Journal of Materials Science & Surface Engineering Vol. 3 (2), 2015, pp 211-214

Contents lists available at http://www.jmsse.org/

Journal of Materials Science & Surface Engineering

Fatigue Life Analysis of Spiral Arm Flexure Bearing

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Article history
Received: 02-June-2015
Revised: 10-July-2015
Available online: 26-Aug-2015

Keywords:
Flexure Bearing,
Finite Element Analysis,
Stress Analysis,
Fatigue Life Analysis.

Abstract

The use of wear-free, frictionless clearance seals in linearly driven miniature cryocoolers has tremendously increased the reliability and life of such units as compared to those using contact type seals. This has been achieved by employing a non-conventional suspension system, called Flexural Suspension or Flexure Bearing. The earliest Oxford Cryocoolers utilized flexure elements with three spiral arms, which is called as spiral arm flexure bearing. In general, the flexure bearing has a threefold design requirements, viz. fatigue strength, radial stiffness and axial stiffness. In this paper, finite element analysis (FEA) of flexure bearing is carried out. A finite element model of spiral arm flexure bearing is obtained by using CATIA and Hypermesh software. This model is analyzed for stress and fatigue life by using MSC Nastran and MSC Patran software with Durability module. The effect of thickness and material variation is analyzed for stress and fatigue life of spiral arm flexure bearing.

The work had been presented *at an* international conference ***Fatigue Durability India 2015**, 28-30th May 2015, JN TATA AUDITORIUM, Indian Institute of Science, Bangalore.*

Introduction

Spiral arm flexure bearings (flexure bearing disc springs) are installed to support the compressor motor assembly and the displacer within the linear Stirling cryocooler. Such flexures are made from thin sheets of beryllium copper alloy, stainless steel alloy or titanium alloy. Research has been carried out to identify fatigue life of all spiral arm flexure bearings of various thickness and materials.

The use of wear-free, frictionless clearance seals in linearly driven miniature crycooler has tremendously increased the reliability and life of such units as compared to those using contact type seals. This has been achieved by employing a non-conventional suspension system, called Flexural suspension or Flexure Bearing, in favour of the commonly used helical spring. As far as crycoolers are concerned, this concept was first used in the Oxford University Crycooler[1] (nominally 1W at 80K) meant for satellite-based cooling applications, requiring very high reliability in performance and a high operating life.

Gaunekar et al.[2] had presented non-dimensional design curves for a flexure disc with spiral arm configuration for a certain range of sizes and strokes. These curves, which were the results of FE analysis of the flexure bearing disc, indicated that the linear relationship between the axial stiffness with the diameter of 50 mm and 250 mm respectively, thickness of those springs was correspondingly 0.21 mm and 1 mm.

Chen et al.[3] had presented a spiral profile design procedure and parameter analysis of flexure design by analytical expressions and comprised with the FE analysis results.

Lee et al.[4] had presented flexure bearing analysis procedures and design charts. According to the design charts developed for the spiral flexure, a preferred design configuration can be chosen according to the material fatigue stress allowable and the flexure stiffness requirements.

Simcock et al.[5] had presented investigation of materials for long life, high reliability flexure bearing springs for Stirling cryocooler applications. By using FEA tools, springs may be designed safely within their performance envelope, removing the need of batch testing. Operating below a known endurance limit, at a statistically established margin, well-designed springs may be assembled into products with a high level of confidence without batch testing.

Yingxia et al.[6] had presented simulation of deformation of diaphragm spring grooved with spiral slits by FE method. Finite element method was used to investigate the deformation pattern, the stress distribution, and the natural frequency of the flexure spring with three spiral slots in the disc plane.

Wenjie Zhou et al.[7] had presented the performance of Oxford and triangle flexure bearings. From the comparisons, the triangle arm flexure bearings can produce a larger radial/axial stiffness ration, which is beneficial for minimizing the size of linear compressors. With the same fundamental parameters, triangle flexure bearings have a larger natural frequency and can therefore be used in high frequency conditions.

Description

A single unit of the flexure bearing under investigation is shown in *Figure 1*. Each 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. Each spiral traverses an angle of 480^o. Twelve peripheral holes are used to clamp the disc rigidly onto a support structure. These are amply oversized with respect to bolt size, to provide freedom in radial positioning, in order to account for any misalignment between the concerned mating components.

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The central hole in the disc allows the shaft to fit in snugly. Minute holes are also provided at the end of the spiral slots to relieve stress concentration. The schematic assembly of a compressor unit using flexure bearings is shown in *Figure 2*. Sufficient numbers of flexure discs are stacked parallel, one beside the other to obtain the desired axial and radial stiffnesses. Two such stacks are used for dynamic stability. Unlike the helical spring, which has the buckling tendency, the flexure bearing has a very high radial stiffness. During actual operation in linear compressor the flexures are deflected by 5 mm at central position due to 5 mm stroke of piston rod having operating frequency of 50 Hz.

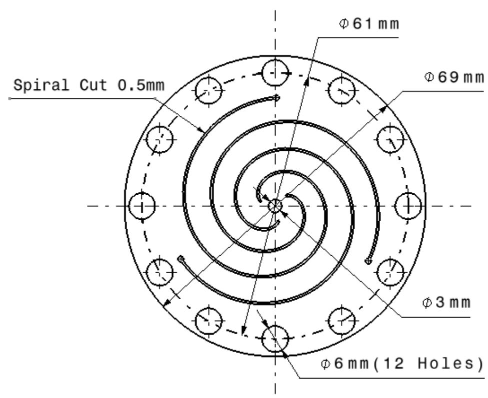

Figure 1: Spiral arm flexure bearing under investigation

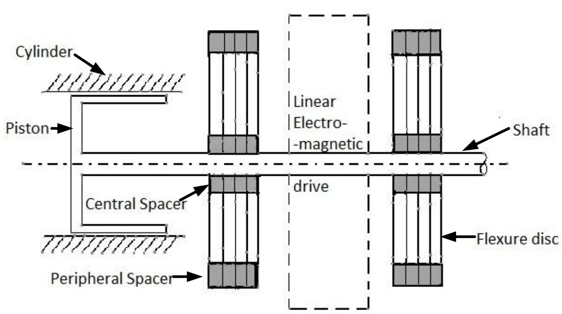

Figure 2: Schematic assembly of the flexure bearing

Design Requirements of Typical Flexure Bearing

In general, the flexure bearinng has a threefold design requirement, viz. fatigue strength, radial stiffenss and axial stiffness.

Fatigue Strength

Each arm of a flexure disc is subjected to alternating stresses at a frequency equal to the operating frequency of the linear motor. For a given axial displacement, the location and magnitude of the maximum stress in a disc are dependent upon the spiral profile, diameter and thickness of the disc. This maximum stress value should fall well below the endurance limit of the material selected in order that the disc has virtually infinite life.

Radial Stiffness

The radial stiffness of the flexure bearing assembly should be high enough to support the clearance seal under the weight of the suspended mass which consists mainly of the piston-shaft sub-assembly, coil and coil support.

Axial Stiffness

In order to minimize the power drawn by the linear motor of the compressor, the moving mass should resonate on the combined spring effect of the gas spring above the piston and the flexure bearing below it. The axial stiffness of the flexure bearing is normally kept substantially low compared to the stiffness of the gas spring above the piston so as to minimize the moving mass which determines the level of vibrations in the unit.

Design Parameters

The design parameters of the flexure disc are-

a) the geometry of the spiral profile
b) the element thickness
c) the total number of discs in each stock

Given the requirements of stroke and both axial and radial stiffness, an ideal disc should operate fully within the endurance limit of the material so as to have an infinite fatigue life. For a chosen spiral profile and size, the remaining two design variables, viz. element thickness and number of elements, should satisfy the design requirements. The analysis of such a flexure bearing is quite complex in nature as it falls into a category of large deflections with shear, bending and torsion all acting simultaneously on arms which in general have a variable width.

Fatigue Life and Stress Analysis of a Flexure Bearing by FEA

Since exact analysis of the flexure element is not possible the Finite Element Method (FEM) of analysis was chosen. Initially the model is created with the help of *CATIA V5R20* software for a particular thickness and particular material. Then meshing has been carried out with the help of *Altair Hyper Mesh v9.0* software for working in *MSC Nastran 2013* software as shown in *Figure 3.* Following various elements are used. Total number of nodes are 3310.

1. Element Type: CQUAD4, No. of Elements: 2537
2. Element Type: CTRIA3, No. of Elements: 335
3. Element Type: RBE2, No. of elements: 13

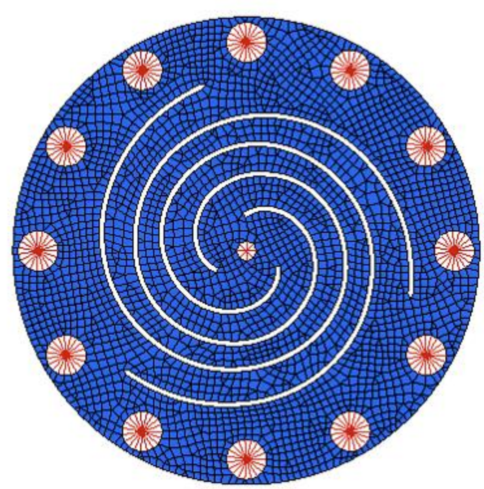

Figure 3: Mesh Model of Spiral Arm Flexure Bearing

The different materials used for spiral arm flexures are tested in Test Laboratory for mechanical properties like UTS. The properties are shown in *Table 1*. These properties are used while analyzing the spiral arm flexure by using MSC Nastran software for stress and fatigue analysis.

Table 1: Material Properties of Spiral Arm Flexure Bearings

Sr. No.	Material	Density (kg/m^3)	Young's Modulus of Elasticity (GPa)	UTS (MPa)	Poisson's Ratio (μ)
1	Beryllium Copper Alloy BeCu 17200	8100	130	678	0.3
2	Stainless Steel Alloy AISI 304	8000	200	1192	0.29
3	Titanium Alloy Ti-6Al-4V Grade 1	4510	105	309	0.37

Then this model is solved for stress analysis in MSC Nastran software. Stress analysis results are shown in the *Figure 4*.

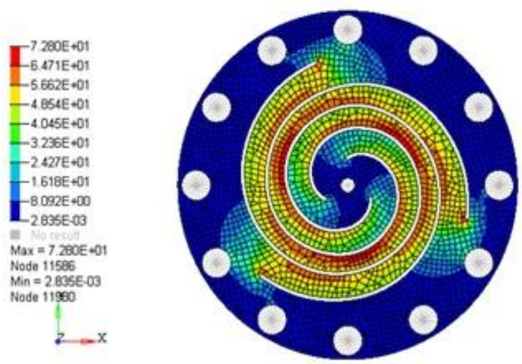

Figure 4: Stress Distribution in Spiral Arm Flexure Bearing of BeCu Alloy with thickness of 0.2 mm

The *MSC Patran 2013* software having *Durability Module* is used for obtaining fatigue life of spiral arm flexure bearing. The load cycle used is as shown in *Figure 5*. This represents loading from zero deflection of flexure to different deflections. The working deflection at central position of flexure is considered as 5 mm. As deflection is increased stress value in the flexure increases and we get a limiting value of stress value which changes infinite fatigue life of flexure bearing to finite life of flexure bearing. The stress value corresponding to this fact is called as fatigue strength of that flexure bearing. *Figure 6* shows fatigue life distribution across the spiral arm flexure bearing.

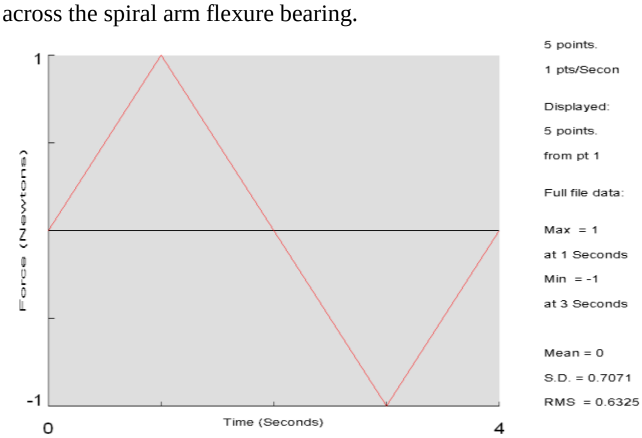

Figure 5: Load Cycle for Fatigue Life Calculation of Spiral Arm Flexure Bearing

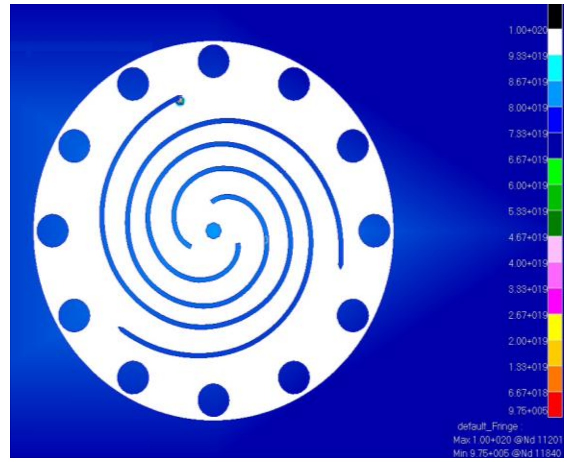

Figure 6: Spiral Arm Flexure Bearing in Patran 2013, showing Fatigue Life Distribution of BeCu Alloy with thickness of 0.15 mm

Results and Discussion

Figure 7 shows maximum strss variation of spiral arm flexure bearing of BeCu alloy, SS alloy and Ti Alloy materials with thickness variation from 0.1 mm to 0.5 mm in steps of 0.05 mm.

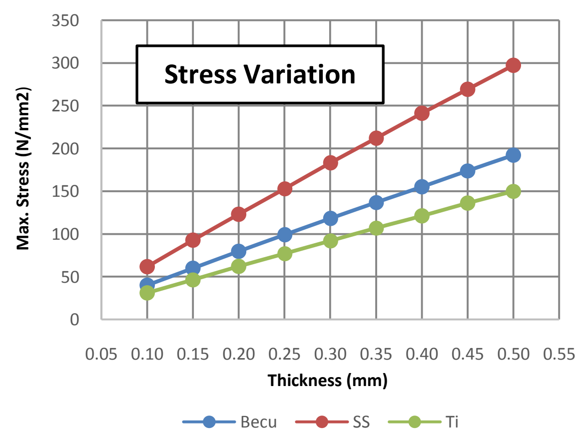

Figure 7: Maximum stress variations of Spiral Arm Flexure Bearing of BeCu Alloy, SS Alloy and Ti Alloy materials with thickness variation

Figure 8 shows S-N Curve of BeCu Alloy Spiral Arm Flexure Bearing with thickness variation from 0.1 mm to 0.5 mm in steps of 0.05 mm.

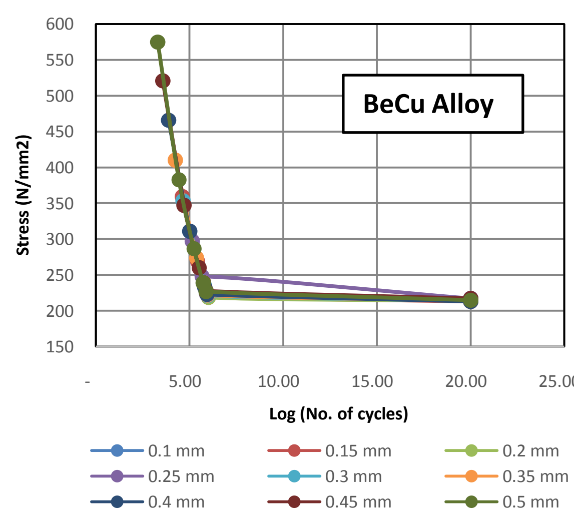

Figure 8: S-N Curve of BeCu Alloy Spiral Arm Flexure Bearing with thickness variation

Figure 9 shows S-N Curve of SS Alloy Spiral Arm Flexure Bearing with thickness variation from 0.1 mm to 0.5 mm in steps of 0.05 mm.

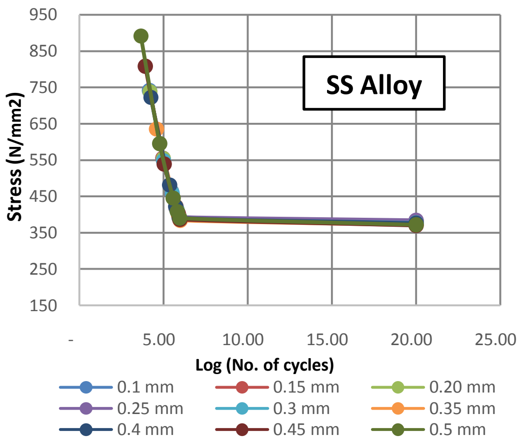

Figure 9: S-N Curve of SS Alloy Spiral Arm Flexure Bearing with thickness variation

Figure 10 shows S-N Curve of Ti Alloy Spiral Arm Flexure Bearing with thickness variation from 0.1 mm to 0.3 mm in steps of 0.05 mm. For thicknesses of 0.35 mm and above Ti alloy material is failing for 5 mm piston stroke. Hence we could not obtained infinite life of flexure bearing for 0.35 mm and above thicknesses for 5 mm piston stroke.

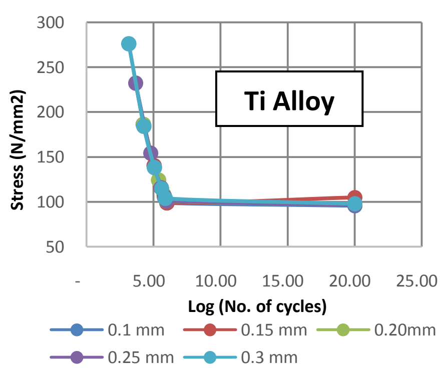

Figure 10: S-N Curve of Ti Alloy Spiral Arm Flexure Bearing with thickness variation

Figure 11 shows S-N curve of Spiral Arm Flexure Bearing of 0.15 mm thickness and of BeCu alloy, SS alloy and Ti alloy materials. Similar S-N Curves of all these material's flexure bearings are obtained with thickness variation from 0.1 mm to 0.5 mm in steps of 0.05 mm. The nature of graphs are similar to *Figure 11*, in all these cases.

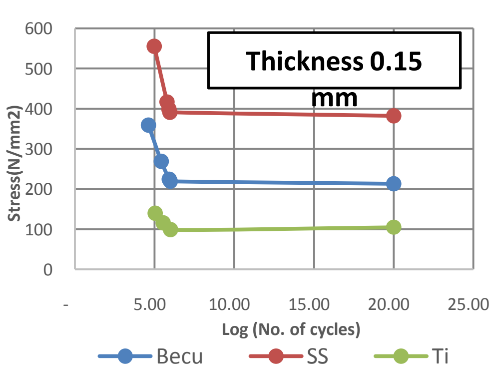

Figure 11: S-N curve of Spiral Arm Flexure Bearing of 0.15 mm thickness and of BeCu Alloy, SS Alloy and Ti Alloy materials

Conclusions

We have successfully obtained the Fatigue life of spiral arm flexure bearing having different thicknesses and different materials.

1. For same thickness of three materials and for 5 mm piston stroke, higher stresses are induced in SS alloy spiral arm flexures as compared to other material flexures. (Refer *Figure 7*)
2. For same material and for 5 mm piston stroke if the thickness is varied between 0.10 mm to 0.50 mm, then fatigue strength remains unaltered for BeCu alloy and SS alloy flexures. (Refer *Figure 8*, *Figure 9*)
3. For 5 mm piston stroke, BeCu alloy and SS alloy flexures give infinite life for thickness between 0.1 mm to 0.50 mm, however Ti alloy flexures give infinite life for thickness between 0.1 mm to 0.3 mm. (Refer *Figure 8*, *Figure 9* and *Figure 10*)
4. Spiral Arm flexures made from SS Alloy gives better fatigue strength as compared to Beryllium copper alloy and Titanium alloy for the same thickness of flexures.(Refer *Figure 11*)
5. Fatigue strength of Spiral arm flexures made from BeCu Alloy is 213 MPa, SS Alloy is 370 MPa and Ti Alloy is 95.8 MPa. For maximum induced stress below the fatigue strength of material, we get the infinite fatigue life for all these flexures. (Refer *Figure 11*).

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