly matches and having error in percentage up to 12-15%.

The work was mainly concentrated on stress and strain analysis, in future; it can also be done with fatigue strength as main area of concentration.

REFERENCES

[1] Davey, G. "Review of the Oxford cryocooler". Advanced. Cryogenic. Engineering, 35, Plenum Press, New York, pp. 1423-1430 (1990).

[2] Marquardt, E., Radebaugh R. and Kettal P., "Design equations and scaling laws for linear compressor with flexure spring." Proc. 7th Intl. Cryocooler conf., pp.783-395, New Maxico (1992).

[3] K. G. Narayankhedkar,,A. S. Gaunekar and S.L. Bapat, "Dynamic and thermodynamic analysis of doubly motorized miniature String cryocooler using double coil linear motors" Cryogenics, Vol. 34, pp. 43-50 (1993).

[4] AjitGaunekar, "Analysis, design development and experimental testing of a linear motor driven, miniatures, split Stirling cryocooler using flexure bearings for satellite based cooling applications.", Ph. D. Thesis, Department of mechanical Engineering I.I.T. Bombay. (1996).

[5] Gaunekar A. S., GoeddenhenrichT, and Heiden C., " Finite element analysis and testing of flexure bearing elements", Cryogenics , 36 , pp. 359-364 (1996).

[6] B.J.Huang,Y.C.Chen, System Dynamics and control of linear compressor frequency adjustment, transaction of the ASME,Vol.124,March 2002

[7] TonnyBenschop, JeroenMullie, Peter Bruins, Jean Yves Martin, "Development of a 6W high reliability cryogenic cooler at Thales cryogenics", CEC12 (2001).

