Shape Factor Optimization and Parametric Analysis of Spiral Arm Flexure Bearing Through Finite Element Analysis Studies

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***Abstract*— Numerous studies have been made and published to understand the behavior of spiral arm eccentric flexure bearings in various conditions. Well-designed eccentric flexure bearings play a vital role in longevity of cryocoolers. Shape factor is a design parameter which influences the performance and life of the flexure bearings to a larger extent. We have made an attempt to study the design of spiral arm eccentric flexure bearings used in cryocoolers using MatLab codes. Behavioral performance of flexure bearings is obtained by changing the arm shape factor f and spiral end angle. Besides, the effect of shape factor is also studied for a specific axial displacement apart from other important design requirements like induced maximum stress, axial stiffness and radial stiffness. Simulations thus carried out using ANSYS- ® have been plotted. Experimental validation has been done for a flexure bearing of specific dimensions. In general for an optimal shape factor, maximum stress developed in the flexure bearing for any given displacement will have a minimum value.**

Keywords—Cryocoolers, Flexure Bearing, Shape factor.

I. INTRODUCTION

Flexure springs or flexure bearings are widely used in linear resonant compressors and linearly driven miniature cryocoolers because they integrate the use of spring and bearing in one single part. As far as cryocoolers are concerned this concept was first used in Oxford University cryocoolers [1] reported in mid 1980s. Because of the advantage of no rubbing parts, the life and reliability of the system increases tremendously in comparison with using conventional systems i.e. helical springs and linear bearings. Three-arm flexure spring is the most common arrangement among different designs, the two and four-arm design schemes are also being adopted by many research groups. Figure 1 shows the common types of flexure bearing designs adopted in many cryocoolers driven by linear motor compressor. To start with, electromagnetic analysis carried out to optimize the dimensions of linear motor components paves the way for deciding the geometric dimensions of the flexure bearings. As a result of research and development of moving magnet linear compressor in IISc, flexure dimensions are fixed to be having 40mm outer diameter and 5mm hole at the Centre for accommodating piston shaft. Other characteristic flexure dimensions have been optimized based on the analysis for a design requirement of 3mm axial displacement. Wong et al[2], have analyzed the optimized spiral flexure for dynamic stress and concluded that dynamic stress should be considered in flexure fatigue life prediction; Marquardt et al[3], have reported scaling laws and appropriate equations for the design of linear arm flexures. Gaunikar et al [4], had presented non dimensional curves for a flexure disc with spiral arm configuration for a certain range of sizes and piston strokes.

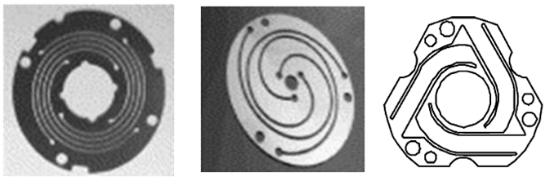

Fig 1: a) four arm concentric b) Three arm Eccentric c) Linear arm Flexure Bearings.

Lee et al [5], had developed an Automated analysis procedure for predicting the flexure performance of any given flexure configuration and for a variety of design requirements. N.Chen et al [6], put forward a universal design method of spiral flexure and parameter analysis through finite element analysis and results are verified experimentally. This paper attempts for the first time, use of MatLab codes and GSD Macros, written for CATIA V5, for eccentric spiral arm flexure bearing design. Besides the method of choosing optimum shape factor for different spiral angles is reported.

II. BASIC DESIGN

Parametric equation in polar coordinates for generating eccentric spiral arm profile is,

$$R = R_i + (R_o - R_i) * [\Theta/\Theta_0 + f * \sin(2\pi\Theta/\Theta_0)] \quad \text{.....(1)},$$

where

R is the polar distance from the centre of the flexure,

Θ/Θ_0 is the relative sweep angle ratio,

Ri is the active inner radius,
Ro is the active outer radius,
f is the arm shape factor.
In the present work with the intension to make design changes easier and faster Matlab code is written to form spiral coordinate points using equation 1. In the next step these polar coordinate points are converted to Cartesian coordinates, and then imported to GSD macro support provided for CATIA users, which can directly generate the geometric points in CATIA. Figure 2 shows the spiral coordinate points generated using macros.

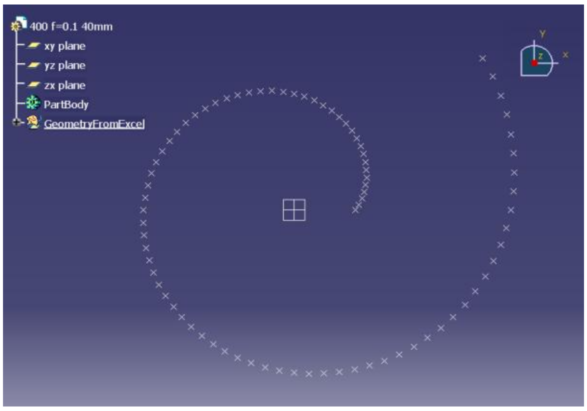

Fig 2: Geometric points generated using Macros

These geometric points are joined using 'spline' followed by simple geometric operations like offset, circular pattern etc. are done to obtain complete model of flexure bearing. Fig 3 (a, b, c, d, e, f) shows three arm flexure bearings with different shape factors (0 to 0.1 in steps of 0.02) modeled in CATIA using above procedure. Use of MatLab codes and GSD support system tremendously saved the time required for the design.

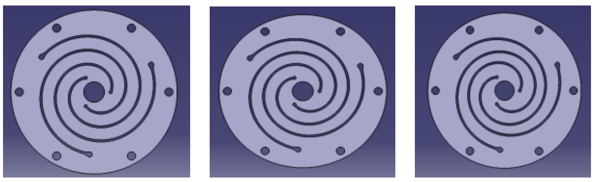

Fig3: a) f=0 b) f=0.02 c) f=0.04

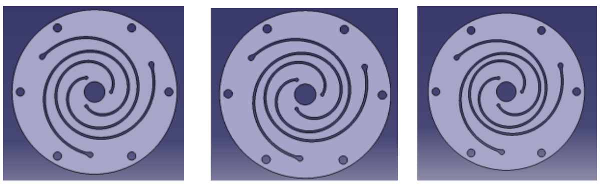

Fig 3: d) f=0.06 e) f=0.08 f) f=0.1

Also it is observed from figure 3 that, the material width between spiral slots decreases as the shape factor increases. Fig 4 shows the relationship between shape factor and minimum material width between two successive slots for different spiral angles.

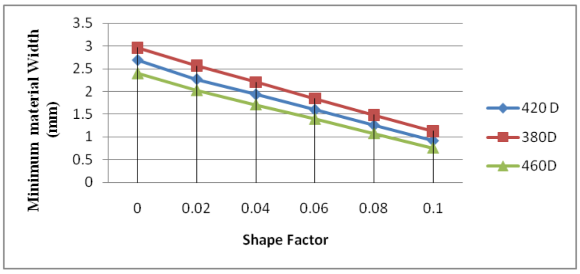

Fig 4: Relationship between shape factor and minimum material width between two successive slots.

III. DESIGN REQUIREMENTS

Flexure springs has to be design optimized to have very high fatigue life, this can be achieved by restricting the maximum stress developed in flexure bearing under loading to be well within the fatigue limit of flexure material under consideration. Flexure springs should have High radial stiffness to maintain uniform annular gap between cylinder wall and piston to avoid friction and to minimize blow-by losses. Besides, flexures springs should have less axial stiffness because of the larger axial displacement.

IV. FINITE ELEMENT ANALYSIS

The finite element method had been proved to be the effective way to analyze the performance of flexure spring[3,4]. Geometrically Nonlinear static structural analysis has been conducted with personal computer, based on advanced finite element software (ANSYS 11®). The mesh is successfully refined and convergence study is conducted for each model before deciding the final mesh. Total 18 flexure geometric models are considered for the analysis consisting of three different spiral angles and each spiral angled design is studied for six different shape factors.

Boundary Conditions

All nodes on the outer periphery were fully constrained by putting all six degrees of freedom to zero. The inner nodes were constrained so that rotation about same plane is not possible, then all the inner nodes are displaced by 3mm displacement value perpendicular to the disc plane and the simulation is carried out. The steps are repeated once again with the addition of 10µm radial displacement so as to obtain solution for such combined loading, (Axial + Radial). 10 µm displacements along radial direction represent the gap between cylinder inner wall and piston. Fig 5 shows the FEA model with all the boundary conditions imposed for 460^0 spiral angle with a shape factor of zero. Table 1 shows the flexure material specifications used for the analysis.

Flexure material	Spring Steel
Young's modulus	210E3 MPa
Poisson's ratio	0.313
Density	7850Kg/m^3
Fatigue strength	870MPa

Table 1: Specifications Used in the Analysis

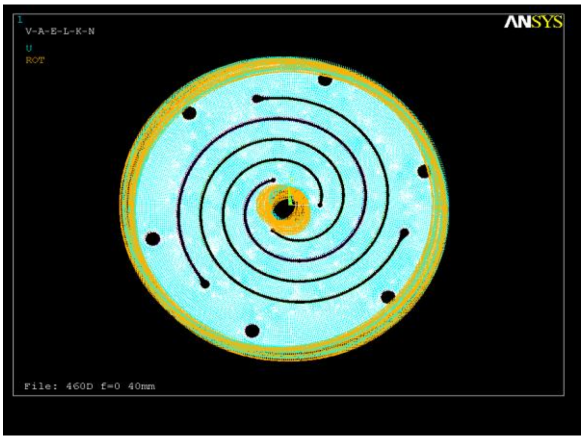

Fig 5: Flexure bearing model (460˚ spiral angle, f=0) with applied boundary conditions

V. RESULTS AND DISCUSSIONS

Increase in shape factor decreases the material width between any two successive spiral slot (Fig 2), this makes the flexure bearing to become more flexible to axial and radial loads. Hence Increase in shape factor resulted in decrease in axial and radial stiffness values for all spiral angles.

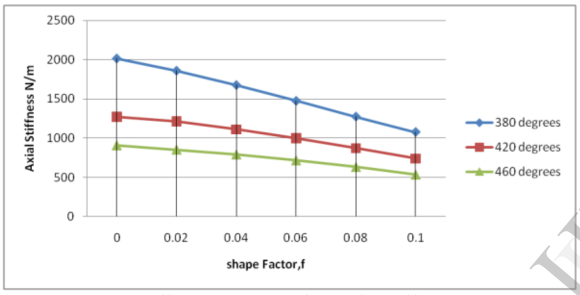

Fig 6(a): Axial Stiffness v/s Shape factor for different spiral angles.

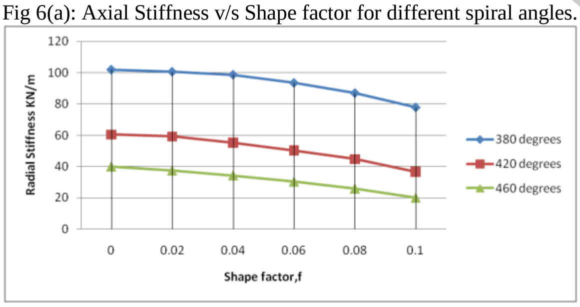

Fig 6(b): Radial Stiffness v/s Shape factor for different spiral angles.

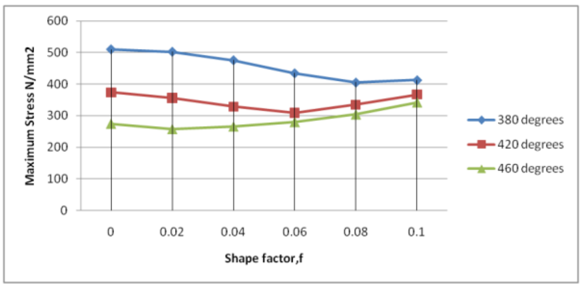

Fig 6(c): Maximum induced Stress (Von-Mises) v/s Shape factor for different spiral angles.

Lesser values of axial stiffness can be obtained at higher shape factors as shown in fig 6(a), but Choosing very high values of shape factor is not recommended. Because radial stiffness decreases along with decrease in axial stiffness as shown in fig 6(b), which intern may can cause out of axis movement in piston shaft, resulting in friction between cylinder wall and piston head. Hence choosing optimum shape factor which results in minimum axial stiffness and still can take more radial load for the given radial displacement is a challenging task. Fig 6(c) shows the variation of maximum Von-Mises stress for different spiral angles and shape factors. For a shape factor of zero (uniform material width (fig 3(a)), we observed the undesirable stress concentration at the ends of the spiral arm of the flexure bearing [2]. Further increase in shape factor resulted in decrease in maximum stresses in flexure bearing, as well as the stress gets distributed uniformly throughout the spiral arms. This trend continues till a particular value of shape factor is reached. Further increase in shape factor resulted in increase in maximum vonmises stress in flexure bearing. Hence optimum shape factor can be chosen at the value where it results in minimum Von-Mises stress in flexure bearing, provided the radial stiffness is quite high such that it can prevent the out of axis movement. From fig 6(c) for spiral angles of 380˚, 420˚, 460˚ it is obvious that the shape factor values of 0.08, 0.06, and 0.04 respectively will lead to minimum possible von-Mises stress in flexure bearings. Hence these values of "f" are chosen to be optimum values of shape factor for those particular spiral angles provided all other design parameters are same.

VI. EXPERIMENTAL RESULTS

Photochemical etching process has been used to fabricate the required flexure profile as it is inexpensive and residual stresses induced during the process will be minimum. Based on the FEA simulations, it is decided to start with the fabrication of flexure profile having shape factor of f=0.1 as it shows the most optimal performance in terms of maximum stress induced and the axial stiffness for a spiral angle of 340 Degrees. Fig 7(a) shows the flexure spring thus etched and flexure stack for the linear compressor shown in Fig 7(b) were carefully selected after verifying for any etching defects using SEM (Scanning Electron Microscope).

Fig 7 (a, b): Flexure Bearing and Dual apposed linear compressor developed in IISc [7]

To assess the correctness of FEA simulations, axial stiffness values are verified experimentally by measuring deflection v/s load. For the test, a linear motor configuration with an identical moving mass assembly of the operating system as shown in Fig 8 is used.

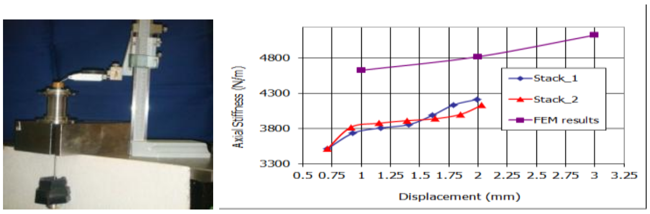

Fig 8 (a, b): Experimental setup and validation graphs.

The experimental axial stiffness value obtained is about 90% of FEM value. Finally the flexures are fatigue tested for 10^8 cycles at 1.25 times of the designated stroke and no failure of the flexure stack was experienced during the tests. More experiments are underway to test the flexure springs with different shape factors and Spiral angles based on above analysis.

VII. CONCLUSIONS

Use of Mat lab codes and GSD macros for designing eccentric type spiral arm flexure bearings has been introduced for the first time to make the design process simpler and faster. Effect of Shape factor, on important design requirements like Induced Maximum stress, axial stiffness and Radial stiffness are studied for different spiral angles. Results of the analysis shows that same shape factor value cannot be used for different spiral angled designs intern it has to be optimized in terms of design requirements. Also a simple and effective way to choose optimum shape factor for any spiral angle is put forward through stress analysis.

REFERENCES

1. Wong TE, Pan RB, Johnson AL. "**Novel linear flexure bearing**". In: Proceedings of the 7th international cryocoolers conference. 1992. p.675–698.
2. Wong, T.E., Pan, R.B., H.D., Sve, C., Galvan, L. and Wall, T.S. "**spiral flexure bearing**" proc 8th cryo conf (1994) 305-311.
3. Marquardt E, Radebaugh R**. "Design optimization of linear-arm flexure bearing"**. In: Proceedings of the 8th international cryocoolers conference. 1994. p. 293–304.
4. A.S.Gaunekar, T.Goddenhenrich, and C. Heiden, "**Finite element analysis and Testing of flexure bearing elements**" Cryogenics, volume 36,pp.359-364,1996.
5. Lee CC, Pan RB. "**Flexure bearing analysis procedures and design charts"**. Cryocoolers, vol. 9. New York: Plenum Press; 1997. p. 413–420.
6. N.Chen, X. Chen, Y.N.Wu, C.G.Yang, L.Xu "**Spiral profile design and parameter analysis of flexure spring"**. Cryogenics 46 (2006) 409-419.
7. Jacob S., Ramanarayanan V, Karunanithi R., Narasimham GSVL., Damu C et al. **"Development and Testing Linear Motor Compressor Driven miniature Pulse Tube Cooler"**, 23rd International Cryogenic Engineering Conference, Wroclaw, Poland (July 2010).