

Selection and Analysis of Spherical Plain Bearing

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Abstract

Spherical plain bearings are used to accommodate shafts or rods with varying amount of misalignment. They are designed to manage high radial loads and to provide consistent performance under condition of extreme speed and critical stresses. One of the applications of this bearing is vehicle suspension. There is wide range of spherical plain bearing which can be used to withstand high alternating loads, shock loads and radial loads with unilateral load direction. In this seminar, the aim is to select proper type spherical plain bearing as a replacement to ball joint which is going to accommodate at the knuckle end of upper and lower arm of front suspension of QUAD BIKE and to carry out analysis of spherical plain bearing by using FEM method. It has been seen that there is better performance of spherical plain bearing than the ball joint and having wide range of load carrying capacity. The study consists of selection of bearing size which includes, 1) Load rating 2) Rating life 3) Specific bearing load 4) Temperature conditions and 5) Design of bearing arrangements. Load rating includes static as well as dynamic load rating. Dynamic load rating is used for calculations when the bearing is subjected to dynamic stress. Dynamic stresses occur when tilting, oscillatory or rotational movements are made under load, e.g. resulting from vibrations, or loads which are applied alternately at high frequency. Static load rating represents the load which is taken by bearing when static contact stress of bearing contact surface reaches the material stress limit and it remain stationary under load and it is considered when dynamically loaded bearings are subjected to heavy shock loads. For accounting the loads, lots of research and efforts have been done in this field to reduce mechanical failures and

improved sustainability by using spherical plain bearing. Fatigue analysis is carried out on solid-Thinking Inspire 2018 and analytical modelling in CATIA V5.

Keywords: CATIA; Finite element method, HYPERWORKS, Spherical plain bearing, Static and dynamic load rating

INTRODUCTION

A bearing is nothing but one that bears the load and reduces the friction.

A bearing is a machine element used to support and guide a rotating, oscillating or sliding shaft, pivot or wheel. Whenever a shaft rotates, it needs a bearing for smooth, effective operation.

A bearing is designed to-

- Reduce friction
- Support a load (both radial and axial)
- Guide moving parts-wheel, shafts

LITERATURE REVIEW

Xiumei Wang et.al [1] has studied “Numerical investigation on press forming of self-lubricating spherical plain bearing.” In this work, the press forming of outer ring in a typical spherical plain bearing is numerically investigated using finite element modelling and simulations and focus on the final press-formed shape of the ring

after spring back as well as contact pressure and residual stress. Simulation results are found consistent with those from the actual measurements of bearing.

Simulation results are found consistent with those from the actual measurements of bearing osculation and they can provide theoretical guides for enhancing performance of the spherical plain bearings and improving the shape of the outer ring blank or the die design consequently. In the press forming simulation, the plastic deformations occurred in the outer and inner rings after the press forming can be scaled by the effective plastic strains obtained from the simulations. The inner ring has relatively lower deformations compared to those in the outer ring.

Xin Fang et.al [2] has studied “Newly developed theoretical solution and numerical model for conformal contact pressure distribution and free-edge effect

in spherical plain bearings.” In this work, based on the Fang model for complete spherical surfaces, they proposes a new accurate theoretical solution for calculating the conformal contact pressure in spherical plain bearings.

The study investigated extensive 3D finite element (FE) models of bearings verified by optical experiments and then further improved. Because the bearing width is finite, free-edge effect, which causes a deviation in pressure between the two directions, is prominent. This effect varies with the clearance and load and produces a high stress gradient and concentration near the edges. There is an approximate linear relationship between the pressure concentration factors and contact radius proportion. The contact pressure distribution is an important parameter for the design and wear of spherical plain bearings.

Classical solutions cannot be used to describe them in spherical plain bearings. Furthermore, the contact surfaces in bearings are incomplete spherical surfaces and the maximum radius of the contact region exceeds the width of the outer ring. Additionally, there are no constraints on the edge

surfaces of the outer ring during operation, which will result in the free-edge effect.

Lei CAO et.al [3] has studied “Three-dimensional Thermal Analysis of Spherical Plain Bearings with Self-lubricating Fabric Liner”. In this work, Finite element analysis and corresponding experimental comparisons of temperature were performed to investigate the thermal behaviour of spherical plain bearings with self-lubricating fabric liner. A sequentially coupled, 3D, thermo-mechanical finite element analysis model of the bearing system was built up, in which the steady-state temperature distribution from the thermal analysis was applied as a body load to the structural model. As a result, the maximum steady-state temperature of 78.1°C, von Mises stress of 299MPa, displacement of 0.0806mm along Z axis of the bearing are presented, together with the maximum contact pressure of 324MPa which are significant in the structural design and optimization of these bearings.

The effect of temperature rise on the contact pressure distribution is discussed. In order to simulate the practical work conditions of the

aerospace mechanical bearings, the ambient temperature changes and material property changes with temperature rise should be taken into consideration.

Jianjun Lu et.al [4] has studied “Numerical analysis of self-lubricating radial spherical plain bearings and investigations on fatigue damage mechanisms of the liner”. In this work, with ABAQUS software, a 3D simplified model for finite element analysis (FEA) was established for a radial spherical plain bearing with Kevlar/PTFE fabric woven liner, which can simulate different motions of the inner ring. Motion directions, friction forces and wear depths can cause offset distributions of higher concentrated contact pressures on the liner.

The amplitudes or offset directions of higher contact pressures change periodically during motions. Different fatigue damage mechanisms such as surface fatigue wear, impact effect, and sweep deformation of the liner were analysed. Fatigue behaviours of the liner were validated by experimental results. The conclusion was, for self-lubricating radial spherical plain bearings, distributions and amplitudes of contact

pressures as well as deformations of the liner can be influenced by the liner performances, the motion types of the bearings, the friction forces and wear depths during work process. Under tilt motion in liner width direction, friction force can cause off set distribution of higher concentrated contact pressures, which can also be induced by wear depth in liner circumference direction under swivel motion condition.

M. Qiu et.al [5] has studied “Effects of Oscillation Frequency on the tribological properties of self-lubrication Spherical Plain Bearings with PTFE woven liner”. In this work, the working principle and performance characteristics of a self-made oscillation tribo-tester were introduced. Under different oscillation frequency and pressure conditions, the friction coefficient and linear wear of spherical plain bearings with polytetra fluoroethylene (PTFE) woven liner were investigated. By using scanning electron microscopy (SEM) and energy dispersive spectroscopy (EDS), the wear mechanism of the spherical plain bearing was analysed.

The results indicate that with the oscillation frequency increases, the friction coefficient of spherical plain

bearings decreased while the wear loss increased. In the high frequency of 4.8Hz, the reduction ranges of the friction coefficient is not obvious with the contact pressure increased, but the effect of contact pressure on the wear of the spherical plain bearing is great. During the experiment, adhesive wear, abrasive wear took place on the surfaces of woven liners. The conclusion was, the influence of the pressure on linear wear loss at the high-frequency is greater than that of low-frequency, but opposite for the effect of the pressure on friction coefficient. At a lower oscillation frequency, the main wear mechanism is adhesive wear. However, abrasive wear has occurred at a higher oscillation frequency.

To select and analyse spherical plain bearing which is going to accommodate at the knuckle end of upper and lower suspension of ATV QUAD BIKE.

Selection of spherical plain bearing:

- i. The requirement is to select spherical plain bearing having inner diameter = 10mm and which can sustain the loads acting on it and which is going to accommodate on the knuckle of suspension arm of ATV QUAD BIKE.
- ii. For that purpose, first Free Body Diagram (FBD) is drawn to find out the forces acting on the bearing.

PROBLEM STATEMENT

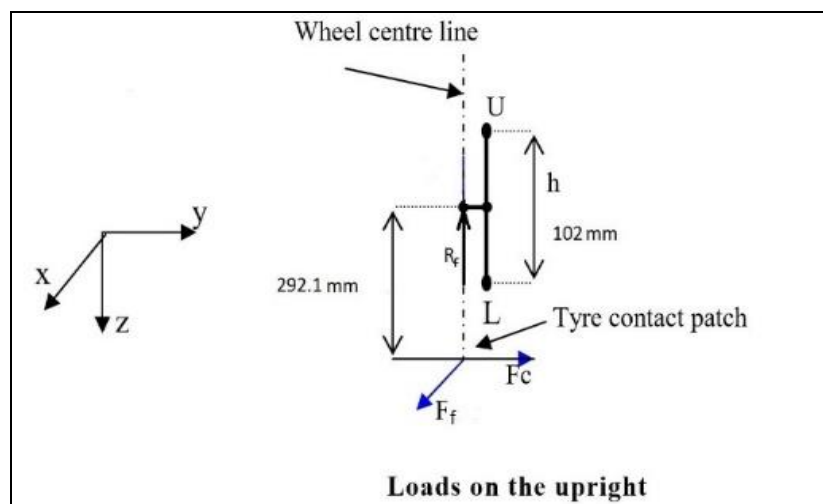


Fig.3.1.1- Free Body Diagram

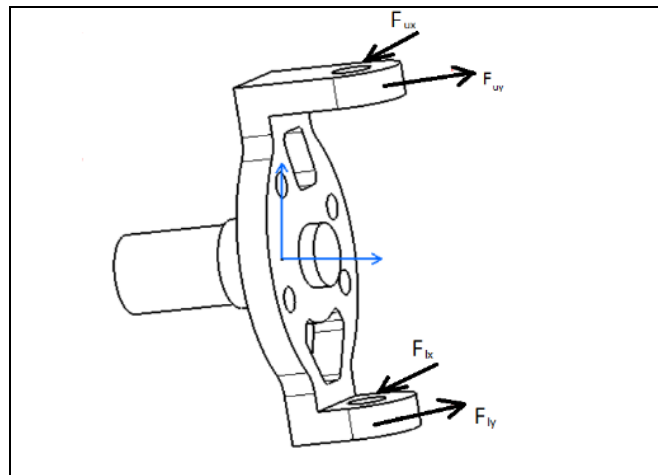


Fig.3.1.2- Forces acting on knuckle

Forces on knuckle end:

Different load cases

1) Running condition:

Driving torque = 550 Nm (max. At 1st gear)
Normal reaction = $R_f = 784.575 \text{ N}$

2) Cornering:

Driving torque = 550 Nm
Lateral forces = $F_c = 450.279 \text{ N}$
Normal reaction = 735.45 N

3) Braking:

Normal reaction = 453.46 N (Longitudinal weight transfer)
Braking torque = 550 Nm

Force at u:

$$F_{uy} = \frac{Mc}{0.081} = 2435.55 \text{ N (in y direction)}$$

$$F_{ux} = \frac{Mf}{0.081} = 6365.92 \text{ N (in x direction)}$$

Forces at l:

$$F_{ly} = \frac{Mc}{0.076} = 2595.78 \text{ N (in y direction)}$$

$$F_{lx} = \frac{Mf}{0.076} = 6784.73 \text{ N (in x direction)}$$

iii) These are the forces acting on the knuckle end of upper and lower suspension arm which ultimately are the forces acting on the spherical plain bearing.

iv) Forces F_{ux} and F_{uy} are the forces acting on the bearing which is placed at the knuckle end of upper suspension arm and F_{lx} and F_{ly} are the forces acting on the bearing at knuckle end of lower suspension arm.

v) Spherical plain bearing should be selected in such a way that it should sustain these loads and also should sustain tilting of arm without failure.

3.1.2 Selection of bearing:

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SPHERICAL BUSHINGS
Steel-on-steel Spherical Bushings



GE ··· G

(Shaft dia. 6 – 120mm)

Shaft dia. mm	Identification number		Mass (Ref.) kg	Boundary dimensions mm								Permissible tilting angle degree	
	Without seals	With seals		d	D	B	C	d ₁	r ₁ ⁽¹⁾ mm	r ₂ ⁽¹⁾ mm	α	α ₁	
6	GE 6G	—	0.010	6	16	9	5	13	0.3	0.3	21	—	
8	GE 8G	—	0.015	8	19	11	6	16	0.3	0.3	21	—	
10	GE 10G	—	0.022	10	22	12	7	18	0.3	0.3	18	—	
12	GE 12G	—	0.041	12	26	15	9	22	0.3	0.3	18	—	
15	GE 15GS	GE 15GS-2RS	0.059	15	30	16	10	25	0.3	0.3	16	13	
17	GE 17GS	GE 17GS-2RS	0.083	17	35	20	12	29	0.3	0.3	19	16	
20	GE 20GS	GE 20GS-2RS	0.155	20	42	25	16	35.5	0.3	0.6	17	16	
25	GE 25GS	GE 25GS-2RS	0.215	25	47	28	18	40.7	0.6	0.6	17	15	
30	GE 30GS	GE 30GS-2RS	0.330	30	55	32	20	47	0.6	1	17	16	
35	GE 35GS	GE 35GS-2RS	0.400	35	62	35	22	53	0.6	1	16	15	
40	GE 40GS	GE 40GS-2RS	0.515	40	68	40	25	60	0.6	1	17	14	
45	GE 45GS	GE 45GS-2RS	0.660	45	75	43	28	66	0.6	1	15	13	
50	GE 50GS	GE 50GS-2RS	1.50	50	90	56	36	80	0.6	1	17	16	
60	GE 60GS	GE 60GS-2RS	2.05	60	105	63	40	92	1	1	17	15	
70	GE 70GS	GE 70GS-2RS	3.00	70	120	70	45	105	1	1	16	14	
80	GE 80GS	GE 80GS-2RS	3.60	80	130	75	50	115	1	1	14	13	
90	GE 90GS	GE 90GS-2RS	5.41	90	150	85	55	130	1	1	15	14	
100	GE 100GS	GE 100GS-2RS	6.15	100	160	85	55	140	1	1	14	12	
110	GE 110GS	GE 110GS-2RS	9.70	110	180	100	70	160	1	1	12	11	
120	GE 120GS	GE 120GS-2RS	15.5	120	210	115	70	180	1	1	16	15	

Notes⁽¹⁾ Minimum allowable values of chamfer dimensions r₁ and r₂.
⁽²⁾ When spherical bushings are used with full tilting angle, the shaft shoulder dimension must be less than the maximum value of d₁.
 Remarks 1. GE: G has no oil hole. Others are provided with an oil groove and two oil holes on the inner ring and outer ring, respectively.
 2. No grease is prepacked. Perform proper lubrication.

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SPHERICAL BUSHINGS
Steel-on-steel Spherical Bushings

Fig.3.1.2.1-Selection of bearing from manufacture's catalogue [5]

From the above table bearing to be selected is GE 10G (Steel on Steel).

3.1.3 Properties of GE 10G:

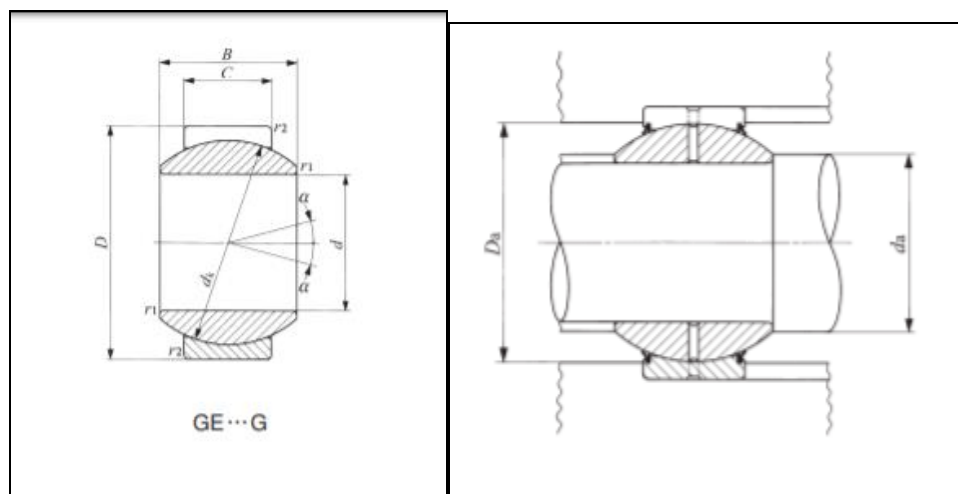


Fig.3.1.2.2-Dimensions of bearing [5] Fig.3.1.2.3-Mounting Dimensions [5]

Table 3.1.2.1-Specifications of bearing

Sr. No	Dimensions	Value
1.	Mass, m	0.022 kg
2.	Inner diameter, d	10 mm
3.	Outer diameter, D	22 mm
4.	Width, B	12 mm
5.	Sphere diameter, d_k	18 mm
6.	Internal clearance, C	7 mm
7.	Permissible angle of tilt, α	18°
8.	Max. inner mounting diameter, d_a	12.5 mm
9.	Max. outer mounting diameter, D_a	19.5 mm
10.	Static load capacity, C_s	74,100 N
11.	Dynamic load capacity, C_d	12,400 N

3.1.4 Theoretical calculation of GE 10G bearing:

Operating parameters:

i) $F_r = F_{ux} = 6365.95$ N (Maximum radial force)

ii) $F_a = F_{uy} = 2435.55$ N (Maximum axial force)

iii) Dynamic load rating, $C_d = 12,400$ N

iv) Static load rating, $C_s = 74,100$ N

v) Operating Temperature, $t = 1000^\circ\text{C}$

Solution:

1) Dynamic load capacity:

The dynamic load capacity C_{dt} considering the influence of bushing temperature can be obtained from the following equation using the temperature factor.

$$C_{dt} = f_t \cdot C_d$$

where, C_{dt} : Dynamic load capacity considering temperature increase N

f_t : Temperature factor (Refer to fig.3.4)

C_d : Dynamic load capacity N (Refer to fig.3.3)

Type of bushing		Temperature °C					
		- 30	+ 80	+ 90	+ 100	+ 120	+ 150
		+ 80	+ 90	+ 100	+ 120	+ 150	+ 180
Steel-on-steel	Without seals	1	1	1	1	1	0.7
	With seals	1	—	—	—	—	—
Maintenance-free	Without seals	1	1	0.9	0.75	0.55	—
	With seals	1	—	—	—	—	—

Fig.3.1.4.1-Temperature factor (ft) [5]

Hence, for 1000C temperature condition
and steel on steel without seal bearing,

$$f_t = 1$$

$$C_{dt} = 12,400 * 1 = 12,400 \text{ N}$$

2) Static load capacity:

From table 3.1,

Static load capacity of GE 10G, $C_s = 74,100 \text{ N}$

3) Equivalent radial load:

The equivalent radial load can be obtained
from the following formula,

$$P = F_r + Y F_a$$

where, P : Equivalent radial load N

F_r : Radial load N

F_a : Axial load N

Y : Axial load factor

Now,

$$F_a / F_r = 2435.55 / 6356.92 = 0.38$$

F_a / F_r	0.1	0.2	0.3	0.4	0.5	> 0.5
Type of bushing						
Steel-on-steel	1	2	3	4	5	Unusable
Maintenance-free	1	2	3	Unusable		

Fig.3.1.4.2-Axial load factor (Y) [5]

So, Y = 4

$$P = 6365.92 + (4 * 2435.55)$$

$$P = 16108.12 \text{ N}$$

4) Life of bearing:

As the actual life is affected by many
factors such as the material of the sliding
surface, the magnitude and direction of
load, lubrication, sliding velocity, etc.,
the calculated life can be used as a
practical measure of expected service
life.

I. Confirmation of pV value:

ii) Before attempting to calculate the life,
the operating conditions should be within
the permissible range by referring to the
pV diagram (fig.3.6).

The contact pressure p and the sliding velocity V are obtained from the following formulae,

$$p = 100 \cdot P / C_{dt}$$

$$p = 29.9 \text{ N/mm}^2 \dots \dots \dots \text{contact pressure}$$

$$V = 5.82 \times 10^{-4} d_k \cdot \beta \cdot f$$

where,

V : Sliding velocity, mm/s

β : Oscillating angle, degrees

f : Number of oscillations per minute

For this particular application,

$2\beta = 40^\circ$ and oscillation half angle = 20° and $f = 60$ turns per minute

Hence, $V = 12.5712 \text{ mm/sec}$

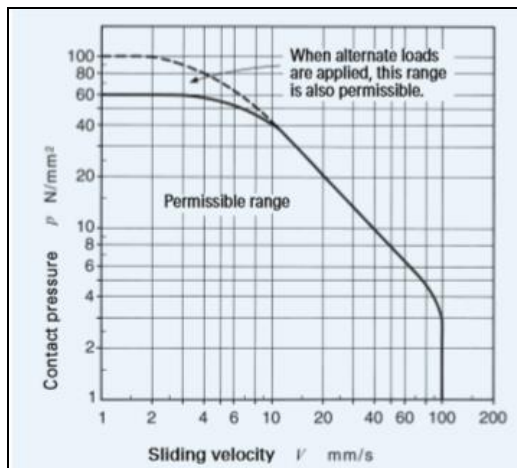


Fig.3.1.4.3-pV diagram of steel on steel spherical bearing

II. Life calculations:

The life of steel-on-steel spherical bushings can be calculated from the following formulae,

$$G = 3.18 \cdot b_1 \cdot b_2 \cdot b_3 \cdot (C_{dt}/P)^2 \times 10^5 \div (d_k \cdot \beta)^{0.5}$$

Where,

G : Life (Total number of oscillations)

b_1 : Load directional factor

b_2 : Lubrication factor

b_3 : Sliding velocity factor

For alternate loads, $b_1 = 5$, $b_2 = 15$ (considering regular lubrication), $b_3 = 1.5$

$G = 1.11 \times 10^6$ number of oscillations =

G = 1 million oscillations

.....total no. of oscillations of bearing before failure

And,

$$L_h = G / (60 \cdot f)$$

Where,

L_h : Life in hours, h

f : Number of oscillations per minute

$L_h = 278$ hours

.....total no. of hours of bearing before failure

3.2 Modelling of spherical plain bearing:

Modelling of spherical plain bearing is carried out in modelling software CATIA V5.

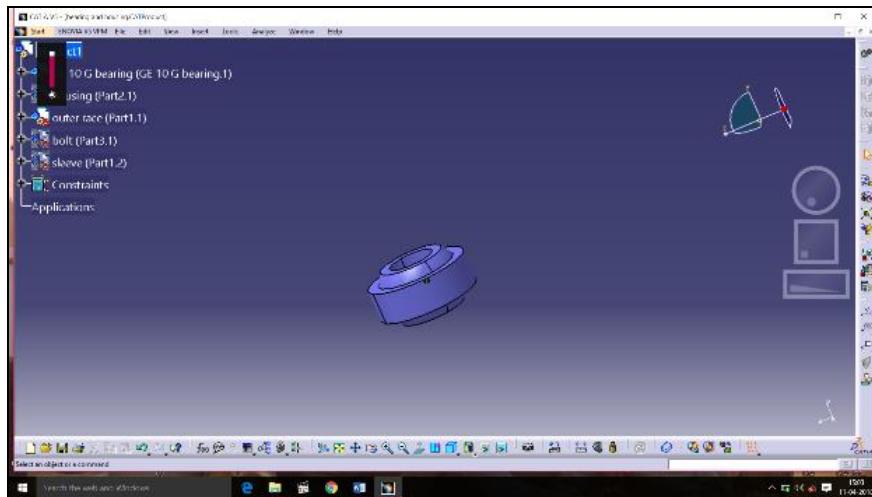


Fig.3.2.1-Modelling of spherical plain bearing

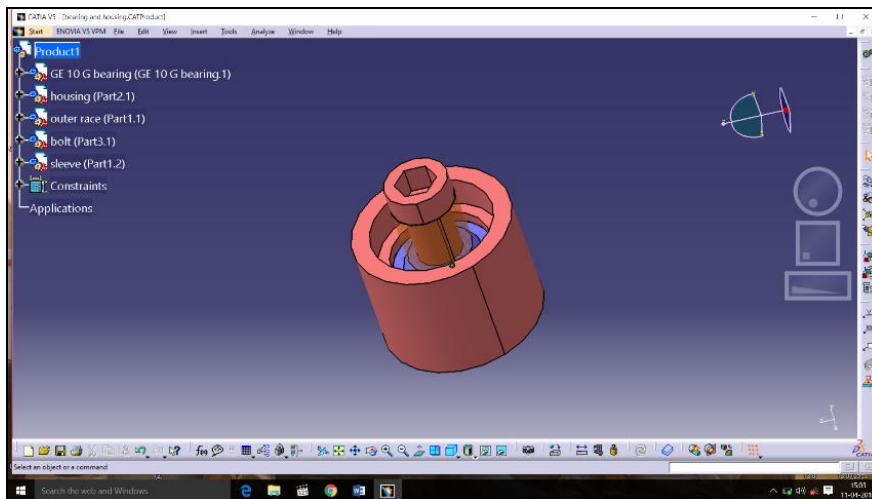


Fig.3.2.2-Assembly

The assembly consists of

- i) GE 10G bearing which consists of inner ring and outer ring
- ii) Housing for bearing
- iii) M10 bolt
- iv) Sleeve

3.3 Fatigue analysis of spherical plain bearing GE 10G:

Fatigue analysis of spherical plain bearing is carried out in solid-Thinking Inspire(2018).

Step 1: Active and lock parts of assembly



Step 2: Application of forces



Both radial and axial forces are applied on the bearing which are perpendicular to each other and maximum values of forces which were calculated earlier are applied on the bearing.

Step 3: Actuate the motion

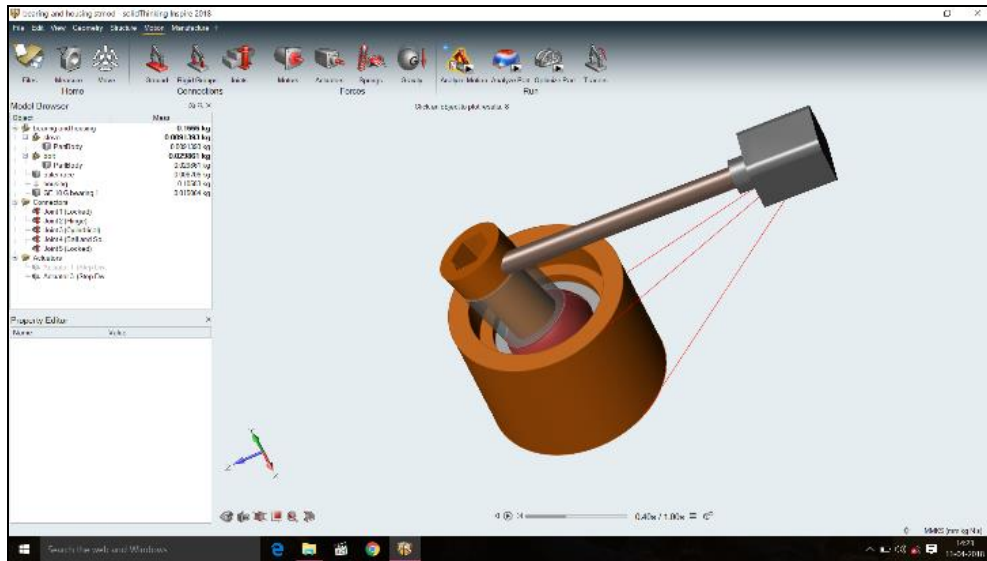


Fig.3.3.3-Actuate the motion

To actuate the motion, an actuator is provided to bolt which is inserted in the inner ring of bearing. So, here the tilting of inner ring will occur after actuation.

Step 4: Analysis of motion

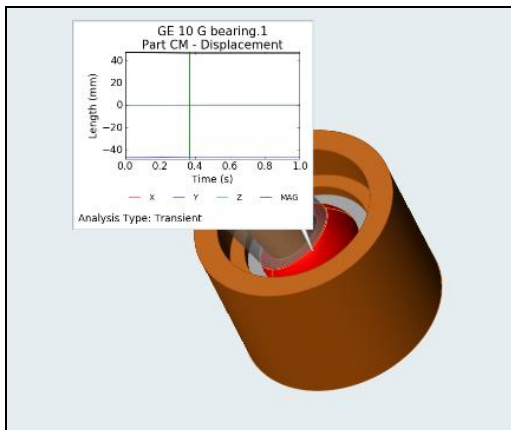


Fig.3.3.4-Graph of length vs time

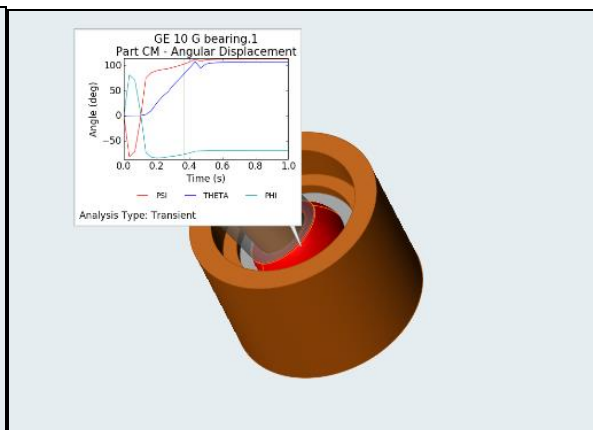


Fig.3.3.5-Graph of angle vs time

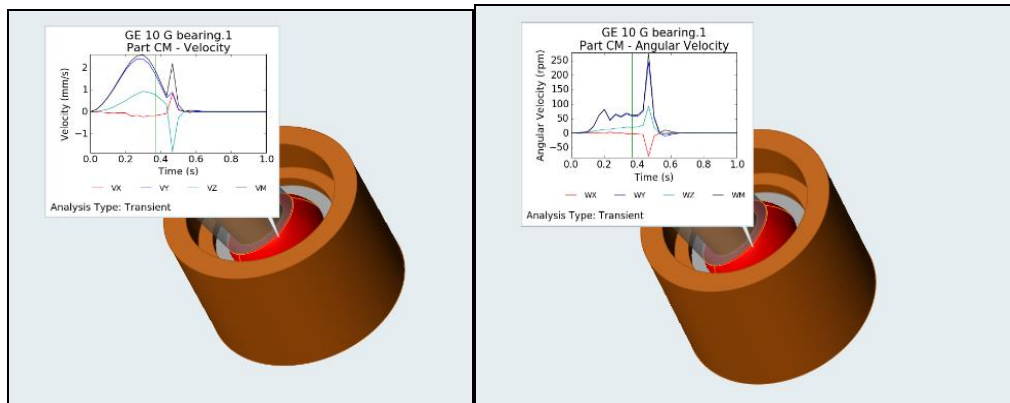


Fig.3.3.6-Graph of velocity vs time

Fig.3.3.7-Graph of angular velocity vs time

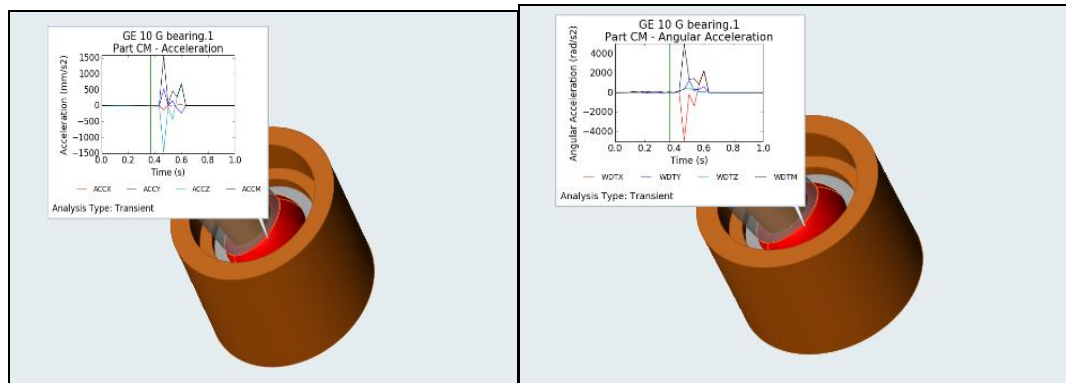


Fig.3.3.8-Graph of acceleration vs time

Fig.3.3.9-Graph of angular acceleration vs time

Above graphs shows the motion of inner ring with the tilting of bolt due to actuation of motion.

Step 5: Load analysis of bearing

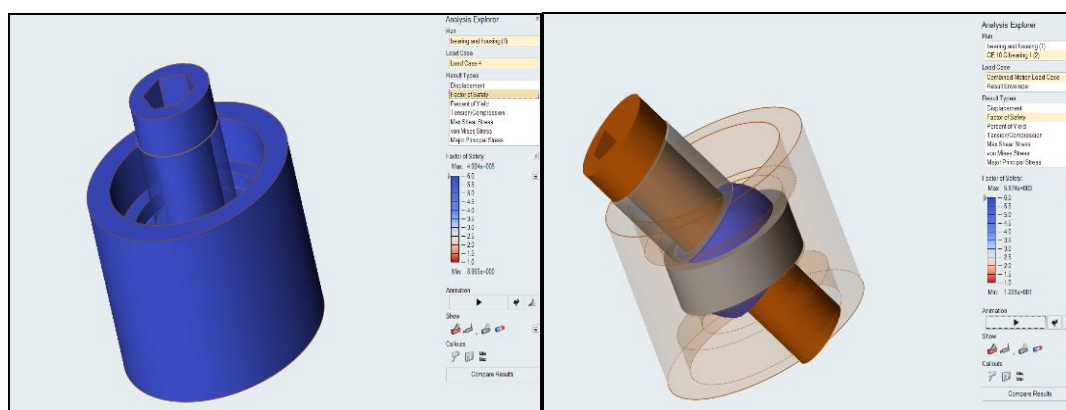


Fig.3.3.10-Results of factor of safety

So, from the above figure, it can be concluded that the bearing is safe for 18° angle of rotation with the application of maximum forces.

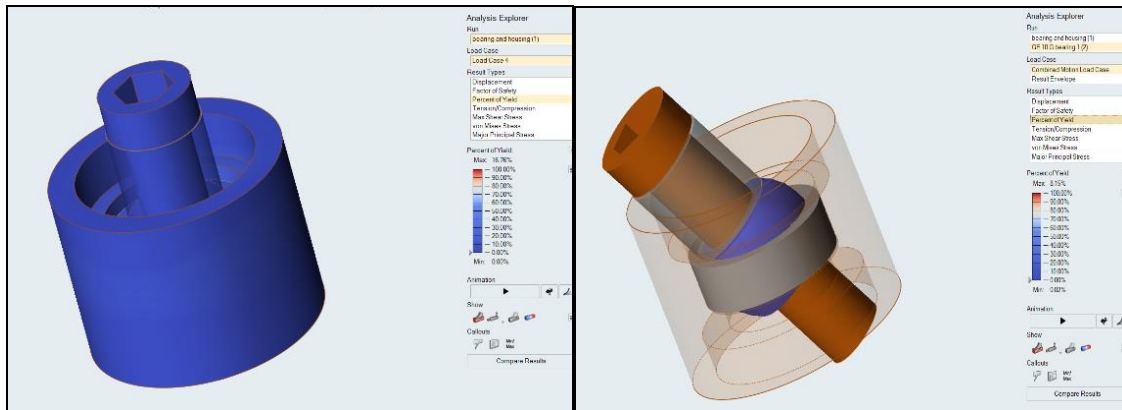


Fig.3.3.11-Analysis of percent of yield

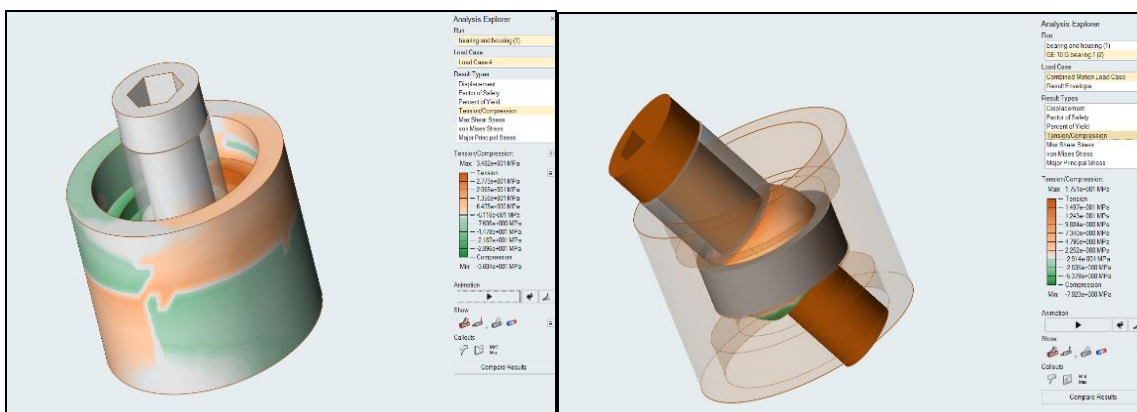


Fig.3.3.12-Analysis of tension/compression

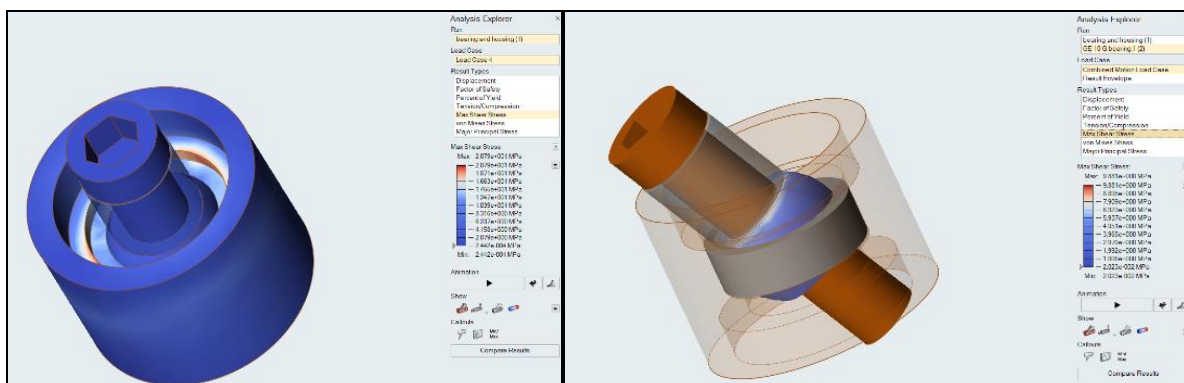
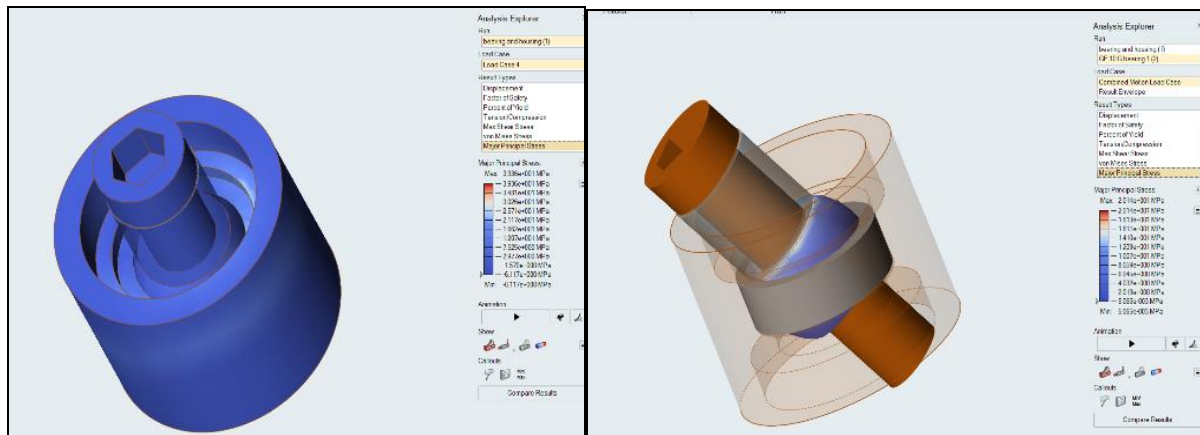


Fig.3.3.13-Analysis of maximum shear and principle stress



So, from all the above results, it is very clear that during dynamic motion, spherical plain bearing GE 10G can sustain all the stresses developed and the bearing remains safe.

Step 6: Modal analysis of bearing

In modal analysis the motion of the bearing is divided into number of segments which is the operator's choice. The modes of frequency will be adjusted accordingly.

Here, the motion is divided into 10 seconds and for every seconds results are obtained.

For 10th second, factor of safety and major principle stresses are evaluated and the results obtained are as follows :

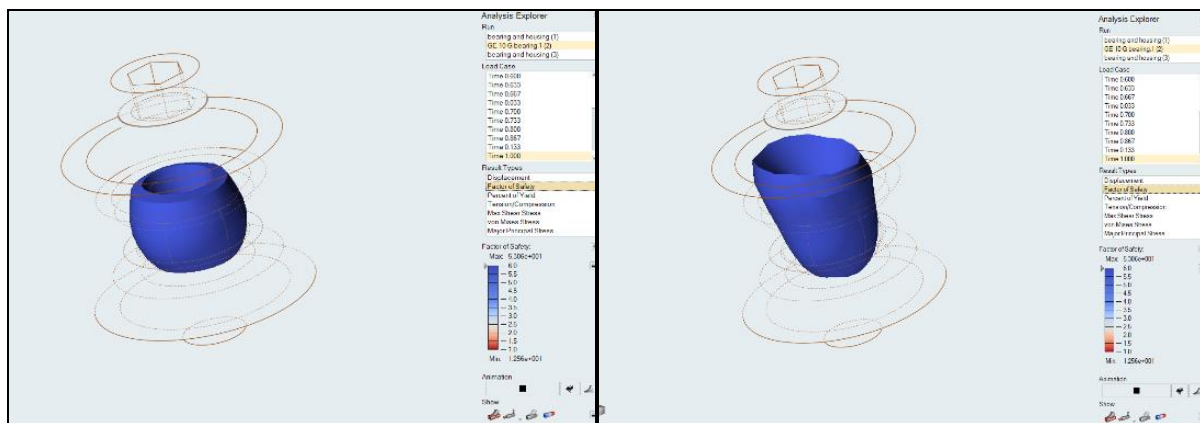


Fig.3.3.14-Bearing frequency on 10th second Fig.3.3.15-Factor of safety of bearing on 10th second

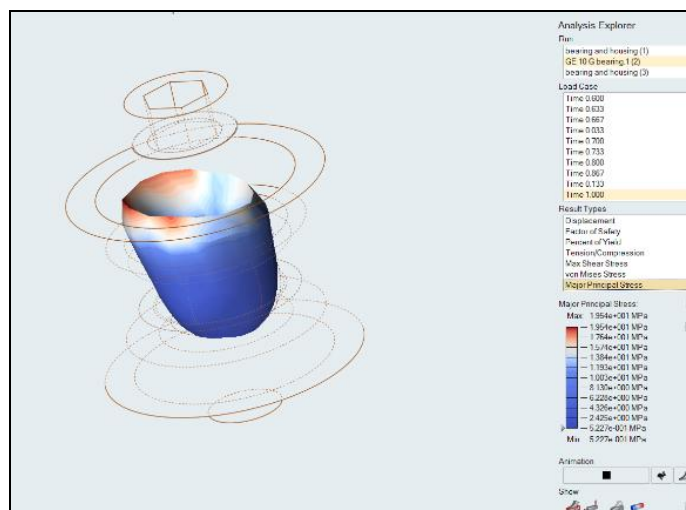


Fig.3.3.16-Major principle stress on 10th second

RESULT AND DISCUSSION

So, from the fatigue analysis it is very clear that spherical plain bearing GE 10G is safe and can be used when the suspension arm is tilted at an angle. Also it shows overall good performance. Maximum number of oscillations that this bearing can make without failure are 1 million oscillations and maximum life of bearing in hours is 278 hours without failure, when the bearing is used in application of ATV vehicle.

CONCLUSION

Spherical plain bearing can be widely used when there is a need to transmit motion when the shaft is tilted at an angle. They are designed to manage high radial loads and to provide consistent performance under condition of extreme speed and critical stresses. They can be used in the

suspension arms of ATV vehicle as permissible angle of tilt in case of spherical plain bearing is more and also have good performance under high radial loads.

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