ISSN 2250-3153

Design, Development and Mechanical Analysis of Sun/Moon Simulator Facility

Chetan Kumar D.S.[*], Prof, B.H.Vasudevamurthy[**], Mr.Krishnaprasad A.K[***]

[*] IV Sem, M.Tech, Department of Mechanical Engg., SIET, Tumkur-572106
[**] Professor & HOD, Department of Mechanical Engg., SIET, Tumkur-572106
[***] Head, MSD-SDA, Scientist/Engineer-SG, LEOS-ISRO, Bengaluru-560058

Abstract- The Sun/Moon simulator facility is used in earth sensor lab for interface checking, mapping checks and other purpose used in geostationary satellites. This has been mounted in the earth lab in Laboratory for Electro Optics Systems (LEOS), ISRO. The overall design has been carried out in rational design. First we select suitable material for designing lead screw which fulfils the design requirements. A suitable AC motor along with a gear is selected, and bevel gear is used to convert rotary motion into a reciprocating motion of the simulator. To locate the simulator to desired location a horizontal beam has been designed with standard diameter which meets the design requirements & this simulator should travel freely.

Index Terms- Lead Screws. Bevel Gear, Helical Gear box, Electric motor (AC&DC), Sun/Moon simulator

I. Introduction

The geo-stationary orbit is such a position for a satellite that it keeps pace with the rotation of the Earth. These platforms are covering the same place and give continuous near hemispheric coverage over the same area day and night [9]. These satellites are put in equatorial plane orbiting from west to east. Its coverage is limited to 70oN to 70oS latitudes and one satellite can view one-third globe (Fig.1.1). As a result it is continuously located above the same geographical position.

According to Shannon in 1975, Simulation is “the process of designing a model of a real system and conducting experiments with this model for the purpose either of understanding the behaviour of the system or of evaluating various strategies (within the limits imposed by a criterion or set of criteria) for the operation of the system.”

As our project is used in the earth lab in LEOS-ISRO, for sun/moon simulator facility we are using a lead screw. With the help of lead screw and its movement the sun/moon simulator facility moves up and down. A lead screw is the device which can transmit the linear motion by the action rotary motion. A lead screw is such a part which when rotates gives the linear motion. A bevel gear is used to transmit a horizontal motion into linear motion [5, 8]. The speed of the screw depends on upon the motor and a gear box is connected on it.

In this project the sun/moon simulator has to be moved 4000mm along x-axis and 3000mm along y-axis. The y-axis movement is achieved by using two lead screws driven by a common double ended shaft AC-drive motor using two gear boxes. The x-axis movement is achieved by DC-drive motor using a fixed beam connected to the lead screws.

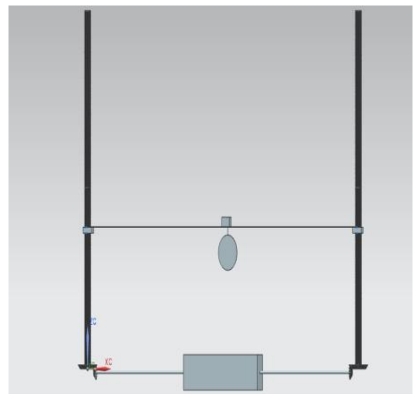

Fig1.1 Block diagram of Sun/Moon simulator facility

II. DESIGN CONSIDERATIONS

Following are the design considerations in this dissertation work

- Selection of materials for lead screw and Bevel gear.
- Check the buckling load of the lead screws
- Design of lead screws.
- Design of bevel gear.
- Selection of suitable AC/DC motor for horizontal and vertical movement of Sun/Moon simulator facility.
- Selection of suitable gearbox (Helical gear box) for the rotation of lead screw.
- Designing of X-movement translator.

This project has been carried out for the need of Earth lab in LEOS (ISRO). In this the vertical movement of Sun/Moon simulator facility is about 3000mm and horizontal movement is about 4000mm.

For vertical movement we can use lead screw, Acme thread which is driven by a double ended shaft AC motor with two helical gear box and the rotary motion is transferred to rotating motion by bevel gears. The design calculation of lead screws and bevel gears are as follows.

III. THEORETICAL ANALYSIS

A. CALCULATION FOR LEAD SCREW

i) Buckling Load for Column

ISSN 2250-3153

Euler's formula for Buckling, [4]

$P_{cr} = \frac{\pi^2 CEI}{l^2}$

Where P_{cr} = Critical load in Buckling (N).

E = Young's modulus (N/mm^2).
I = Moment of Inertia (mm^2).
l = length of the screw rod (mm).
C = Factor depends on different end conditions {Table-5 [4]}
= 4

- For the material EN-47 steel, Buckling load is as follows.

Given E= $210 x 10^3$ N/mm^2.

$I = \frac{\pi 100^2}{32} = 981.78mm.$

l = 3000mm.

$P_{cr} = \frac{\pi^2 CEI}{l^2} = \frac{\pi^2 4x210x10^3\,x981.78}{3000^2} = 904.38N$

We select EN-47 steel for lead screw because of its strength for buckling load and cost of the material and availability of the material in the local market.

ii) Design of Lead Screw

For Acme thread the values [2] that were chosen are shown below which were safe, when the allowable stresses of the screw material is considered.

Consider:

Major diameter (d) =100mm, Minor/Core diameter (d_c) = 97.5mm,

Weight (W) = 1962N, Pitch (p) = 10,

Mean diameter (d_m) = $\frac{d+dc}{2} = \frac{100+95.5}{2} = 97.5mm$

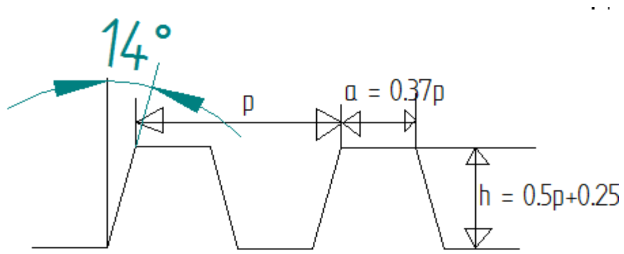

Fig 2.1 Form of Acme Thread

a = 0.37p = 0.37x10 = 3.7mm,
h = 0.5p+0.25 = 0.5x10+0.25 = 5.25mm,
$\tan^{\propto} = p/\pi d_m = 10/\pi x97.75 = 0.03$
$\propto = 1.7^0$.

The torque to overcome friction at the thread surfaces of the screw is given by [2].

$T = Wd_c \left[\frac{\tan\propto + \tan\varphi}{1-\tan\propto \tan\varphi}\right]$

Where tan Φ = μ/cosβ,

At β = 14.5^0 and μ = 0.15 {Table18.5 [2]}
tan Φ = 0.15/cos14.5 = 0.15

Hence torque due to upper screw is given by [2].

$T_1 = 1962x97.5 \left[\frac{0.03+0.15}{1-0.03x0.15}\right] = 34677.44N/mm^2$

Torque due to lower end of the screw is given by [2].

$T_2 = T_1/2 = 34677.44/2 = 17338.5\ N/mm^2$

Shear stress on the screw due to torque is,

$T = \frac{16T1}{\pi d_c^3} = \frac{16x34677.44}{\pi\ 95.5^2} = 19.36\ N/mm^2$

iii) Design of Screw Follower

The screw follower has the function of moving up and down the thread of the screw depending on the motion of the screw. The screw follower was threaded and it is fixed tom the lead screw. Assuming the screw follower moves uniformly y on both the lead screw, the bearing pressure [2] is given by,

$\sigma_b = \frac{W}{\frac{\pi}{4}(d^2-d_c^2)n}$

Where σ_b = Safe bearing pressure = 1.5N {Table 18.6 [2]}

$n = \frac{1962x4}{\pi(100^2-95.5^2)x1.5} = 1.89$

Using FOS value of 1.5n = 1.5x1.89 = 12.25 = 12 Threads.
Height of the screw follower, r = n*p = 120mm.

Design choice for r = 120mm.

Shear stress induced in screw was,

$\tau_s = \frac{W}{\pi n a d_c} = \frac{1962}{\pi x123.7x95.5} = 0.148\ N/mm^2$

Shear stress induced in the follower was,

$\tau_{sf} = \frac{W}{\pi n a d} = \frac{1962}{\pi x123.7x100} = 0.1406\ N/mm^2$

Since τ_s and τ_{sf} were below the bearing pressure of 18 N/mm^2 [4, 8], Hence design for the screw follower was safe. The tearing strength of the screw follower was [2].

$\delta_t = \delta_t/FS = \frac{W}{\frac{\pi}{4}(D_1^2-D_0^2)}$

$200/2 = \frac{1962}{\frac{\pi}{4}(D_1^2-100^2)}$

$D_1 = 100.02mm$

Hence the choice of 120mm is robust and safe.

B) DESIGN OF STRAIGHT BEVEL GEARS.

The bevel gears are used to transmit power form gear box to lead screws. The design of pinion and driven bevel gears are as follows [3].

- Pressure angle $\alpha = 20^0$ full depth involutes
- Moderate medium shock and 3hrs/day

Hence service factor $C_s = 1.25$ {Table 2.33[3]}

- Pinion material as Alloy steel, case hardened (SAE2320)

From Table 2.16[3] $\sigma_{01} = 345\ N/mm^2$

Select minimum number of teeth on pinion to avoid interference for 20^0 full depth involutes systems $z_1 = 15$ {Table 2.92[3]}.
$i = n_1/n_2 = = z_2/z_1 = = d_2/d_1$
Number of teeth on gear $z_2 = n_1/n_2 * z_1 = 300/10 \times 15 = 45$
Gear ratio $i = n_1/n_2 = 300/10 = 3$
For right angle bevel gear i.e., $\sigma = 90^0$
Pitch angle of pinion $\delta_1 = \tan^{-1}(1/i) = \tan^{-1}(1/3) = 18.435^0$
Pitch angle of gear $\delta_2 = \tan^{-1} i = \tan^{-1} 3 = 71.565^0$
Formative number of teeth in a straight teeth bevel pinion
$z_{v1} = z_2/\cos\delta_2 = 15/\cos 18.435^0 = 15.8114$
Formative number of teeth in a straight teeth bevel gear
$z_{v2} = z_2/\cos\delta_2 = 45/\cos 71.565^0 = 142.3021$
Lewis form factor for 20^0 full depth Involute system
$y = 0.154 - 0.192/z_v.$
Form factor for pinion $y_1 = 0.09632$
Form factor for gear $y_2 = 0.1476$
To select gear material equate $\sigma_{01}y_1$ to $\sigma_{02}y_2$
i.e., $345 \times 0.09632 = \sigma_{02} \times 0.1476$; $\sigma_{02} = 225.138\ N/mm^2$
From Table 2.16[3], select the gear material such that its value of σ_{02} must be nearer to $225.138\ N/mm^2$. Hence select steel, SAE1030, heat treated.
Hence $\sigma_{02} = 220\ N/mm^2$

(a)Identify the weaker member

Particulars	σ_o	y	$\sigma_o y$	Material	Remarks
Pinion	345	0.096	33.23	Alloy steel case hardened SAE2320.	
Gear	220	0.147	32.47	SAE 1030 Heat treated steel	weaker

As $\sigma_{02}y_2 < \sigma_{01}y_1$, gear is the weaker member. Therefore design should be based on gear.

(b)Design

(i) Tangential tooth load $F_t = \frac{9550*1000*PC_s}{nr}$ where r in mm
Hence tangential tooth load of the weaker member $F_{t2} = \frac{9550*1000*PC_s}{n_2 r_2}$
Pitch circle radius of gear $r_2 = d_2/2 = mz_2/2 = m*45/2 = 22.5m$

$F_{t2} = \frac{9550*1000*25*1.5}{400*22.5m} = \frac{39791.67N}{m}$
(ii) Tangential tooth load from Lewis equation
$F_t = \sigma_0 C_v b Y_m \left(\frac{R-b}{R}\right)$

Tangential tooth load of the weaker member
$F_{t2} = \sigma_{02} C_v b \pi y_{2m} \left(\frac{R-b}{R}\right)$

Cone distance $R = \frac{m}{2}\sqrt{z_1^2 + z_2^2} = \frac{m}{2}\sqrt{15^2 + 45^2} = 23.717m$

For face width $\frac{R}{4} < b > \frac{R}{3}$

$\frac{R}{3} = \frac{23.717m}{3} = 7.9m$....... (1)

The face width of the bevel gear is generally taken as 10m or $\frac{R}{3}$ whichever is smaller

As $\frac{R}{3} = 7.9m < 10m$, take face width b = 8m

Hence $F_{t2} = (220)(K_v)(8m)(\pi \times 0.1476)(m)\left(\frac{23.717m - 8m}{23.717m}\right) = 540.83m^2 K_v$.... (2)

Equating the equation (1) and (2)

i.e., $\frac{39791.67N}{m} = 540.83m^2 k_v$

Therefore, $m^2 k_v = 73.576$....... (3)

Mean pitch line velocity of the weaker member $v_m = \frac{\pi d_2 n_2}{60000} = \frac{\pi m z_2 n_2}{60000} = \frac{\pi * m * 45 * 400}{60000} = 0.9425m$

By trial & error method,
Assume m =6mm

$v_m = 0.9425 \times 6 = 5.655m/sec$

Therefore, Velocity factor $C_v = K_v = \frac{3}{3+5.655} = 0.34662$
From equation (3)

$(6^3)(0.34662) \geq 73.756$
$74.787 < 73.756$

Hence suitable, Therefore module m = 6mm

(c) Dimensions
From Table 2.1[3] $\alpha = 20^0$ Full depth
Addendum $h_a = 1m = 6mm$
Dedendum $h_f = 1.25m = 7.5mm$
Working depth $h^1 = 2m = 12mm$
Total depth $h = 2.25m = 13.5mm$
Tooth thickness $s = \frac{\pi m}{2} = 9.425mm$
Minimum clearance $c = 0.25m = 1.5mm$

Pitch circle diameter of pinion $d_1 = mZ_1 = 90mm$
Pitch circle diameter of gear $d_2 = mZ_2 = 270mm$
Outside diameter of pinion $d_{a1} = d_1 + 2h_{a1} = 102mm$
Outside diameter of gear $d_{a2} = d_2 + 2h_{a2} = 282mm$
Dedendum circle diameter of pinion $d_{f1} = d_1 - 2h_{f1} = 75mm$
Dedendum circle diameter of gear $d_{f2} = d_2 - 2h_{f2} = 255mm$
For addendum angle, $\tan v_a = \frac{2h_a \sin\delta_1}{d_1} = \frac{2*6*\sin 18.435}{90}$
Therefore, Addendum angle $v_a = 2.414^0$
For dedendum angle, $\tan v_f = \frac{2h_f \sin\delta_1}{d_1} = \frac{2*7.5*\sin 18.435}{90}$
Therefore, Dedendum angle $v_f = 3.107^0$
Face angle of pinion $\theta_{f1} = \delta_1 + v_a = 20.849^0$
Face angle of gear $\theta_{f2} = \delta_2 + v_a = 73.979^0$
Cutting angle of pinion $\theta_{c1} = \delta_1 - v_f = 15.418^0$
Cutting angle of gear $\theta_{c2} = \delta_2 - v_f = 68.548^0$
Tangential took load $F_t = 6631.945N$
Face width $b = 8m = 48mm$
Cone distance $R = 23.717m = 142.3mm$
Mean pitch line velocity $v_m = 5.655m/sec$
Velocity factor $C_v = K_v = 0.34662$
Service factor $C_s = 1.5$

IV. CONCLUSION

The design and development of a Sun/Moon simulator facility using lead screw and bevel gear was carried out successfully to meet the required design standards. The vertical movement is achieved by the two lead screws driven by double ended shaft speed control AC motor (7.5Hp or 10Hp) with two helical gear box and two pairs of bevel gears. The horizontal movement is achieved by a simply supported beam connected to the lead screws and a DC motor. The horizontal movement is about 4000mm and vertical movement is about 3000mm is achieved. This type of mechanism affords plenty of scope of modifications for further improvements and operational efficiency, which should make it commercially available and attractive. Thus it is recommended for the engineering industries (lifting mechanism) and agricultural field (to lift bore well motors).

ACKNOWLEDGMENT

I wish to thank Dr.K.SUKUMARAN, Principal, SIET, Tumkur, for having inspired and encouraging us throughout my PG course.

I am grateful to Mr. B.H.Vasudevamurthy, HOD, and project guide, Dept. of MECH, SIET, Tumkur, for sharing his knowledge and experience with us and his valuable advice and supervision at every stage of the project.

I would like to express my sincere gratitude and grateful to Mr.A.K.Krishnaprasad, Head, MSD-SDA, Scientist/Engineer-SG, LEOS-ISRO, Bengaluru, for his excellent guidance, constant supervision and encouragement for the project work and also my sincere acknowledgement to Mr.Anandachandran.G, Engineer-SF, Head, Mechanical Fabrication Section for his guidance & co-operation.

I convey my thanks to Er.Sampath Kumar L, Lecturer and PG coordinator and also to Er.Meghani mollies Reddy, Sr.Lecture, Dept of ME SIET, Tumkur.

I convey my special thanks to Dr.Vijaya Vittala, Dean (R&D), HMSIT, Tumkur, for his valuable suggestions and guidelines during my project work.

Finally, I would like to express my sincere thanks to our beloved parents, family members, and friends and all Teaching and non-teaching Staff members of Department of Mechanical Engineering, SIET for their valuable guidance and support.

REFERENCES

[1] "The design, construction and testing of a vertical squeeze cassava pulp dewatering machine", H.D. Olusegun, T.K. Ajiboye, Published in IJEAS, Vol.2, Issue 1(2010)27-43.

[2] Machine design data hand book Volume-I, By Dr. K.Lingiah, Second edition, Year of publication- 1983, Suma publishers, Bengaluru.

[3] Machine design data hand book Volume-II, By Dr. K.Lingiah, Sixth edition, Year of publication- 1984, Suma publishers, Bengaluru.

[4] CMTI, Machine Tool Design Hand Book by Dr. Ramachandran, year of publication-1982, Ta[15] A text book of Machine Design by R S Khurmi and J K Gupta Year of publication-2012, S Chand and company, New Delhi.

[5] Machine Design, Hall, Holowenko, Laughlin (Shum's outline series) adopted by S K Somoni, Tata Mc. Graw Hill Publishing Component Ltd. New Delhi, Special Indian edition-2008

[6] Strength of Materials by R S Khurmi, 23rd edition, S Chand and company, New Delhi. McGraw-Hill Publishing Co. Ltd. New Delhi.

[7] Joseph, G. 1996. Imaging Sensors for Remote Sensing, Remote Sensing Reviews, 13: 257-342

[8] Mechanical Engineering Design, Fifth Edition, Author-Joseph Edward Shigely,Charles R Mischke,Publication-Mc Graw Hill International Edition.

AUTHORS

First Author – Chetan Kumar D.S., IV Sem, M.Tech (Machine Design), Shridevi Institute of Engineering & Technology, Tumkur-572104, Karnataka, India. E-mail: chetan.mmd@gmail.com, Mob: +91-9844033826

Second Author – Prof., B.H.Vasudevamurthy,Professor & HOD, Department of Mechanical Engg., Shridevi Institute of Engineering & Technology, Tumkur-572104, Karnataka, India.

Third Author – 3) Mr.Krishnaprasad A.K., Head, MSD-SDA, Scientist/Engineer-SG, LEOS-ISRO, I Phase, Peenya Industrial Area, Bengaluru-560058, Karnataka, India.