International Journal of Mechanical Engineering and Robotics Research

ISSN 2278 – 0149 www.ijmerr.com
Vol. 3, No. 1, January 2014

Research Paper

MODELING AND ANALYSIS OF DEFORMATION ON A FLEXURE BEARING IN LINEAR COMPRESSOR

Vennapusa V Sivareddy[1]*

*Corresponding Author: Vennapusa V Sivareddy, ✉ sivareddy.vennapusa1@gmail.com

Flexure bearing is a new concept and used for precision applications such as Programmable Focusing Mechanism (PFM), linear compressor, etc. These bearings are compact and inexpensive. A flexure bearing is designed for specific applications. These designed can usually be done with the advanced design tool like Fine Element Analysis. With the advent of computers Fine Element Analysis has become the most suitable tool for the engineering analysis where the conventional approach is not suitable, geometric complexity are involved etc. This bearing contains three slots having 120° apart and 12 peripheral holes are used to clamp the disc rigidly onto a support structure. One central hole made for movement of shaft. The bearings are commonly made by Aluminum material we calculate the deformations in theoretical and FEA method by using Ansys Software. In this paper we are modeling the flexure bearing by using copper material we are calculate the deformations in theoretical and FEA method by using Ansys Software. We are analysis the deflection deformation on the both materials of bearing. We are analysis the deflection deformation on Flexure Bearing at different load (means 1 N to 5 N). Using software's like CATIA and PROE, modeling of flexure bearing done. Also make Fine Element Method analysis on it by using Ansys software and lastly, this project considers the fatigue life criteria for flexure bearing and tries to optimize it. We are made design calculation and Fine Element Analysis for flexure bearing to make appropriate model and draw the performance charts for linear compressor bearing by comparing the power consumption with different temperatures, and capacity.

Keywords: Flexure bearing, Deformation, Linear compressor

INTRODUCTION

A bearing is any of various machine elements that constrain the relative motion between two or more parts to only the desired type of motion. This is typically to allow and promote free rotation around a fixed axis or

[1] Tadipatri Engineering College, Kadapa Road, Veerapuram Village, Tadipatri Mandal, Anantapur Dist., Andhra Pradesh, India.

free linear movement; it may also be to prevent any motion, such as by controlling the vectors of normal forces. Bearings may be classified broadly according to the motions they allow and according to their principle of operation, as well as by the directions of applied loads they can handle.

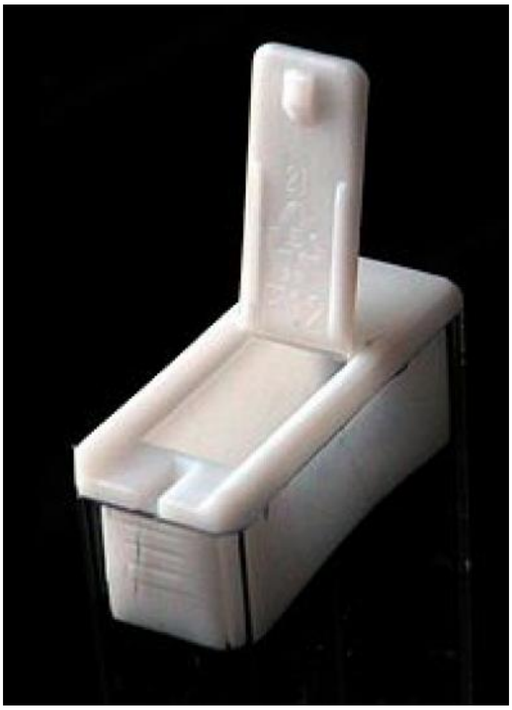
The term "bearing" comes ultimately from the verb "to bear",and a bearing is thus a machine element that allows one part to bear another, usually allowing (and controlling) relative motion between them. The simplest bearings are nothing more than bearing surfaces, which are surfaces cut or formed into a part, with some degree of control over the quality of the surface's form, size, surface roughness, and location (from a little control to a lot, depending on the application). Many other bearings are separate devices that are installed into the part or machine. The most sophisticated bearings, for the most demanding applications, are very expensive, highly precise devices, whose manufacture involves some of the highest technology known to human kind.

Flexure Bearing

A flexure bearing is a bearing which allows motion by bending a load element. A typical flexure bearing is just one part, joining two other parts. For example, a hinge may be made by attaching a long strip of a flexible element to a door and to the door frame. Another example is a rope swing, where the rope is tied to a tree branch.

Flexure bearings have the advantage over most other bearings that they are simple and thus inexpensive. They are also often compact, light weight, have very low friction,

Figure 1: A Living Hinge (a Type of Flexure Bearing), on the Lid of a Tic Tac Box



and are easier to repair without specialized equipment. Flexure bearings have the disadvantages that the range of motion is limited, and often very limited for bearings that support high loads.

A flexure bearing relies on the bearing element being made of a material which can be repeatedly flexed without disintegrating. However, most materials fall apart if flexed a lot. For example, most metals will fatigue with repeated flexing, and will eventually snap. Thus, one part of flexure bearing design is avoiding fatigue. Note, however, that fatigue is important in other bearings. For example, the rollers and races in a rolling-element bearing fatigue as they flatten against each other.

Flexure bearings can give very low friction and also give very predictable friction. Many other bearings rely on sliding or rolling motions, which are necessarily uneven because the bearing surfaces are never perfectly flat. A flexure bearing operates by bending of materials, which causes motion at microscopic level, so friction is very uniform. For this reason, flexure bearings are often used in sensitive precision measuring equipment.

Flexure bearings are not limited to low loads, however. For example, the drive shafts of some sports cars replace cardan universal joints with an equivalent joint called a rag joint which works by bending rubberized fabric. The resulting joint is lighter yet is capable of carrying hundreds of kilowatts, with adequate durability for a sports car.

Many flexure bearings are combined with other elements. For example, many motor vehicles use leaf springs. The spring both holds the position of the axle as the axle moves (flexure bearing) and provides force to support the vehicle (springing). In many cases it is not clear where flexure bearing leaves off and something else takes up. For example, turbines are often supported on flexible shafts so an imperfectly balanced turbine can find its own center and run with reduced vibration. Seen one way, the flexible shaft includes the function of a flexure bearing; seen another, the shaft is not a “bearing”.

Linear Motion Bearings

A linear-motion bearing or linear slide is a bearing designed to provide free motion in one dimension. There are many different types of linear motion bearings and this family of products is generally broken down into two sub-categories: rolling-element and plane.

Rolling-Element Bearing

A rolling-element bearing is generally composed of a sleeve-like outer ring and several rows of balls retained by cages. The cages were originally machined from solid metal and were quickly replaced by stampings. It features smooth motion, low friction, high rigidity and long life.

Ball Bearing Slides

Also called “ball slides”, ball bearing slides are the most common type of linear slide. Ball bearing slides offer smooth precision motion along a single-axis linear design, aided by ball bearings housed in the linear base, with self-lubrication properties that increase reliability. Ball bearing slide applications include delicate instrumentation, robotic assembly, cabinetry, high-end appliances and clean room environments, which primarily serve the manufacturing industry but also the furniture, electronics and construction industries. For example, a widely used ball bearing slide in the furniture industry is a ball bearing drawer slide.

Commonly constructed from materials such as aluminum, hardened cold rolled steel and galvanized steel, ball bearing slides consist of two linear rows of ball bearings contained by four rods and located on differing sides of the base, which support the carriage for smooth linear movement along the ball bearings. This low-friction linear movement can be powered by either a drive mechanism, inertia or by hand. Ball bearing slides tend to have a lower load capacity for their size compared to other linear slides because the balls are less resistant to wear and abrasions. In addition, ball bearing slides are limited by the need to fit into housing or drive systems.

Roller Slides

Also known as crossed roller slides, roller slides are non-motorized linear slides that provide low-friction linear movement for equipment powered by inertia or by hand. Roller slides are based on linear roller bearings, which are frequently criss-crossed to provide heavier load capabilities and better movement control. Serving industries such as manufacturing, photonics, medical and telecommunications, roller slides are versatile and can be adjusted to meet numerous applications which typically include clean rooms, vacuum environments, material handling and automation machinery.

Consisting of a stationary linear base and a moving carriage, roller slides work similarly to ball bearing slides, except that the bearings housed within the carriage are cylinder-shaped instead of ball shaped. The rollers crisscross each other at a 90° angle and move between the four semi-flat and parallel rods that surround the rollers. The rollers are between "V" grooved bearing races, one being on the top carriage and the other on the base. The travel of the carriage ends when it meets the end cap, a limiting component. Typically, carriages are constructed from aluminum and the rods and rollers are constructed from steel, while the end caps are constructed from stainless steel.

Although roller slides are not self-cleaning, they are suitable for environments with low levels of airborne contaminants such as dirt and dust. As one of the more expensive types of linear slides, roller slides are capable of providing linear motion on more than one axis through stackable slides and double carriages. Roller slides offers line contact versus point contact as with ball bearings, creating a broader contact surface due to the consistency of contact between the carriage and the base and resulting in less erosion.

Plain Bearing

Plain bearings are very similar in design to rolling-element bearings, except they slide without the use of ball bearings.

Dovetail Slides

Dovetail slides, or dovetail way slides are typically constructed from cast iron, but can also be constructed from hard-coat aluminum, acetal or stainless steel. Like any bearing, a a dovetail slide is composed of a stationary linear base and a moving carriage. a Dovetail carriage has a v-shaped, or dovetail-shaped protruding channel which locks into the linear base's correspondingly shaped groove. Once the dovetail carriage is fitted into its base's channel, the carriage is locked into the channel's linear axis and allows free linear movement. When a platform is attached to the carriage of a dovetail slide, a dovetail table is created, offering extended load carrying capabilities.

Since dovetail slides have such a large surface contact area, a greater force is required to move the saddle than other linear slides, which results in slower acceleration rates. Additionally, dovetail slides have difficulties with high-friction but are advantageous when it comes to load capacity, affordability and durability. Capable of long travel, dovetail slides are more resistant to shock than other bearings, and they are mostly immune to chemical, dust and dirt contamination. Dovetail slides can be motorized, mechanical or electromechanical. Electric dovetail slides are driven by a number of different devices, such as ball screws, belts and cables, which are powered by functional motors such as stepper motors, linear motors and hand wheels. Dovetail slides are direct contact systems, making them fitting for heavy load applications including CNC machines, shuttle devices, special machines and work holding devices. Mainly used in the manufacturing and laboratory science industries, dovetail slides are not ideal for high-precision applications.

Why Flexural Bearings?

A bearing is a rotating component which is placed between to parts to allow them to move easily. A flexure bearing is just that it allows to parts to move with each other with no trouble. There are several different types of bearings and they all use a different shape to move the two parts, for example ball bearings use little balls, needle bearings use very thin needle tubes, roller bearings use tubes as well but are larger than needle size.

A flexure bearing allows the motion of two parts by bending a load element. What does this mean? It means it takes two materials and fixes them together by a flexure bearing and this bearing allows them to move. Think of a door on a hinge. The door needs to be attached the door frame but it also needs to be opened and closed and so a hinge is used. The door and door frame are the two materials and the hinge is the flexure bearing allowing it to move freely.

Flexure bearings are great because they are very simple to manufacture especially compared to some other types of bearings and they are easy and cheap to replace and so maintenance isn't a large issue. Flexure bearings have many advantages including they do not jitter or wobble as they are fixed into place, this minimalists the risk of damage to the bearing and the two rigid parts, they can operate in a vacuum, they have virtually an unlimited life span if the atmosphere is not corrosive, they can work in high and low temperatures, and they don't make a noise when they are operating. Of course these types of bearings have some limitations and downfalls for example they can only be used on materials that do not disintegrate after being repeatedly flexed, their angular excursion is limited, they are more expensive than ball bearings, and they are harder to install.

All bearings have their up and down sides to them and flexure bearings are the same, they are great for some applications where in others another bearing is better suited.

DESIGN CONSIDERATION

Today still most of the linear coolers are driven by a moving coil motor. Moving coils are well known form the loudspeaker technology and are a standard technology today. The motor can easily be calculated according to the Lorentz Force Law; also high motor efficiencies can be obtained as eddy current losses are low. However, the moving coil concept is associated with several drawbacks which can be overcome when changing to moving magnets.

The Benefits are

- Coil can be placed outside the helium vessel, i.e., no out gassing into the working gas
- No electrical feed through needed
- No flying current lead: eliminates potential failure mechanism and reduces complexity.

The goals for the compressor design were:

- Compliance with the One Watt Linear interface (SADA2): diameter 60.45 mm, length < 123 mm
- Lowest complexity and less part for low fabrication cost.
- Low amount of out gassing materials inside helium vessel.
- Suitable for operation of all relevant Stirling and Pulse Tube cold fingers.
- Implementation of efficient alignment process for serial production.
- Motor impedance compatible to current linear coolers.
- Suitable as form, fit and function replacement for SADA2.

To allow an easy exchange with the common 1 W linear coolers the new compressor SF100 has a cable exit for both motor halves on one end of the compressor, located on the same side as the fill port. This concept also allows a compressor version with integrated cooler control electronics, see Figures 2 and 3. The achieved reduction in complexity can be seen in Table 1, some key figures are compared for the Flexure Bearing Moving Magnet compressor SF100 with the conventional moving coil compressor SL100. Although achieving the same performance data, the compressor length was reduced by more than 4 mm, the weight was reduced a little.

Table 1: Comparison of Design Characteristics Between Linear Compressor and Flexure Bearing Compressor

Compressor Type	Units	SL100 (Moving Coil)	SF100: Moving Magnets, Flexure Bearings
Number of Parts	%	100	44
Thread Joints	%	100	29
Adhesive Joint	%	100	40
Welding Joint	%	100	375
Weight Non-Metallic Components in Helium Vessel	%	100	3
Compressor Weight	g	1720	1680
Compressor Length	mm	122.2	117.9

Figure 2: Flexure Bearing Spring and FEM Simulation for Max. Geometrical Stroke

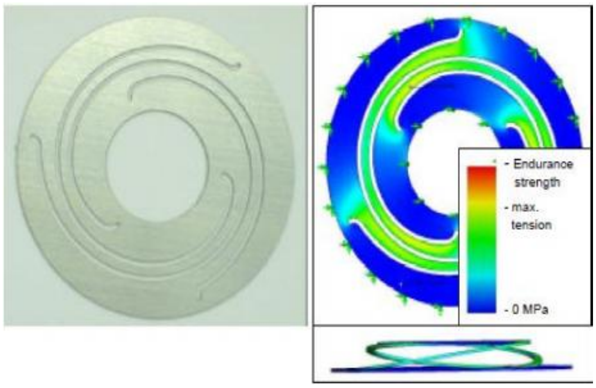
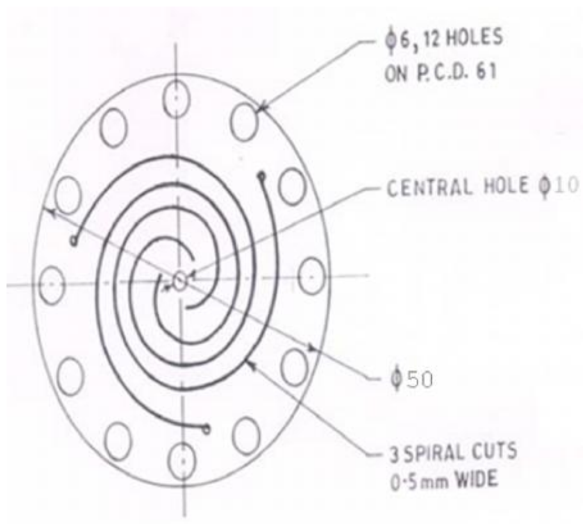


Figure 3: Typical Flexure Unit of the Suspension System



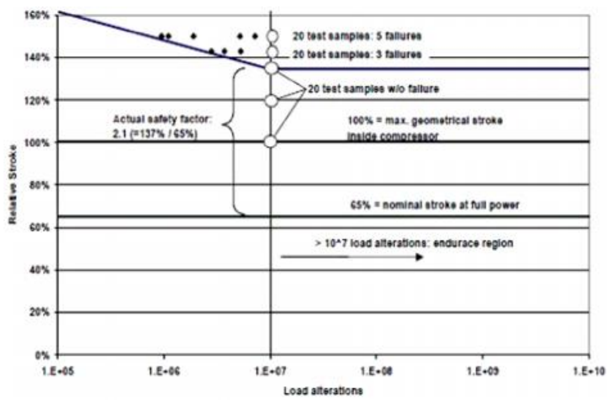
Flexure Bearing Design

A high end spring material was chosen for the springs to achieve the highest radial stiffness for the given spring diameter and needed axial stiffness. The optimized spring design was developed by systematic FEM simulation at both AIM and also external institutes. Figure 1 shows the optimized spring design and a result from FEM calculation. The highest strain within the spring at max geometrical piston stroke is at least 35% below the specified fatigue limit of the material. This ensures that the spring is operated well within the endurance level of the material.

For highest compactness of the compressor a welding joint for the spring was developed. All relevant parameters like material and tolerance for spring and adjacent compressor parts as well as welding details had to be considered to ensure robust welding seam in serial production. Also the heat distribution zone with impact on the material characteristics inside the spring had to be evaluated.

To verify the simulation results the welded spring was qualified and a Wohler curve was generated. This diagram is shown in Figure 4. For each stroke 20 springs were tested. During the tests the actual stroke was monitored with a laser stroke sensor. The results confirm more than 35% margin in tension to the endurance level at max geometrical stroke. For nominal stroke at full input power (= 65% of max stroke) the safety factor is about 2.1 and thus sufficient.

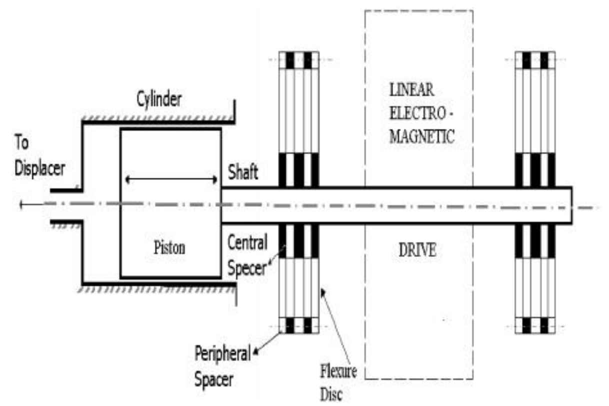
Figure 4: Wohler Curve for Flexure Bearing Spring (Mounting Condition Identical to Cooler)



Design Requirements of Compressor Flexure Bearing

The radial stiffness of the flexure bearing assembly (Figure 18) should be high enough to support the clearance seal under the effect of the suspended mass which comprises mainly of the piston-shaft sub assembly, coil and coil former. In order to evaluate the radial stiffness requirement for each of the flexure discs that constitute the two stacks of the whole bearing assembly, a simplified model proposed and piston deflection formulated.

Figure 5: Assembly of the Flexure Bearing



Piston Alignment

To ensure contact-less operation of the piston inside the sleeve over full stroke and typical operating conditions an active alignment process was developed. Heart of the alignment is a computerized alignment—and welding station, sees Figure 6. In the alignment station a precision of the adjustment of about 1 μm is achieved. After finishing the alignment, the components are welded directly in the aligned position. This process was optimized to allow a fast alignment in serial production.

Figure 6: Computerized Alignment and Welding Station

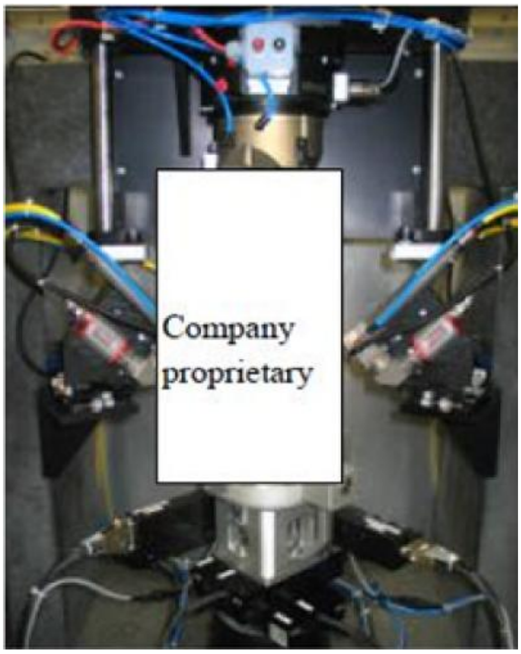


Figure 7: SF100A with 14 mm Pulse Tube Cold Finger

Figure 8: Outline Dimension of Cooler SF100A

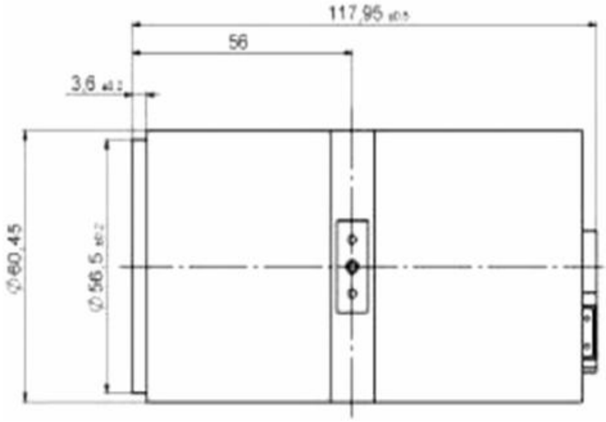


Figure 9: Flexure Bearing Compressor SF100A with 13 mm AIM Sleeve Coldfinger

Figure 10: Outline Dimension of Cooler SF100B

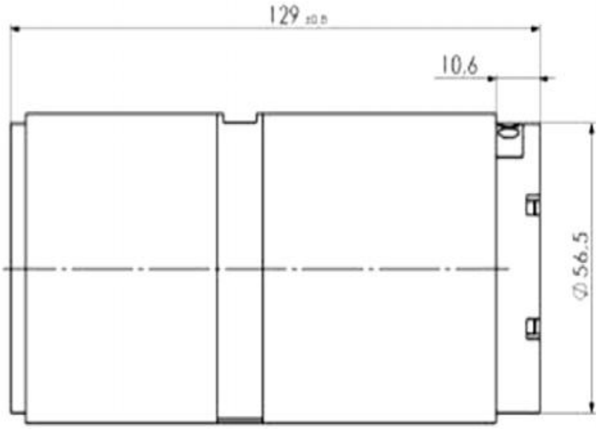


Figure 11: Flexure Bearing Compressor SF100B

PERFORMANCE DATA

The compressor is optimized for operation with different types of Sterling and Pulse Tube cold fingers:

- ½" SADA cold finger
- 13 mm AIM sleeve cold finger
- 12 mm AIM standard cold finger
- 14 mm AIM Pulse Tube cold finger

Smaller cold fingers can be used as well, however, to keep resonance condition inside the compressor and to achieve a high motor efficiency a reduction in piston diameter is needed. The compressor design takes account for that. As part of the qualification efforts a detailed performance mapping was carried out. Performance data for the ½" SADA cold finger are shown in Figures 9 to 12 show the cooling capacity vs. the AC input power for three different ambient temperatures with curves for different cold tip temperatures each.

All given data are measured with the same charge pressure at constant operating frequency (55 Hz), i.e., without any adaptation

Figure 12: Input Power for Constant Heat Load of 400 mW vs. Ambient Temperature for 77 K, 84 K and 95 K

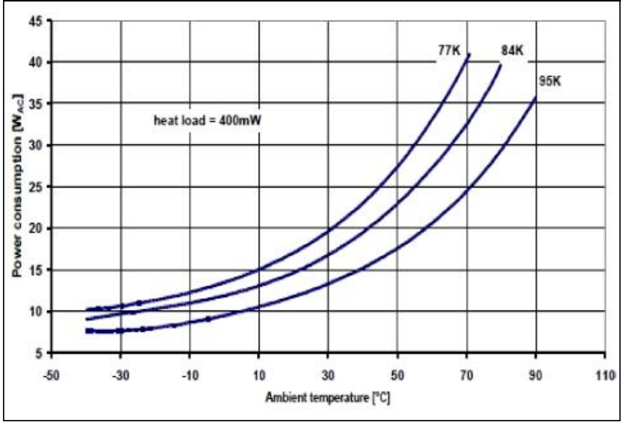


Figure 13: Cooling Capacity for 71 °C Ambient Temperature, Constant Coldtip Temperature

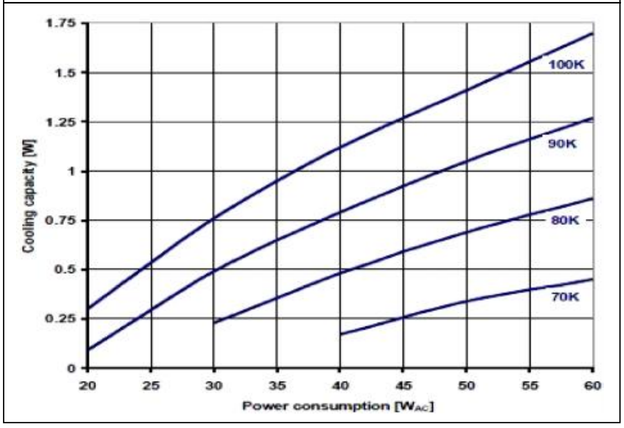


Figure 14: Cooling Capacity for 23 °C Ambient Temperature, Constant Coldtip Temperature

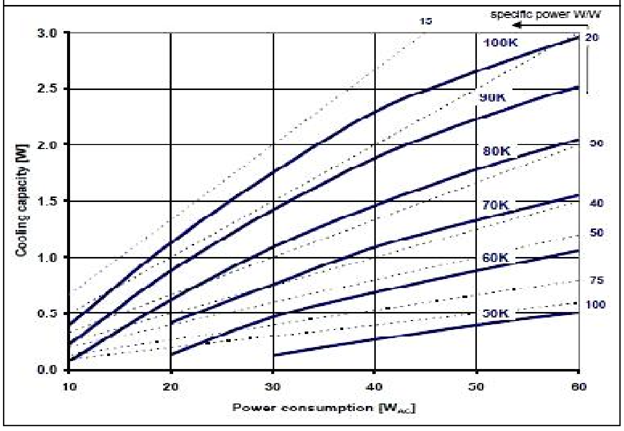


Figure 15: Cooling Capacity at 80 K, Curves for Constant Ambient Temperature

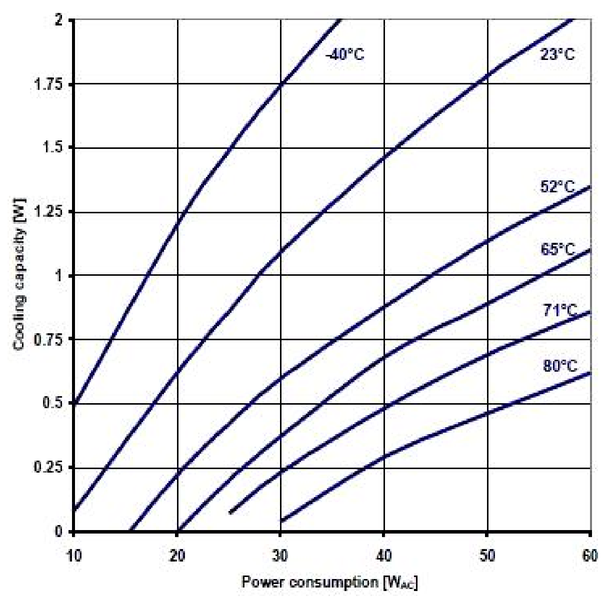
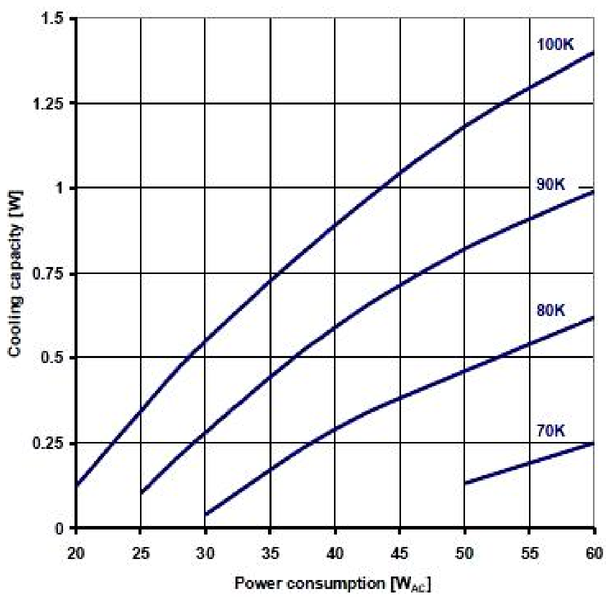


Figure 16: Cooling Capacity for 80 °C Ambient Temperature, Constant Coldtip Temperature



for ambient or cold tip temperature. When changing these parameters, a further improvement can be achieved. The performance data of the AIM 13 mm sleeve cold finger are slightly better compared to the SADA cold finger. This is achieved by improved heat removal at the expander warm end and a reduced longitudinal conduction due to reduced sleeve length. At 23 °C ambient temperature more than 2 W cooling capacities at 80 K are achieved with 60 W of input power, at 71 °C more than 0.8 W@80 K are available. Compared to the SL100 cooler the cool down time is improved.

The given ambient temperature is the air temperature. Surface temperatures are up to 15 K higher. The given heat load is the power of the electrical heater on a 1,440 J copper thermal mass. Data for the compressor AC input power are shown in the diagrams. The compressor is designed for AC input power of up to 70 W. To allow the cooler operation with typical power supplies, the dynamic impedance is matched to about 4 Ohm. Thus the cooler can be operated with the AIM controller AM7 [2] and also the HTI board as used in the SADA program for instance. The AM7 can be used in an external housing and also as part of the SF100B compressor directly attached to the compressor housing.

ANALYSIS OF FLEXURE BEARING IN ANSYS

Material Data for Flexure Bearing

Copper Alloy

Young's Modulus	1.1e+011 Pa = 1.1 e+5 N/mm^2
Poisson's Ratio	0.34
Density	8300. Kg/m^3
Thermal Expansion	1.8e-005 1/°C

Tensile Yield Strength	2.8e+008 Pa
Compressive Yield Strength	2.8e+008 Pa
Tensile Ultimate Strength	4.3e+008 Pa
Compressive Ultimate Strength	0. Pa
K1	0.238
Thermal Conductivity	401. W/m·°C
Specific Heat	385. J/kg·°C
Relative Permeability	1
Resistivity	1.724e-008 Ohm·m

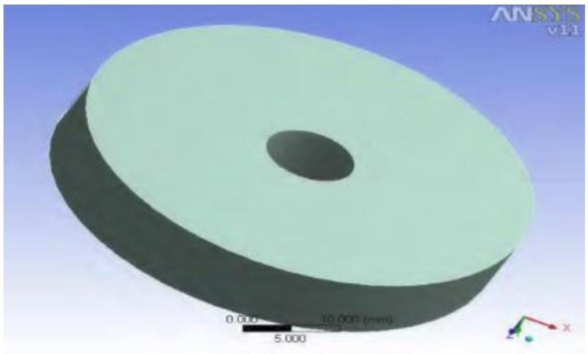
Aluminum Alloy

Young's Modulus	7.1e+010 Pa = 7.1 e+4 N/mm^2
Poisson's Ratio	0.33
Density	2770. kg/m^3
Thermal Expansion	2.3e-005 1/°C
Tensile Yield Strength	2.8e+008 Pa
Compressive Yield Strength	2.8e+008 Pa
Tensile Ultimate Strength	3.1e+008 Pa
Compressive Ultimate Strength	0. Pa
K1	0.238
Specific Heat	875. J/kg·°C
Relative Permeability	1.
Resistivity	5.7e-008 Ohm·m

Analysis of Deformation

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 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. Figure 17 shows a typical disc with hole for which parameter '*a*' and '*b*' have been defined.

Figure 17: Typical Disc



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

$$W_{max} = k*q*a^4/E*h^3$$

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

Given Data for Copper

Outer Diameter a = 50 mm

Inner Diameter b = 10 mm

Thickness = t = 5 mm

Material Used = Copper

Loading Point = at center peripheral surface

Force Applied = 5 N

Co-efficient = $K1$ from data book is enclosed according to a/b

Pressure = $P = F/A$ for Force 5 N

$= F/2 \times \circledR \times R \times t$

$= 5/2 \times \circledR \times 5 \times 5 = 0.0318$ N/mm^2

Wmax = $k1*P*a^4/E*h^3$

$= 0.238*0.0318*25^4/1.1*10^5*5^3$

= 0.00021501 mm

Pressure = $P = F/A$ for Force 4 N

$= F/2 \times \circledR \times R \times t$

$= 4/2 \times \circledR \times 5 \times 5 = 0.0254$ N/mm^2

$W_{max} = k1*P*a^4/E*h^3$

$= 0.238*0.0254*25^4/1.1*10^5*5^3$

= 0.0001717 mm

Pressure = $P = F/A$ for Force 3 N

$= F/2 \times \circledR \times R \times t$

$= 3/2 \times \circledR \times 5 \times 5 = 0.0190$ N/mm^2

$W_{max} = k1*P*a^4/E*h^3$

$= 0.238*0.019*25^4/1.1*10^5*5^3$

= 0.000128 mm

Pressure = $P = F/A$ for Force 2 N

$= F/2 \times \circledR \times R \times t$

$= 2/2 \times \circledR \times 5 \times 5 = 0.0127$ N/mm^2

$W_{max} = k1*P*a^4/E*h^3$

$= 0.238*0.0127*25^4/1.1*10^5*5^3$

= 0.000085 mm

Pressure = $P = F/A$ for Force 1 N

$= F/2 \times \circledR \times R \times t$

$= 1/2 \times \circledR \times 5 \times 5 = 0.0063$ N/mm^2

$W_{max} = k1*P*a^4/E*h^3$

$= 0.238*0.0063*25^4/1.1*10^5*5^3$

= 0.000043 mm

Given Data for Aluminum

Outer diameter a = 50 mm

Inner diameter b = 10 mm

Thickness = t =5 mm

Material Used = Aluminum

Loading Point = at center peripheral surface

Force Applied = 5 N

Co-efficient = $K1$ from data book is enclosed

Pressure = $P = F/A$ for Force 5 N

$= F/2 \times \circledR \times R \times t$

$= 5/2 \times \circledR \times 5 \times 5 = 0.0318$ N/mm^2

$W_{max} = k1*P*a^4/E*h^3$

$= 0.238*0.0318*25^4/7.1*10^4*5^3$

= 0.0003331 mm

Pressure = $P = F/A$ for Force 4 N

$= F/2 \times \circledR \times R \times t$

$= 4/2 \times \circledR \times 5 \times 5 = 0.0254$ N/mm^2

$W_{max} = k1*P*a^4/E*h^3$

$= 0.238*0.0254*25^4/7.1*10^4*5^3$

= 0.0002660 mm

Pressure = $P = F/A$ for Force 3 N

$= F/2 \times \circledR \times R \times t$

$= 3/2 \times \circledR \times 5 \times 5 = 0.0190$ N/mm^2

$W_{max} = k1*P*a^4/E*h^3$

$= 0.238*0.0190*25^4/7.1*10^4*5^3$

= 0.0001990 mm

Pressure = $P = F/A$ for Force 2 N

$= F/2 \times \circledR \times R \times t$

$= 2/2 \times \circledR \times 5 \times 5 = 0.0127$ N/mm^2

$W_{max} = k1*P*a^4/E*h^3$

$= 0.238*0.0127*25^4/7.1*10^{4}*5^3$

$= 0.000133$ mm

Pressure = $P = F/A$ for Force 1 N

$= F/2 \times$ ®:$\times R \times t$

$= 1/2 \times$ ®:$\times 5 \times 5 = 0.0063$ N/mm^2

$W_{max} = k1*P*a^4/E*h^3$

$= 0.238*0.0063*25^4/7.1*10^{4}*5^3$

$= 0.0000659$ mm

Table 2: Comparisition Table for Theoretical Calculations

S. No.	Force	Pressure N/mm^2	For Copper	For Aluminum
			Axial Deformation	Axial Deformation
1.	5 N	0.0318	0.00021501 mm	0.0003331 mm
2.	4 N	0.0254	0.0001717 mm	0.0002660 mm
3.	3 N	0.0190	0.000128 mm	0.0001990 mm
4.	2 N	0.0127	0.000085 mm	0.000133 mm
5.	1 N	0.0063	0.000043 mm	0.0000659 mm

Graph between force Vs Axial Deformation, the force is taken in x-axis and Axial Deformation in y-axis.

Figure 18: Graph Between Force vs. Axial Deformation

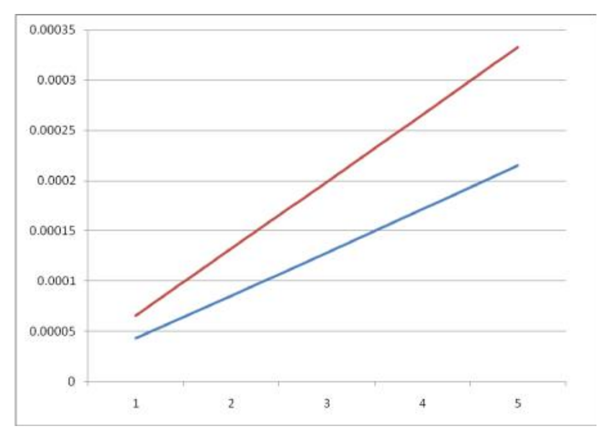


Table 3: Comparison Table for Theoretical Calculations According to the Ansys

S. No.	Force	Pressure N/mm^2	For Copper	
			Axial Deformation	Axial Deformation
1.	5 N	0.0318	0.00021501 mm	0.0003331 mm
2.	4 N	0.0254	0.0001717 mm	0.0002660 mm
3.	3 N	0.0190	0.000128 mm	0.0001990 mm
4.	2 N	0.0127	0.000085 mm	0.000133 mm
5.	1 N	0.0063	0.000043mm	0.0000659 mm

CONCLUSION

We are design the bearing by using copper material, the copper material is stronger than the aluminum according to the material data, presently the bearings are design by the aluminum, and in this project we design the bearing by using copper material. We analysis the deformation on the bearing at different loads.

The deflection deformations are found out in theoretically and FEA Method by using Ansys. Comparing the both results of the Copper and Aluminum, copper is having the more deflection deformations in the Flexure bearing at different loads. Finally the copper bearing is gives the more deflections.

REFERENCES

1. Benschop A A J, Mullié J and Meijers M (2000), "High-Reliability Coolers at Thales Cryogenics", Proc. SPIE 4130, pp. 385-648.

2. Groep vd W L, Mullié J C, Willems D W J and Benschop T (2008a), "Development of a 15 W Coaxial Pulse-Tube Cooler", *Cryocoolers*, Vol. 15, pp. 157-165.

3. Groep vd W, Mullié J, Benschop T, Wordragen v F and Willems D (2008b),

"Design and Optimization of the Coaxial Pulse-Tube Cooler", *Adv. in Cryogenic Engineering*, Vol. 53, pp. 1667-1674.

4. Linder M (2007), "ESA Cryogenics and Focal Plane Cooling", ESA Harmonization Meeting.

5. Mullié J C, Bruins P C, Benschop T and Meijers M (2004), "Development of the LSF95xx 2nd Generation Flexure Bearing Coolers", *Cryocoolers*, Vol. 13, pp. 71-76.