DESIGN OF 12M ANTENNA MOUNT STRUCTURE

By-

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1.0 INTRODUCTION :

A ground based parabolic antenna of 12 m diameter with the mount structure has been designed to withstand the critical wind and thermal loads expected during its service. The reflector, the antenna dish consists of pre loaded parabolic radial spikes and annular members. This report deals with the design of all components / substructure.

2.0 ANTENNA MECHANICAL SYSYEM:

The most important sub system or major components of this Antenna mechanical system are

1. Antenna mount
2. Drive system
3. Reflector and Backup structure
4. Sub-reflector & Mounting arrangement
5. Lightning protection

3.0 PRE-LODED PARABOLIC DISH ANTENNA

The reflector consists of a Central Hub of 4m diameter, Parabolic radial spokes (all spokes are rigidly attached to the hub with the help of Taper blocks), Circumferential and Bracing members, Quadrapod structure forms the backup structure .The panels are made of stainless steel welded mesh of size 6 X 6 X 0.55 mm ϕ. These panels are mounted on the backup structure using studs. This antenna is designed to operate with prime focus feed (size $1m^2$,weighing $\approx$ 60kg) as well as cassegrain feed (weighing $\approx$250kg) .The total weight of the antenna with the hub is 3000kg.

4.0 ANTENNA MOUNT

The Mount has the following sub assemblies:

1 Cradle structure.

2. Yoke structure.

3. Steel shell structure.

4. Conical concrete structure.

5. Foundation for the above.

6. Azimuth drive system.

7. Elevation drive system.

8. Encoder mounting (El and Az).

9. Cable wrap.

5.0 CRADLE STRUCTURE:

This structure is interface between dish structure and mount .The hub is attached to the cradle structure. The cradle is mounted on the two elevation bearing housings. The elevation bull gear is placed in an orthogonal plane to the elevation axis and is attached to the cradle structure. The two elevation bearings, which support the cradle and hub, are held by elevation shaft mounted on the elevation bearing of the yoke structure.

6.0 YOKE STRUCTURE:

The yoke structure consisting of yoke arms is connected to the top of the azimuth bearing plate, the plate is connected to the azimuth bearing which is mounted on to the top of the steel shell structure .Two elevation bearing with housing are attached to the top of yoke arm .In one of the arm the encoder is mounted, on the yoke platform two gear boxes are fixed with the pinion mounted, the pinion is coupled to the elevation bull gear for elevation drive.

7.0 STEEL SHELL STRUCTURE:

This steel structure bottom is bolted down to the top of the concrete base with anchor bolts. To pass the power cables, RF cables etc sufficient passage is provided in the centre. On the top of the steel structure Azimuth bearing with external gear is bolted .Two planetary gear boxes with D.C drive is rigidly attached to shell structure with 180° spacing and they are coupled to the azimuth slewing ring external gear through pinion.

8.0 CONICAL CONCRETE STRUCTURE:

The conical concrete structure with 2m diameter at top and 4m diameter at ground level with a height of 3m supports the complete antenna along with mount and drive system at the top. This structure is designed to with stand the wind force acting on the antenna and completely stable. This conical concrete structure along with foundation meets all the structural stability requirements of the antenna .The space inside the structure is used to accommodate the control system and cable track etc.

Antenna Specification

1.	Dish diameter	:	12m
2.	Focus from apex	:	4.8m
3.	Hub diameter	:	4m
4.	F/D of the dish	:	0.4
5.	Solidity of the dish	:	22.22%
6.	Wire mesh surface area	:	$1210m^2$
7.	Wire mesh size	:	6 X 6X 0.55mm ϕ
8.	Total mass of the panels	:	370kg
9.	Mass of the full dish	:	3000kg
10.	Optics	:	Prime focus feed &
		:	Cassegrain focus feed
11.	Operating frequency	:	0.3 to 10GHZ
12.	Elevation Range	:	0° to 90°
11.	Limit stop positions	:	-0.5° to 92°
12.	Azimuth Range	:	$\pm 270^\circ$
13.	Pointing accuracy	:	Better than 1arc minute
14.	Dish slewing speed	:	Azimuth 40°/min
		:	Elevation 20°/min
15.	Wind speed	:	Operable to 50kmph
			Manor able to 80kmph
			Survival to 150kmph

DESIGN LOADS ON ANTENNA AND MOUNT

Wind load constitutes major component of the total loading on the antennas as these structures are erected in the open field. Wind forces play very significant role in the design and operation of large antennas and satisfactory estimates of these forces is becomes very much essential. The main design loads on Antenna are:

(i) Wind Loads and

(ii) Self Weight.

The resultant wind force and torque on a body immersed in an air-stream can be expressed in the form by the application of Bernoulli's principle and the theories of dimensional analysis

F	=	$\frac{1}{2} p v^2 AC_R$
T	=	$\frac{1}{2} p V^2 AdC_M$
P	=	Mass density of the air stream
V	=	Wind velocity
A	=	Typical area of the body
d	=	Typical dimension of the body
C_R and C_M	=	Dimensionless force and moment coefficients which depends upon the geometrical properties of the body and on Reynolds number. The term $\frac{1}{2}$ **p** $\mathbf{V^2}$ is the dynamic pressure of the undisturbed flow, and is designated "q".

The wind velocities recorded in any location are extremely variable and in addition to the steady wind at any time, there are the effects of gusts which may last for only a short period. In choosing the appropriate wind velocity for the purpose of determining the basic wind pressure due consideration had been given to the degree of exposure appropriate to the location. Wind force is the cause for static and dynamic loading on the antenna due to Drag and Lift force associated with it.

Employing conventional aerodynamic terminology, the force F may be divided into three orthogonal forces: drag, lift and side force, with coefficients designated C_W C_X and C_Y. In equation form:

Drag force	=	$C_D\, q\, A$
Lift force	=	$C_L\, q\, A$
Side force	=	$C_S\, q\, A$
Rolling moment	=	$C_W\, q\, A$
Pitching moment	=	$C_X\, q\, A$
Yawing moment	=	$C_Y\, q\, A$

The wind is assumed to flow only in the horizontal direction; hence the angle $\propto$ which the wind makes with the plane of the reflector rim (the angle of attack) is a function of the altitude angle θ and azimuth angle Ψ relative to the wind stream expressed by

$$\propto \quad = \quad \mathrm{Sin}^{-1}\,(\mathrm{Cos}\,\theta\ \mathrm{Cos}\,\Psi)$$

Due to reflector symmetry the drag coefficient are identical to those of side force for $\theta = 0$ and Ψ variable, lift being zero for $\theta = 0$ and side force zero for $\Psi = 0$. Thus values in lift and side force are not the same for a given Antenna position, but are functions of the angle of attack. In this connection peak values in lift and side force do not occur at the same antenna position.

Similarly, due to symmetry, yawing moment are maximum when the reflector rotates in azimuth Ψ and altitude θ is zero. Similarly pitching moments are highest when the angle Ψ zero and θ is verifying.. In this connection maximum moments in pitch and yaw do not occur at the same antenna position.

The drag, Lift / side forces and yawing / pitching moments acting on the antenna are :

Drag Force	=	$q\,A\,C_D$
Lift / Side force	=	$q\,A\,C_{L,S}$
Yawing / Pitching moment	=	$q\,A\,d\,C_P$

The force and moment coefficients are taken from the paper titled " Large steerable Radio Antennas- Climatological and Aerodynamic Considerations, Annals of New York Academy of Science (1964).

As per I.S:875.

the value of $q = 0.6\,V_z^2$ where V_z is the design velocity of wind in m / sec.

$V_z = V_b\,K_1\,K_2\,K_3$ When V_b basic wind velocity

$K_1\,K_2\,K_3$ Constants

K_1	=	1.05 for 100 KMPH and 1.07 for 150 KMPH		
K_2	=	1.05 (Cat 2, Clause A, Height 15m) (100 & 150 KMPH)		
K_3	=	1.0		
Vz	=	28 x 1.05 x 1.05 x 1.0 =		30.8 m/sec at 100 KMPH
	=	42 x 1.07 x 1.05 x 1.0 =		47.1 m/sec at 150 KMPH
q	=	$0.6\,V_z^2 = 0.6 \times 30.8^2$	=	569 N/m^2 at 100 KMPH
	=	0.6×47.1^2	=	1331N/m^2 at 150 KMPH

The drag, lift and movement coefficients are obtained from the graph for mesh reflector. Since the present 12m Antenna is having central 4m dia with solid panels and the rest with mesh panel, the coefficients obtained for mesh reflector is increased by 17% by using on the weightage for the 4m dia solid and rest with mesh panels.

By using the drag, lift and moment coefficient as arrived, the loading Table 1 is prepared for different elevation angles of antenna and for different wind attack angle.

Table.1

Angle θ	Wind Direction γα	α	C_D	C_L	C_M	100 KMPH Drag Force KN	100 KMPH Lift/ Side Force KN	100 KMPH Mom. KNm	150 KMPH Drag Force KN	150 KMPH Lift/ Side Force KN	150 KMPH Mom. KNm
0°	0°	90°	0.69	0.00	0.00	44.5	00.0	00.0	104.3	00.0	000.0
	90°	0°	0.29	-0.08	0.08	18.9	-05.3	62.0	44.2	-12.5	145.2
15°	0°	75°	0.68	0.12	0.01	43.6	7.7	7.7	102.1	18.1	18.1
	90°	0°	0.29	-0.08	0.08	18..9	-.5.3	62.0	44.2	-12.5	145.2
30°	0°	60°	0.63	0.22	-0.02	40.7	14.0	-15.5	95.3	32.9	-36.3
	90°	0°	0.29	-0.08	0.08	18..9	-.5.3	62.0	44.2	-12.5	145.2
45°	0°	45°	0.59	0.23	-0.05	38.6	15.0	-38.7	89.6	35.2	90.7
	90°	0°	0.29	-0.08	0.08	18..9	-.5.3	62.0	44.2	-12.5	145.2
60°	0°	30°	0.44	0.14	-0.08	28.6	9.2	-62.2	66.9	21.5	-145.2
	90°	0°	0.29	-0.08	0.08	18..9	-.5.3	62.0	44.2	-12.5	145.2
75°	0°	15°	0.35	0.00	0.09	22.7	00.0	69.7	53.3	00.0	163.0
	90°	0°	0.29	-0.08	0.08	18..9	-.5.3	62.0	44.2	-12.5	145.2
90°	0°	90°	0.29	-0.08	0.08	18..9	-.5.3	62.0	44.2	-12.5	145.2
	90°	0°									

In addition to the wind load on reflector, wind load on back up structure, sub reflector, hub and mount assembly also will occur.

Wind load on Backup structure and sub reflector :

The wind load on the backup structure is conservatively computed by calculating the projected surface of the member in the wind direction and multiplying it with the design wind pressure. Similarly the wind pressure on the sub reflector is calculated without considering the shield effect . The wind load on the backup structure and the sub reflector for two different wind pressure are given in the Table.2.

Sl.No.	Sub element / Component	Wind Load (in KN)
1	Wind load on Hub	1.06
2	Wind load on Cradle	0.76
3	Wind load on Bull gear	4.24
4	Wind load on Sub reflector	0.80

The PPD Antenna is connected to the Backup structure in turn supported on Yoke plates through elevation bearing as shown in the Figure, . Forces and moments on Elevation axis for wind speed of 100 and 150 KMPH are calculated and tabulated in Table.3 and Table.4 respectively.

TABLE : 3

Forces and Moments at Elevation Axis Level (Wind speed 100 KMPH)

Sl. No.	Dish Position	Loading case	Force along X-dir (t)	Force along Y-dir (t)	Force along Z-dir (t)	Moment about X-axis (t.m)	Moment about Y-axis (t.m)	Moment about Z-axis (t.m)
01.	Dish facing sky	DL (dish + cradle)			7.5			
02.	Dish facing horizon	DL (dish + cradle)			7.5			
03.	Dish @ Facing sky	WL (front) WL (side)	2.0 -	- 2.18	0.97 0.97	- 0.172	6.44 -	- 7.7
04.	Dish @ 75 deg.	WL (front) WL (side)	2.38 -	- 2.18	- 0.97	- 0.17	8.19 0.99	- 7.743
05.	Dish @ 60 deg.	WL (front) WL (side)	2.86 -	- 2.18	2.42 0.97	- 0.15	8.11 1.91	- 7.79
06.	Dish @ 45 deg.	WL (front) WL (side)	3.86 -	- 2.18	3.0 0.97	- 0.12	6.57 2.70	- 7.82
07.	Dish @ 30 deg.	WL (front) WL (side)	4.07 -	- 2.18	2.9 0.97	- .09	4.86 3.31	- 7.85
08.	Dish @ 15 deg	WL (front) WL (side)	4.36 -	- 2.18	2.2 0.97	- 0.043	4.47 3.70	- 7.87
09.	Dish facing horizon	WL (front) WL (side)	4.45 -	- 2.18	- 0.97	- -	3.825 3.825	- 7.872

TABLE : 4

Forces and Moments at Elevation Axis Level (Wind Speed 150KMPH)

Sl. No.	Dish Position	Loading case	Force along X-dir (t)	Force along Y-dir (t)	Force along Z-dir (t)	Moment about X-axis (t.m)	Moment about Y-axis (t.m)	Moment about Z-axis (t.m)
01.	Dish facing sky	DL (dish + cradle)			7.5			
02.	Dish facing horizon	DL (dish + cradle)			7.5			
03.	Dish @ Facing sky	WL (front) WL (side)	4.68 -	- 5.11	0.25 0.25	- 0.4	16.07 -	- 16.02
04.	Dish @ 75 deg.	WL (front) WL (side)	5.59 -	- 5.11	- 0.25	- 0.39	17.82 0.99	- 16.12
05.	Dish @ 60 deg.	WL (front) WL (side)	6.69 -	- 5.11	3.65 0.25	- 0.35	16.43 1.91	- 16.22
06.	Dish @ 45 deg.	WL (front) WL (side)	8.96 -	- 5.11	5.02 0.97	- 0.28	11.77 2.70	- 16.30
07.	Dish @ 30 deg.	WL (front) WL (side)	9.53 -	- 5.11	4.79 0.25	- 0.20	6.94 3.31	- 16.37
08.	Dish @ 15 deg	WL (front) WL (side)	10.21 -	- 5.11	3.31 0.25	- 0.10	5.53 3.70	- 16.41
09.	Dish facing horizon	WL (front) WL (side)	10.43 -	- 5.11	- 0.97	- -	3.825 3.825	- 16.42

THE BEARING ANALYSIS

Loads on EL bearing due to wind on the PPD antenna, Backup structure and Sub reflector are calculated considering bearing B1 is restrained in Y direction load and bearing B2 is free. The forces on bearing along X, Y, Z direction are calculated for 100 KMPH and 150 KMPH wind speed and presented in the Table-5 and Table-6. Forces and moments from Table 3 and Table 4 are used to prepared Table-5 and Table-6.

Table 5

Design loads on bearings (Wind Speed 100 KMPH)

θ EL	Wind Angle ψ	BEARING B1			BEARING B2		
		X1 (Ton)	Y1 (Ton)	Z1 (Ton)	X2 (Ton)	Y2 (Ton)	Z2 (Ton)
0^0	0^0	2.23	-	3.75	2.23	-	3.75
	90^0	2.28	2.18	3.49	2.28	-	3.49
15^0	0^0	2.18	-	4.1	2.18	-	4.1
	90^0	2.28	2.18	3.51	2.28	-	3.51
30^0	0^0	2.04	-	4.45	2.04	-	4.45
	90^0	2.27	2.18	3.52	2.27	-	3.52
45^0	0^0	1.93	-	4.5	1.93	-	4.5
	90^0	2.26	2.18	3.53	2.26	-	3.53
60^0	0^0	1.43	-	4.21	1.43	-	4.21
	90^0	2.22	2.18	3.54	2.22	-	3.54
75^0	0^0	1.19	-	3.75	1.19	-	3.75
	90^0	2.23	2.18	3.55	2.23	-	3.55
90^0	0^0	1.00	-	3.50	1.00	-	3.50
	90^0	2.10	2.18	3.55	2.10	-	3.55

Table 6

Design loads on bearings (Wind Speed 150 KMPH)

θ EL	Wind Angle ψ	BEARING B1 X1 (Ton)	BEARING B1 Y1 (Ton)	BEARING B1 Z1 (Ton)	BEARING B2 X2 (Ton)	BEARING B2 Y2 (Ton)	BEARING B2 Z2 (Ton)
0^0	0^0	5.22	-	3.75	5.22	-	3.75
	90^0	5.33	5.11	3.13	5.33	-	3.13
15^0	0^0	5.11	-	4.62	5.11	-	4.62
	90^0	5.33	5.11	3.17	5.33	-	3.17
30^0	0^0	4.77	-	5.4	4.77	-	5.4
	90^0	5.31	5.11	3.2	5.31	-	3.2
45^0	0^0	4.48	-	5.51	4.48	-	5.51
	90^0	5.29	5.11	3.23	5.29	-	3.23
60^0	0^0	3.35	-	4.83	3.35	-	4.83
	90^0	5.26	5.11	3.26	5.26	-	3.26
75^0	0^0	2.80	-	3.75	2.80	-	3.75
	90^0	5.22	5.11	3.27	5.22	-	3.27
90^0	0^0	2.34	-	3.13	2.34	-	3.13
	90^0	5.19	5.11	3.27	5.19	-	3.27

ELEVATION DRIVE SYSTEM

8. GENERAL

The elevation drive rotates the reflector and cradle systems about elevation axis, from dish at 0° to zenith in (90°). This rotation complements the rotation of reflector, cradle and yoke about azimuth axis (+/- 270°) so that the dish can be positioned to track any radio source.

8.1 SCHEME OF OPERATION

The Antenna is driven by bull gear system attached to cradle so that the required rotation about elevation axis is achieved. The input to the gearing system is through D.C. Brush less motors. An encoder assembly is also present which helps in positioning the antenna accurately about the elevation axis. The brakes on the motors help to position and retain the antenna in any position as required.

8.2 CONSTRUCTION

The Elevation drive consists of bull gear driven by 2 pinions. The pinions are in turn driven by D.C. Brush less motors through planetary gear boxes.

8.3 DESIGN CONDITIONS AND OPERATING FEATURES

The Elevation drive rotates the antenna about elevation axis from 0^0 to 90^0 (Zenith). The elevation drive broadly consists of three phases.

1. Tracking and slewing of antenna up to 40 Kmph
2. Drive to stow (Wind speed from 40 to 80 Kmph)
3. Stow locking for Survival wind speed of 150 Kmph

8.4 TRACKING AND SLEWING

Two brush less D.C. Motors (one each for either direction of rotation) drives two pinions through two heavy reduction gear boxes. During the tracking one pinion drives the Bull gear, against the counter torque provided by the second pinion. This helps in providing a preload, there by reducing the effect of backlash in the gearing system. The couple torque equal to 10% to 30% of driving torque is provided.

8.5 DRIVE TO STOW

Once the wind speed crosses 40 Kmph. The Antenna will be driven to zenith position and stow locking will be effected by driving stow pin in to receptacles in counter weight. It is expected that stow locking, will be completed before wind speed reaches 80 Kmph. Both the pinion will drive the antenna during drive to stow mode.

8.6 STOW LOCKING

The stow locking of antenna about elevation axis should be done by stow lock pin. This relieve loads on elevation drive components. So that these are not subjected to high loads reaching a maximum at estimated design survival winds.

8.7 BULL GEAR AND PINION DESIGN CONSIDERATIONS

The Bull gear is designed for driving the Antenna against wind speed up to 80 Kmph. However a check is made whether the gear can withstand loads, without causing Catastrophic failure when the dish is caught in a survival wind condition without stows locking.

8.8 DESIGN LOADS ON BULL GEAR / ELEVATION PINION

During tracking and slewing (wind speed < 40 Kmph) one pinion drives against back torque provided by other pinion. The pinion is therefore to resist a maximum value of 1.2 (Wind torque + inertia of dish torque + friction torque of elevation bearing)
During drive to stow (wind speed 40 to 80 Kmph) both the pinions drive the gear to stow position. The sharing of loads between the two pinions is assumed to be in 1: 1 ratio of the driving torque and therefore the pinion has to resist a maximum torque of 0.5 (Wind torque + dead load torque + inertia torque of dish system + friction torque of elevation bearings).

The force acting on the gear sector will also be the same proportion as above. Since at no stage both the pinions can drive the same sector. This is achieved because of angular positioning of the pinions.

8.9 CONSTRUCTION FEATURES

The scheme of arrangement of elevation bearing assembly is shown in Fig.1. Plummer block with bearing is fixed one each on top plate of yoke arms in one line. M.S. Bracket is fixed to cradle. Pin is mounted through the bearing and bolted two ends to M.S. Bracket with sleeve. The dish cradle can rotate about the elevation axis i.e. about pin centerline.

8.10 DESIGN CONDITIONS AND PARAMETERS

The cradle. The bearings, bearing housing, shaft and M.S.bracket are designed for loads corresponding to survival wind speeds of 150 Kmph. Bearing are selected based on

a. Static load capacity for loads at 150 Kmph winds.
b. Dynamic load capacity and life for loads at 80 Kmph winds

The design conditions are as follows:

a. The Drive gear boxes (To which the pinions are attached) are to be fixed to brackets to meet the layout requirements brackets to be designed to the prevailing loads.
b. The ends of bull gear are to be provided with mechanical stops with limit switches.

8.11 ELEVATION BEARING SYSTEM

The Dish, cradle of the Antenna, swivel about the elevation axis. The elevation bearing assembly is necessary for supporting this structure and facilitate its rotation. These bearing system transfer the loads coming on to them to the yoke structure. The antenna's rotation is from 0 to 90° (Zenith) about elevation axis.

BEARING SELECTION

Type of Bearing	:	Spherical roller bearing
Load carrying capacity	:	To carry radial and axial loads as per Design requirements.

The bearing system selected should absorb angular misalignments of +/-2^0. Center to center distance between bearing is 2.8m, speed of oscillation 3.33 rev/hr max slewing speed and 0.0017 rev/hr min tracking speed. Shafts, Brackets and bearing housings are designed for static loads at survival wind speeds of 150 Kmph.

Forces on bearing along XYZ directions are calculated for 100 Kmph Wind Speed refer Table #5 for wind speed 150 Kmph refer #6 .

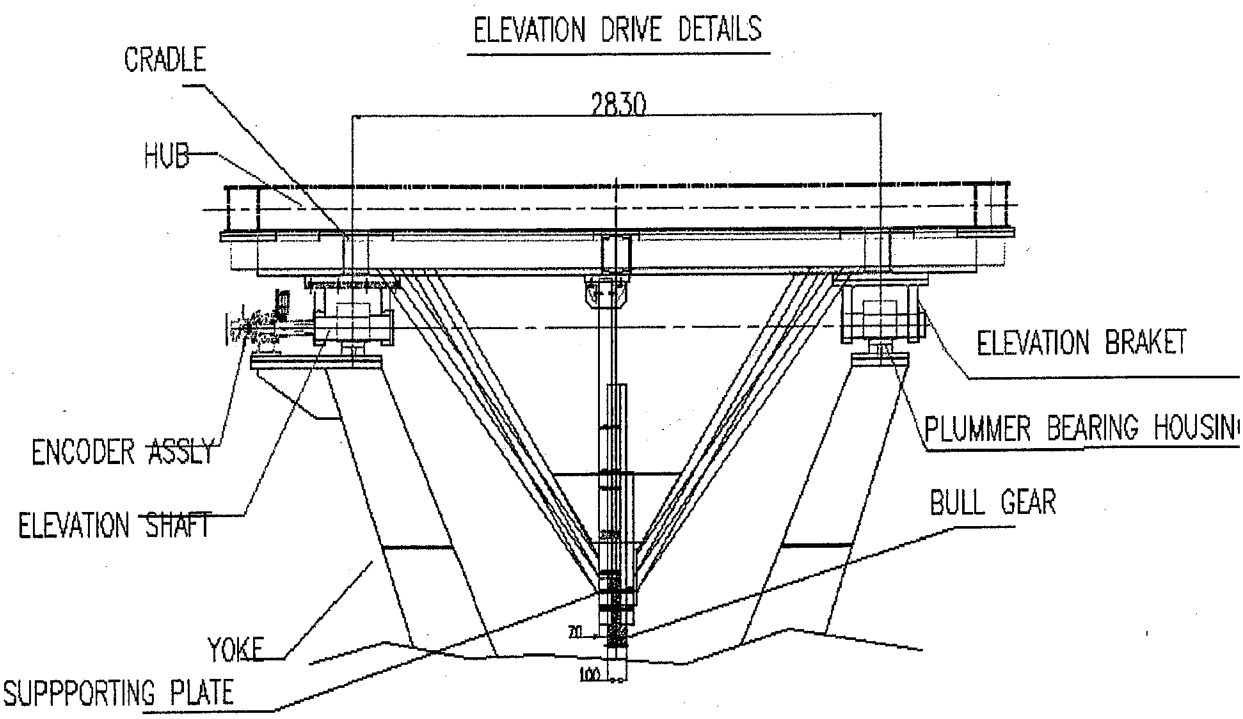


Fig 2

LOADS ON ELEVATION BEARINGS @ 150 KMPH WIND SPEED

Considering the design loads given in table 6 and loads due to braking, stow locking etc for wind speed of 150 Kmph. The axial and radial load are calculated and tabulated in table 7 critical value from table 7 are used for selection elevation bearing.

TABLE 7

	ELEVATION ANGLE ϕ	WIND ANGLE ψ	RADIAL LOAD – t F_R	AXIAL LOAD F_a
1	0^0	0	3.138	0
		90	3.590 t	5.0139
2	15^0	0	4.645	0
		90	7.3931	5.0139
3	30^0	0	6.2351	0
		90	6.2351	5.0139
4	45^0	0	8.4009	0
		90	4.0177	5.0139
5	60^0	0	7.9719	0
		90	6.7408	5.0139
6	75^0	0	9.7762 9.776	0
		90	4.6221	5.0139
7	90^0	0	8.2013 t	0
		90	5.6037	5.0139

Maximum Fr = 9.77 Fa = 5.013

8.12 ELEVATION BEARING ARRANGEMENT

The details of elevation bearing arrangements shown in fig.1 are analyzed for stresses. Forces generated due to the wind loads, the dead loads and reactions due to braking and stow lock are considered here.

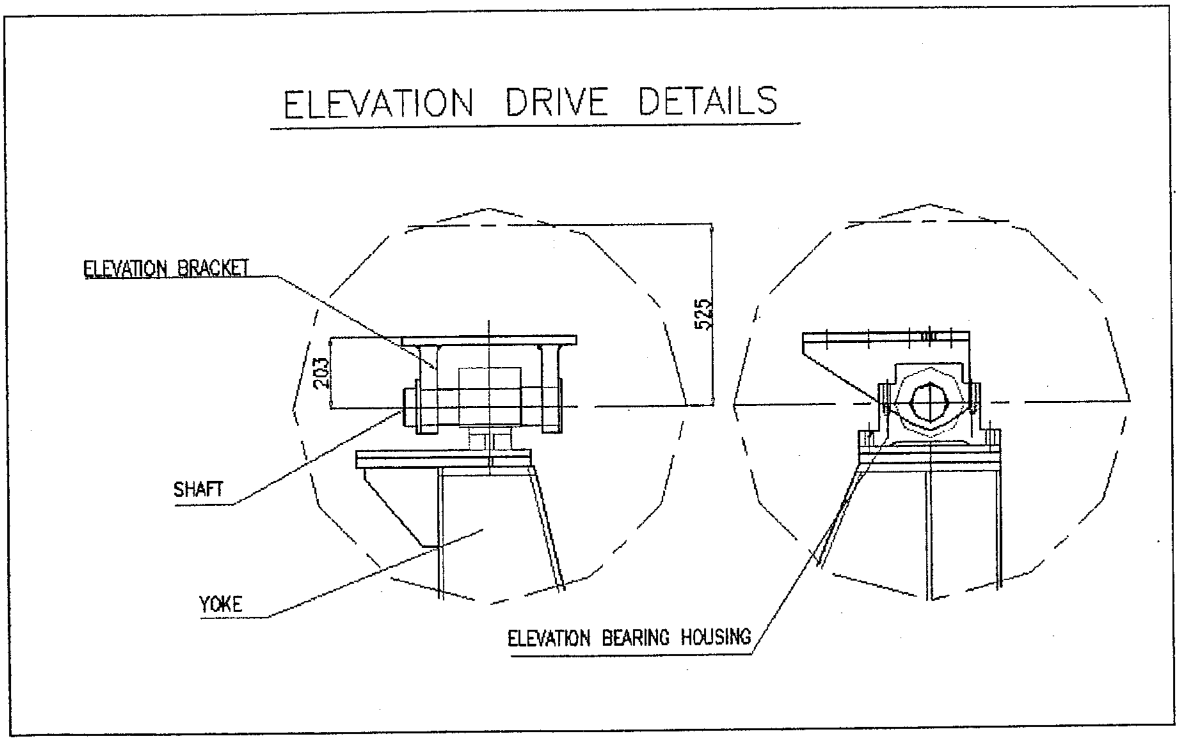


Fig 1 2

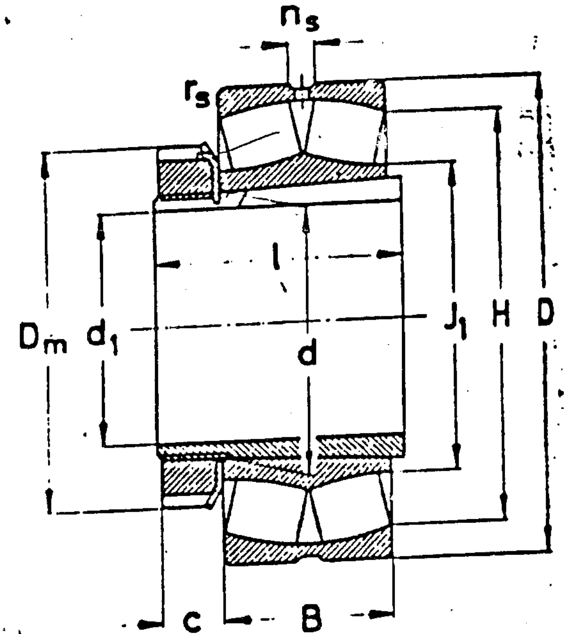
Shafts and Brackets 6

Shafts and brackets are designed for static loads at survival wind speed at 150 Kmph. The brackets are subject to severe bending loads during antenna operation. Considering these loads materials selections are made appropriately.

8.15 ELEVATION BEARING SELECTION

Bearings are selected on the basis of static capacity for consideration of safety from damage due to permanent deformation at survival wind condition. Double row spherical roller bearings are used to support the Antenna Dish.

1) Spherical Roller Bearing 2 no
2) Toroidal Roller Bearing 1 no



Bearing parameters are shown in the Table.
FAG/SKF 22222 E~~S~~K

22222 EK + H322

d	d_1	D	B	rs min	D_M	I	C	n_s	H	J_1
110		200		2.1		77		9.5		129
	100		53		145		21		179	

Dyn C Kn	e	Fa / Fr ≤ e	Fa/Fr >e y^1	Stat Co.kN	Y_0
475	0.25	2.7	4	610	2.7

Fr = Radial Load (Force) = 97,732 kN @ 150 Kmph wind Speed.

Fa = Axial Load = 50.139 kM

$\frac{Fa}{Fr}$ = 50.139/97.732 = 0.513

P_o = Equivalent static load rating

$$= F_r + Y_0 F_a$$
$$= 91.432 + 2.7 \text{ y } 50.139$$
$$= 233 < Co$$

Index of static stressing $f_0 = \frac{C_0}{P_0} = \frac{610}{233}$

$$f_0 = 2.618$$

This is adequate to take care of shock loads.

8.15.1 ESTIMATION OF BEARING LIFE

It is estimated that the antenna will work in the following modes for the following specified periods

- Tracking at 15 Arc mm / min - 50% of useful time
- Slewing at 20^0 / mm (normal) for 30% of useful time
- Slewing at 15^0 / mm (drive to stow) – 20% of useful time

Further it is assured that

a) During tracking ~ $½^0$ travel would be deemed as 1 rev / cycle

b) During slewing ~ 30^0 travel would be deemed as 1 rev / cycle

c) During drive to stow 30^0 would be deemed as 1 rev / cycle

With the above assumptions the number of revolutions made by antenna in one year

$$\frac{N}{\text{year}} = \left[\frac{0.5 \times (15/60)^0}{1/2^0} + \frac{0.3 \times 20^0}{30^0} + \frac{0.2 \times 15^0}{30^0} \right] \frac{\text{rev}}{\text{min}} \quad X$$

$$365 \text{ days} \times \frac{24 \text{ hrs}}{\text{day}} \times \frac{60\text{min}}{\text{hr.}}$$

$$= 289080 \text{ rev / year}$$

Life of Bearing = N years

$$= \underline{63000000} = 217.93 \text{ years}$$

8.15.2 EQUIVALENT DYNAMIC LOADS OF DIFFERENT WIND VELOCITIES

Estimated Dead Load = F_{DL} = 44 kN

Wind Load = Horizontal Component = F_{RH} = 97.7 kN

Vertical component = F_{WL} = 22.8 kN

Wind speed of 150 Kmph

Total vertical load $F_{RV} = F_{WL} + F_{DL}$

The resultant load on bearing = $\sqrt{F_{RH}^2 + F_{RV}^2} = F_R$

8.15.3 RADIAL & RESULTANT LOADS AT DIFFERENT WIND VELOCITIES

Wind velocity Kmph	F_{RH} KN	F_{WL} kN	F_{RV}	F^I_R
25	25	0.62	44.76	44.71
50	10	2.53	46.6	47.64
100	40	10.1	54.2	67.3

a) At 25 kmph

Resultant force = 44.71 = F^1_R

Axial force = $50.139 \times \left[\frac{25}{150}\right]^2$

= 1.393 kN

P_{25} = Eq.Dyn. load = Fr + Y^1 Fa

= 44.71 + 4 x 1.393

= 50.282 kN

b) At 50 Kmph

Resultant force = 47.64 = F^1_R

Axial force = Fa = 50.139 x 0.1111

= 5.57

P_{50} = Eq.Dyn Load = Fr + Y^1 x Fa

= 47.64 + 4 x 5.57

= 69.92 kN

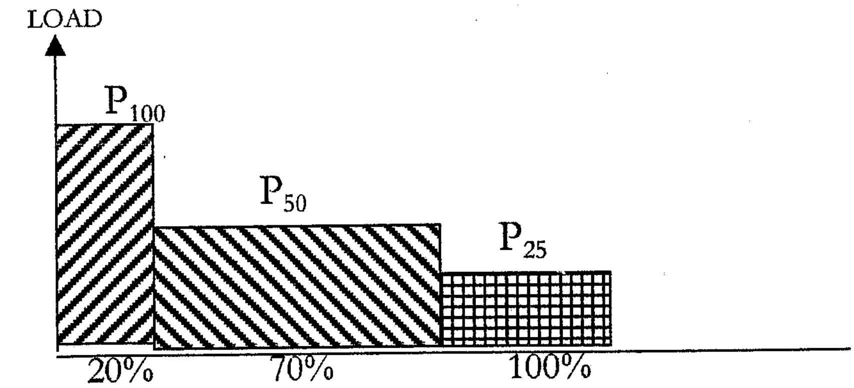
c) At 100 Kmph

Radial force $= 67.3$ kN $= F^1_R$

Axial force $= Fa = 50.139 \times 0.4444 = 22.2$ kN

P_{100} = Eq. Dyn,Load $= 67.3 + 4 \times 22.2 = 156$ kN

8.15.4 LOAD DISTRIBUTION



8.15.5 MEAN EQUIVALENT LOAD

$$P = \left[P_{100}^3 \times \frac{U_1}{U} + P_{50}^3 \times \frac{U_2}{U} + P_{25}^3 \times \frac{U_3}{U} \right]^{1/3}$$

$\frac{U_1}{U} = 0.2$ $\quad \frac{U_2}{U} = 0.5$ $\quad \frac{U_3}{U} = 0.3$ Percentage utilization

$$P = \left[156^3 \times 0.2 + 69.92^3 \times 0.5 + 50.282^3 \times 0.3 \right]^{1/3}$$

$$= 98.9 \text{ kN}$$

Basic rating life of bearing $= L_{10} = \left(\frac{C}{P}\right)^P = \left(\frac{475}{98.9}\right)^{10/3} \times 10^6$

$= 186.71 \times 10^6$ rev

C = Dyn load rating

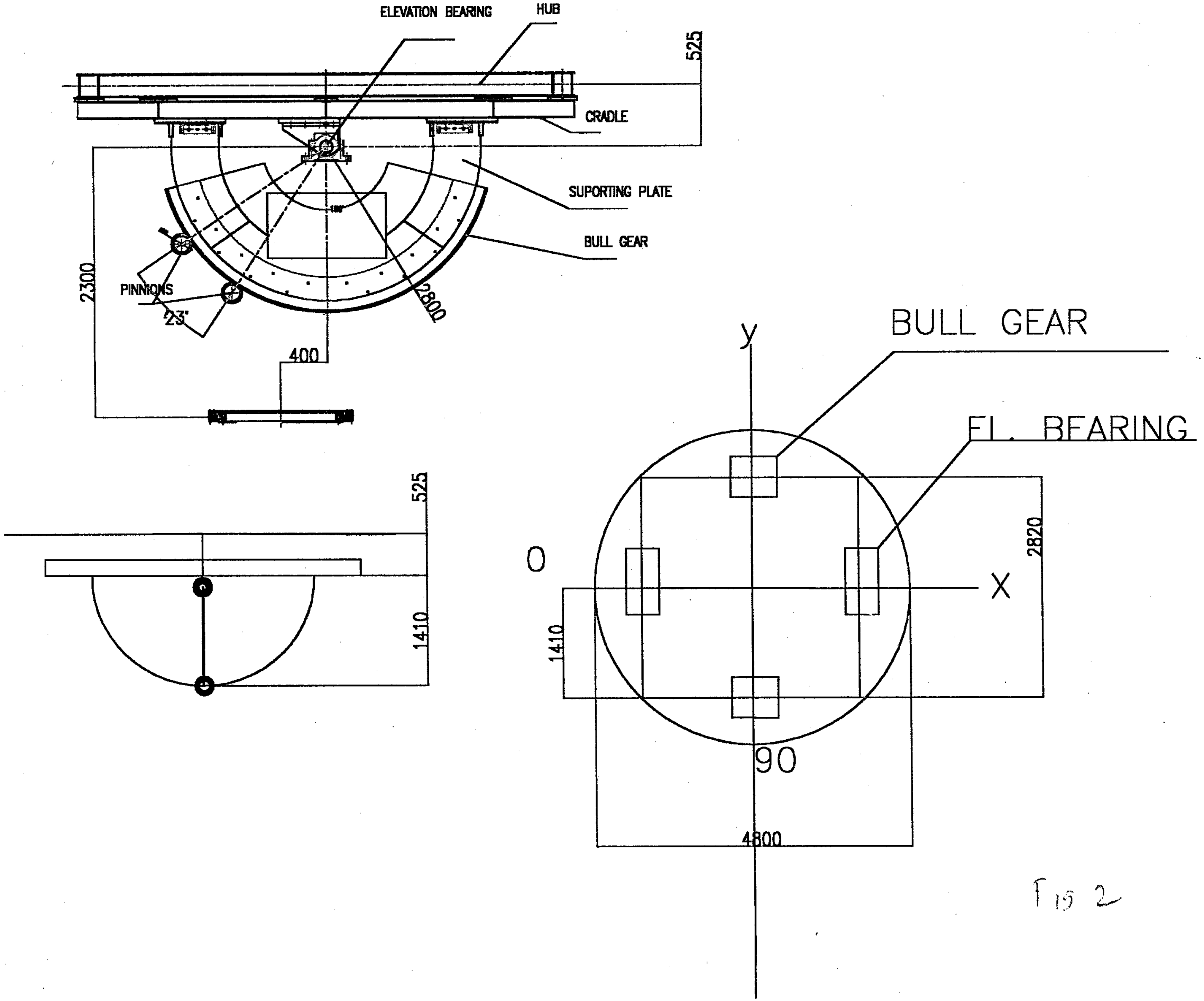
P = 10/3 for roller bearings

ELEVATION DRIVE TORQUE:

Elevation drive is through Bull gear driven by a pinion, attached to gearbox and motor. The load on Elevation drive due to self weight is very less (Balanced) and the wind load torque on EL. Drive is calculated.

Table 9

Wind Speed	ϑ EL	Wind Angle ψ	Wind Torque Tm
150 Kmph		0	-
	90	90	16.84
		0	-
	90	90	7.19
		0	8.2
	75	90	7.15
		0	-4.899
	60	90	7.058
100 Kmph		0	-2.438
	45	90	6.9
		0	.484
	30	90	6.69
		0	1.34
	15	90	6.44
		0	0
	0	90	6.2



ELEVATION BEARING
HUB
525
CRADLE
SUPORTING PLATE
BULL GEAR
2300
PINNIONS
23°
2800
400
525
1410
BULL GEAR
EL. BEARING
y
O
X
2820
1410
90
4800

Fig 2

8.16 ELEVATION AXIS DRIVE:

The load on the elevation drive are mainly is due to wind loads and are calculated as follows:

a) Maximum Torque @ 100KMPH → 10.8 t.m → 10.8×10^4 Nm
Maximum Torque @ 80KMPH → 6.912 t.m → 6.91×10^4 Nm
Maximum Torque @ 50KMPH → 2.700 t.m → 2.7×10^4 Nm

b) Tangential forces on sector gear

@100KMPH → 7.7t $= 7.7 \times 10^4$ N

@80KMPH → 4.93t $= 4.93 \times 10^4$ N

@50KMPH → 1.92t $= 1.92 \times 10^4$ N

c) Inertia Torque = 8.189 Nm.

d) Friction Torque = 40 Nm.

e)

Wind Speed KMPH	Wind Torque Nm	Total Torque Nm	Tangential force N
50	2.70×10^4	2.7048×10^4	1.932×10^4
80	6.91×10^4	6.9148×10^4	4.939×10^4
100	10.80×10^4	10.804×10^4	7.7177×10^4

8.16.1 GEAR BOX SPECIFICATION AND GEAR RATIOS:

Maximum wind speed during operation for Tracking and Slewing is 40 Kmph and when drive to Stow Lock position wind speeds can read to 80 Kmph. Calculations are made taking 80 Kmph wind speeds.

Total Torque at 80 Kmph on Elevation Axis = 6.9148×10^4 Nm.

Bull gear/Pinion ratio = Z_2/Z_1 = 350/20 = 17.5

Torque on Pinion axis = output of gear box

= 6.9148 x 104 / 17.5xη_{gb} = 6.9148/17.5 x 0.8

= 4.939 x 10^3 Nm

= 4.939 KNm.

≐ 5000 Nm (rounded)

Elevation axis Slew Speed = $20^0/360^0$ / min = 1/18 rpm = 0.05555

Assume 1500 RPM of Motor

Total reduction at motor output
speed of 1500 rpm = 1500 RPM/0.05555 = 27,000

Required Gear Box Ratio = 27,000/17.5 = 1542.85

Gear Box ratio nearest available = 1590

Output torque = 5000 Nm

Total gear ratio = 1590 x 17.5

= 27825

Since two pinions and hence two gearboxes share the load at wind speeds of 80 Kmph. It can take up higher wind speeds.

Four stage Planetary Gearbox with ratio 1590 or more is selected with output torque of 5000 Nm.

18.6.3 Elevation Bull Gear Pinion Dimensional Calculations

Gear Data

	Bull Gear	Pinion
Module	8	8
Face Width	100	120
No. of Teeth	$Z_1 = 350$	$Z_2 = 20$
Ref. Dra	2800	160
Pressure Angle	20^0	20^0

Sl. No	Parameter	Gear	Pinion
1	Base Dia = db1 = $Z_1 m \text{Cos}\alpha$	db_1 =2631.139	db_2=150.351
2	Tip Dia $da_1 = m\ [Z_1+2(1\text{-}xg+y)]$	da_1 = 2823.157	da_2=182.197
3	Root dia = $df_1 = m[Z_1\text{-}2(1+c'\text{-}x_1)]$ c= clearance coefficient @m=1 = 0.25 m = 0.25	df_1 = 2786.4	df_2 = 147.36
4	Tooth depth = $h_1 = \frac{1}{2}(da_1\text{-}df_1)$	18.3785	17.4185
5	Working depth = $h' = \frac{1}{2}(da_1+da_2)\text{-}a'$	15.8986	15.8986
6	Pressure angle @ tip $\alpha\ a_1$ $\text{Cos}\alpha\ a_1 = db_1/da_1$	$\alpha a_1 = 21.2537^0$ $21^0\ 15`\ 13``$	$\alpha a_2 = 34.3901$ $34^0\ 23`\ 24``$
7	Transverse contact ratio $\frac{1}{2}\pi\ [Z_1(\tan\alpha a_1\text{-}\tan\alpha') + Z_2(\tan\alpha a_2\text{-}\tan\alpha')]$	1.5856 > 1.2 safe.	

18.6.4 Working Pressure

$\text{Cos}\alpha^1 = a\ \text{Cos}\alpha/a_1 = \alpha = 20.70576662°$ $\quad = 20^0\ 42^1\ 21^{11}$

$a = \{(Z_1+Z_2)/2\}m = 8\ [350+20/2]^0$ $\quad = 1480$

Nominal Backlash $Jn = 0.05 + (0.025 \text{ to } 0.1)\ m$
$= 0.25$ min $\quad$ avg $= 0.55$ mm average
$Jn = 0.85$ max $\quad \cong 0.6$ mm

18.6.5 Modification Factor For Bull Gear and Pinion

$\alpha = 20°$ $Z_1 = 350$ $Z_2 = 20$

$Zm = 350+20/2 = 185$, $X_1 = 0.4$, $X_2 = 0.46$, $X_1+X_2 = 0.86$

For higher bending and wear resistance = X_1+X_2 selected at 0.8 from graph (Ref Azimuth gear-graph read by interpolation at X gear = 0.4 X pinion = 0.46)
= X_1 = X_2

$(X_1 + X_2) / Zm = 0.86/185 = (inv\alpha' - inv\alpha) / \tan\alpha$

$$0.004649 = \frac{(inv\alpha' - 0.014904)}{\tan 20°}$$

$inv\alpha' = 0.016595969$

$\alpha' = 20°42'17''$ from Machinery handbook

1) Center distance modification coefficient = y = (y/Zm) x Zm
= 0.00458 x 185
y = 0.8473

2) Working center distance = a′ = m (Zm+y)
= 8 (185 + 0.8473)
a′ = 1486.7784 m.m

18.6.6. Elevation Bull Gear / Pinion

These values are needed for inspection of gears

Calculation for Base Tangeral Length

$W_k = W_k^* \times m + 2\, m \times \sin\alpha$

For Pinion

W_k for Std Width across 3 teeth = $w_k^* = 7.66044$

Base Tangent length across 3 teeth = W_k

W_k = $W_k^* m + 2m \times \sin\alpha$

= $7.66044 \times 8 + 2 \times 8 \times 0.46 \times \sin 20$

= 63.80078825

W_3 = 63.800 mm

For Gear

Normally Base Tangential length is measured across 10% of teeth + 2 teeth for 350 teeth it will be 37 teeth.

$$W_k \text{ for 37 teeth} = m \cos\alpha\,[(K-0.5)\,\pi + Z \,\mathrm{inv}\,\alpha] + 2\,mx \sin\alpha$$

$$W_k = 8 \cos 20\,[(37 - 0.5)\,\pi + 350 \times \mathrm{inv}20] + 2 \times 8 \times 0.4 \times \sin 20$$

$\boxed{= 903.426 \text{ mm}}$ inv20 = 0.014904

Accuracy class – DIN 10

	Gear	Pinion
Tolerance on Wk = fw = pinion	-0.23 -0.47	0.15 pinion 0.3
Tooth to tooth error f_r^{11}	0.063	0.04
Radial runout f_r	0.180	0.125
Involute Profile error f_f	0.08	0.05
Adjacent Pitch error f_t	0.08	0.05
Cumulative Pitch error F_t	0.28	0.16
Tolerance on center distance F_a	± 0.16	
Base Pitch Error f_e	0.08	0.05
Total composite error double flank F_1^{11}	0.18	0.125
Tooth alignment error f_B		0.058

8.17 ANALYSIS OF BULL GEAR & PINION

Following are calculations & analysis of bull gear & pinion from strength point of view, for pitting resistance & bending.

8.17.1 Parameters

	Pinion	Bull gear
Module	8	8
Number of teeth	20	350
Pressure angle	20	20
PCD ref Dia	160 mm	2800 mm
Face width	100 mm	100 mm
Profile shift	0.46	0.4
Material	C45 $40Ni_2Cr_1Mo25$ IS 1570 Induction/Flame hardened to 400 BHN or 15 Ni_2Cr_1Mo15 IS 1570 or $20MnCr_5$ IS 1570 Case hardened	C45 IS 1570 or equivalent Hardened to 300 BHN

18.7.2 Due to profile shift in pinion

The operating pressure angle changes to ϕ_r

$\cos\phi_r = a\cos\phi / a'$

$\phi_r = 20.7057°$

$inv\phi_r = inv\phi + (2Kx \tan\phi)/(Z_1+Z_2)$

Z_1 = pinion No of teeth

Z_2 = gear No of teeth

Working or Operating Center Distance = a'

$a' = m\,(Zm + y)$

$= 8(185 + 0.8473)$

$= 1486.778$

Recommended Nominal Backlash
Refer Machinery hand book Page 1870, AGMA recommendations for center distance 58 inches (1480 mm and diameteral pitch
$P = 25.4/m = 25.4/8 = 3.175$
A backlash of 0.045 to 0.065 inches (1.143 to 1.65 m) is recommended. These Backlash toleraıce contains Allowance for gear expansion due to differential in operating temperatures etc.

8.17.3 Operating pinion radius $= d/2.$

$$d/2\ (1 + Zgear/Zpinion) = 1486.778$$
$$d/2\ (1 + 350/20) = 1486.778$$
$$\therefore d = 160.7328$$

Operating Pressure angle $= 20.7057° = 20°42'20''$
Operating Gear radius $= 160.7328/2 \times 350/20$
$= 1406.412$
Pinion OD $= da_1$ $= m\ [Z_1 + 2(1 - xg + y)]$
$= 8\ [20 + 2(1 - 0.4 + 0.8473)]$
$= 182.197$

8.17.4 Geometry factors for pitting resistance

Geometry factor for Pitting resistance for Elevation Gear and Pinion pair as per AGMA 20001 – B88 standard ref machinery Hand Book P1834 onwards. This is for the analysis of Elevation Bull gear/pinion to check for pitting resistance. As per the above standard and procedure.

Note: Certain variables are made dimensionless by dividing by module m_n.

Calculation for Pitting resistance Basic geometry factor (Various nomenclature as per Machinery Handbook).

a. Gear ratio m_G $= n_2/n_1 = 350/20 = 17.5$ --------------1
   Where n_1 = Pinion No of teeth
   Where n_2 = Gear No of teeth

b. Standard reference Pinion Pitch radius (Spur gear) $= R_1$
   $= n_1/2 = 20/2 = 10$-----------2

c. Standard (ref) gear pitch radius $= R_2 = R_1 \times m_G$

$= 10 \times 17.5$

$= 175$ ----------------------------3

d. Standard transverse Pressure angle $\phi = \tan^{-1} (\tan \phi_n/\text{Cos}\psi)$

{Where ϕn = std normal pressure angle=20° $= \tan^{-1} (\tan 20°)$

ψ = helix angle = 0} $= 20°$------------4

e. Pinion Base radius $Rb_1 = R_1 \text{Cos}\phi$ $= 10\text{Cos}20°$

$= 9.3969$ ---------------5

f. Gear base radius $= Rb_2$ $= Rb_1 \times m_G$

$= 9.3969 \times 17.5$

$= 164.4462$ -------------6

g. Operating transverse pressure angle $\phi r = \text{Cos}^{-1} [(Rb_2+Rb_1)/cr]$

$= 20.7057662$

C_r = operating center distance/unit module

$= 1486.7784/8$

$= 185.8473$--------------7

h. Transverse base pitch $= P_b$ $= (2\pi Rb_1)/n_1$

$= (2\pi x 9.3969)/20$

$= 2.9521$-----------------8

i. Normal base pitch $= Pn = \pi \text{Cos}\phi_n$ $= 2.9521$----------------9

j. Base helix angle ψ_b $= \text{Cos-}' (P_N/P_b)$

= 0 spurgear-------------10

k. Calculation of distances C_1 to C_6 along the mesh of gears

$C_6 = Cr \cdot \text{Sin}\phi_r$ { C_r=operating center distance/unit module (Ref. Machinery Hand book)

$= 185.8473 \text{ Sin } 24.7058$

$= 65.7098$ ϕ_r=operating transverse pressure angle

$= 20.7058°$}

$= 65.7098$--11

l. $C_1 = \pm [\, C_6 - (R_{o2} - R_{b2})^{0.5} \,]$

$+ [\, 65.7098 - \sqrt{(176.3673^2 - 164.4462^2)} \,]$

Ro_2 = add m - radius of gear/unit module

$= [\, 65.7098 - 63.7960 \,]$

$C_1 = 1.91378$--12

m. $C_3 = C_6/m_G+1 = 65.7098/17.5+1 = 3.55188$-----------------13

n. $C_4 = C_1 + P_b = 1.91378 + 2.9521$

$C_4 = 4.86588$---14

o. $C_5 = \sqrt{(Ro_1^{\,2} - Rb_1^{\,2})} = \sqrt{11.4473^2 - 9.3969^2}$

Ro_1 = Addendum radius of pinion/unit module

$= 91.5785 \div 8$

$= 11.4473125$

$Rb_1 = 9.3969$-----ref(5) above

$C_5 = 6.5375$---15

p. $C_2 = C_5 - P_b$

$= 6.5375 - 2.9521$

$= 3.5854$---16

q. Active length of contact $Z = C_5 - C_1$

$= 6.5375 - 1.9138$

$Z = 4.6237$--------------------17

r. Transverse contact ratio $= m_p$

$m_p = Z/P_b$ $= 4.6237/2.9521$

$m_p = 1.56624$ --------------------18

s. For spur gears with $m_p < 2$

Minimum length of line contact = Lmin

Lmin = F = Effective face width/ unit module

= 100/8

Lmin = F = 12.5---19

t. Load sharing ratio = m_N = F/Lmin

m_N = 1 for spur gears--------------------20

u. Pitting resistance Geometry factor I

$I = (\cos \phi_r \times C\psi_2)/(1/\rho 1 + 1/\rho 2)\, d\, m_N$

$= \cos 20.7058 \times 1/(1/3.5854 + 1/62.1244) \times 20.096 \times 1$

$I = 0.1578$ ---21

Note: C_ψ = helical overlap factor
= 1 for spurgear

ϕr = 20.7058 Operating Transverse .Pr angle

d = pinion operating pitch dia

$= 2C_r/m_G+1$

$= 2 \times 185.8473/17.5 + 1$

$= 20.0916$

$\rho 1$ and $\rho 2$ = Radii of curvature of pinion and gear profiles

$\rho 1$ = C_2 for spur gear , $\rho 2 = C_6 - \rho 1$

8.12.4 Pitting Resistance

$$Sc = C_p \left(\frac{W + Ca}{Cv} \times \frac{Cs}{d_F} \times \frac{Cm\ Cf}{I} \right)^{1/2}$$

Sc = Contact stress No. MP_a

Cp = Elastic co efficient $(MP_a)^{1/2}$

$$= \sqrt{\frac{1}{\pi \left(\frac{1-\mu p^2}{Ep} + \frac{1-\mu g^2}{Eg} \right)}}$$

$$= \sqrt{\frac{1}{\pi \left(\frac{1-0.3^2}{210000} + \frac{1-0.3^2}{210000} \right)}}$$

$\mu_p = \mu_g = 0.3$
Poisons Ratio
$Ep = Eg = 2.1 \times 10^5$ MP_a

$= 191.65\ (MP_a)^{1/2}$

C_a = Overload factor = 1.25 for uniform power source and Moderate Shock ref 'Dudley'

C_v = Dynamic factor =1 fig 3 page 1844 Machinery's H B as the pitch line Velocity is low.

V_t = Pitch line Velocity

$$= \frac{\pi \times \eta_\phi \times d}{60{,}000} \text{ m/Sec}$$

$$= \frac{\left(\pi \times \frac{20}{360} \times 17.5\right) \times 160}{60 \times 1000} = 0.008145 \text{ m/Sec}$$

$\therefore C_v = 1$

C_s = Size factor = 1.0

d = Operating pinion PCD = 160.73 mm (d in mm)

F = Face width = 100 mm

C_m = Load distance factor

C_m = 1.646 When transverse load distribution factor =1 = C_{mt}

= then $C_m = K_m = C_m$

C_f = Surface condition Factor = 1.0

$C_{mt} = 1 + \frac{G \times et \times F}{2 \times Wt} = C_m$

G = Tooth Stiffness constant

$= 1.0 + 1.4 \times 10^4 \text{ MP}_a$

et = Total lead mismatch between mating teeth in loaded condition

≈ 0.1 mm

F face width = 100

$$Cm = 1 + \frac{1 \times 10^4 \times 0.1 \times 100}{2 \times 77.1428 \times 10^3}$$

$= 1 + 0.64576$

$= 1.64576$

I = 0.1578

Tangential Load W_t

$$W_t = \frac{1000 \times P}{V_t} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots 1$$

$$= \frac{T}{R} = \frac{\text{Torque}}{\text{Radius}}$$

$$W_t = \frac{10.8 \times 10^4}{1.4} =$$

$W_t = 77.1428 \times 10^3$ N

W_{t20} = 3085.7 N

W_{t40} = 12357 N

W_{t60} = 49286 N

W_{t100} = 77142.85 N

8.17.4.1 Tangential Load at different Wind speed. Table

Wind Speed	Pinion torque Kgm	Tangential Load Wt (N)
20 Km ph	0.43×10^3	3.085×10^3
40 Km ph	1.73×10^3	12.357×10^3
80 Km ph	6.9×10^3	49.286×10^3
100 Km ph	10.8×10^3	77.1428×10^3

a. **At 20 Kmph**

$$Sc = Cp\left[\frac{Wt \times Ca}{Cv} \times \frac{Cs}{df} \times \frac{C_m C_f}{I}\right]^{1/2}$$

$$= 191.65\left[\frac{Wt \times 1.25}{1} \times \frac{1}{160.73 \times 100} \times \frac{1.646 \times 1}{0.1578}\right]^{1/2}$$

$$= 191.65 = [\, W_t \times 8.1121 \times 10^{-4} \,]^{1/2}$$
$$= 5.458\, [\, Wt \,]^{1/2}$$
$$= 303.21\ MP_a$$

b. **At 40 Kmph**

$$Sc = 5.458 \sqrt{12357} = 606.78\ MP_a$$

c. **At 80 Kmph**

$$Sc = 5.458 \sqrt{49286} = 1211.82\ MP_a$$

d. **At 100 Kmph**

$$Sc = 5.458 \sqrt{77142.85} = 1516\ MP_a$$

8.17.4.2 **Ref. Machinery H = and Book P 1839. Fig 2**
Sac for Surface Hardness BHN 400

Sac = 1091.5 MPa

$$Sc \leq Sac \frac{C_L C_H}{C_T C_R} = S'ac$$

Where C_L = Life factor Ref Dudley
= 1.05 for 20 & 40 Kmph
= 1.45 for 80 Kmph
= 1.5 for 100 Kmph

C_H = Hardness Ratio factor
= 1.065 ref Machinery HB P 1850
for Hardness ratio of Pinion
to gear = 1.333

C_T = Temp. factor = 1

C_R = Reliability factor = 1 Upto 80 Kmph for 1 failure in 100
= 0.85 for above 100 Kmph

Wind Vel Kmph	Sc	Sac
20	303	1220.6
40	607	1220.6
80	1212	1685.5
100	1516	2051

Applying Minor Rule

Ni = No of Permissible cycles
η_i = No. of actual cycles

$$\Sigma \frac{\eta i}{Ni} \leq 1$$ The Design life can be achieved

	Sc	Sac	Ni	η_1
20 Kmph	303	1220	10^7	4.2×10^6
40 Kmph	607	1220	10^7	1.5×10^6
80 Kmph	1212	1605	10^7	0.03×10^6
100 Kmph	1516	2051	Neglected	

$$\Sigma \frac{\eta_1}{Ni} = \Sigma\, 0.42 + 0.15 + 0.3 = 0.87$$

Since $\Sigma \frac{\eta i}{Ni} \leq 1$

CONCLUSION

Pinion meets the life criteria for pitting

8.17.5 Bending Stress Calculations (Tooth Loading)

Wind Vel Kmph	Torque Nm	Tangential Load (N)
20	0.432×10^4	3085.7
40	1.728×10^4	12357
80	6.91×10^4	49280
100	10.8×10^4	77142.85

Total Moment of Inertia about elvm axis $= 14078\ Kg\ m^2 = J$

Angular velocity of azimuth @ 20^0/ min $= \frac{2\pi n}{60} = \frac{2\pi\ 20 / 360}{60} = \omega_1$

Ang. Accln $= \alpha = \frac{\omega_1}{t} = \frac{5.817 \times 10^{-3}}{10} = 5.817 \times 10^{-4}\ rad/Sec^2$

Accln time @ 10 Sec

Inertia Torque $= J\,\alpha = 14078 \times 5.817 \times 10^{-4}\ Kg\ mtr^2 \times rad/sec^2 = 8.189\ Nm$

Total Vertical Weight

Antenna	=	3000 kg
Cradle	=	1500 kg
Bull gear, Counter balance wt	=	3000 kg
Bracket & Misc	=	500 kg
Total Weight	=	8000 kgs = 80,000 N

Friction torque @ Bearings = $80000 \times \frac{0.01}{2} \times 0.1 = 40$ Nm

Max torque acting on the elevation gear

$$= 10.8 \times 10^4 \text{ Nm} + 8.188 \text{ Nm} + 40\text{Nm}$$

$$= 108048 \text{ Nm}$$

Tangential force acting on pinion

$$F_t = \frac{\text{Total torque}}{\text{Pitch Radius}} = \frac{108048}{114}$$

$$= 77177 \text{ N}$$

Pitch line Velocity at Max Speed $= \dfrac{\pi n d}{60{,}000}$

$= 0.008145$ m/sec

Velocity factor $= e_v = \dfrac{3.05}{3.05 + 0.008145}$

$= 0.997$

Form factor $f = 0.154 - \dfrac{0.9n}{20}$

$= 0.1084$

Bending stress on the Pinion $f_b = \dfrac{F_t}{\pi \times b \times m \times e_v \times f}$

$$f_b = \frac{77177}{\pi \times 100 \times 8 \times 0.997 \times 0.1084} = 284.13\ N/mn^2 \quad f_b = \frac{F_t}{271.62}$$

f_{b100} = 284.13 N/mm^2 F_{t20} = 3085.7+8.18 + 40 = 3137 N

f_{b20} = 11.55 N/mm^2 F_{t40} = 12357 + 8.18 + 40 = 12405N

f_{b40} = 45.67 N/mm^2 $F_{t\,80}$ = 49280 + 8.18 + 40 = 49328

f_{b80} = 181.6 N/mm^2

Allowable stress in continues loading = 380 N/mn^2 - for C45 IS 1570 direct Hardening steel for $40N_12Cr$ 1Mo 28 IS 1570 alloy that direct hardening steel.

Note : Loads at Wind speed 80 to 100 Kmph are shared equally by two pinions and therefore the stress are halved.

i.e. $f_b100 = \frac{284.13}{2} = 142\ N/mm^2$

$f_b\ 80 = 181.6 = 90.8\ N/mm^2$

CONCLUSION

Pinions are very safe & has safety factor of 2.6 for severely loaded condition

8.17.6 Surface endurance stress calculations

Since the velocity of rotation is very low dynamic loads can be taken as equal to static loads & wear load = Dynamic load = F_w

$F_w = F_t$

Fw20 = 3137 N

Fw40 = 12405 N

Fw80 = 49328 N

Fw100= 77177 N

Surface endurance stress = f_e

$$f_e = \sqrt{\frac{Fw \times 1.4 \times E}{b \times q \times m \times Z_2 \times 2 \sin\phi}} \qquad q = \frac{2 Z_2}{Z_1 + Z_2}$$

$$f_e = \sqrt{\frac{Fw \times 1.4 \times 210000}{100 \times 1.892 \times 8 \times 20 \times 2 \sin 20 \times 7057}} \qquad = \frac{2 \times 350}{350 + 20} = 1.892$$

$$f_e = \sqrt{Fw \times 13.734} = 3.705 \sqrt{Fw}$$

$$f_{e20} = 3.705 \times \sqrt{3137} = 207.57 \ N/mm^2$$

$$f_{e40} = 3.705 \times \sqrt{12405} = 412.76 \ N/mm^2$$

$$f_{e80} = 3.705 \times \sqrt{49328} = 823.09 \ N/mm^2$$

$$f_{e100} = 3.705 \times \sqrt{77177} = 1029.54 \ N/mm^2 \approx 1030 \ N/mm^2$$

Endurance limit depends on surface hardness

Surface Hardness in BHM leaded $= \dfrac{1030 + 70}{2.75} \approx 400$ BHN

The pinion therefore to be hardened to 400 BHN
and the gear to be hardened to 300 BHN

Permitted surface
Endurance limit $=$ BHN x 400 – 10,000
$=$ 400 x 400 – 10,000
$=$ 150000 p/Sqm
$=$ 10563.3 kg/cm^2
$=$ 1056.3 N / mm^2

Endurance loads @ wind speeds 80 to 100 Kmph are shared equally by two pinions and therefore the loads & stress are halved.

i.e. $$f_{e\,80} = \frac{823.09}{2} = 414.545 \ \ N/m^2$$

$$f_{e100} = \frac{1629.54}{2} = 514.77 \ \ N/m^2$$

CONCLUSIONS

The bull gear and pinion are safer and has safety factor of 2 in the severely loaded condition.

FINAL CONCLUSIONS

Ele. Bull gear / Pinion	Module 8
Face width	100 mm
Tangential Force tooth load	77.177 KN @ 100 Kmph 49.328 KN @ 80 Kmph Load sheared by two pinions equally
Bending stress max	142.N/mm^2
Max permitted bending stress	380 N /mm^2
Safety factor	2.6
Max surface endurance stress	414.5
Max permitted surface	1056 M/mm^2
Safety factor	2
Surface Hardness of Pinion	400 BHN
Material of Pinion	40Ni 2Cr 1Mo 28 1S 1570
Surface Hardness of Bull gear	300 BHN
Material of Bull gear	C45 or equivalent – IS 1570

8.8.1 CALCULATION OF STRESS IN SHAFT

Radial force = 9.14 tons per bearing
Axial force = 5.014 tons

Shaft size is arrived at after a progressive iterations and sized @ ϕ 100

The selection parameter, the polar movement of

a) Inertia $= \dfrac{\pi D^4}{64} = 490.8\ cm^4$

b) The bending movement = 34275 Kg/cm

c) The bending stress generated = 3.49 Kg/mm^2

Allowable bending stress for material of shaft = c 45 IS1570 = 11.7 Kgs/mm^2

Shaft is safe under bending

d) Bearing pressure generated in at the

Ends in the brackets $= \dfrac{\text{Load}}{\text{Bearing area}} = 0.914\ Kgs/mm^2$

Allowable bearing Pr in MS = 3.49 Kgs/mm^2

e) Shear stress in the shaft - $\dfrac{\text{load}}{\text{Shear Area}}$ $=1.16\ Kgs/mm^2$

Allowed shear stress for C_{45} = 703 Kgs/mm^2

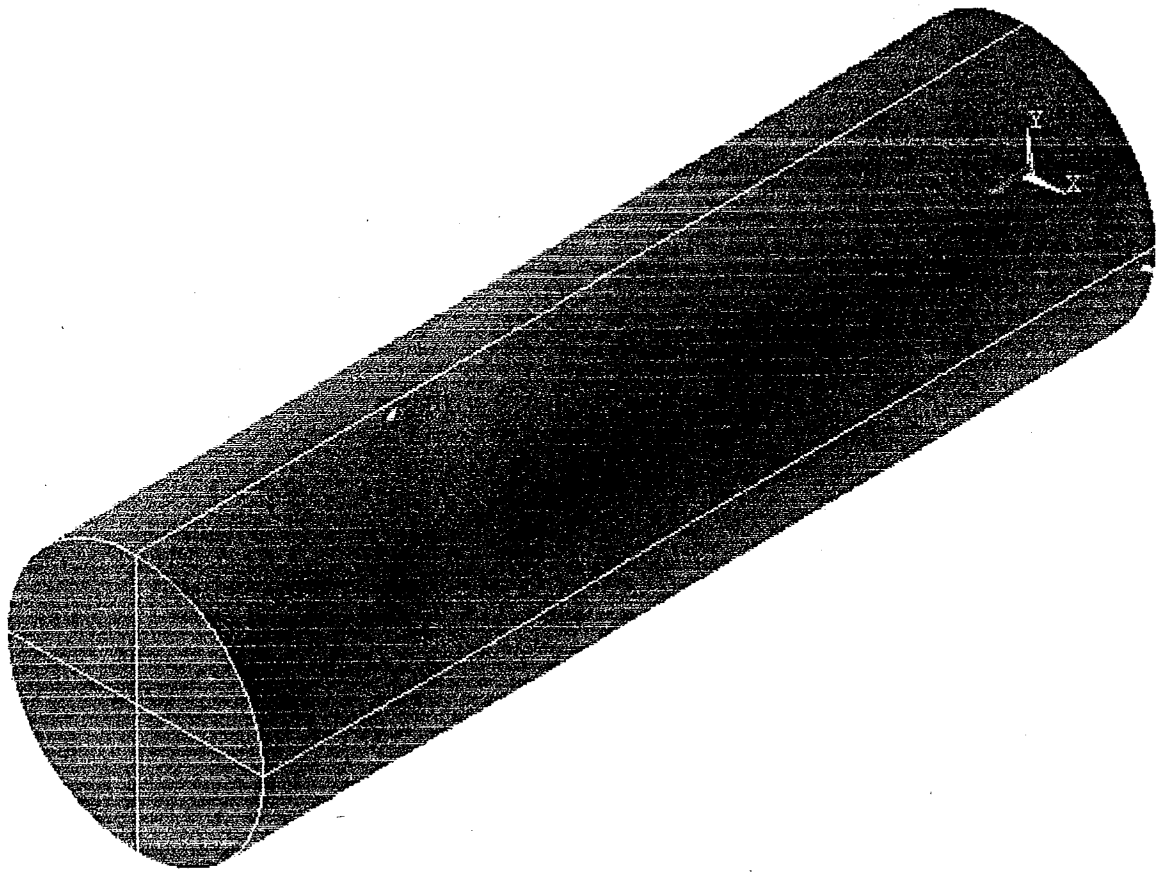
Radial force at 50 Kmph = 1015 kg

Max.deflection $\dfrac{Wl^3}{192\ EI}$ = 0.138 mm

Max.deflection @ 80 kmph = 0.355 mm

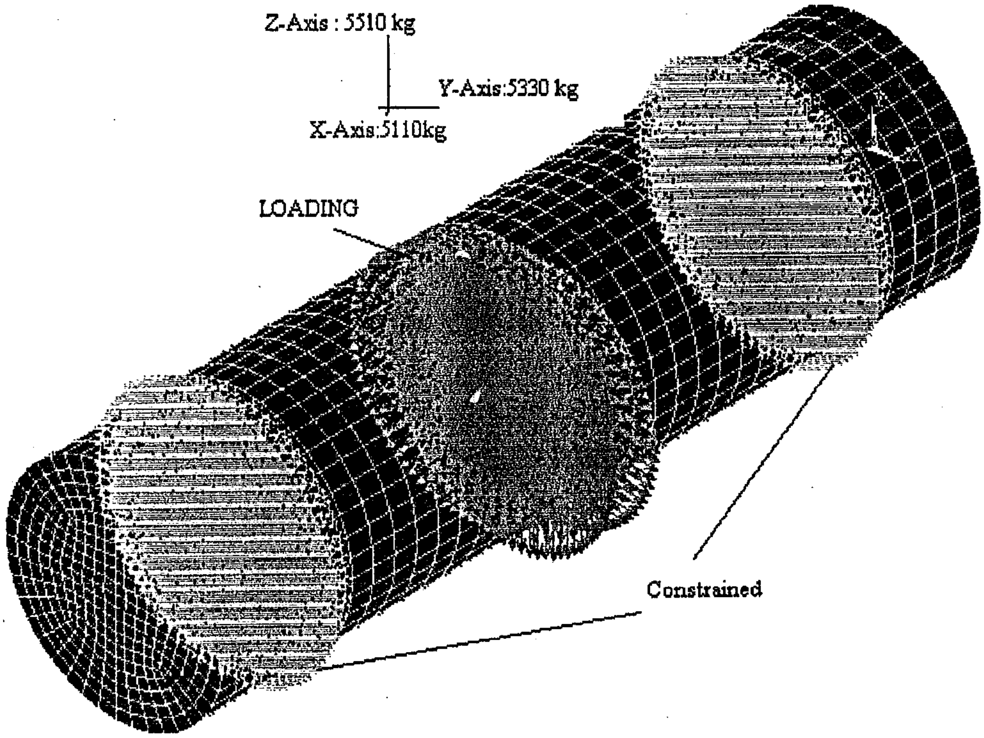
@ a Radial force of 2600 Kgf

STRUCTURAL ANALYSIS FOR SHAFT



SHAFT MODEL

STRUCTURAL ANALYSIS FOR SHAFT



SHAFT MESH MODEL

WEIGHT OF SHAFT: *21.5 kg*

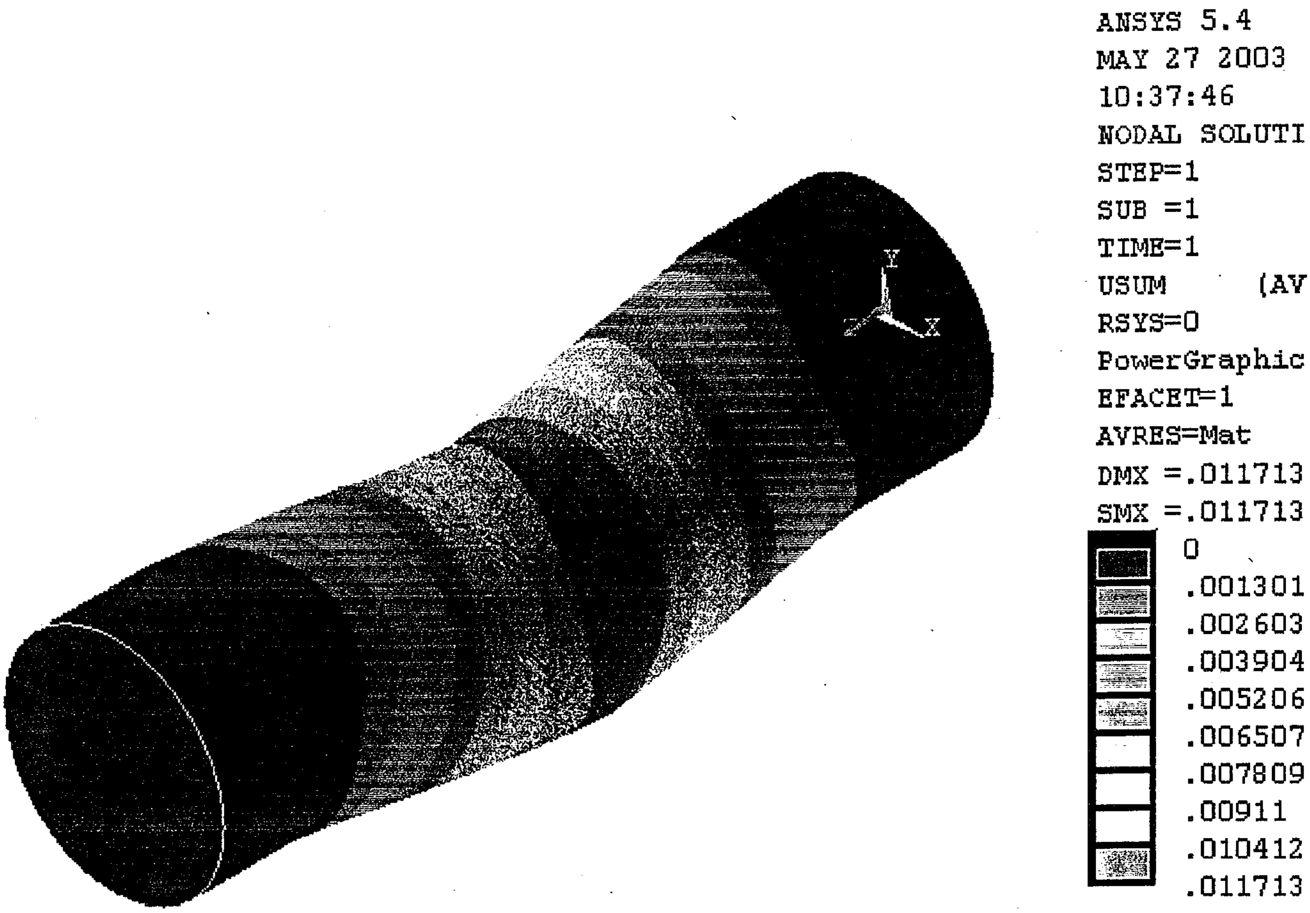
LOADING CONDITION: (REFER SHAFT MESH MODEL Fig.)

Z-AXIS: *5510 kg*

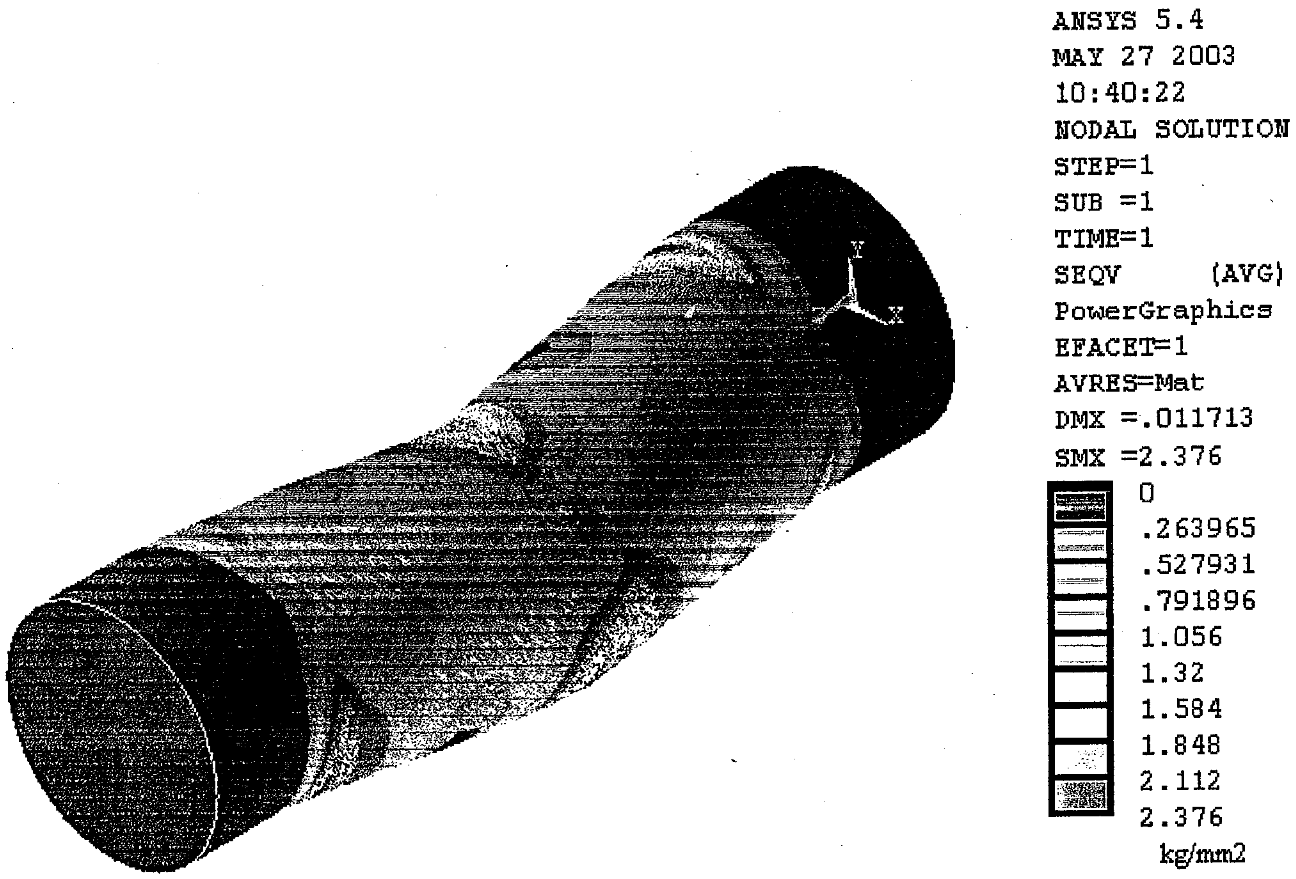
Y-AXIS: *5330 kg*

X-AXIS: *5110 kg*

STRUCTURAL ANALYSIS FOR SHAFT



STRUCTURAL ANALYSIS FOR SHAFT



8.13 CALCULATION OF BOLT STRENGTH AT PLUMMER BLOCK

Force $F = 9.14$ tons
$= 9140$ Kgf

Ref Fig Fig Sheet

$Q = 3006$ Kgf

4 nos. M_{20} bolts are used

Load shared by two bolts

a) Load per bolt = 1503 Kgf @ A

Max. load capacity M20 x 1.5 class 12.9 = 8000 Kgf

∴ bolts are safe

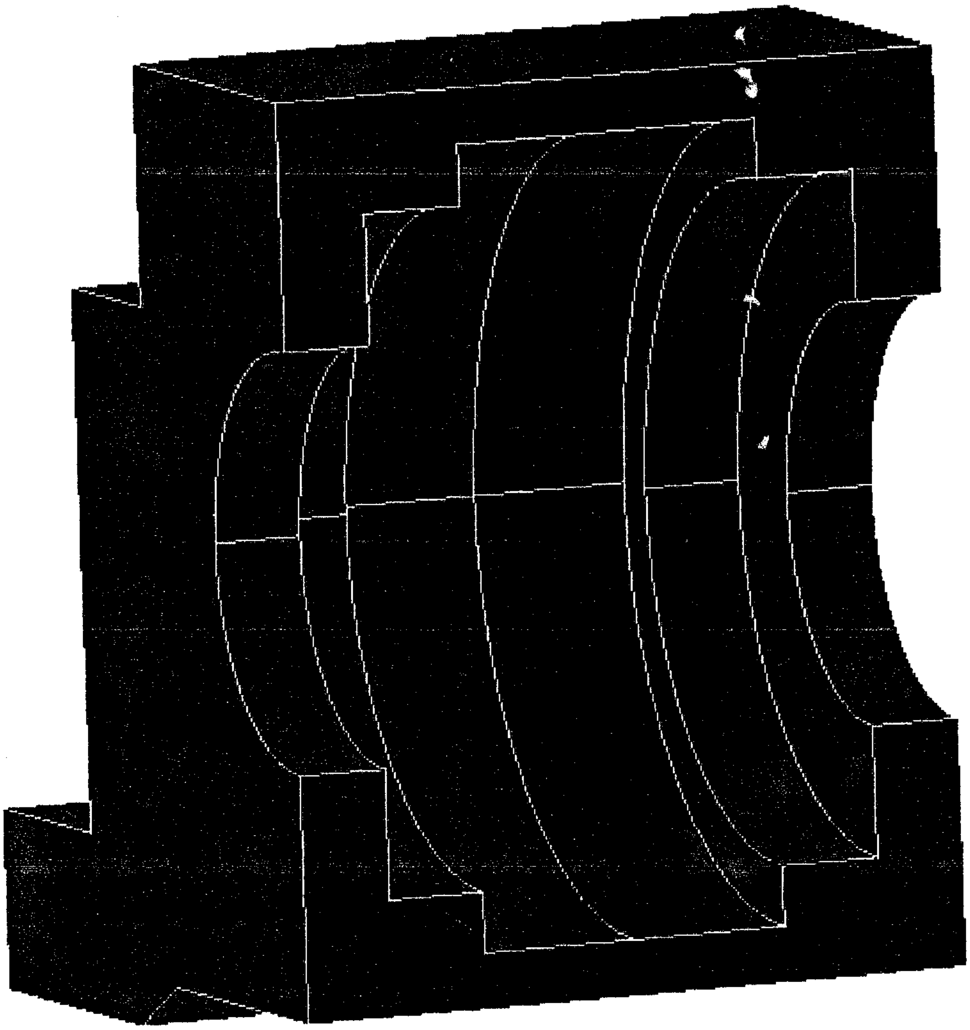
Let us use bolts of good make like UNBRAKO – Class.12,9

b) Shear capacity of bolt = $\frac{\text{Load}}{\text{n x Shear Area}}$ = 4 nos. bolt

This is also safe = 814 Kgf/mm^2

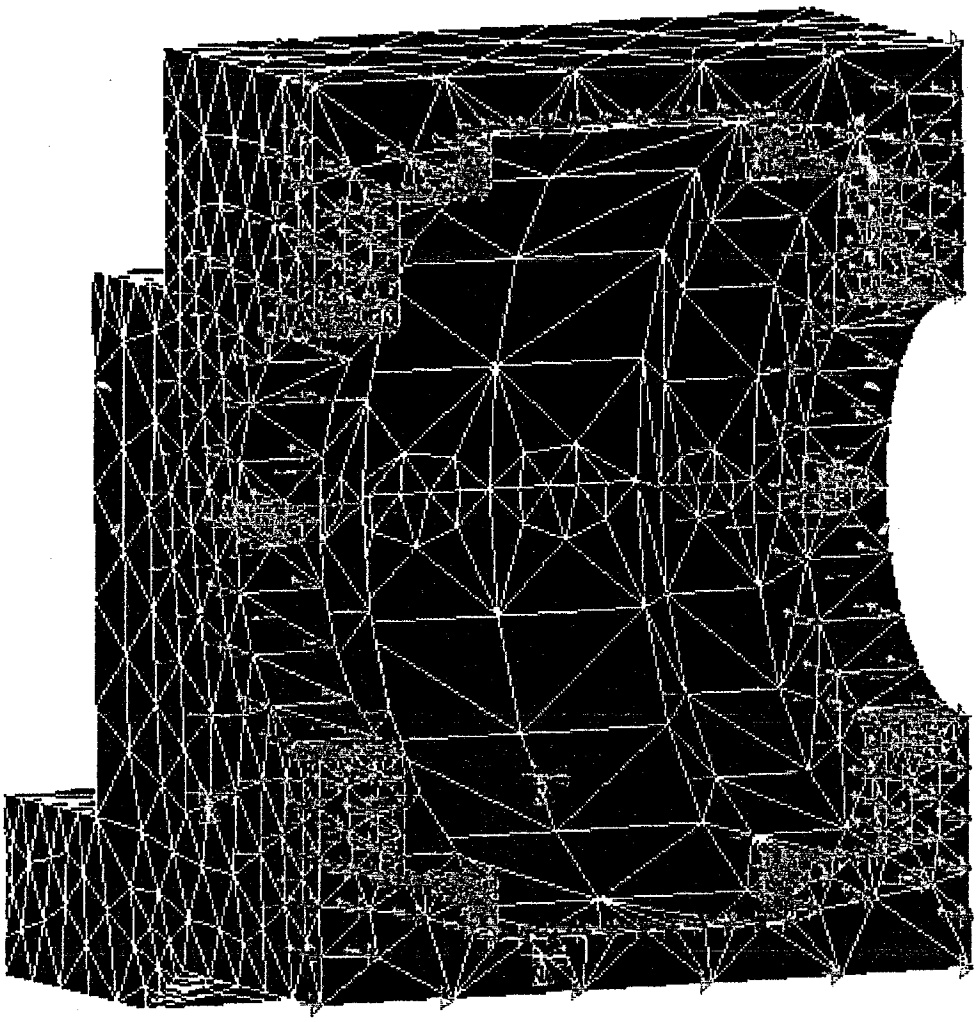
Plummer blocks can be located to position by using corner blocks as shear loading of bolts are not desirable. Also bottom of Plummer block to be thicker.

STRUCTURAL ANALYSIS FOR PUMMERBLOCK



PLUMMERBLOCK

STRUCTURAL ANALYSIS FOR PUMMERBLOCK



PLUMMER BLOCK MESH MODEL

WEIGHT OF PLUMMER: *21.5 kg*

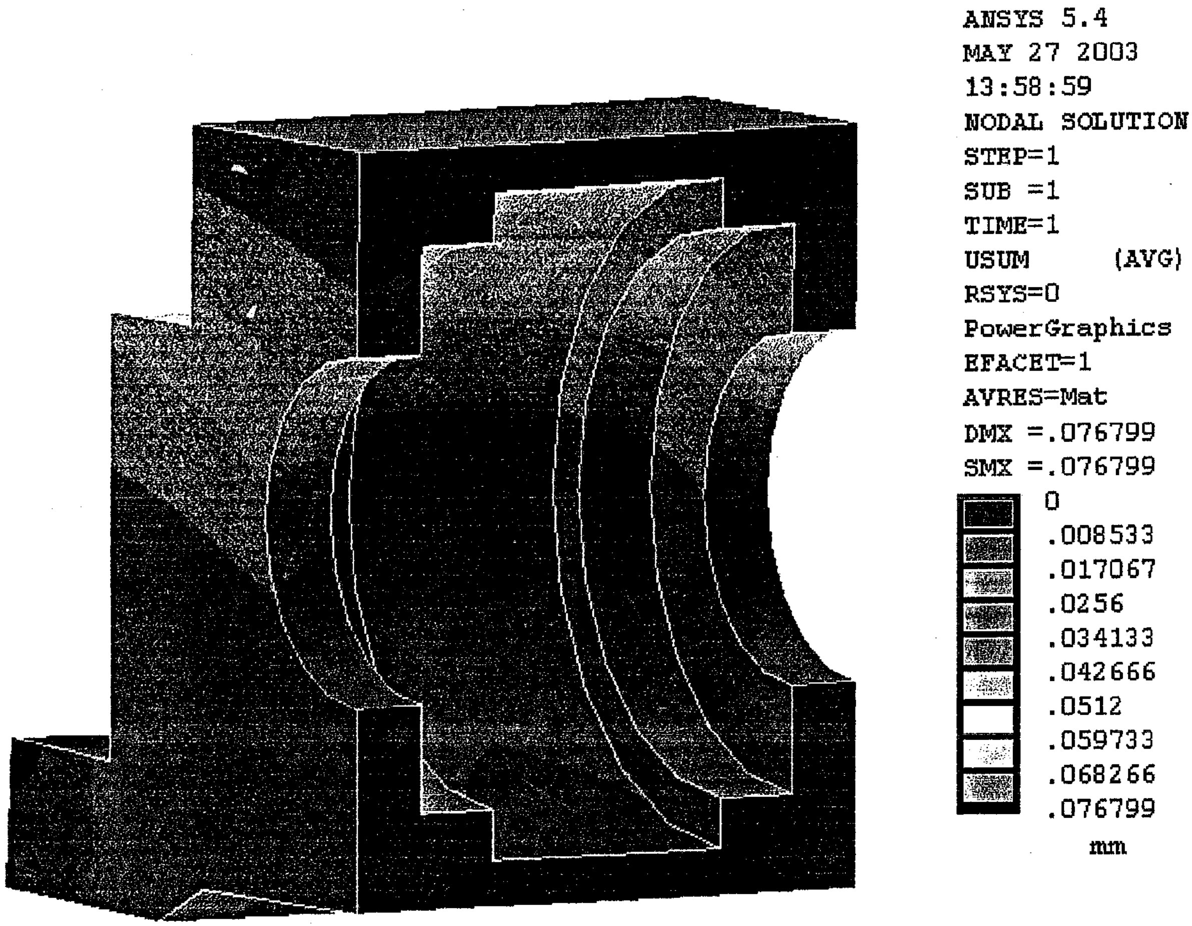
LOADING CONDITION:

Z-AXIS: *5510 kg*

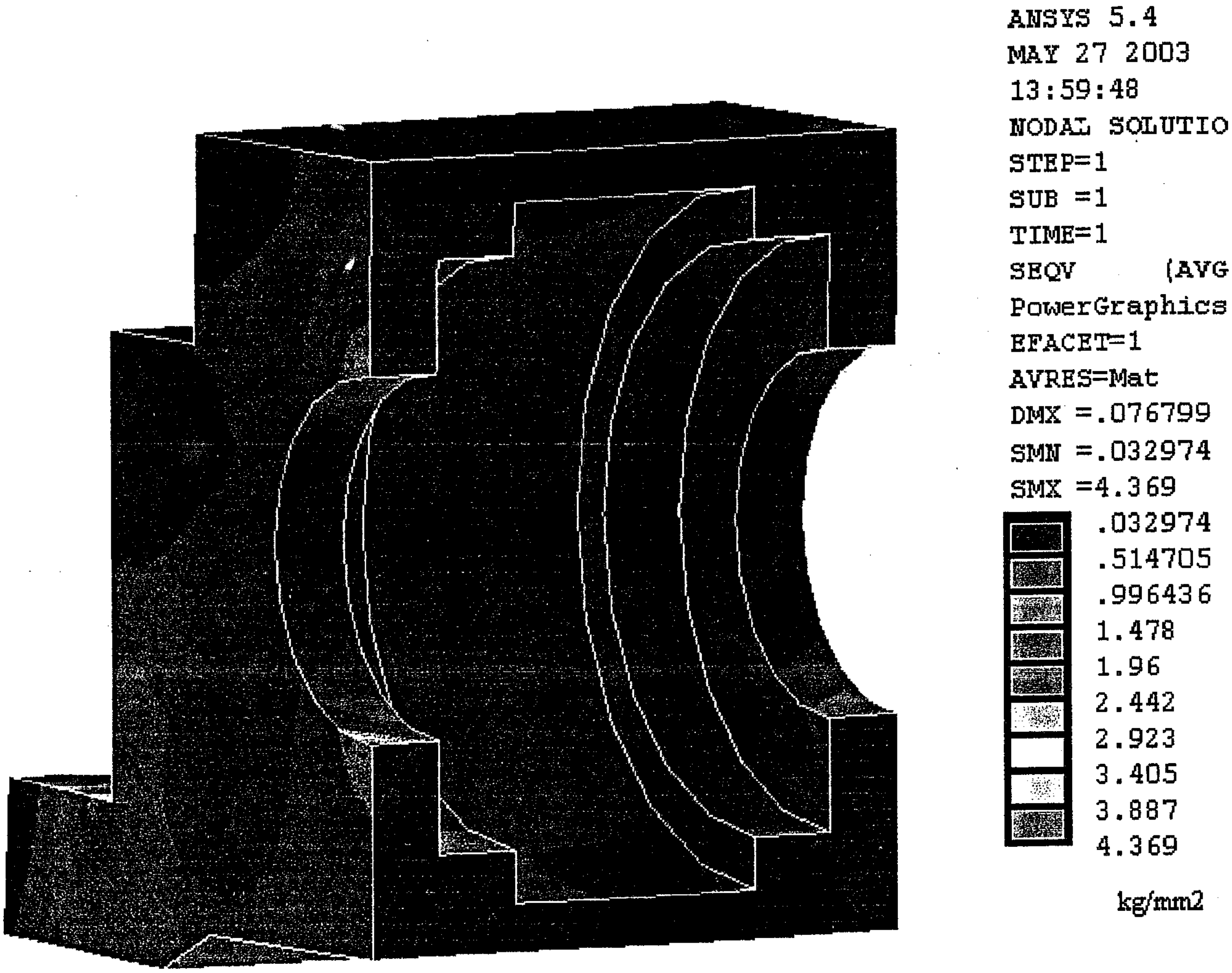
Y-AXIS: *5330 kg*

X-AXIS: *5110 kg*

STRUCTURAL ANALYSIS FOR PUMMERBLOCK



STRUCTURAL ANALYSIS FOR PUMMERBLOCK



8.14 CALCULATION OF THE BRACKETS USED FOR MOUNTING THE ELEVATION BEARING

Ref Fig sheet

a) Bearing pressure at A = Load / Area = 0.914 Kgf/mm^2
Allowed Bearing Pressure = 5.9 Kgf/mm^2

b) Shear Stress @ weld zone C = 1.19 Kgf/mm^2
Allowed shearing stress = 4.4 Kgf/mm^2 Length of weld = 48 mm
Size of weld = 4mm No. of sides welded = 2

Bending moment = Fa x 17.8 = 89231.4 Kgcm = 500 cm^4

c) Bending stress = f = 4.46 Kgf/mm^2
Bracket is Safe in bending
Thin block is optimized in FEM model

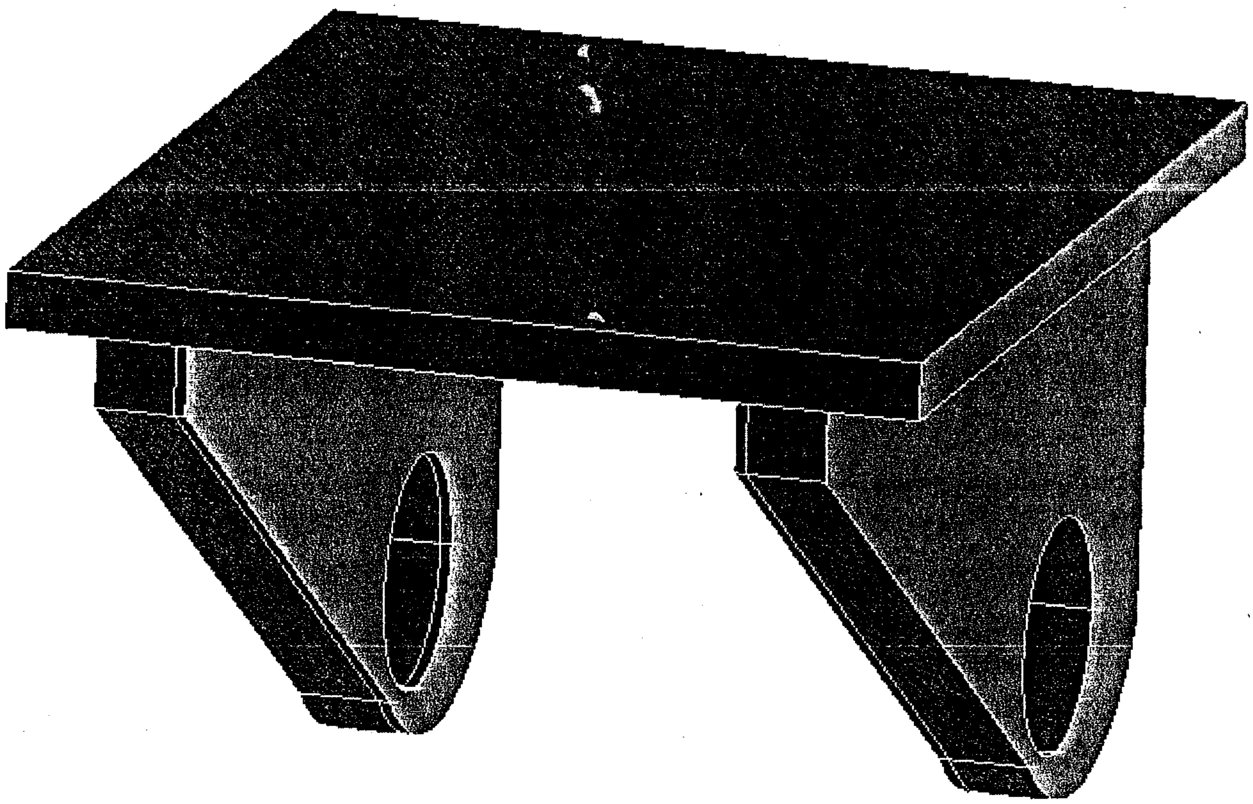
d) Shear strength of welded joint = Load / Shear area
= 5.013 Kgf/mm^2
Size of weld = 10m Length of weld = 5 cm
No. of sides welded = 2

e) Stress generated in the weld = 1.414 x moment / size of weld b x length of weld (b+h) h = thickness of plates = 5

= 5.663 Kgf/mm^2

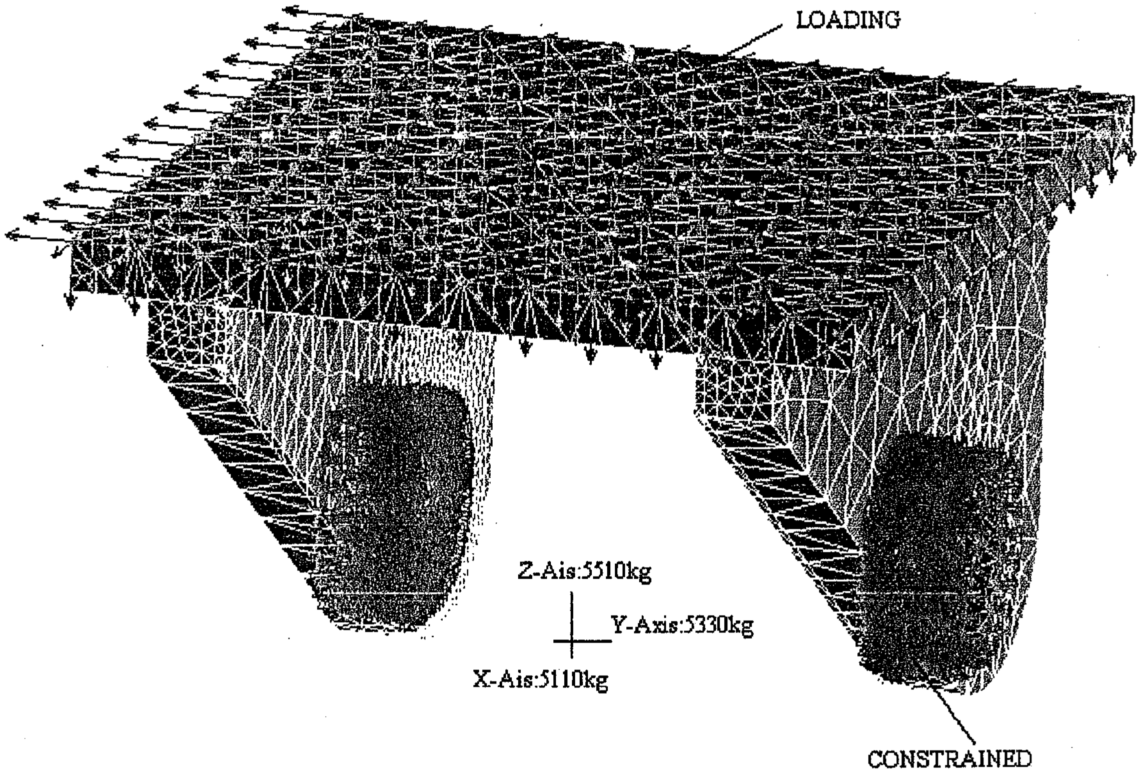
what is ...

STRUCTURAL ANALYSIS FOR BRACKET



BRACKET MODEL

STRUCTURAL ANALYSIS FOR BRACKET



BRACKET MESH MODEL

WEIGHT OF BRACKET: *82.5 kg*

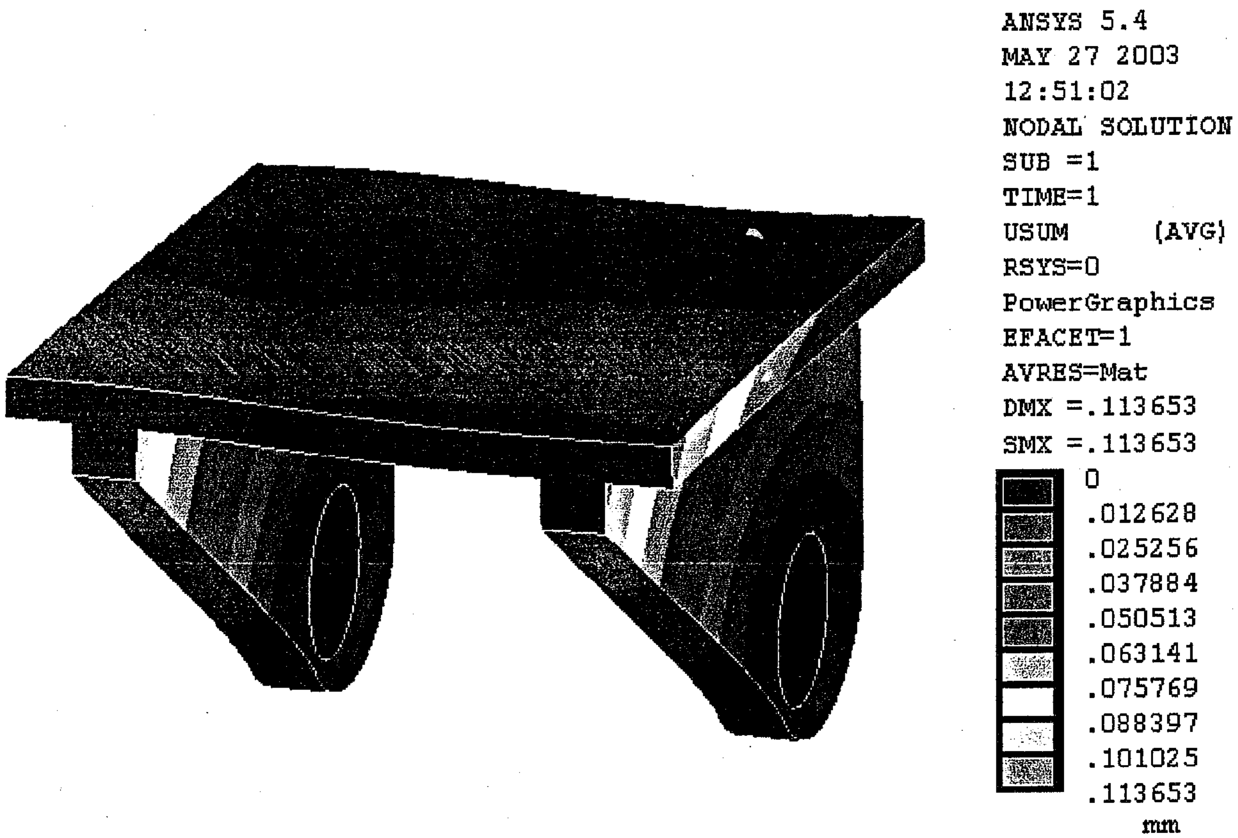
LOADING CONDITION: (REFER BRACKET MESH MODEL Fig.)

Z-AXIS: *5510 kg*

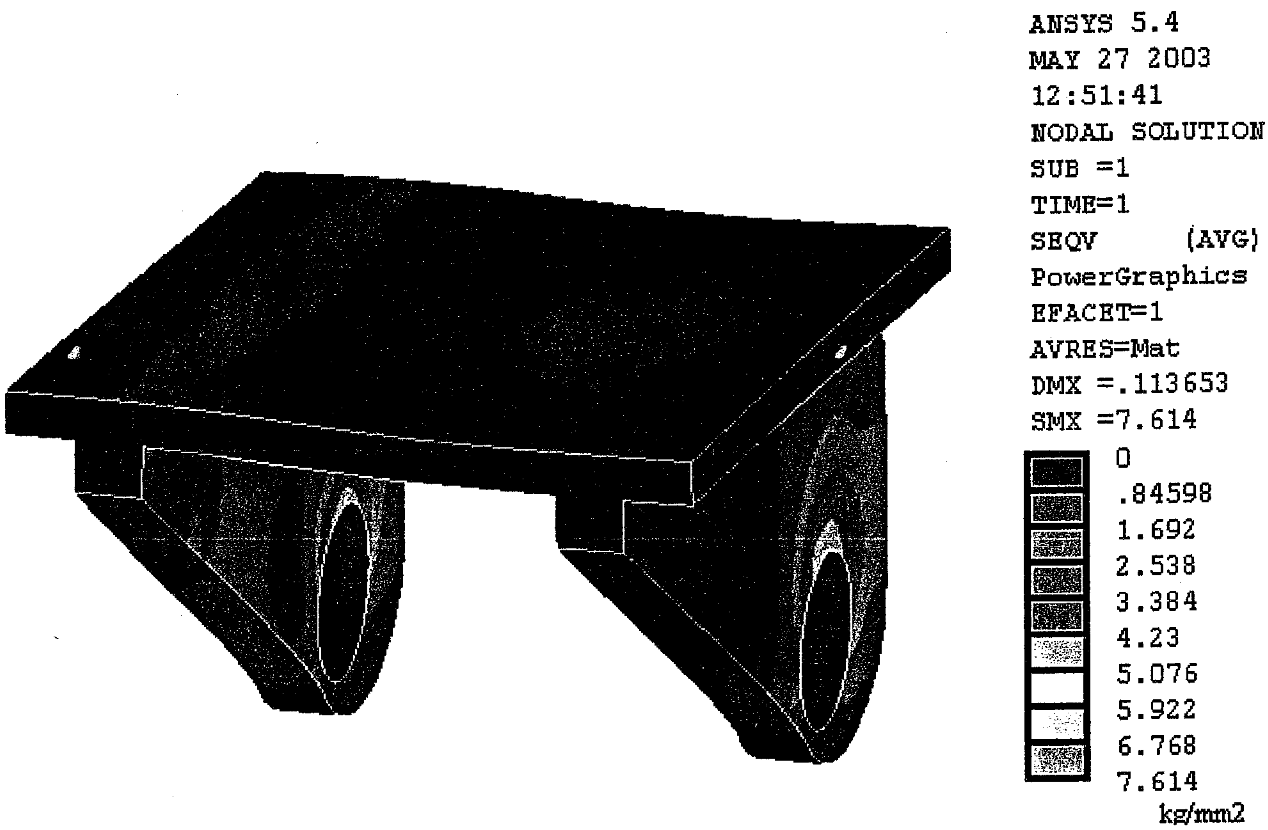
Y-AXIS: *5330 kg*

X-AXIS: *5110 kg*

STRUCTURAL ANALYSIS FOR BRACKET



STRUCTURAL ANALYSIS FOR BRACKET



7. AZIMUTH DRIVE SYSTEM

GENERAL

The Azimuth drive rotates the yoke and reflector about Azimuth axis for ± 270^0. This rotation along with rotation of elevation axis is to position the dish in the desired direction.

7.1 CONSTRUCTION FEATURES

The azimuth drive system consists of a slewing ring with external gear. The yoke is bolted to the azimuth top plate. The two gear boxes are mounted at 180 deg. Separation on the Steel Shell structure where slew ring is supported. The pinions mounted on the out put shaft and each meshes with slewing ring gear. Here pinions are fixed and bull gear on slewing ring rotates. On each gear box a DC bush less motor is mounted which is coupled with the input shaft of gear box and provides the driving torque.

7.2 OPERATING FEATURES

The azimuth drive rotates the dish about azimuth axis through ± 270^0. The tracking operation about the azimuth axis is done up to a wind velocity of 40 Kmph. while for wind speed above 40 Kmph and upto 80 Kmph wind speed antenna can be rotated in slew mode. The drive system is locked by brakes on drive motors.

7.3 TRACKING & SLEWING MODE

During the tracking mode one of the pinions drives the bull gear while other opposes the rotation by a bucking torque equal to 10 –30% of drive torque.

During slewing mode both the pinions drive the bull gear in the same direction. It is assumed that load sharing by both pinions are equal.

7.4 AZIMUTH BEARING ANALYSIS

T0he azimuth bearing is designed to support vertical and horizontal loads on the cylindrical mount which are caused by wind load and self weight etc.

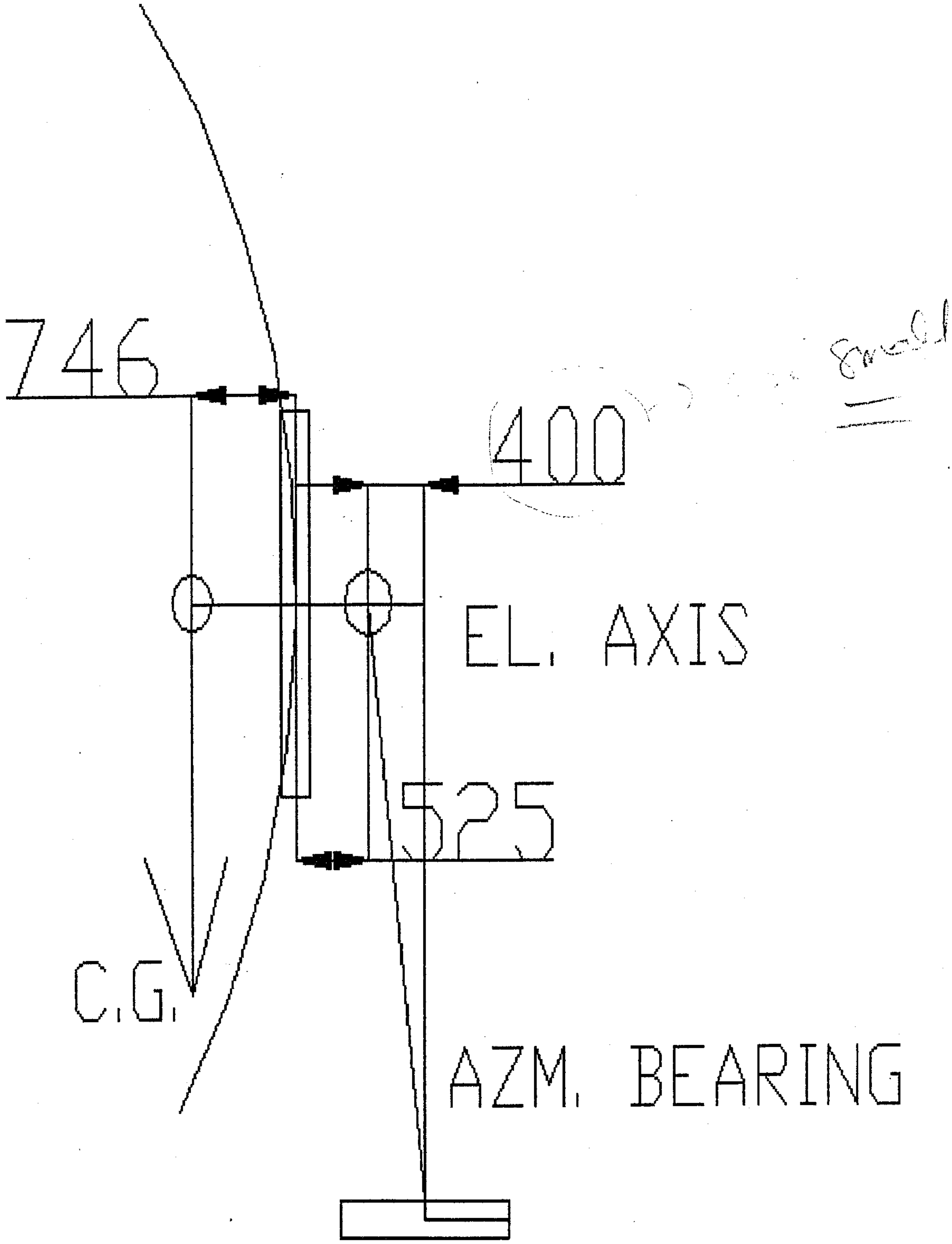


Fig No 3...

7.4.1 THE LOAD ON AZIMUTH BEARING DUE TO WIND

The loads on azimuth bearing such as moment, thrust and radial force due to wind are calculated for 150 Kmph wind speed and presented in the table 8

7.4.2 LOAD ON AZIMUTH BEARING DUE TO WIND

Wind speed 150 Kmph

Table 8

EL	Wind Angle ψ	Thrust T Ton	Radial R Ton	Moment Tm
0	0	0.00	10.43	23.9
	90	-1.25	4.42	24.68
	180	0.00	12.36	-28.42
15	0	1.81	10.21	28.65
	90	-1.25	4.42	26.14
	180	-0.91	-12.36	-25.23
30	0	3.29	9.53	24.34
	90	-1.25	4.42	27.49
	180	-1.81	-10.55	-16.44
45	0	3.52	8.96	19.58
	90	-1.25	4.42	28.65
	180	-2.72	-8.96	-6.88
60	0	2.15	6.69	8.22
	90	-1.25	4.42	29.54
	180	-2.72	-8.05	-5,6
75	0	0.00	5.33	35.77
	90	-1.25	4.42	30.10
	180	-2.15	-6.33	-0.55
90	0			
	90	-1.25	4.42	30.30
	180			

Load on Azimuth Bearing due to Self Weigh

Self Weight of main Reflector with Cradle Structure (Ws) = 4.5 t
Self Weight of Yoke structure with Counter Weight (Wy)= 6.5 t
(Wz) =11 t

From fig no …

$\gamma = 0$
β=80 deg
l1=1271mm
l2=2300 mm

θ EL	γ	My tm	Wz ton
0	0	-4.996	-11t
15	15	-4.880	
30	30	-4.5	
45	45	-3.980	
60	60	-3.104	
75	75	-2.184	
90	90	-1.197	

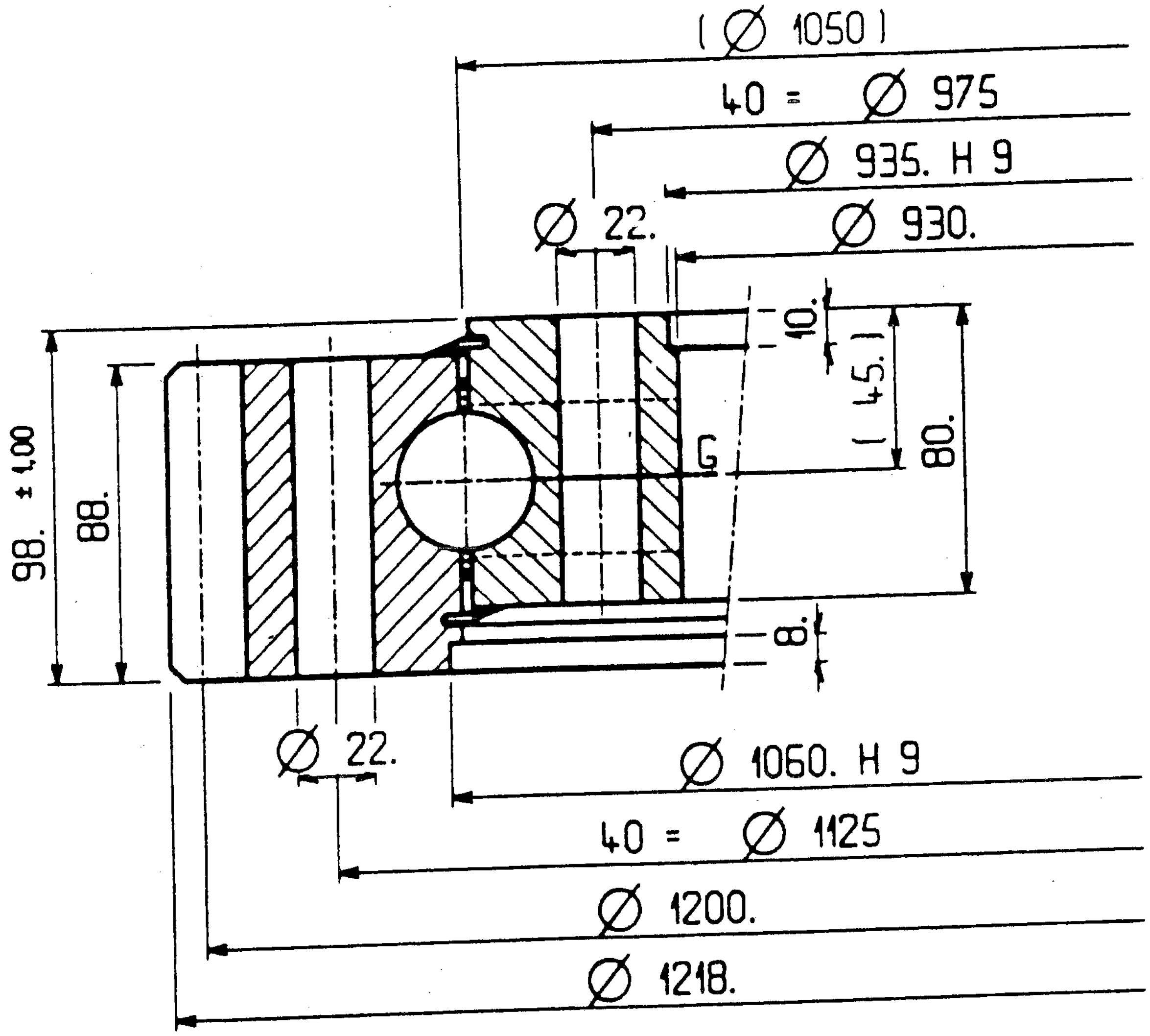
Table 11

7.4.3 Combined effect on wind load and self weight on azimuth bearing for 150 Kmph

Table 12

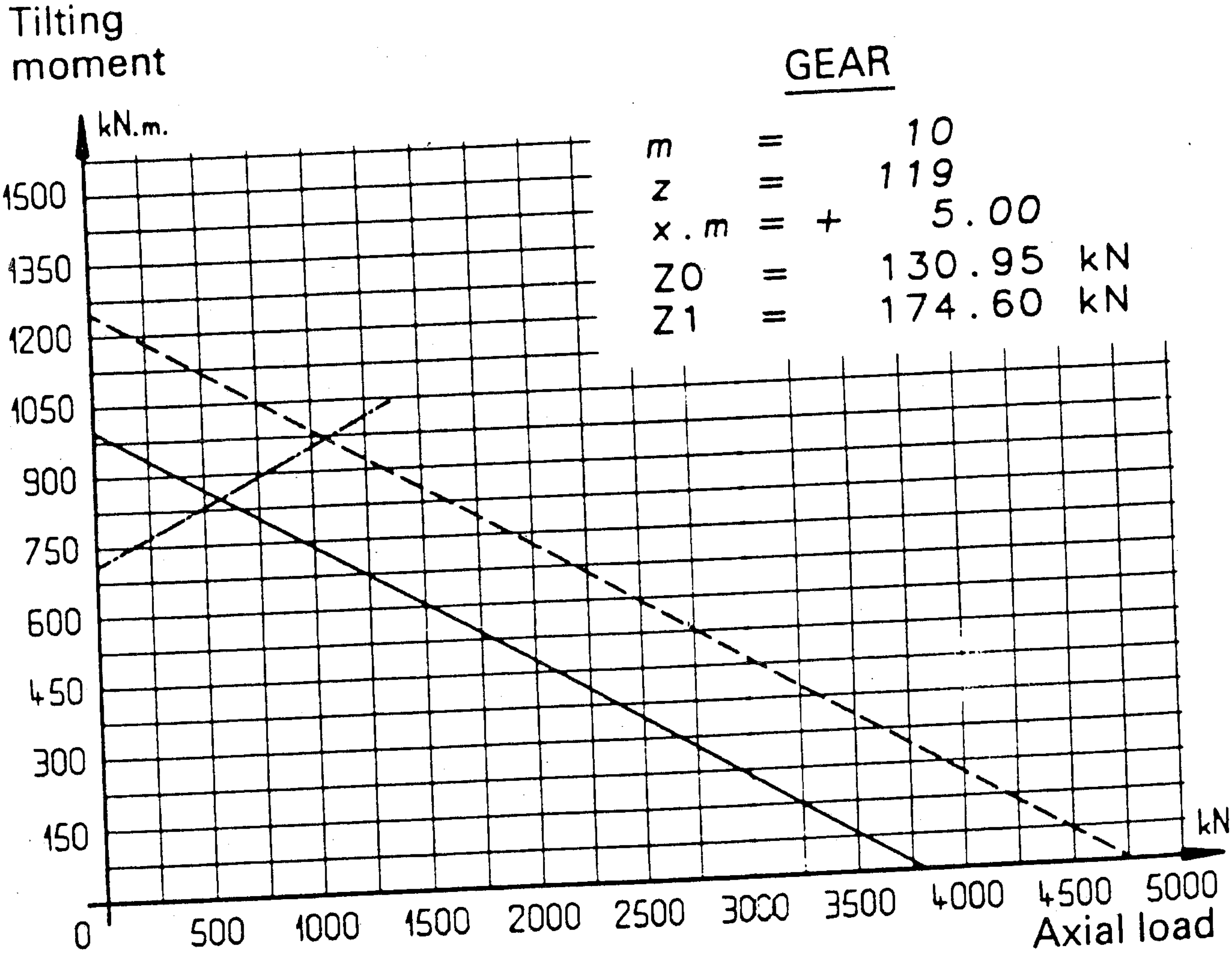
θEL	Wind Angle ψ	Thrust T Ton	Radial R Ton	Moment Tm
0	0	-11.0	10.43	18.91
	90	-12.25	4.42	19.69
	180	-11.00	-12.36	-23.43
15	0	-9.19	10.21	23.77
	90	-12.25	4.42	21.26
	180	-11.91	-12.36	30.11
30	0	-7.71	9.53	5.03
	90	-12.25	4.42	22.99
	180	-12.81	-10.55	-20.94
45	0	-7.48	8.96	15.60
	90	-12.25	4.42	24.67
	180	-13.72	-8.96	-12.94
60	0	-8.85	6.69	5.11
	90	-12.25	4.42	26.43
	180	-13.72	-8.05	-8.70
75	0	-11.00	5.33	32.90
	90	-12.25	4.42	27.91
	180	-13.15	-6.33	-2.73
90	0			
	90	-12.25	4.42	29.10
	180			
		Thrust		
	Max.	–13.72	-12.36	32.9

Azimuth bearing no.01, 1050, 00ZZ00 is selected on the above max. forces and moment



Rollix slewing ring 01, 1050, 00 22 00.

Fig 4



Axial load → 375 kn } Maxi.
tilting moment → 900 kn

Fig 5

7.5 LOAD ON AZIMUTH DRIVE

Maximum yawing moment	@ 100 Kmph	= 6.2 tm
	@ 80 Kmph	= 3.968 tm
	@ 50 Kmph	= 1.55 tm

The moment wind speed crosses 40 Kmph, antenna will be stowed to safe position for azimuth axis will be driven by both the pinions which will aid each other. Since both pinions are sharing the max. load thus they will be loaded @ 50% of max. capacity. Hence, for drive torque calculations wind speed of 50 Kmph is considered.

Maximum yawing moment @ 50 Kmph	= 1.55 + t_m	= 1.55×10^4 N_m
Maximum drag force @ 50 Kmph	= 0.473 +	= 0.473×10^4 N
Moment due to drag force @ 50 Kmph	= 0.4375 +m	= 0.4376×10^4 N_m
Total torque @ 50 Kmph	= 1989 Kgm	= 19890 N_m
Pitch radius of slewing ring	= 0.6m	
Tangential force @ gear	= 3315 Kgf	= 53150 N

Slew speed = 40° / min. = 40/360 = 1/9 rpm = 0.11111 rpm

Ratio of bull gear to pinion = 119/18 = 6.61111

Total gear ratio for input motor rpm of 1500 = 1500/0.11111 = 13500

7.5.1 REQUIRED GEAR BOX RATIO

	= 13500/6.6111	= 2042
Output torque @ azimuth axis	= 1989 / 6.61111 x η	= 376 Kgm = 3760 Nm
	= Output torque of Az gear box	

7.5.2 BASED ON ABOVE CALCULATIONS FOLLOWING GEAR BOX HAS BEEN SELECTED.

4 stage planetary gear box with ratio of 2159 nearest ratio or more is required with out put torque of 400 kgm or 4000

Balador brush less DC motor is used to drive the gear box

Continuous stall torque	12
Peak torque	36
Power	1.9 @ 1500 rpm

Input to gear box = 4000 / 2154 = 2.65 ≈ 3 Nm

Motor output = 4000 /2154 x η

7.6 AZIMUTH PINION & SLEWING RING GEAR

7.7.1 Azimuth Pinion

Analysis of Azimuth Pinion

Parameters	**Pinion**	**Gear on Slewing ring**
Module mm	10	10
No. of Teeth	$Z_1 = 18$	$Z_2 = 119$
Pressure Angle	20^0	20^0
P.C.D mm	180	1190
Face Width	95	88
Profile Shaft factor	0.5	0.5
Material		

7.7.2 Due to profile correction in pressure the operating pressure angle changes to ϕ_r

$$\cos \phi_r = \frac{a \cos \phi}{a^1}$$

Where

a = Ref. Centre Distance

a^1 = Working Centre Dist.

ϕ = Std pressure angle = 20

$a = \frac{(Z_1 + Z_2)}{2} \times m = Z_m \times m$

$\frac{(18 + 119)}{2} \times 10 = 68.5810$

= 685 mm

$$a^1 = m\,(Z_m + y)$$

$$= 10\,(68.5 + 0.9534)$$

y = Centre Distance Modification co-efficient

$= \frac{y}{Z_m} \times Z_m$

7.6.1 Azimuth Pinion

Parameters	Pinion	Gear on Slewing ring
Module mm	10	10
No. of Teeth	$Z_1 = 18$	$Z_2 = 119$
Pressure Angle	20^0	20^0
P.C.D mm	180	1190
Face Width	95	88
Profile Shift factor	0.5	0.5
Material	40Ni&Cr1Mo28 IS 1570 Induction Hardened Flame Hardened to 570 BHM 15 Ni X_r 1Mo15 Case Hardened	

7.6.2 Due to profile correction in pressure the operating pressure angle changes to ϕ_r

$$\cos \phi_r = \frac{a \cos \phi}{a^1}$$

Where

a = Ref. Centre Distance

a^1 = Working Centre Dist.

ϕ = Std pressure angle = 20

$$a = \frac{(Z_1 + Z_2)}{2} \times m = Z_m \times m$$

$$\frac{(18 + 119)}{2} \times 10 = 68.5810$$

$= 685$ mm

y = Centre Distance Modification co-efficient

$$= y \times \frac{Z_m}{Z_m}$$

$= 0.013917618 \times 68.5$

$= 0.953356833$

$$a^1 = m(Z_m + y)$$

$$= 10 (68.5 + 0.9534)$$

$$= 694.534$$

$$\cos \phi_r = \frac{685 \times \cos 20}{694.534.1}$$

ϕ_r = 0.926793857
= 22.05966091
= $22^0\ 3^1\ 35^{11}$

Operating center Distance

a^1 = m (Zm + y)
= 694.534 mm

Recommended nominal back lash ref. Machinery Hand Book Page 1870, AGMA recommendations for center Distance 27.34 inches. (694.5 mm and diametral pitch $\frac{25.4}{m} = \frac{25.4}{10} = 2.54$

a back lash of 0.03 to 0.04 inches (0.762 to 1.016 mm) is recommended.

7.7.2.1 For further calculations the operating center distance = 694.534

Operating Pinion Radius = d /2

$$\frac{d}{2}\left(1+\frac{Z_2}{Z_1}\right) = 694.534$$

$$\frac{d}{2}\left(1+\frac{119}{18}\right) = 694.534$$

$\therefore$ d = 182.505 mm

Radius = 91.253 mm

Operating Pressure angle

ϕ_r = $22^0\ 3^1\ 35^{11}$

Operating Gear radius $\frac{182.505}{2} \times \frac{119}{18}$ = 603.281 mm

Pinion OD = da_1 = m [Z_1 + 2 (1 - x_2 + y)]
= 10 [18 + 2 (1 – 0.5 + 0.9534)]
= 209.068 mm

7.6.3 Geometry factor for Pitting resistance for Azimuth Gear & Pinion Pair as per AGMA 20001 <88 standard ref Machinery Hand Book P1 1834.

Note : Certain variables are made dimensionless by dividing by module m_n
Calculation for Pitting Resistance basic Geometry factor
(Nomenclature as per Machinery HB)

a) Gear Ratio $m_G = \frac{n_2}{n_1} = \frac{119}{18} = 6.611111$ 1

Where n_1 = Pinion no of Teeth
n_2 = gear no. of teeth

b) Standard (reference pinion) pitch radius $= R_1 = \frac{n_1}{2} = \frac{18}{2} = 9$ 2

c) Std (ref) Gear pitch Radius $R_2 = R_1 \times m_G$
$= 9 \times 6.6111$
$= 59.5$ 3

d) Std.Transverse Pr.Angle $\phi = \tan^{-1} (\tan \phi_n / \cos\Psi)$
Where
ϕ_n = Std normal Pr angle = $\tan^{-1} (\tan 20)$
$= 20^0$
Ψ = helix angle = 0 4

e) Pinion Base Radio $= Rb_1 = R_1 \cos \phi$
$= 9 \cos 20$
$= 8.4572$ 5

f) Gear Base Radius $= Rb_2 = Rb_1 \times m_G$
$= 55.9117$ 6

g) Operating Transverse Pressure angle $\phi_r = \cos^{-1}\left(\frac{Rb_2 + Rb_1}{C_r}\right)$

C_r = Operating Centre Dist / unit module
$= 694.534 / 10$

$= 69.4534$

$\phi_r \quad = 22.0566$ 7

h) Transverse Base Pitch P_b $\quad = (2\pi Rb_1)/n_1$

$= 2\pi \times 8.4572 / 18$

$= 2.9521$ 8

i) Normal Base Pitch $\quad = P_N \quad = \pi Cos\ \phi n$

ϕn = Std normal pr angle

$= 2.9521$ 9

j) Base helix angle $\psi b - Cos^{-1} \frac{P_N}{P_b} = 0$ (Spur Gear)......................................10

k) Calculation of distances C_1 to C_6 along the length of mesh of gears

$C_6 = C_r\ Sin\ \phi_r$ $\qquad$ C_r = Operating Center Distance per unit module

$= 69.4534 \times Sin22.05966$ $\qquad$ $= 69.4534$

ϕ_r = Operating Transverse pressure angle

$= 22.05966$

$= 26.08474$ 11

l) $C_1 \quad = \pm\ [C_6 - (Ro^2_2 - Rb^2_2)^{0.5}]$

$= +\ [26.08474 - \sqrt{60.953352 - 55.91172}]$

$= 1.81145$ 12

$R0_2$ = Add.radius of gear / Unit module

$\frac{(1219.067)}{2} / 10$

$= 60.95335$

m) $C_3 = \dfrac{C_6}{m_G + 1}$ $\qquad = \dfrac{26.08474}{6.61111 + 1}$

$= 3.42719$ 13

n) $C_4 = C_1 + P_b$ $\qquad = 1.81145 + 2.9521$

$= 4.76355$ 14

o) $C_5 = \sqrt{R_{01}^2 - R_{b1}^2}$

$= \sqrt{10.45335^2 - 8.4572^2}$

R_{01} = Addm Radius of pinion / unit modules

$= \frac{(209.067)}{2} / 10$

$= 10.45335$

$= 6.14396$ 15

p) $C_2 = C_5 - P_b$

$= 6.14396 - 2.9521$

$= 3.19186$ 16

q) Active Length of Contact = Z

$Z = C_5 - C_1$

$= 6.14396 - 1.81145$

$= 4.3325$ 17

r) Transverse contact ratio = m_p

$m_p = \frac{Z}{P_b} = \frac{4.3325}{2.9521} = 1.4676$ 18

s) For Spur Gears with $m_p < 2$

Minimum Length of contact = L_{min}

L_{min} = F = Effective face width / unit module

$= 88 / 10$

$L_{min} = F = 8.8$ 19

t) Load Sharing Ratio

$m_N = \frac{F}{L_{min}}$

$m_N = 1$ for Spur gears 20

u) Pitting Resistance Geometry Factor = I

$$I = \frac{[\cos\phi_r . C_\Psi^2]}{\left(\frac{1}{\rho_1} + \frac{1}{\rho_2}\right) dm_N}$$

C_Ψ = Helical overlap factor
= 1 for spur gears

ϕ_r = Operation of transverse Pr. Angle ϕ
= 22.05966

d = Pinion operating pitch Dia

$$= \frac{2\,Cr}{m_G + 1}$$

$$= \frac{2 \times 69.4534}{6.61111 + 1}$$

= 18.2505

ρ_1 & ρ_2 Radius of curvature of Pinion & gear profiles

ρ_1 = C_1 = 1.81145

ρ_2 = $C_2 - \rho_1$
= 3.19186 - 1.81145
= 1.38041

$$I = \frac{\cos 22.05966 \times 1^2}{\left(\frac{1}{1.81145} + \frac{1}{1.38041}\right) \times 18.2505 \times 1}$$

= 0.039783

I = 0.039783..21

7.7 PITTING RESISTANCE CALCULATIONS

$$Sc = Cp \left(\frac{W + Ca}{Cv} \times \frac{Cs}{dF} \times \frac{Cm\,C_f}{D} \right)^{1/2}$$

Sc = contact stress No. Mp_a

Where C_p = Elastic co-efficient $(Mp_a)^{1/2}$

$$= \sqrt{\frac{1}{\pi \left(\frac{1 - \mu P^2}{Ep} + \frac{1 - \mu g^2}{Eg} \right)}}$$

= 191.68 $(M\,Pa)^{1/2}$

μ_p, μ_g = 0.3 Poisons ratio of gear & pinion

Ep = Eg
= Youngs Modulus
= 2.1×10^5 MP_a

C_a = Over load factor

= 1.25 for uniform Power source and Moderate Shock Ref : Dudley "Practical Gear Design"

C_v = Dynamic Factor = 1 …….. Fig 3 Page 1844 Machinery Hand Book

As the pitch line velocity is low

V_t = Pitch line Velocity

$$= \frac{\pi \times n_p \times d}{60{,}000}$$

$$= \frac{\dfrac{(\pi \times \underline{40} \times 6.6111) \times 180}{360}}{60{,}000}$$

= 0.006923 m / sec

$\therefore C_v$ = 1

C_s = Size factor =1

D = Operating Pinion P_{cd} = 182.505 mm

F = Face width = 88 mm

C_m = Load distribution factor

Where transverse load distribution factor = 1

$C_m = C_{mf}$ = Km

$$C_{mf} = 1 + \frac{G \times e_t \times F}{2 \times Wt}$$

G = Tooth stiffness Constant

= 1.0 – 1.4 x 10^4 MP_a Select lower value

= 1 x 10^4

e_t = total Load mismatch between mating teeth in loaded condition

= 0.1

$$C_m = 1 + \frac{1 \times 10^4 \times 0.1 \times 8^8}{2 \times Wt}$$

$$= 1 + \frac{44000}{157274} = 1.27976$$

$$W_t = \frac{Torque}{Radius}$$

Wt_{20} = 6291 N

$$Wt_{50} = \frac{23400}{1190/2} = 39328 \text{ N}$$

$$Wt_{80} = \frac{59890}{0.595} = 100655 \text{ N}$$

Wt_{100} = 157274 N

7.7.1 Torque and Tangential Loads Summary

Wind Speed Kmph	Torque Kgm	Tangential Load Wt, N
20	374	6291
50	2340	39328
80	5989	100655
100	9358	157274

7.7.2 Contact Stress No Sc

$$Sc = Cp\left(\frac{W_t \times C_a}{Cv} \times \frac{Cs}{dF} \times \frac{C_m.C_f}{I}\right)^{1/2}$$

$$= 191.0\left(\frac{W_t \times 1.25}{1} \times \frac{1}{182.505 \times 88} \times \frac{1.27976 \times 1}{0.039783}\right)^{1/2}$$

$$= 9.5571\,[\,W_t\,]^{1/2}$$

	Wind speed Kmph	Sc
7.7.2.1	20	760.5
7.7.2.2	50	1901.6
7.7.2.3	80	3042
7.7.2.4	100	3802

7.7.3 Sac = 400 x BHN – 16,000

The slewing ring gear is hardened to 55 Rc ≈ 600 BHN

∴ Sac = 400 x 600 – 10,000

= 2,30,000 Psi Ref. Valance & Doughty

= 16197 Kgf / cm^2

= 1620 MP_a

$$Sc \leq Sac \quad \frac{C_L \; C_H}{C_T \; C_R} = S^t ac.$$

C_L = Lite factor ref " Dudley Practical gear design.

= 1.05 – 20 – 40 Kmph

= 1.4 – for 50 Kmph

= 1.45 – for 80 Kmph

= 1.5 – for 100 Kmph

C_H = Hardness Ratio factor
= 1.02 ref Machinery HB Page 1850 for Hardness ratio of pinion to gear
= 1.09 – 1.2
C_T = Temp factor = 1
C_R = Reliability factor = 1 upto 50 Kmph for 1 failure in 100
= 0.80 for above 80 Kmph

7.7.4 Table for S^1 ac

	Wind Vel Kmph	Sc	S^1ac
4.21	20	760.5	1735
4.22	50	1901.6	2313
4.23	80	3042	2995
4.24	100	3802	3098

We can operate the Dish upto 80kmph and should not operate beyond 80 Kmph from pitting resistance point of view. Since beyond 50 Kmph two pinions share the load equally. Loads & stress will be halved & sac < S^1ac for 80 & 100 Kmph.

7.7.5 Applying Minors Rule

N_i = Number of permissible cycles
n_i = no. of actual cycle

	Sc	Sac	N_1	ni
20 Kmph	760.5	1735	10^7	4.2×10^6
50 Kmph	1901.5	2313	10^7	1.5×10^6
80 Kmph	3092	2995	10^5	0.03×10^6
100 Kmph	3802	......	......	....

$$\Sigma \frac{n_1}{N_1} = 0.42 + 0.15 + 0.3 = 0.87$$

CONCLUSION

Since $\Sigma \frac{n_1}{N_1} < 1$ the Pinion meets the life criteria for petting .

7.8 BENDING STRESS CALCULATIONS

Following are Bending stress calculations

Wind Speed Kmph	Torque N_m	Tangential force N
20	3740	6291
50	23400	39328
80	59690	100655
100	93580	157274

7.8.1 Total moment inertia about Azimuth Axis 37984 Kgm^2

$$\text{Ang Vel} = \frac{2\pi n}{60} = \frac{2\pi \times 40/360}{60} = 0.01164 \text{ Rad/sec} = \omega$$

$$\text{Ang Accln} = \alpha = \frac{\omega}{T} = \frac{0.61164}{10} = 1.1636 \times 10^{-3} \text{ Rad / Sec}^2$$

7.8.2 Inertia torque $= I\alpha = 37984 \times 1.1636 \times 10^{-3}$

$= 44.196$ Nm

7.8.3 Friction Torque = Total Axial Load x Co-eff of friction touch x radius of slewing ring bearing

= 100 KN x 0.01 x 1.05 / 2

= 0.525 KNm 1 x2

= 525 Nm

7.8.4 Max Torque acting on the slewing ring gear

Wind Speed Kmph	Torque Nm	Tangential force @ gear PCD(N) F_t
20	4309	7182
50	23969	39948
80	60259	160432
100	94149	156915

Torque is total of wind, Inertia & friction Torques

7.8.5 Pitch line Velocity at Max Speed

$$= \frac{\pi \eta d}{60{,}000} = 0.006923$$

7.8.6 Velocity Factor $C_v = \frac{3.05}{3.05 + 0.006923}$

$$= 0.9977$$

7.8.7 Form Factor

$$= f = 0.154 - \frac{0.912}{Z_1}$$

$$= 0.154 - \frac{0.912}{18}$$

$$= 0.103333$$

7.9 Bending Stress on this Pinion

$$f_b = \frac{F_t}{\pi \times b \times m \times e_v \times f} = \frac{F_t}{\pi \times 88 \times 10 \times 0.9977 \times 0.1033}$$

$$= \frac{F_t}{285}$$

7.9.1 $f_{b20} = 25.2\ N/mm^2$
7.9.2 $f_{b50} = 140.2\ N/mm^2$
7.9.3 $f_{b80} = 352.4\ N/mm^2$
7.9.4 $f_{b100} = 550.6\ N/mm^2$

Allowable stress as = 380 N/mm^2 in C45, & Other alloy steels as per IS I570

Note : Loads at Wind speeds 80 to 100 Kmph are equally shared by two pinions and therefore Loads and stresses are halved.

i.e. $f_b\ 80 = \frac{352.4}{2} = 176.4\ \text{N / mm}^2$

& $f_b\ 100 = \frac{558.2}{2} = 275.3\ \text{N / mm}^2$

Pinions are safe and has a factor of safety = 2.1 @ 80 Kmph and a safety factor of 1.38 @ 100 Kmph

7.10 Surface endurance stress calculations

Since the velocity of Rotation is very low Dynamic loud can be taken to be equal to static loads and wear loads.

$\therefore Fw = Ft$

7.10.1	Fw_{20}	=	7182 N
7.10.2	Fw_{50}	=	39948 N
7.10.3	Fw_{80}	=	100432 N
7.10.4	Fw_{100}	=	156915 N

7.10.5 Surface endurance stress = f_e

$$f_e = \sqrt{\frac{Fw \times 1.4 \times E}{b \times q \times m \times z \times 2 \sin\phi_r}}$$

$$q = \frac{2 \times Z_2}{Z_1 + Z_2} = \frac{2 \times 119}{18 + 119}$$

$$= 1.7372$$

$$f_e = \sqrt{\frac{Fw \times 1.4 \times 210000}{88 \times 1.7372 \times 10 \times 18 \times 2 \times \sin 22.05966091}}$$

$$f_e = 3.7715 \sqrt{Fw}$$

7.10.5.1 $f_{20} = 3.7715 \sqrt{7182} = 319.6\ N / mm^2$

$f_{50} = 3.7715 \sqrt{39948} = 753.8\ N / mm^2$

$f_{80} = 3.7715 \sqrt{100432} = 1195.4\ N / mm^2$

Actual $fe_{80} = \frac{1195}{2} = 595\ N/mm^2$ /gear pinion

$f_{100} = 3.7715 \sqrt{159116} = 1504\ N / mm^2$

Actual $Fe_{100} = \frac{1494}{2} = 747$ N/mm2 /gear pinion

Note : Max endurance Stress = f_{50} = 753.8

$\approx 754\ N / mm^2$

Max Allowable Surface

Endurance Stress @ 400 BHN $\approx$ 43 H R_c gear is equal to 150000 psi = 1056.3 N / mm^2

$$\text{BHN Hardness required} = \frac{754 + 70}{2.75} = 300\ \text{BHN}$$

The Pinion therefore could be hardened to 300 - 400 BHN

Cradle

Cradle is important interfacing structure between the dish and yoke structure. Cradle structure is made up of structural channels and box sections. From the bottom of the cradle structure elevation bearing are supported at two points and bull gear is attached diametrically opposite points. This provides elevation motion for the dish by 0 – 90°. On top of the cradle structure Hub of the dish is rigidly fixed by bolting at places using plates. One plate is welded to Hub and other to the cradle.

Hub Modification

This is a close ring type structure with box section of 200mm x 200mmx8mm thickness and has 4.20μ OD and 3.8μ ID. Parallelism is maintained between top and bottom plate by machining the hub to 0.1mm.

On top portion every 15° taper (24 nos.) blocks are rigidly fixed by bolts. The taper blocks are used to clamp the radial spokes of the PPD. Quadripod structure to support the focus is mounted on radial spoke at orthogonal axis. This axis should correspond to elevation and bull gear axis. It is important to weld the plates at the bottom of the hub to maintain the orthogonality. The corresponding reference line on the top of hub to be transferred to the bottom of hub very accurately by suitably indexing, and maintain the orthogonality as said above. Then mounting plates at the bottom of the hub welded at placed. Welding to the done to minimise the distortions on hub by the process of staggered welding. If required giving a machining cut to get the levelled surface and parallelism.

On top of cradle 8 plates are welded at every 45° to each other. Suitable plates are mounted for elevation axis and bull gear mounting. Cradle is machined to achieve parallelism and orthogonality of the elevation and bull gear axis at bottom side. Brackets for elevation axis and mounting plate for bull gear axis are bolted by suitably aligning the two axis. This is done by several mechanical fixers to assist the assembly.

Cradle structure has been designed such that it is a self supporting structure and does not on the extreme loads to hub.

Cradle analysis has been done considering maximum wind load (drag) when dish at horizon at 150kmph. This is to make sure the structural stability of cradle. However, the situation should not arise. Since the dish will be drive to safe position as wind velocity exceeds 40kmph.

ANSYS

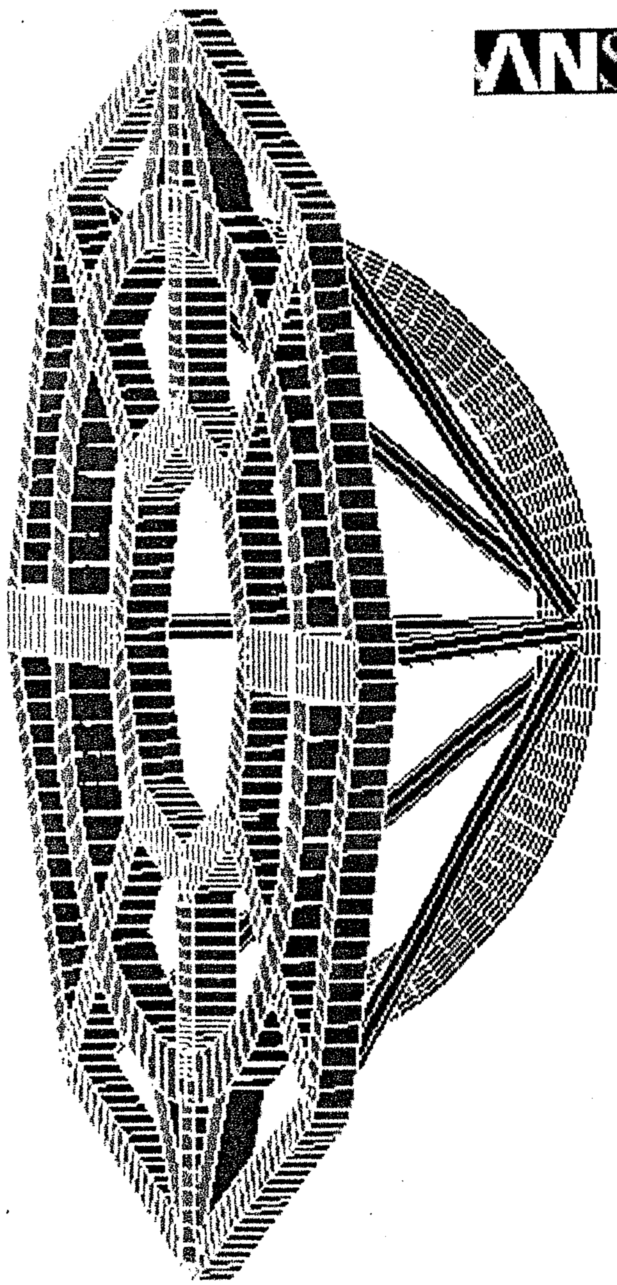
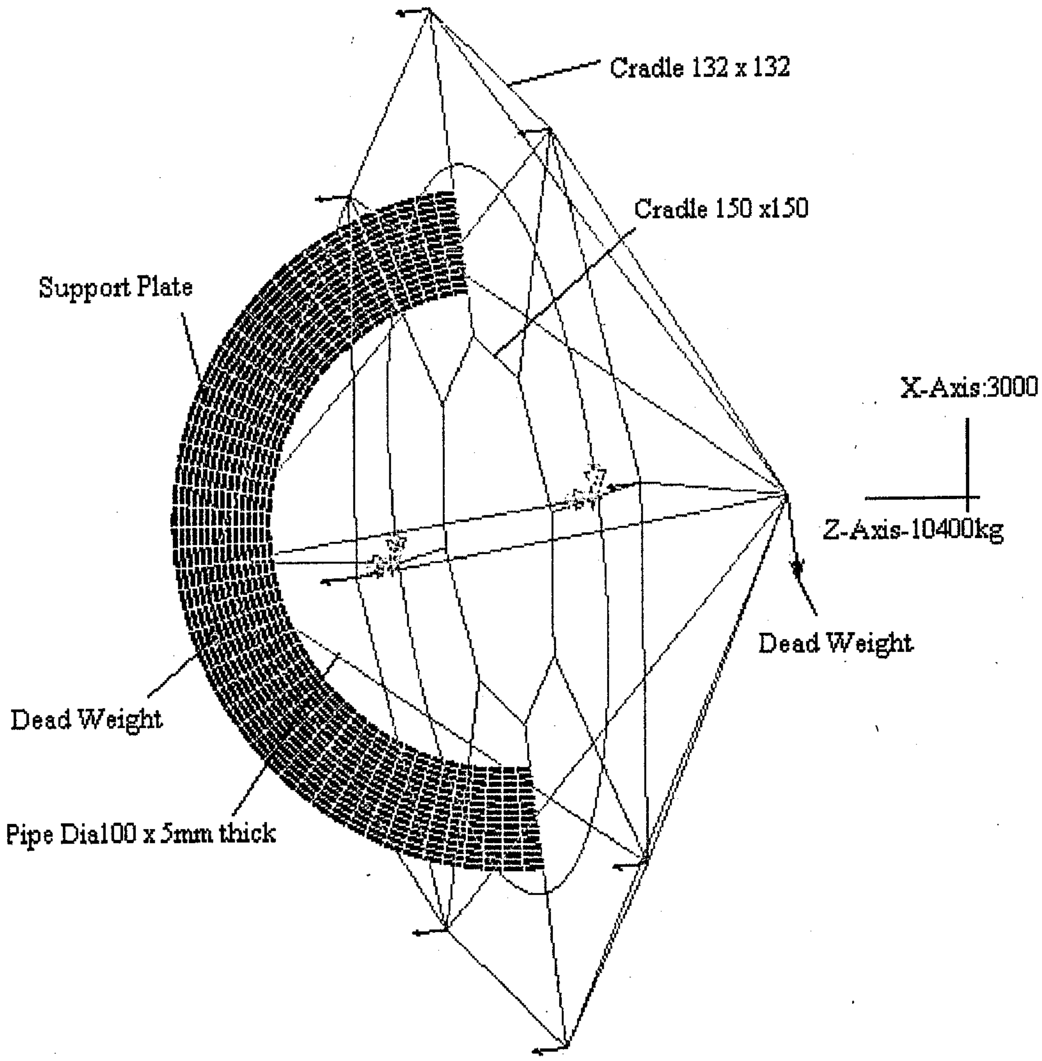


Fig 6

STRUCTURAL ANALYSIS FOR CRADLE



CRADLE MESH MODEL

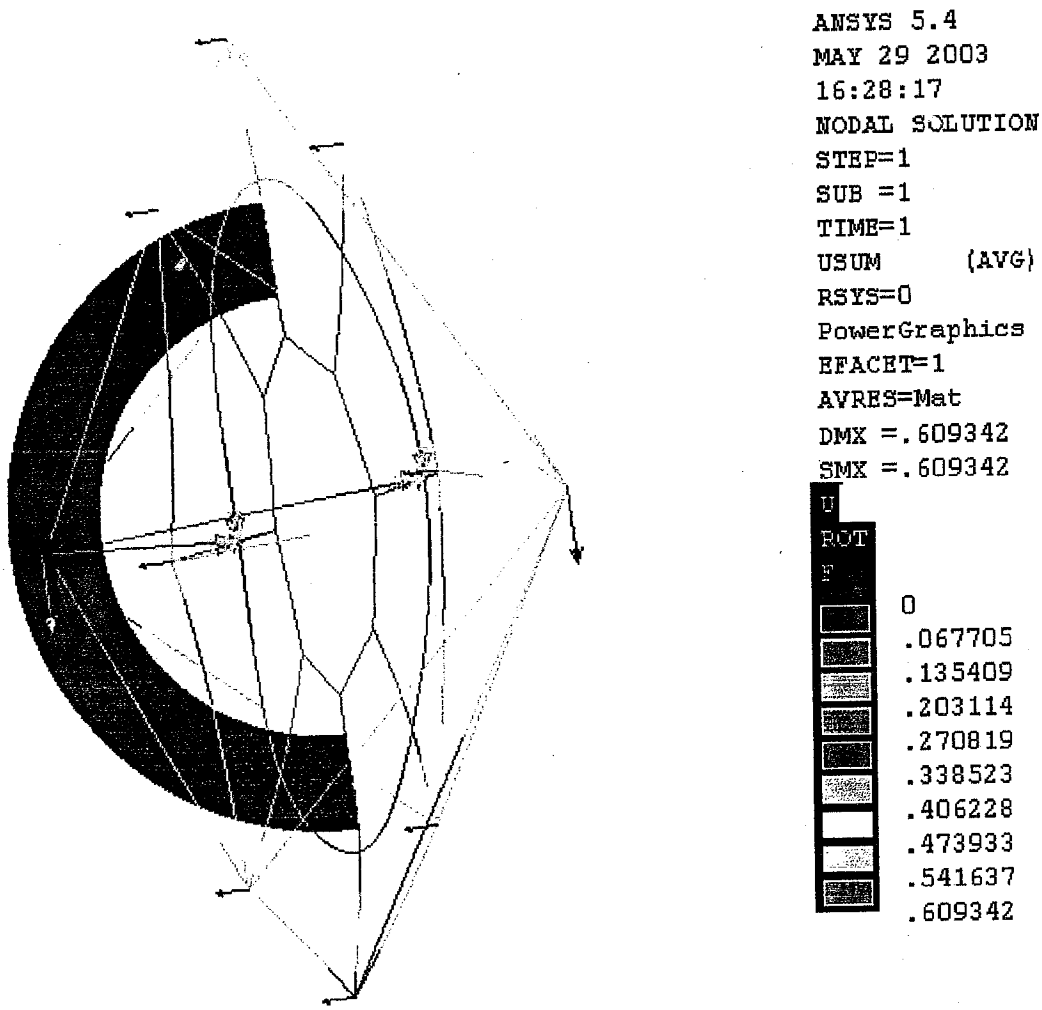
WEIGHT OF CRADLE: *1000 kg*

LOADING CONDITION: (REFER SHAFT MESH MODEL Fig.)

Z-AXIS: *10400 kg (150 kmph)*

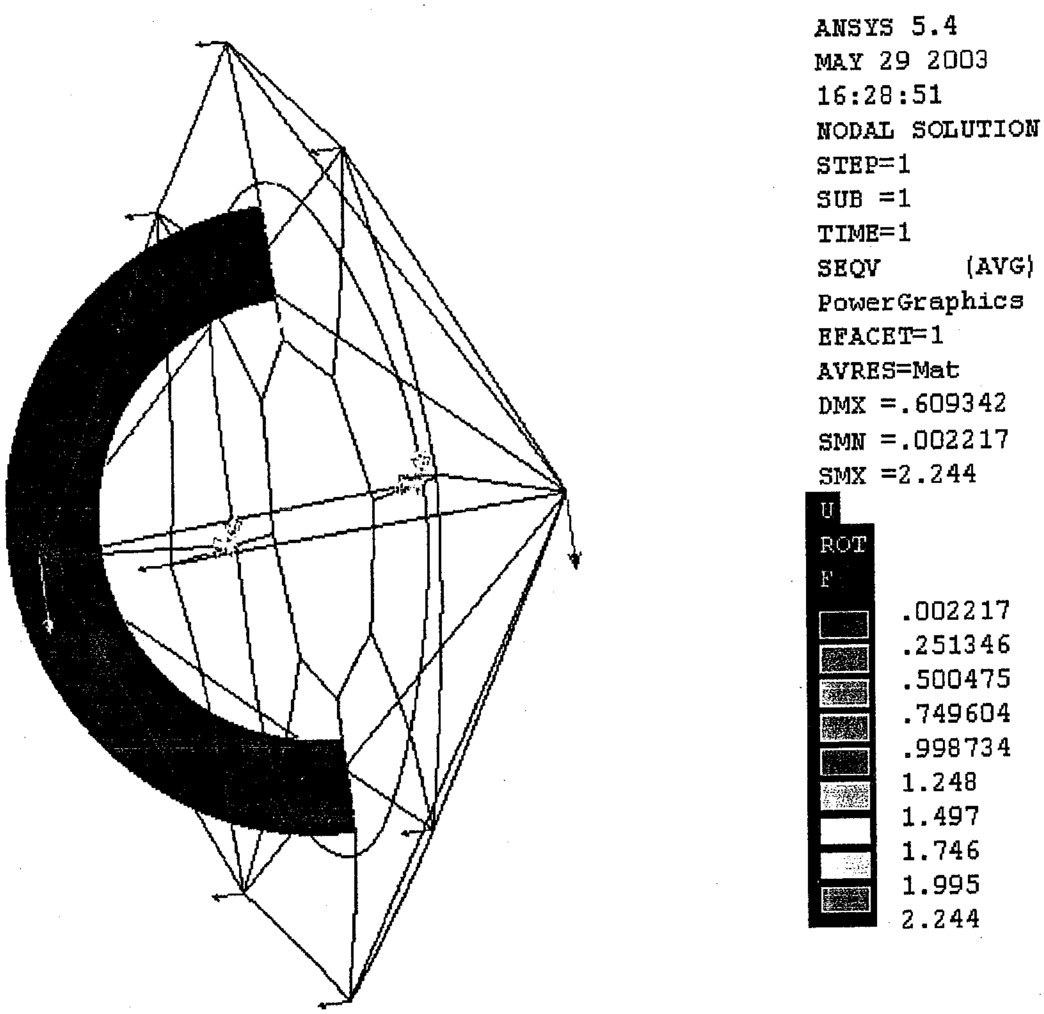
X-AXIS: *3000 + 3000 kg*

STRUCTURAL ANALYSIS FOR CRADLE



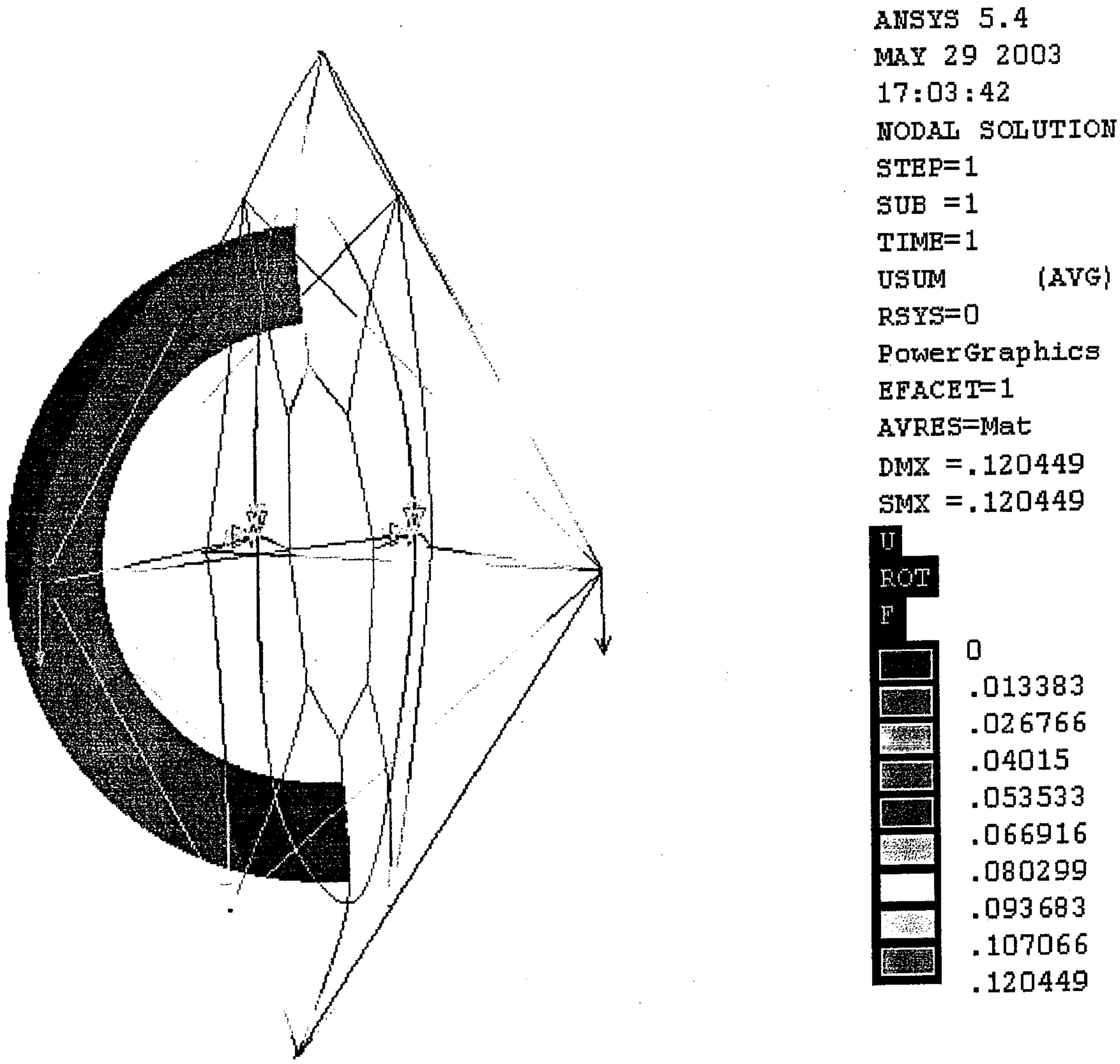
RESULTS: DEFLECTION - **(100 kmph)**

STRUCTURAL ANALYSIS FOR CRADLE



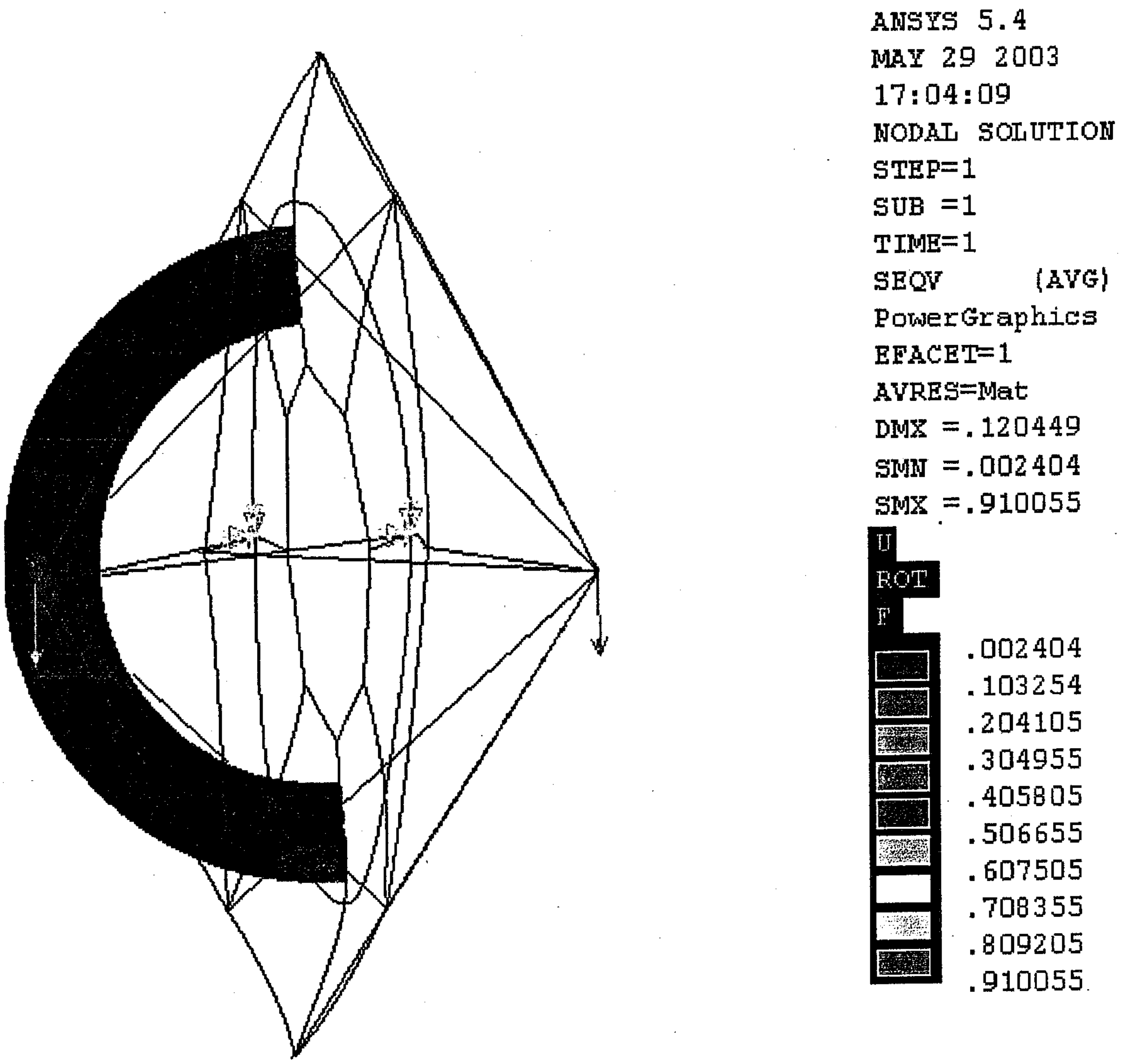
RESULTS: STRESSES - 2.2kg/mm^2**(100 kmph)**

STRUCTURAL ANALYSIS FOR CRADLE



RESULTS: DEFLECTION - *0.12 mm* (**50 kmph)**

STRUCTURAL ANALYSIS FOR CRADLE



RESULTS: STRESSES - 0.91kg/mm^2 (**50 kmph)**

Yoke

Yoke is a fork like structure mounted on azimuth bearing at the bottom, at the top of forks plumber blocks are mounted to accommodate the bearings for the elevation axis of the dish. The plumber block bearings and the bull gear are coaxial. The plumber block axis on the yoke top has an offset of 400mm with reference to azimuth slew ring bearing. Bearing on two arms of the yoke structure takes the dish forces and these are transferred through azimuth bearings to steel shell structure. Bearings used in the plumber blocks are self aligned type to facilitate ease of assembly. The yoke base is made rigid to minimise the effects of deviations due to different mechanical loading like mounting of gear boxes, working platforms, railing approach ladders, etc.

For yoke, analysis has been done taking critical XYZ forces on elevation bearing from table no.   at wind speed of 150kmph. These bearing loads are taken by yoke arms structure. The design has been carried out to minimise the deflections generated on yoke structure.

STRUCTURAL ANLYSIS OF YOKE:

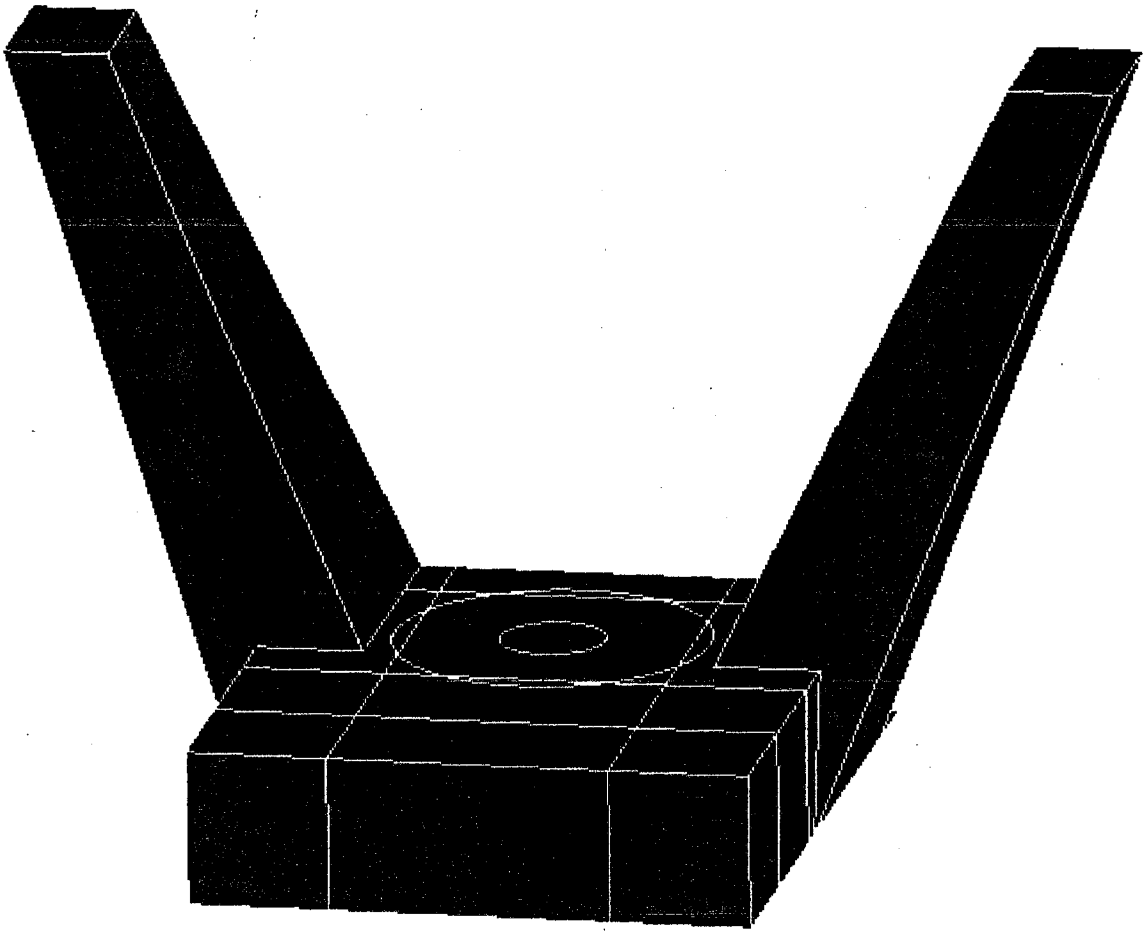


Fig. 1 *YOKE MODEL*

WEIGHT OF YOKE: 3200 kg

STRUCTURAL ANLYSIS OF YOKE:

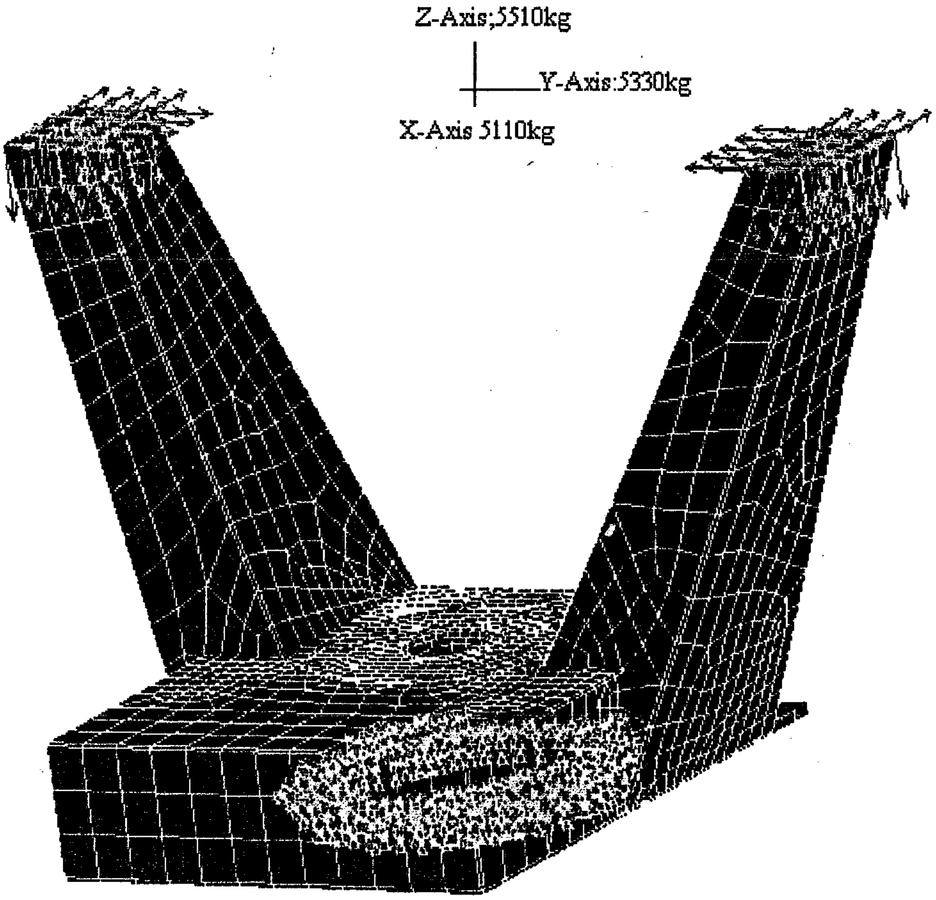


Fig. 2 ***YOKE MESH MODEL***

MATERIAL: St – 42

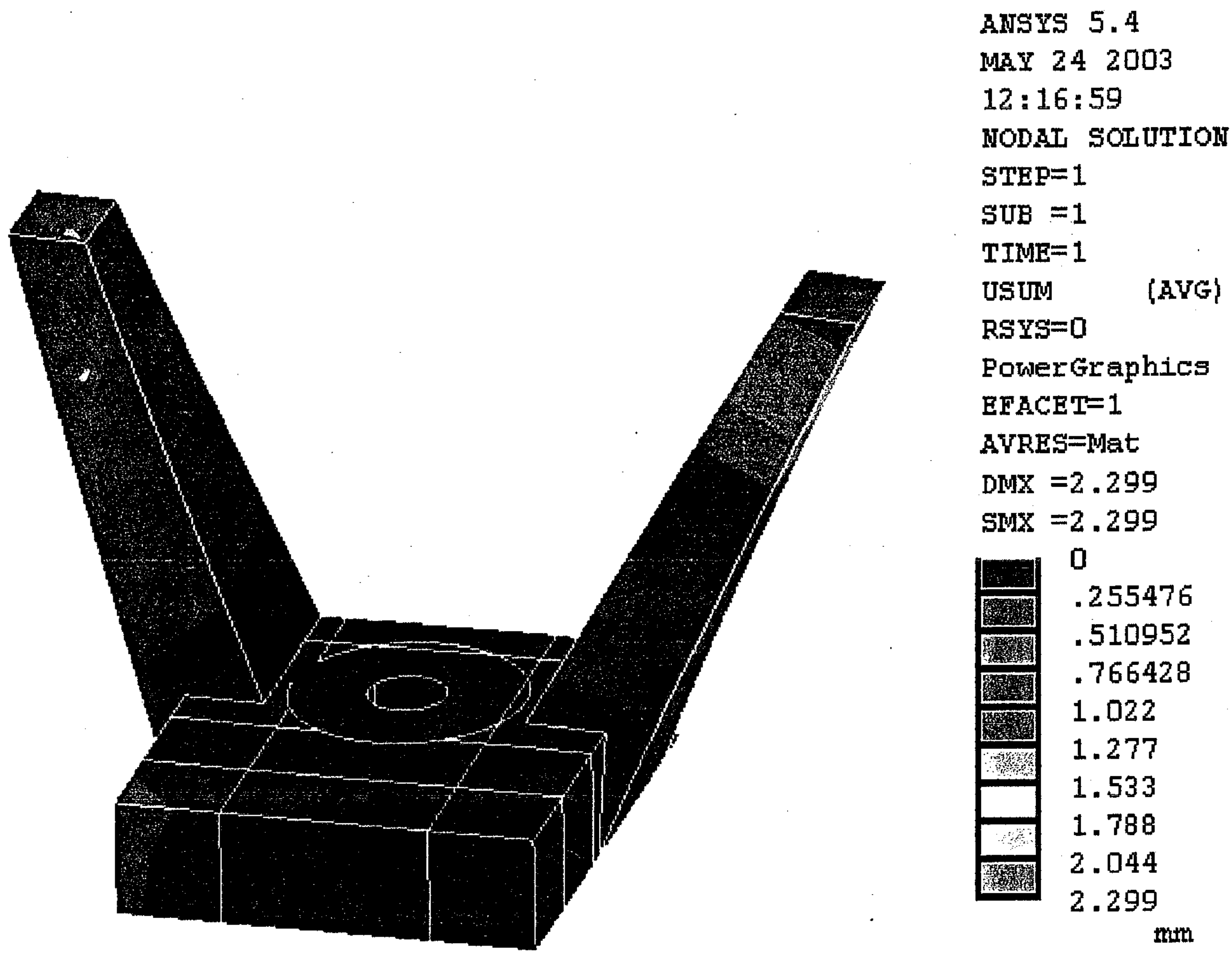
MATERIAL PROPERTY:

MODULUS OF ELASTICITY : 2.1E04 kg/mm^2
POISSONS RATIO : 0.3

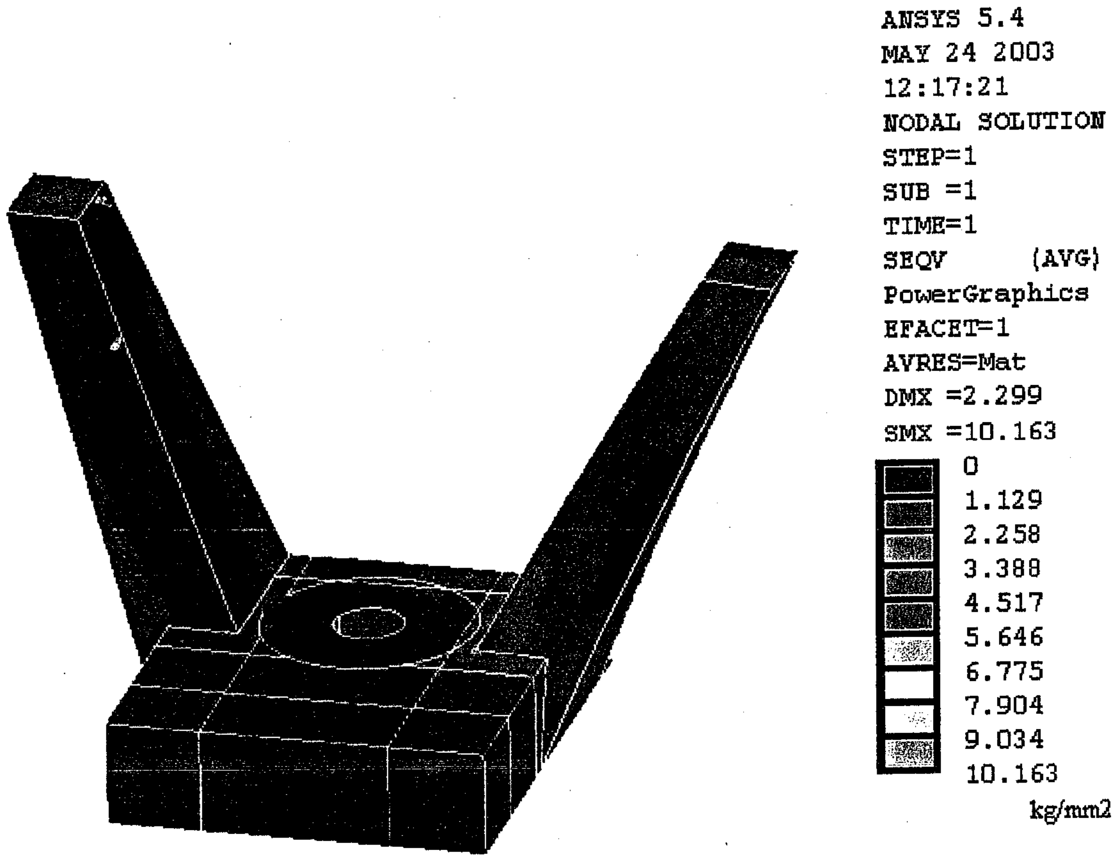
LOADING:

: 5510 kg/mm^2 (Z-AXIS)/Column
: 5330 kg/mm^2 (Y-AXIS)/ Column
: 5110 kg/mm^2 (X-AXIS)/ Column

STRUCTURAL ANLYSIS OF YOKE:



STRUCTURAL ANLYSIS OF YOKE:



Steel Shell Structure

This structure is interface between RRC conical base structure and the yoke. Base of the shell structure is bolted and levelled on to the RRC base structure with 24 bolts. On the top plate of shell structure, slewing ring bearings rack is bolted.

Two sets of pinions mounted on planetary gearbox at 180° apart drives the slewing ring. These Gearboxes are rigidly mounted on the outer walls of the shell structure.

The shell structure is well ribbed and rigidly constructed to take all the loads.

Steel Shell Structure

This Structure is interfaced between RCC Conical base Structure and the Yoke. Base of the Shell Structure is bolted and levelled on to the RCC base Structure with anchor bolts. On the Top of the Shell Structure Azimuth Slewing ring bearing inner race is bolted. This Structure also provides Space for the Power Cables, RF Cables to pass through. The Azimuth Bearing external gears are driven by Two plannetary gear boxes with Servo ~~boxes~~ motors. The gearbox with the Servo motors are rigidly on to the Steel Shell Structure at 180° appart Both are Coupled to the Azimuth Slewing Ring external gear via respective Pinions.

The Steel-Shell Structure is well ribbed and of rigid construction to take all the loads. ~~of~~ The Structure has been ~~ale~~ analysed for Dead Loads, moments, gear box reactions.

Finite Element Analysis Report

The Intermediate Structure has 2 plates and a cylindrical shell structure.

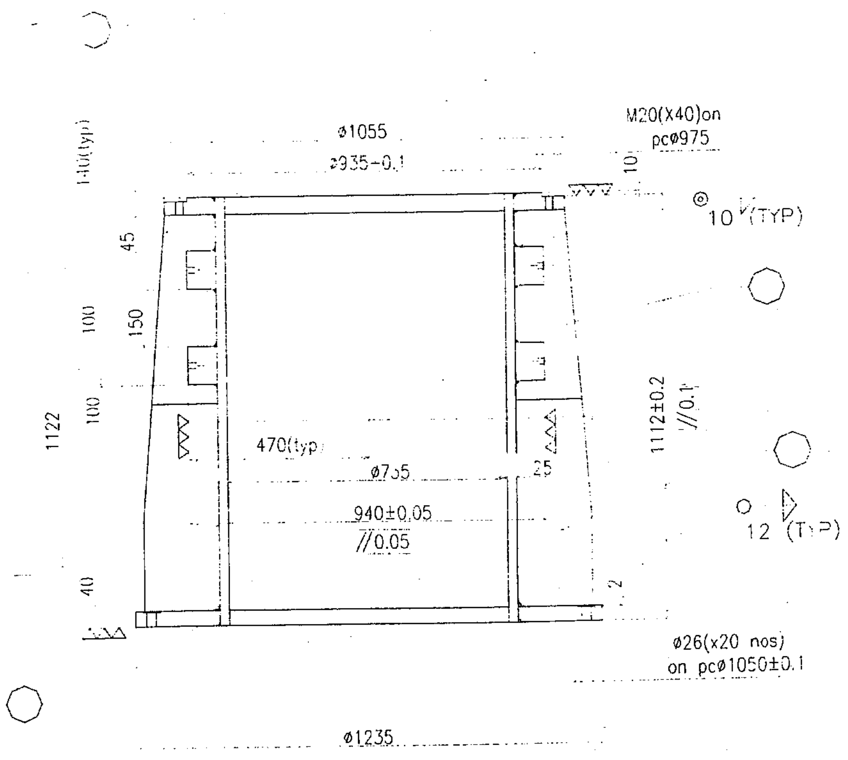


Fig 2

Material Properties taken into considerations:

E=2.1e4 Kg/mm^2 (Young's Modulus)
NUXY =0.3 (Poisson's Ratio)

Other Inputs provided are as follows:

The Vertical Force coming on the Structure = 12T(12000Kg)
The Shear Force on the top plate is = 13.4T(13400Kg)
The Moment About the vertical Axis is =15T-m (15e6 Kg-mm)
The Moment at the bottom plate (2) = 48 T-m (48 x 10^6 Kg-mm)
The Tangential load due to motor Mounting on the cylindrical structure =15.5 T(15596.7 Kg)

The Weight of the Motor +Gear mounting =300kg on each block

ELEMENT TYPE 2 IS SHELL63 ELASTIC SHELL INOPR

ELEMENT TYPE 3 IS BEAM4 3-D ELASTIC BEAM INOPR

CURRENT NODAL DOF SET IS UX UY UZ ROTX ROTY ROTZ
THREE-DIMENSIONAL MODEL

M O D E L   I N F O R M A T I O N

Solid model summary:

	Largest Number	Number Defined	Number Selected
Keypoints	616	616	616
Lines	1440	1440	1440
Areas	1046	1046	1046
Volumes	232	232	232

Finite element model summary:

	Largest Number	Number Defined	Number Selected
Nodes	17106	17106	17106
Elements.	12327	12325	12325
Element types	3	3	
Real constant sets.	2	2	
Material property sets. . .	2	2	
Coupling.	9	9	

B O U N D A R Y   C O N D I T I O N   I N F O R M A T I O N -----------------

	Number
Constraints on nodes.	450
Forces on nodes	1498

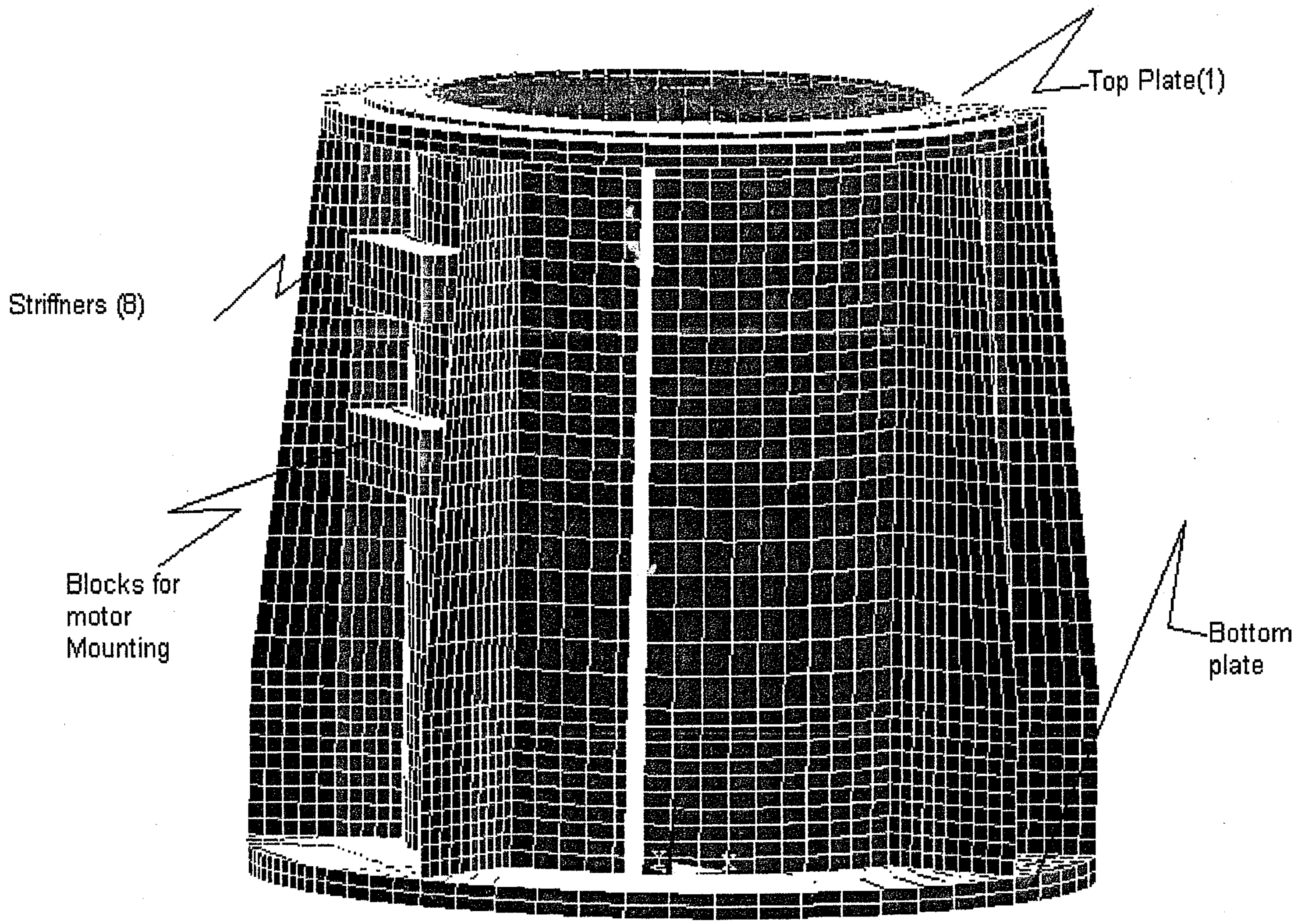
R O U T I N E   I N F O R M A T I O N ---------------------------------------

Current routine. . . . . .General Postprocessor (POST1)
Active coordinate system . . . . 1 (Cylindrical)
Display coordinate system. . . . . 0 (Cartesian)
Analysis type. . . . . .Static (steady-state)

LIST NODAL FORCES FOR SELECTED NODES 1 TO 18522 BY 1
CURRENTLY SELECTED NODAL LOAD SET= FX FY FZ MX MY MZ

NODE	LABEL	REAL	
Loads on the top plate			
1	FX	-22.0394737	(Shear load on each of node)
1	FZ	-19.7368421	(Vertical load on each node)
18522	MZ	15000000.0000	(Moment of structure about Z-axis)
Loads due to motor mounting			
4156	FZ	-13.6363636	(Dead weight of motor mounting on each node)
4169	FY	177.227273	(Tangential load due to motor on each node)
Loads at the Bottom plate			
18521	MY	48000000.0000	(Moment of bottom plate about Y-axis)

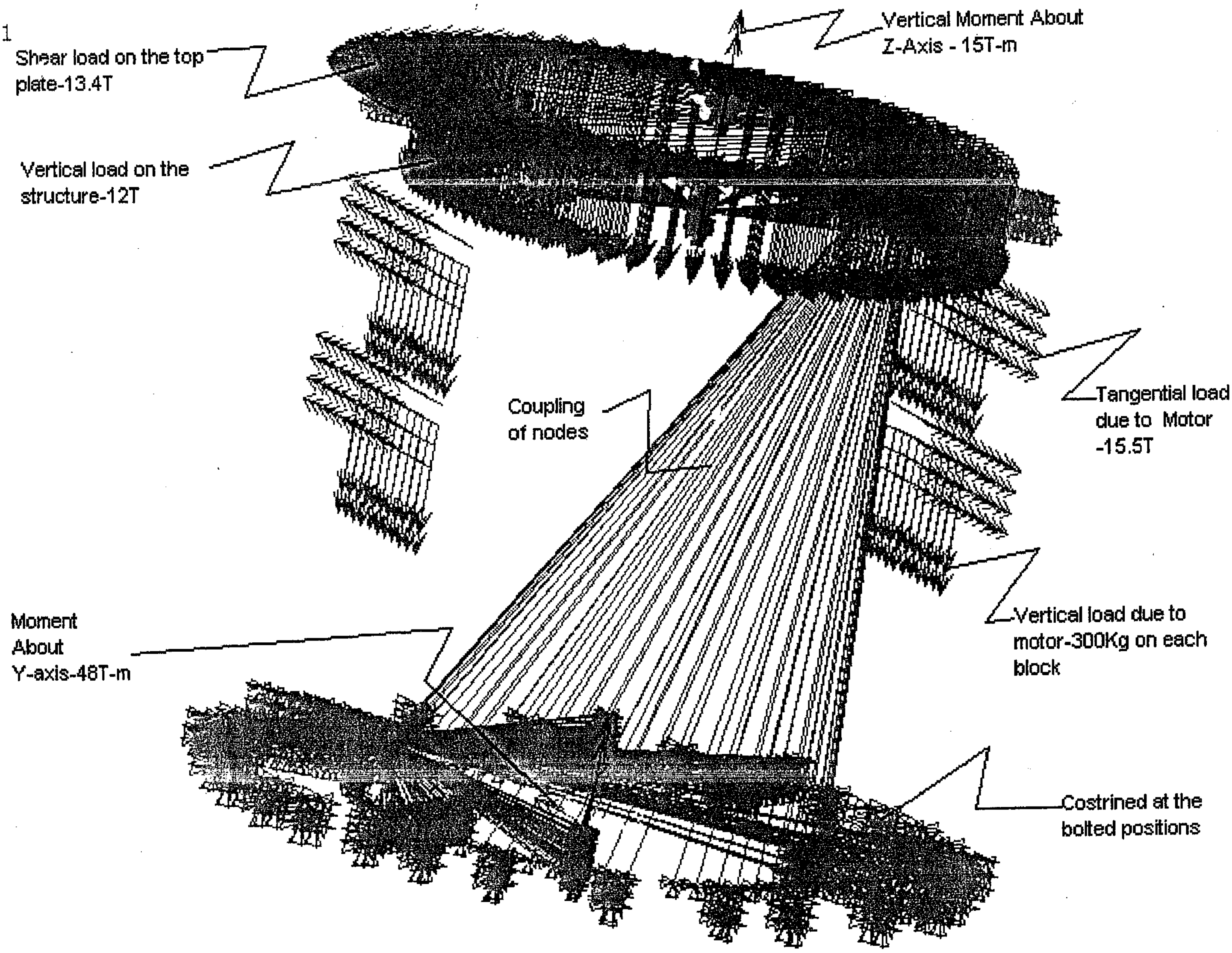
Solid Model with Meshing



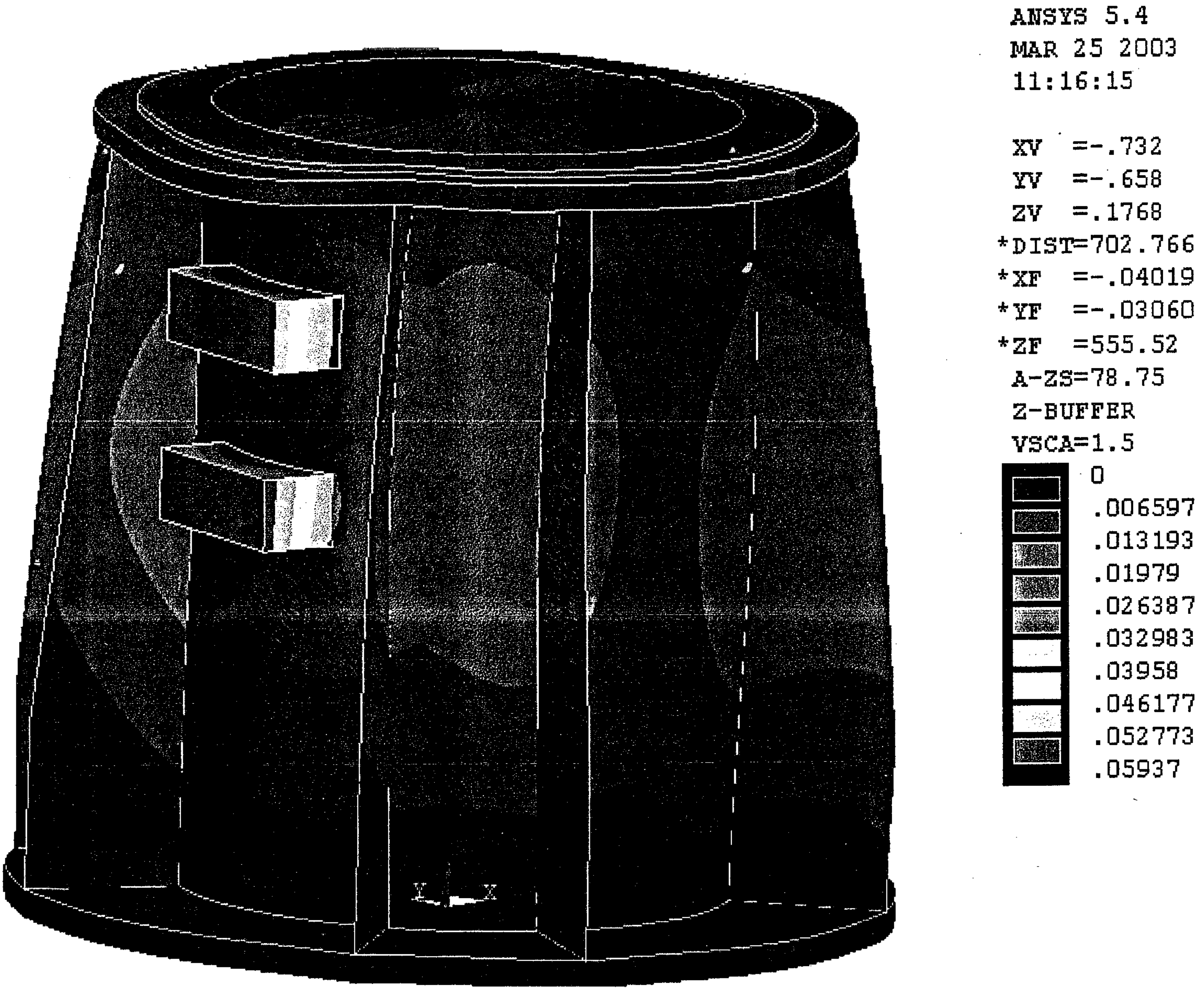
The Model consists of solid brick elements for the plate 1,2,3. All the volumes are mapped mesh so as to retain the element shape (i.e. keeping the Aspect ratio within the limit. The stiffeners 4,5 are modeled with shell element and the beam elements added at the axis of the structure so as to apply the moment about Z&Y-axis.

The Boundary conditions applied

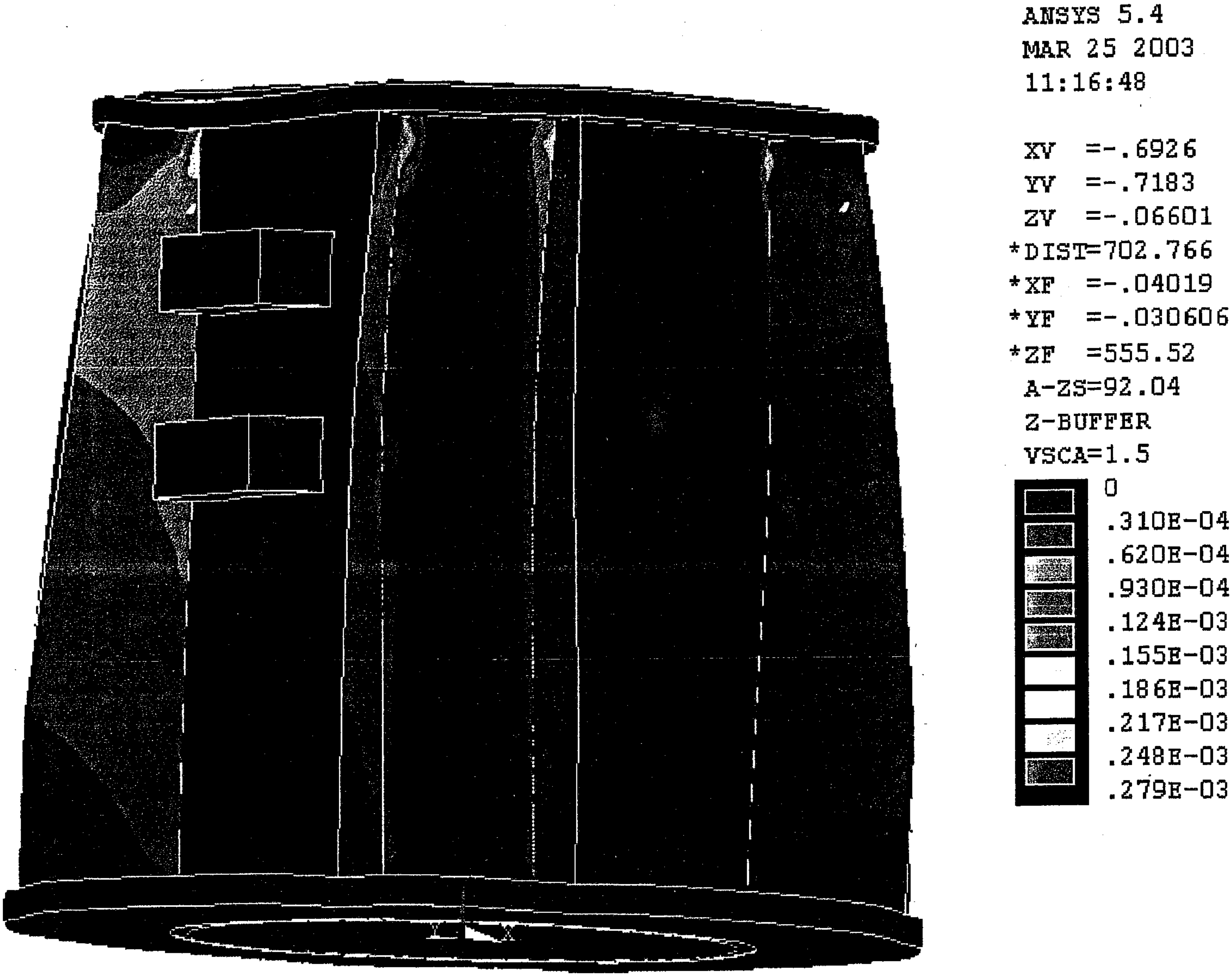
The loading conditions are as per the given specification and the fixing condition is where the bolt is assembled with concrete structure



Maximum displacement vector which is the sum of translation in X, Y, Z directions. And is found to be 0.059 mm.



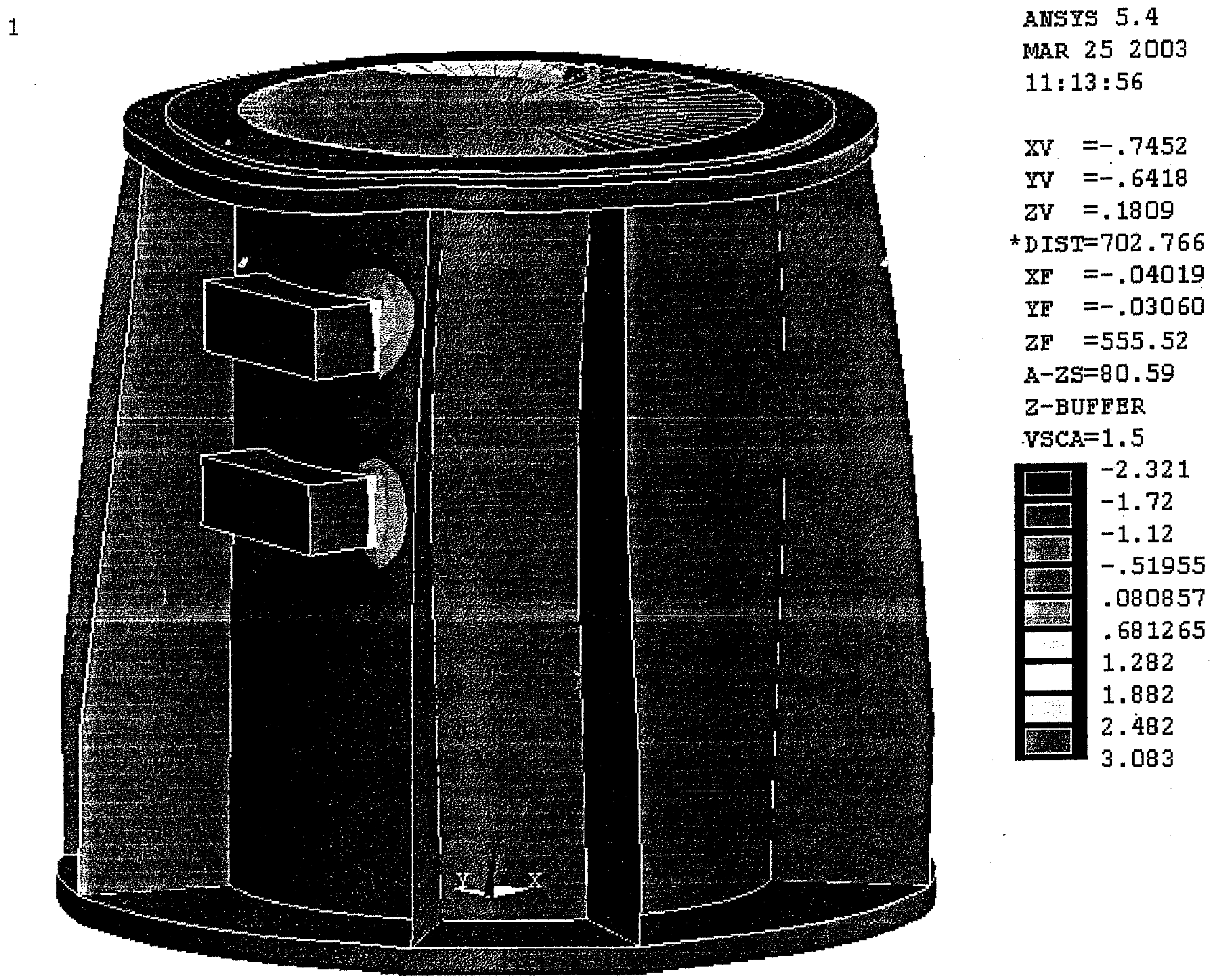
Maximum angular displacement vector which is the sum of rotation in X,Y,Z directions.
And is found to be 0.279e-3 mm. which is within the limit.



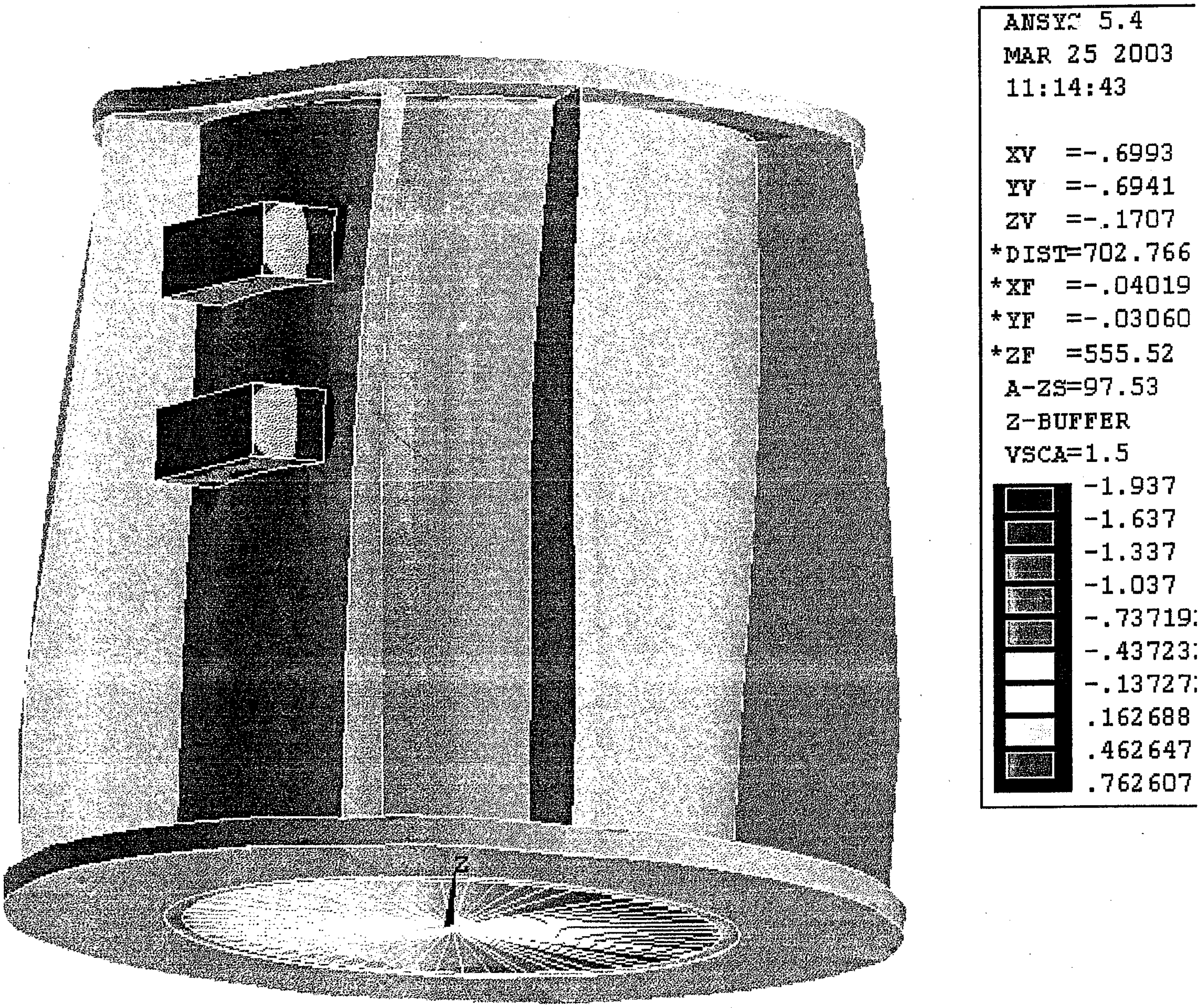
The Direct stresses measured along the Z-axis are

a) Maximum Tension Stress= 3.038 Kg/mm2 (From C.M.T.I Handbook Max allowable tensile stress for structural steel IS 1570 is- 7 Kg/mm2 .)

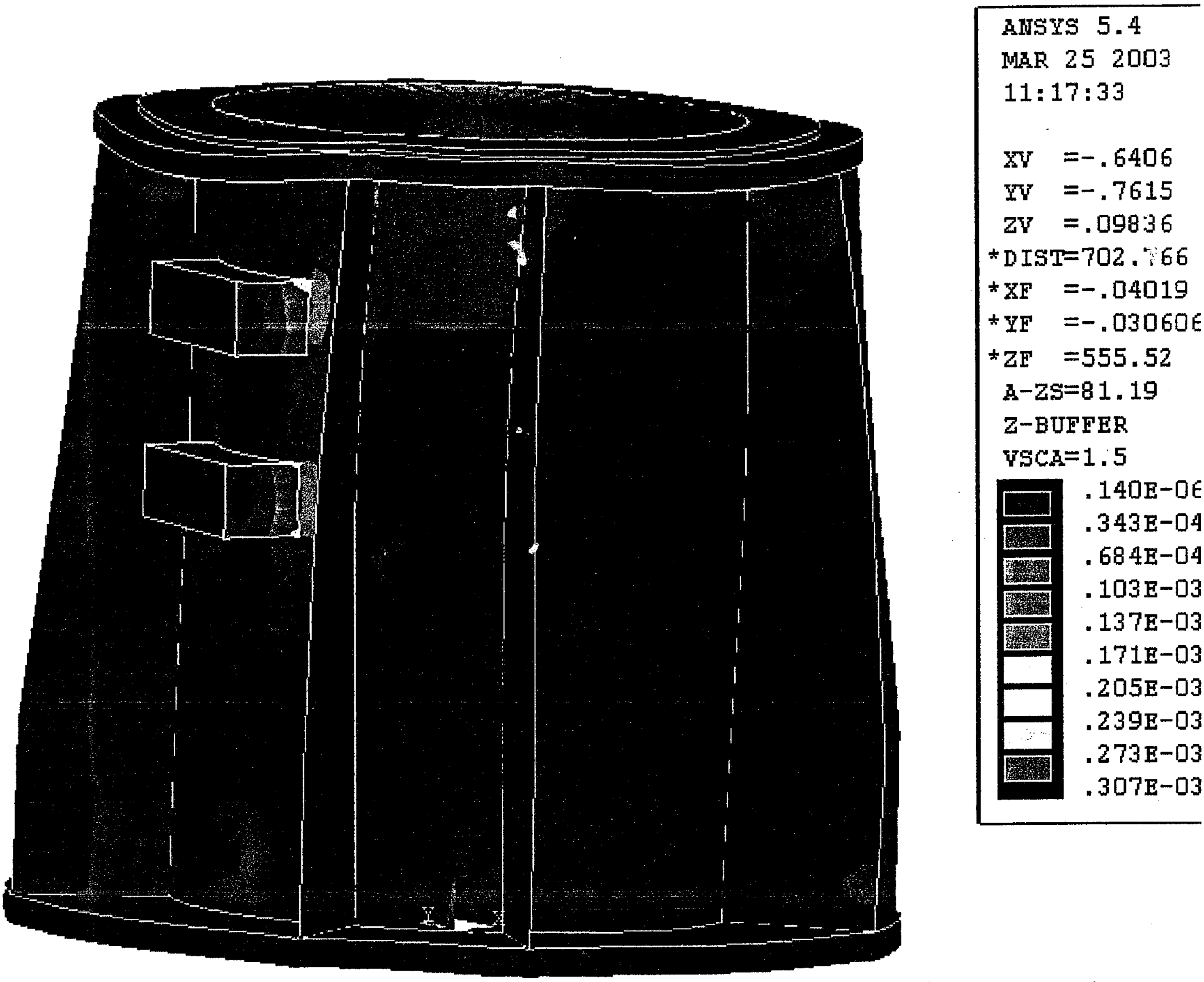
b) Maximum Compression Stress=2.321Kg/mm2 (From C.M.T.I Handbook Max allowable compressive stress for structural steel IS 1570 is 10.5Kg/mm2.)



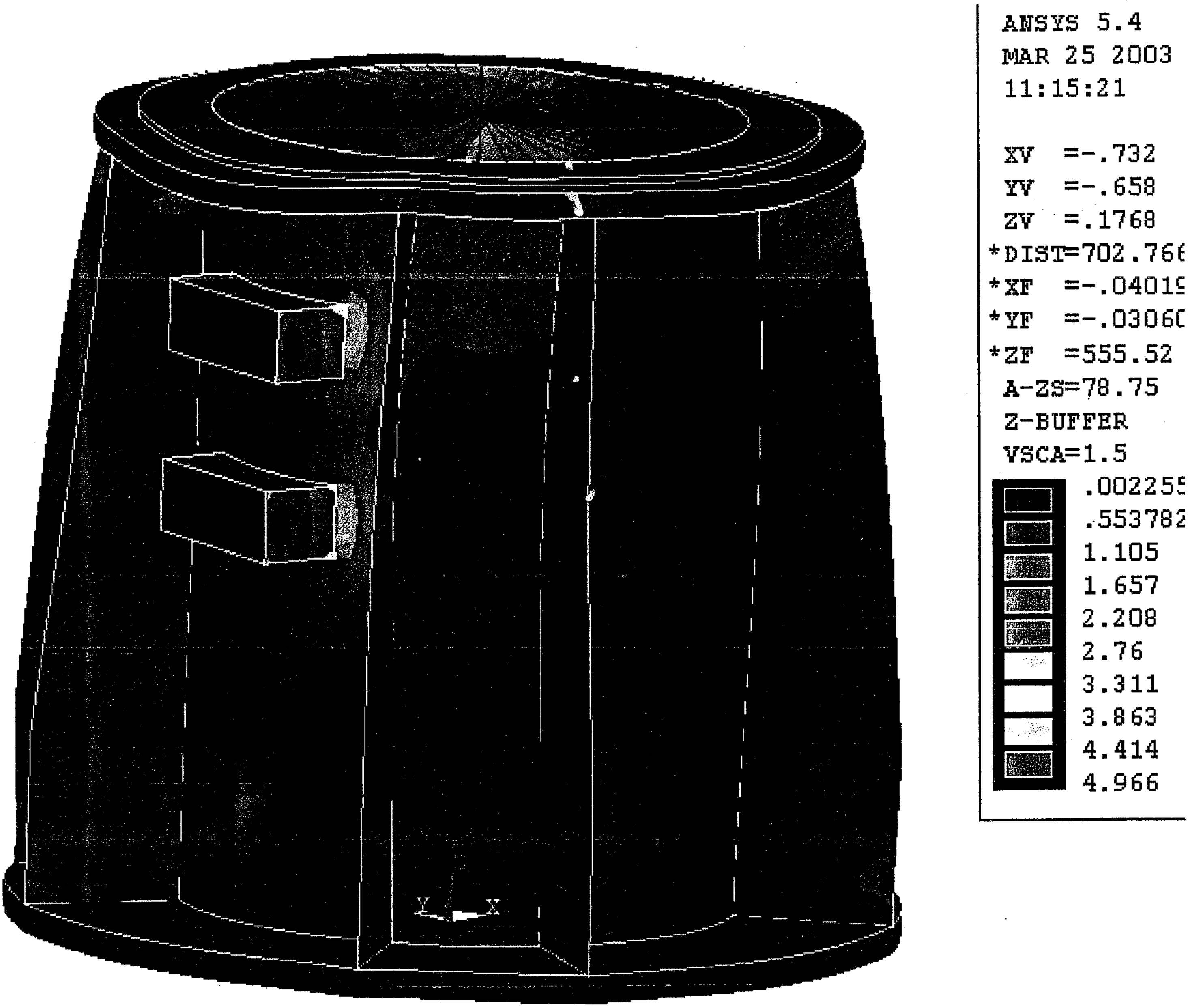
Maximum shear force found at the bottom of the plate where the bolts are assembled the maximum shearing force observed is 3.2Kg/mm2 (4.4kg/mm2 as per C.M.T.I. Handbook)



Von mises Maximum elastic strain



Von Mises stress failure criteria



Conclusions

- Maximum displacement vector which is the sum of translation in X, Y, Z directions, and is found to be 0.059 mm. which is within the limit.
- Maximum angular displacement vector which is the sum of rotation in X, Y,Z directions. And is found to be 0.279e-3 mm. which is within the limit.

  The Direct stresses measured along the Z-axis are
- Maximum Tension Stress= 3.038 Kg/mm2 (From C.M.T.I Handbook Max allowable tensile stress for structural steel IS 1570 is- 7 Kg/mm2.)
- Maximum Compression Stress=2.321Kg/mm2 (From C.M.T.I Handbook Max allowable compressive stress for structural steel IS 1570 is 10.5Kg/mm2.)

Encoder systems

General

Antenna rotations about elevation and azimuth axis need to be monitored so as to provide feed back for correct positioning of antenna.

Antenna movements about the elevation and azimuth axis are monitored by 17 bit encoders. Arrangements are made in the mechanical design to accurately relate antenna movements to the output of encoders in a 1:1 relationship. The outputs of encoders give feedback to drive system to enable them to position the antenna in the required position.

The offtakes (for the encoder drives) which senses antenna motion, also serve as the sensors for limit switching operations. Limit switches are used to limit antenna travel and points for both elevation and azimuth axis.

Azimuth encoder assembly and limit switches:

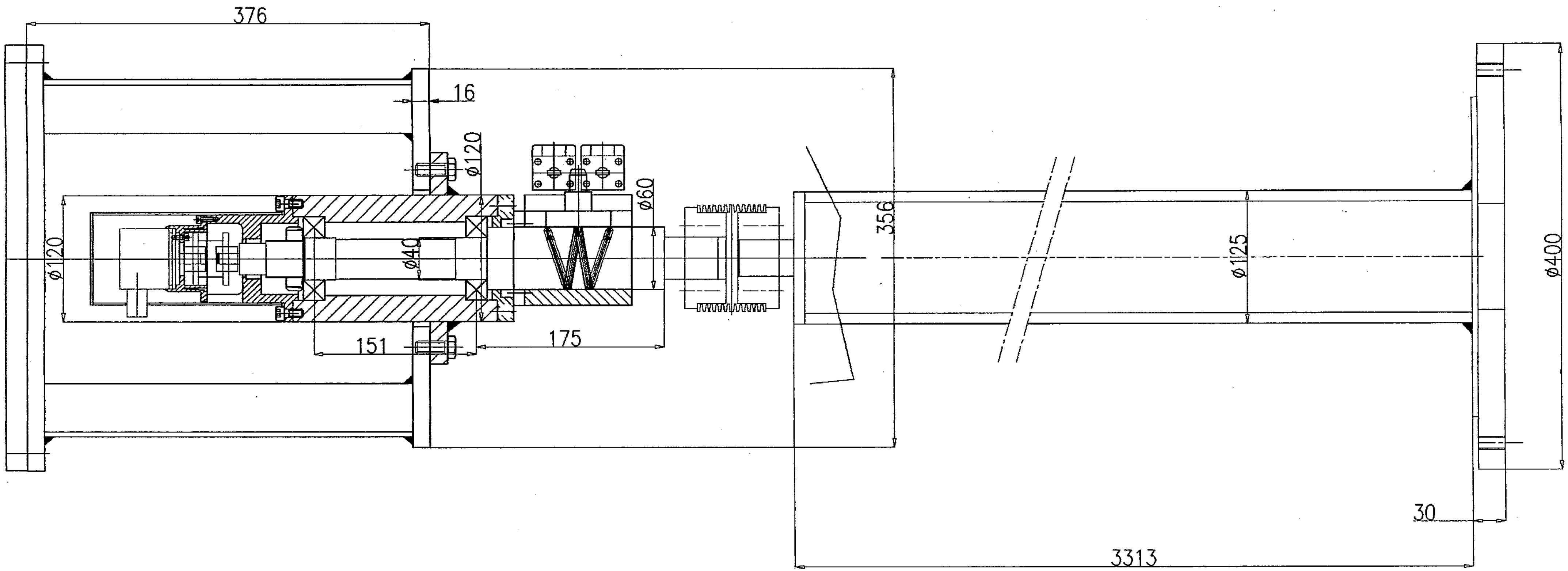
The rotation of the antenna about the azimuth axis needs to be monitored for control of antenna. This is done by providing an absolute encoder, the shaft of which rotates in 1:1 relationship with the azimuth rotation. The azimuth encoder assembly also incorporates an end point switching arrangement to limit the antenna movement to ± 270° of specified mean.

The offtake for the monitoring of azimuth axis rotation is the centre of rotation at the bottom of the yoke base. A central member (pipe) fixed to yoke base and centred with true azimuth axis, transfers signal to the encoder located lower down on 1 meter RCC shaft on the floor of RCC pedestal. The shaft is supported at lower end by bearings so that only the rotation is transferred to the encoder and not any radial or axial forces. The encoder is connected to the shaft by means of a torsionally axially flexible rigid coupling.

Four precision limit switches are provided on a shaft to limit the antenna travel. These limit switches are operated by cam fixed on helix grooves arrangement. Electrical circuitry is designed around these switches so as to effectively limit the end points of antenna travel.

Forces acting on shaft and bearings arise mainly due to misalignment during assembling and erection of main tubular shaft, maximum care while designing of encoder assembly unit has been taken to take care to prevent misalignment force being passed on to encoder unit.

Size of the intermediate shaft and support bearings are accordingly designed. 40mm shaft is adequate and bearing SKF/FAJ 32208, taper roller bearings are selected so that both radial and thrust loads are taken care. Bearings are in back-to-back arrangement.



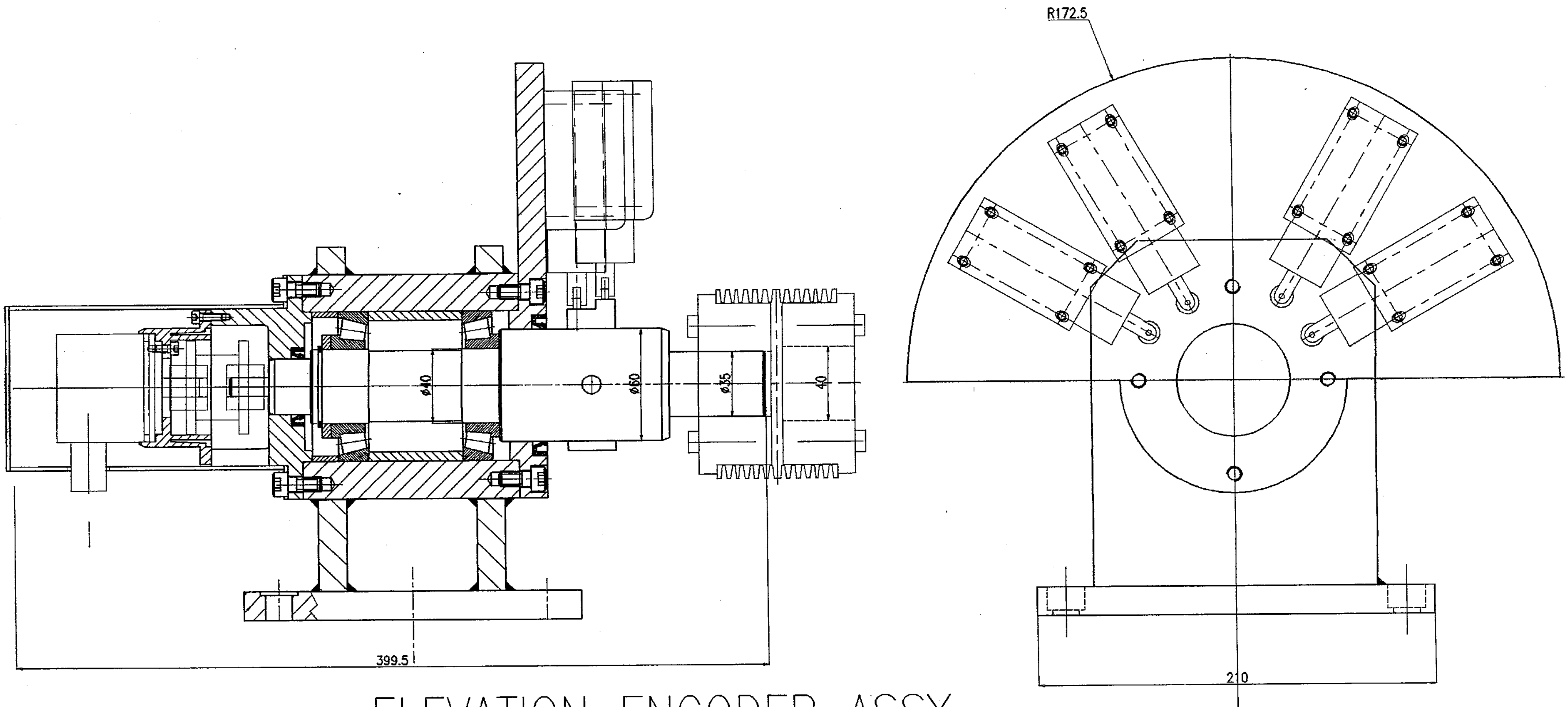
AZIMUTH ENCODER ASSY.

Elevation encoder assembly and limit switches

The rotation of the antenna about the elevation axis needs to be monitored so as to provide feedback for antenna control. An absolute encoder is provided for this purpose. The elevation encoder assembly also incorporates end point switching arrangement to control and signal the limits of antenna movement. The rotary motion of the antenna about the elevation axis is transferred via a rigid coupling, capable of substantial angular misalignment, to a motion transmitting shaft concentric with the main shaft of the elevation bearing assembly.

General arrangement elevation encoder

The inner motion transmitting shaft is supported on precision bearings. The encoder is connected tot he other end of this shaft through a tosionally rigid coupling. By this arrangement it is ensured that undesirable forces are not transmitted to the encoder. This shaft has also mounted on it, the sensing arrangement for limit switching. Totally 4 switches are mounted, 2 for one end bearing and other 2 for the other end.



ELEVATION ENCODER ASSY.

CABLE WRAP SYSTEM

Cable wrap systems are necessitated due to the antenna movements, namely:

- Rotation reflector about the elevation axis in tracking and slewing operations.
- Rotation of antenna structure consisting of yoke and reflector about the azimuth axis.

Cable wrap systems encompass power, control, RF and IF cables.

Cable wrap systems are designed, within limitations imposed on each of the movements to ensure that cable do not kink, twist or bend beyond limits and do not get entangled during movements of the antenna.

Cable wrap systems with design limits for antenna movements

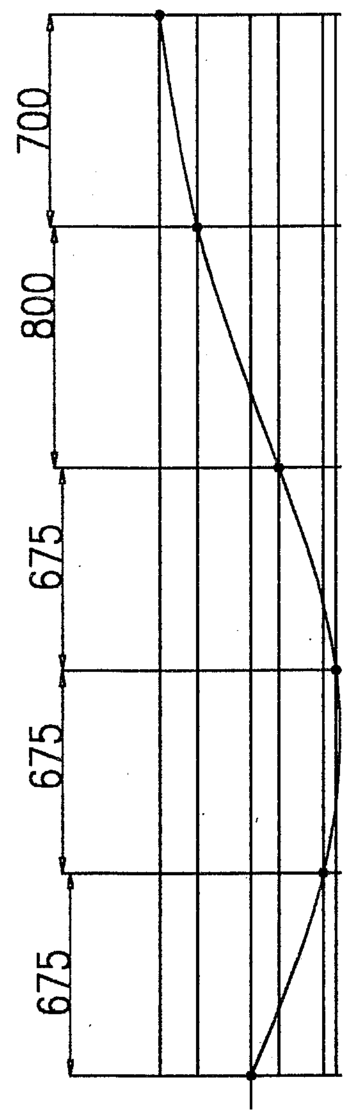
Elevation cable wrap	.-5 to 90°
Azimuth cable wrap	±270°

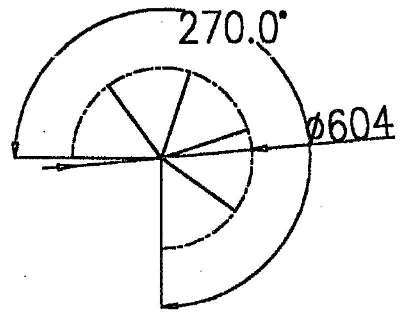
Azimuth cable wrap

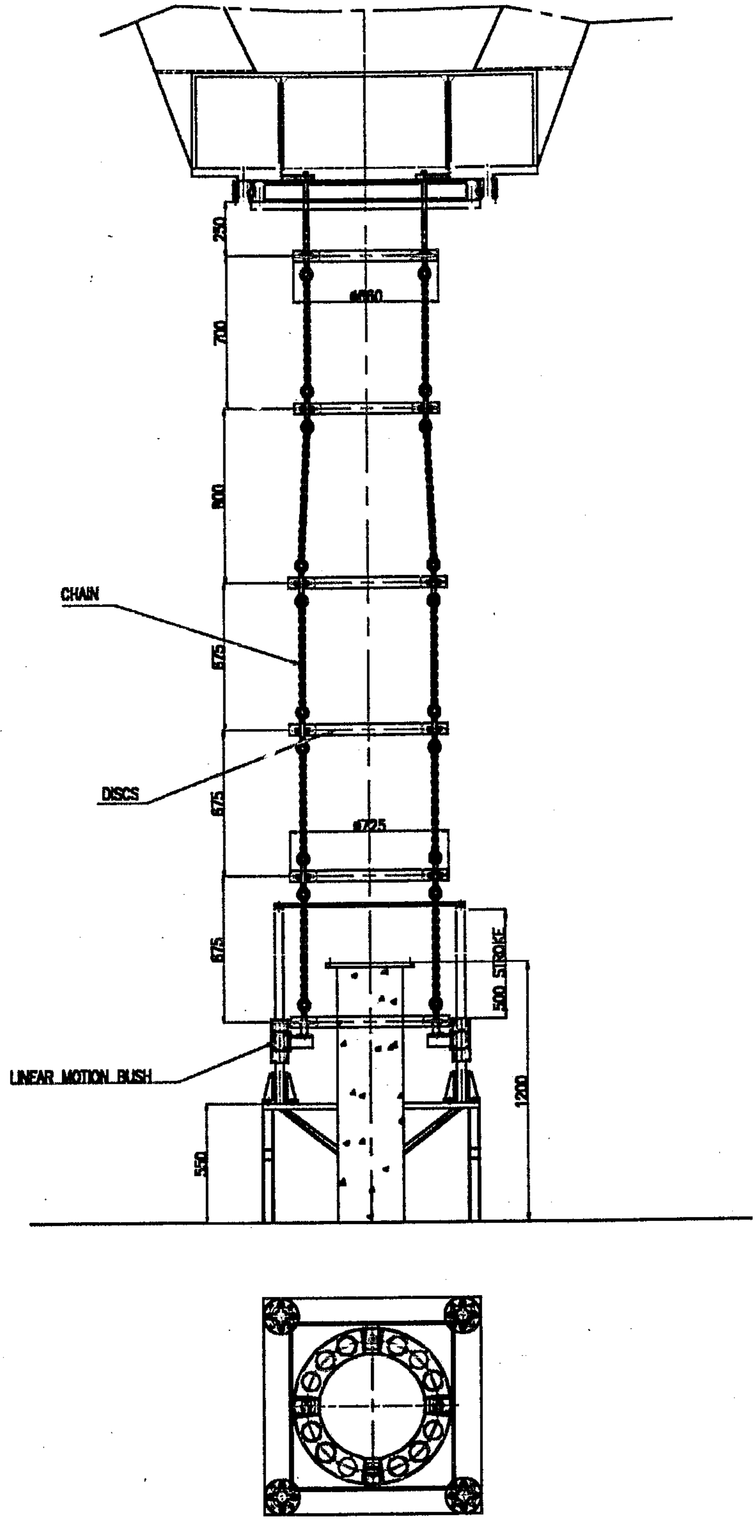
The system consists of six angular rings suspended in a arrangement and separated from each other by chains. The top ring is supported on brackets from the yoke base (concentric with azimuth axis) and revolves with it. The lower most ring is constrained to allow movements only axially. This movement is guided by 4 guide rods with linear brushes.

Cables are made to pass through holes in a ring. The holes serve to space cables along the periphery of an imaginary circle and prevent entanglement. During the azimuth slowing, the cable and chains supported intermittently, from an approximately helix along an imaginary surface. The cable covering is protected by abrasion and wear (due to axial movements) by means of teflon brushes fixed to rings.12 holes per ring for a maximum of 12 sets of cables are allowed for in this design, approximately 2 to 3 cables per hole.

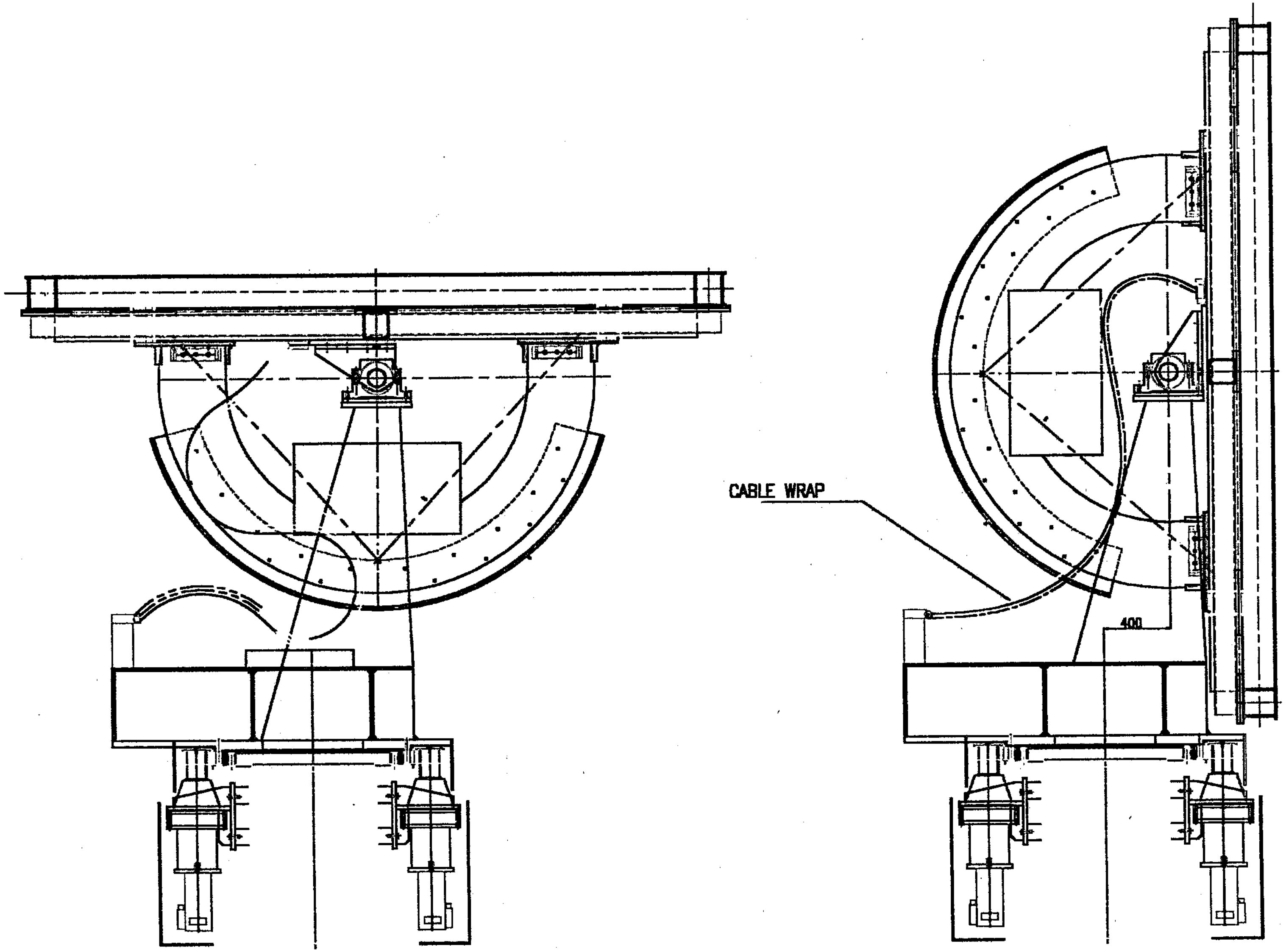
AZIMUTH CABLE WRAP CURVE





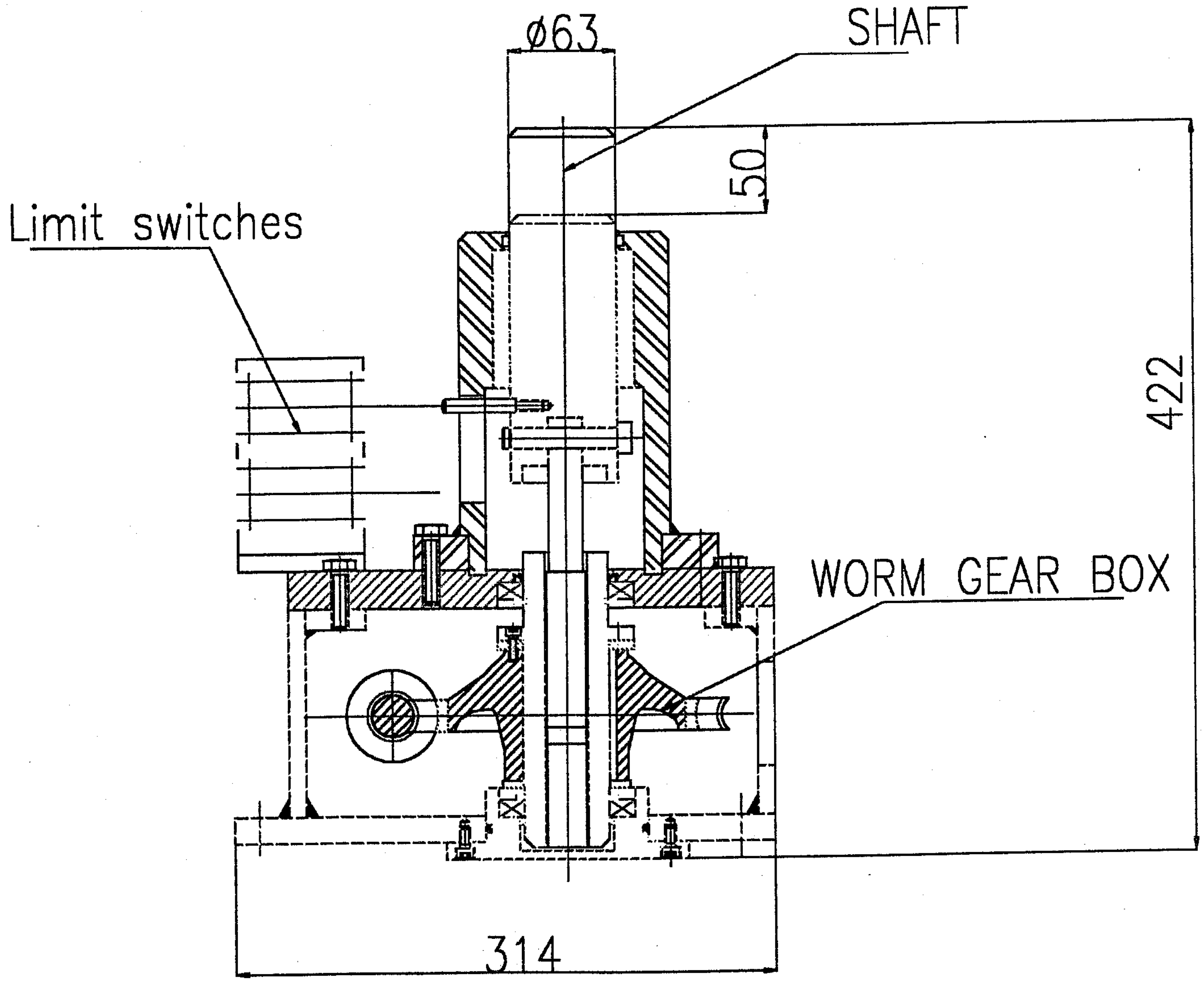


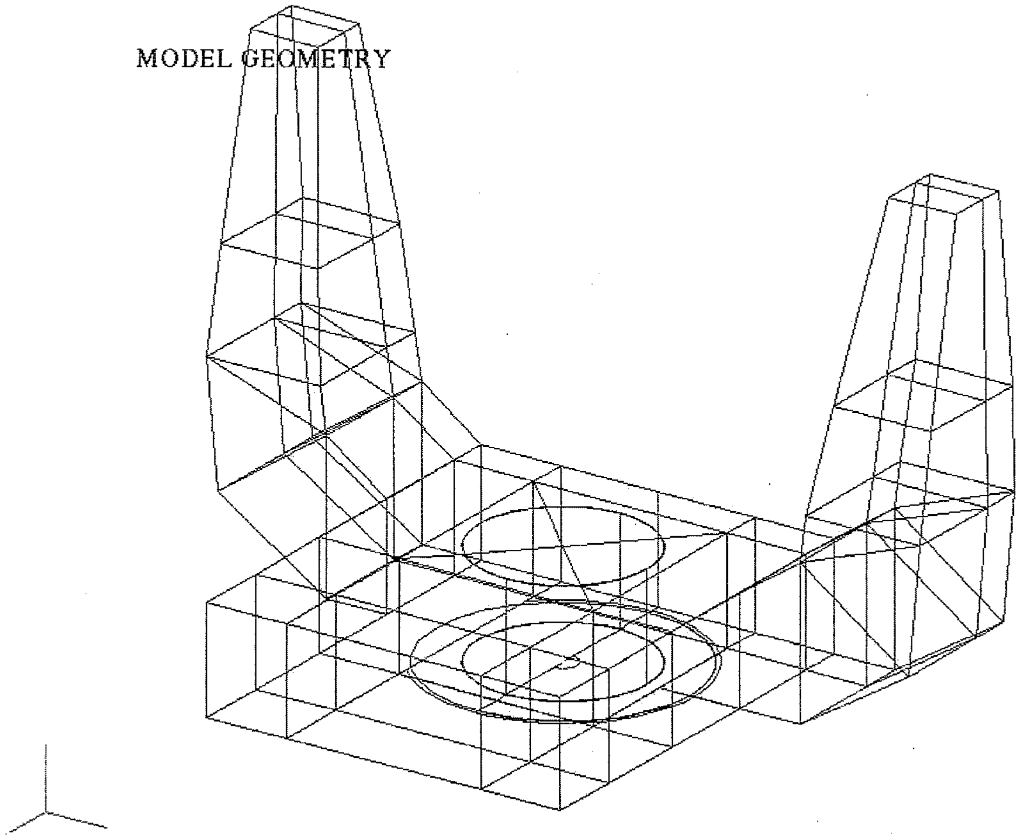
AZIMUTH CABLE WRAP

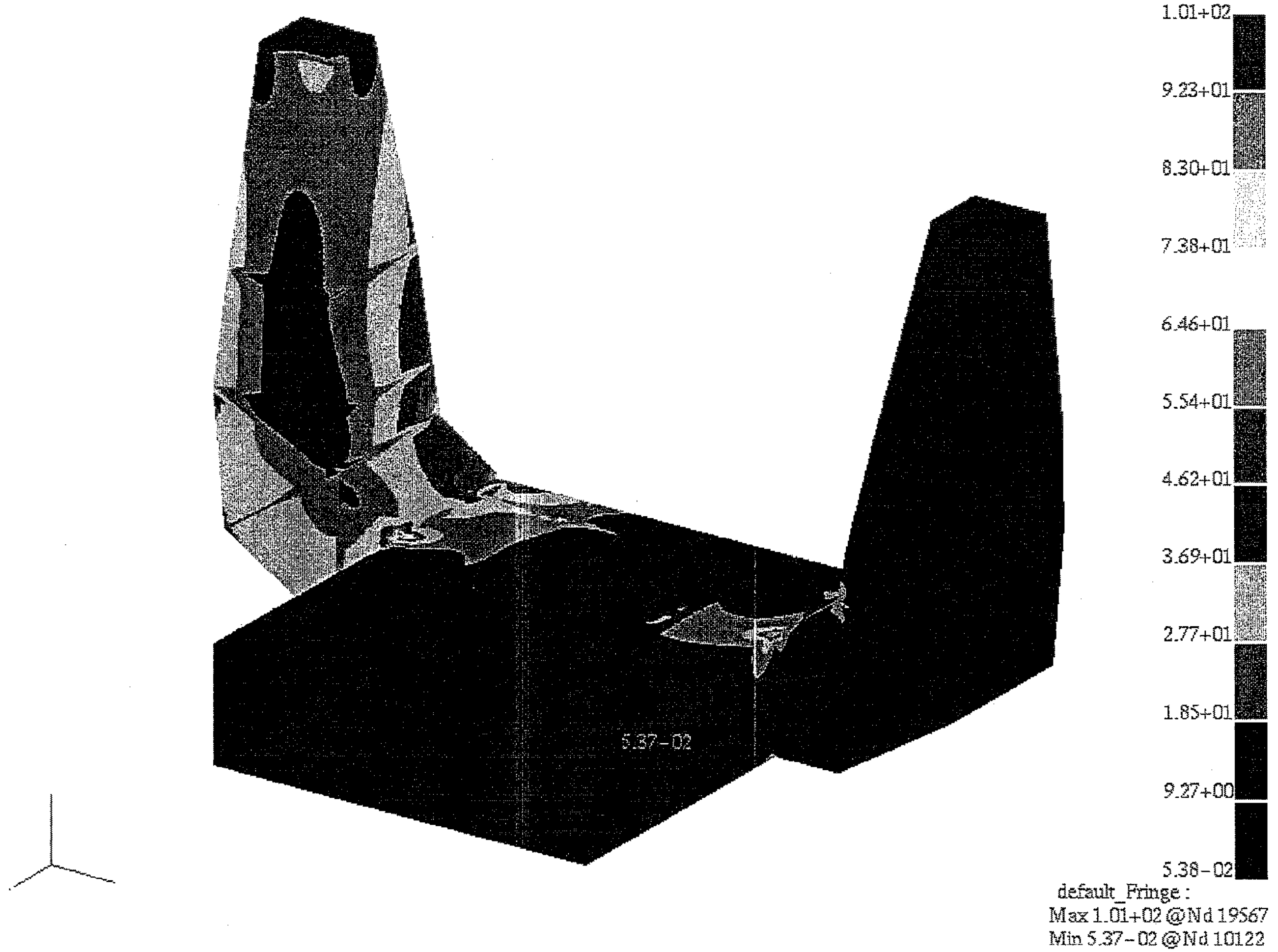


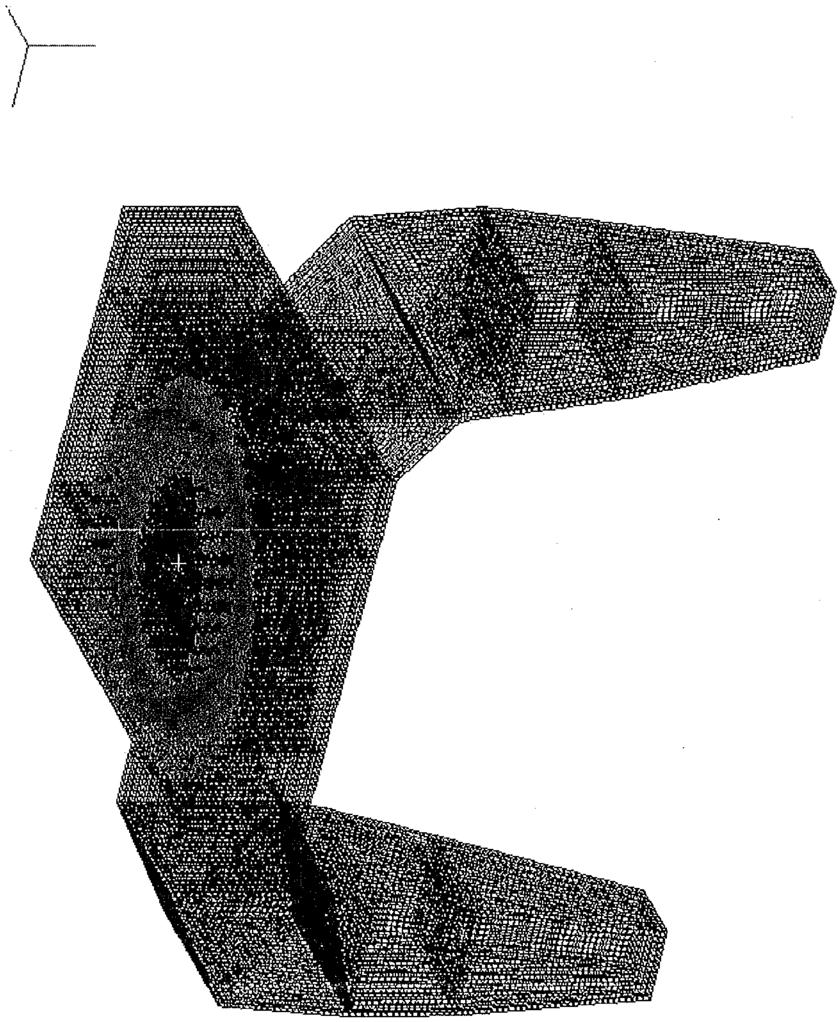
ELEVATION CABLE WRAP

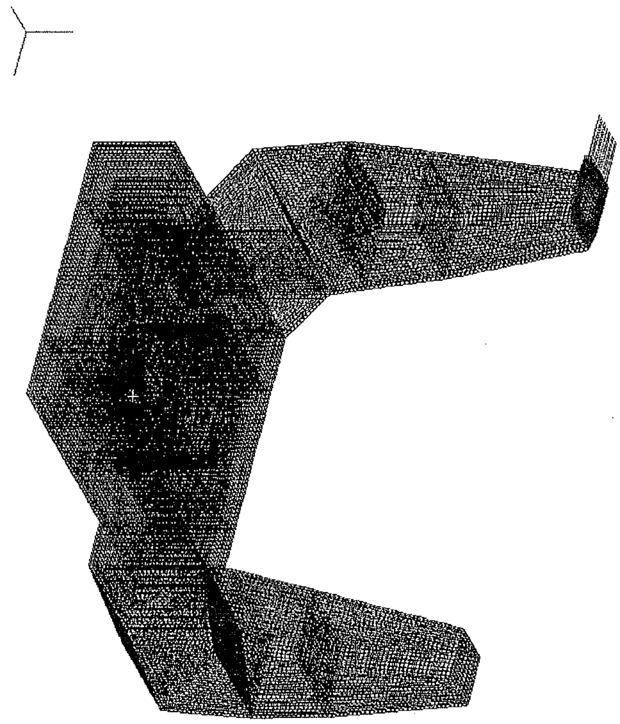
STOWLOCK ASSLY

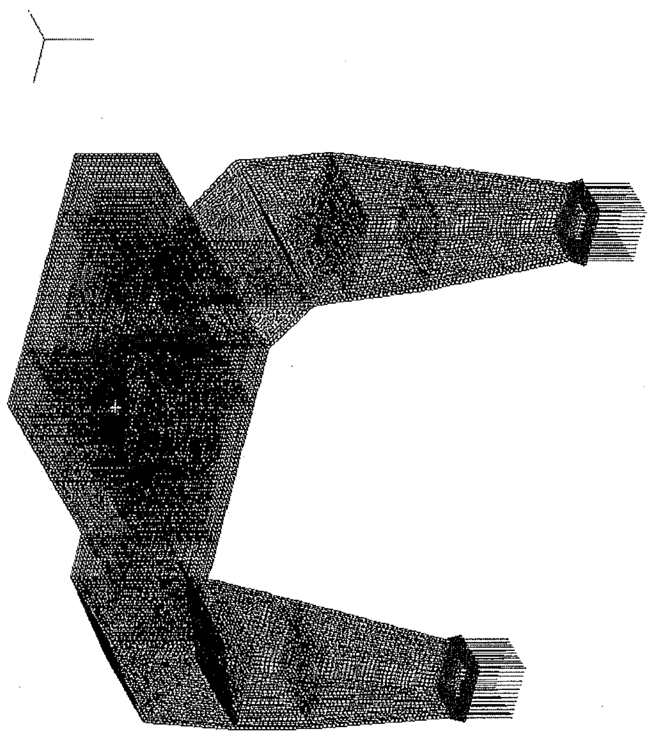


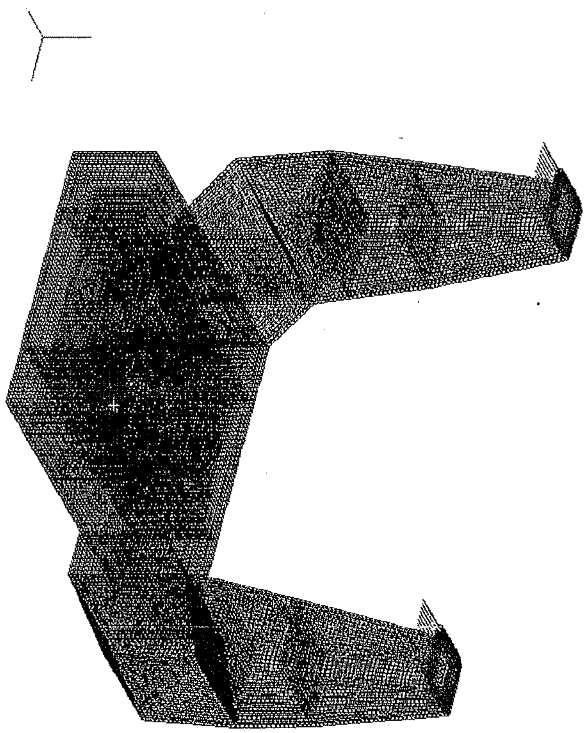

MODEL GEOMETRY


5.37-02
1.01+02
9.23+01
8.30+01
7.38+01
6.46+01
5.54+01
4.62+01
3.69+01
2.77+01
1.85+01
9.27+00
5.38-02
default_Fringe :
Max 1.01+02 @Nd 19567
Min 5.37-02 @Nd 10122









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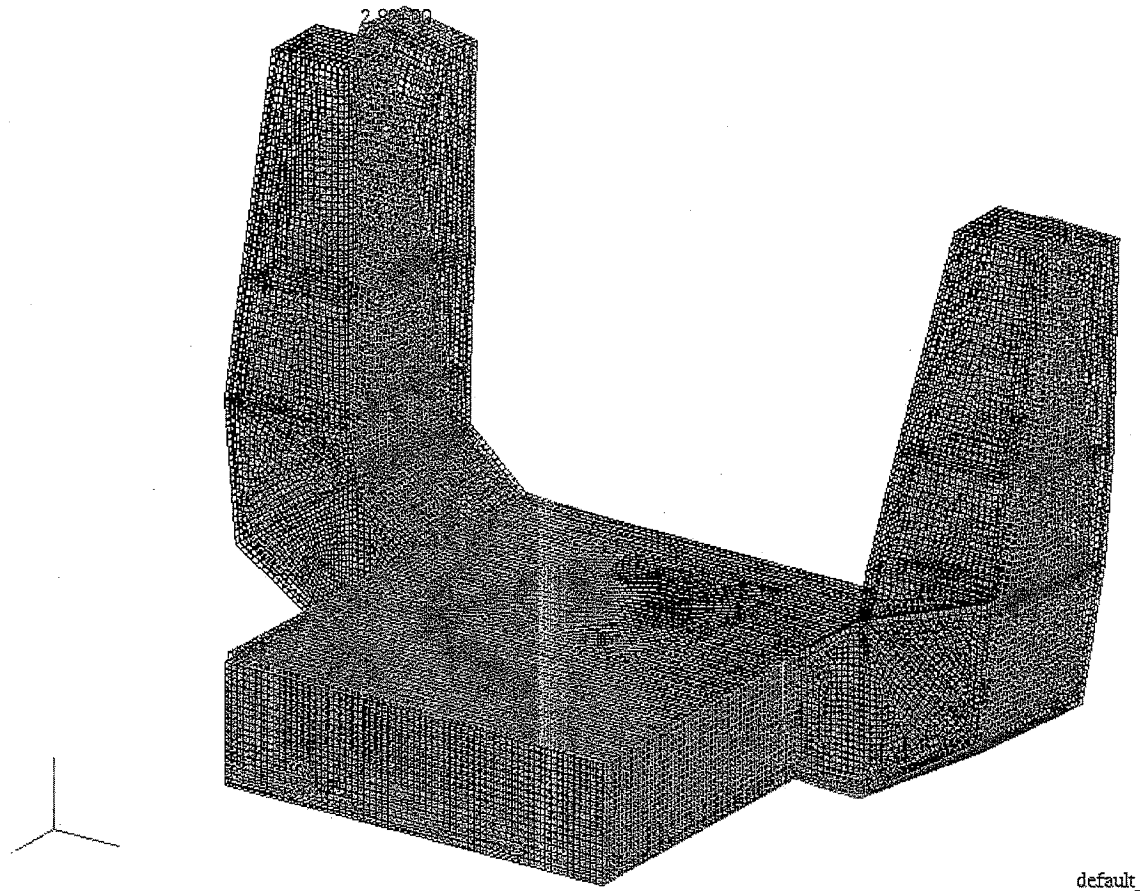
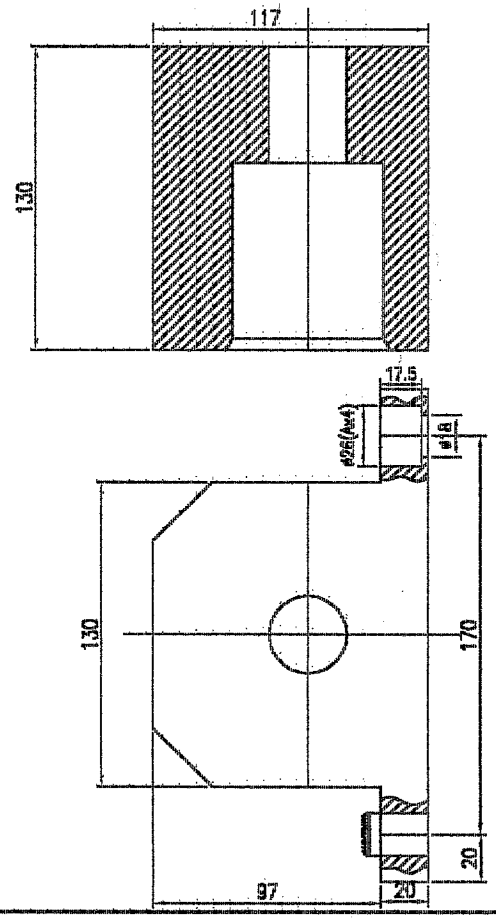
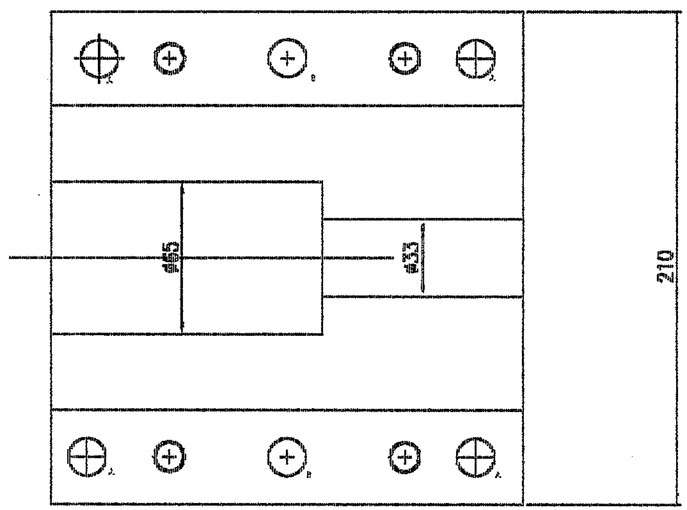
Deform:Default, Static Subcase, Displacements, Translational, (NON-LAYERED)



default_Deformation :
Max 2.92+00 @Nd 21422





NOTE:
Encoder selected is ROC 417.
131072 pulses per revolution
By Calculation 9.88 sec/pulse(angular sec/pulse) or 0.002746 degress/pulse.
i.e 0.062 mm clearance/pulse is sufficient.

Also backlash value is 0.85mm max which includes the encoder clearance /pulse, Therefore 1mm clearance is sufficient for the pin to enter the plunger block.

PLUNGER BRACKET

12M PPD RADIO TELESCOPE

STRUCTURAL ANLYSIS OF STOVELOCK:

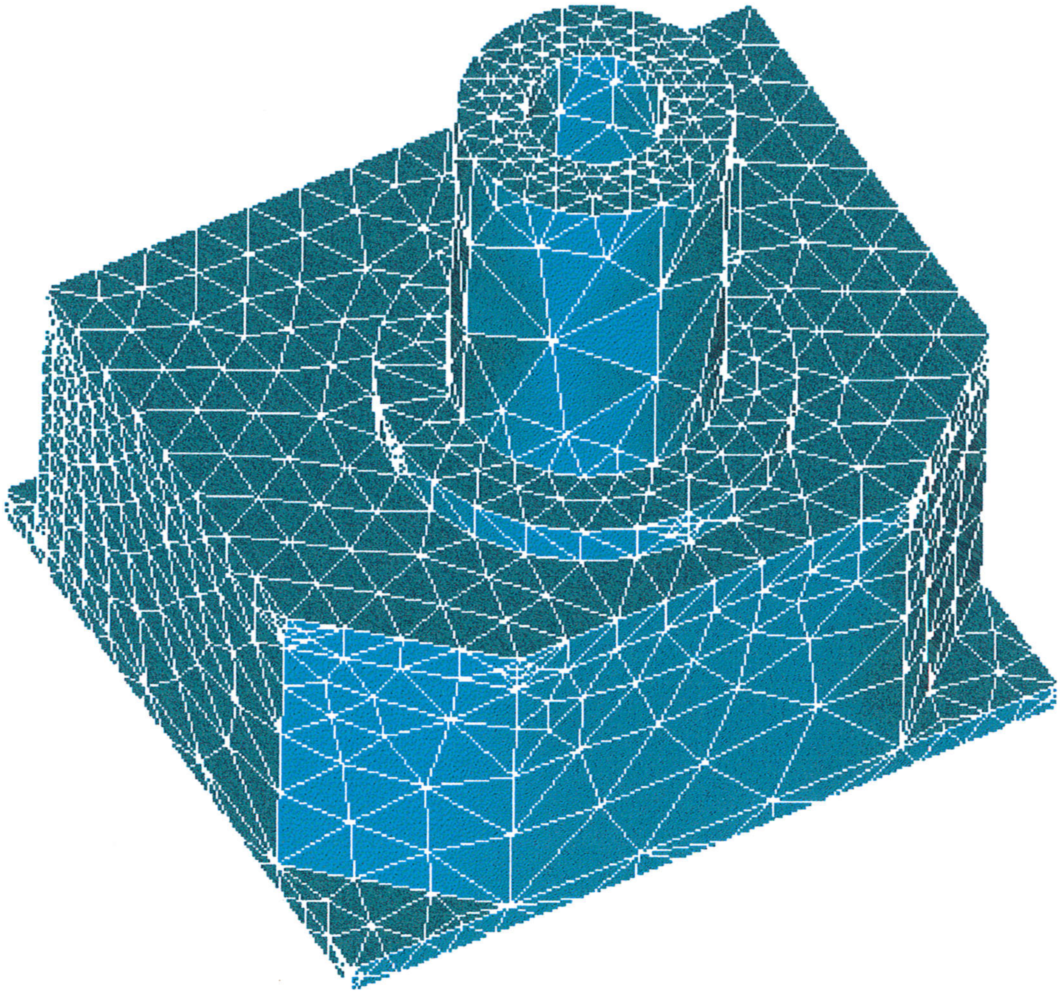


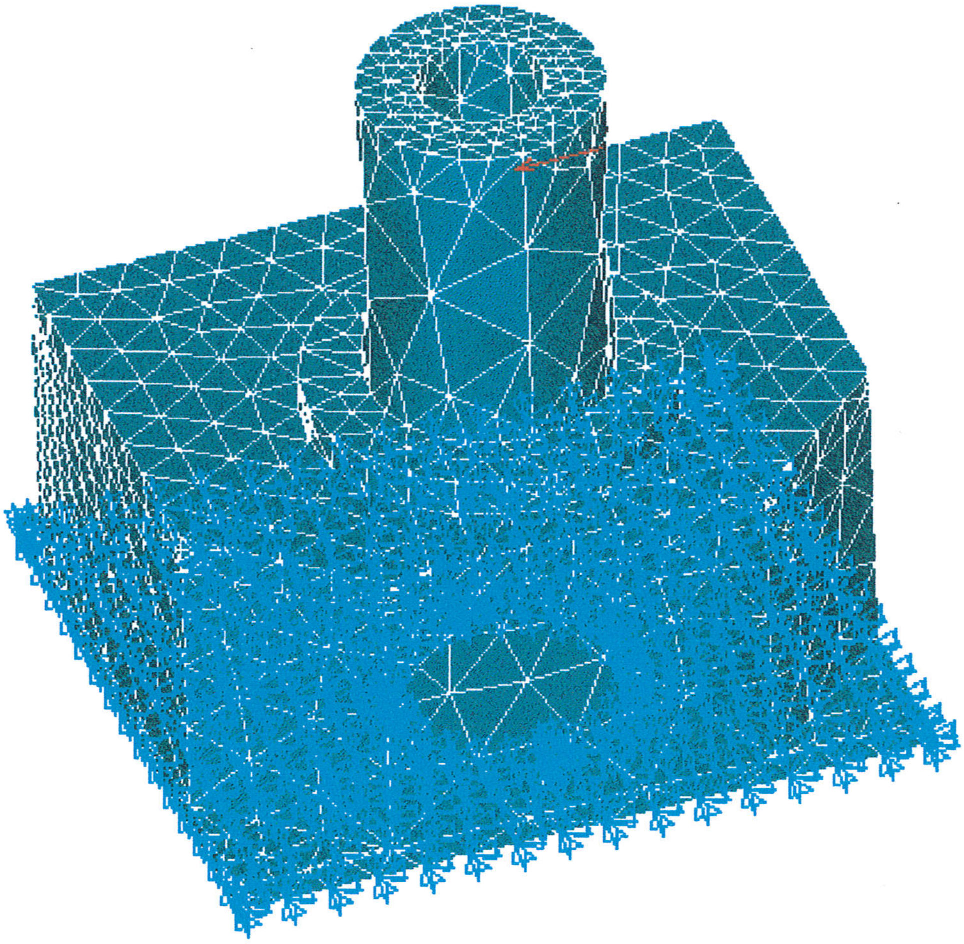
Fig. 2 ***STOVELOCK ASSEMBLY MESH MODEL***

MATERIAL: St – 42

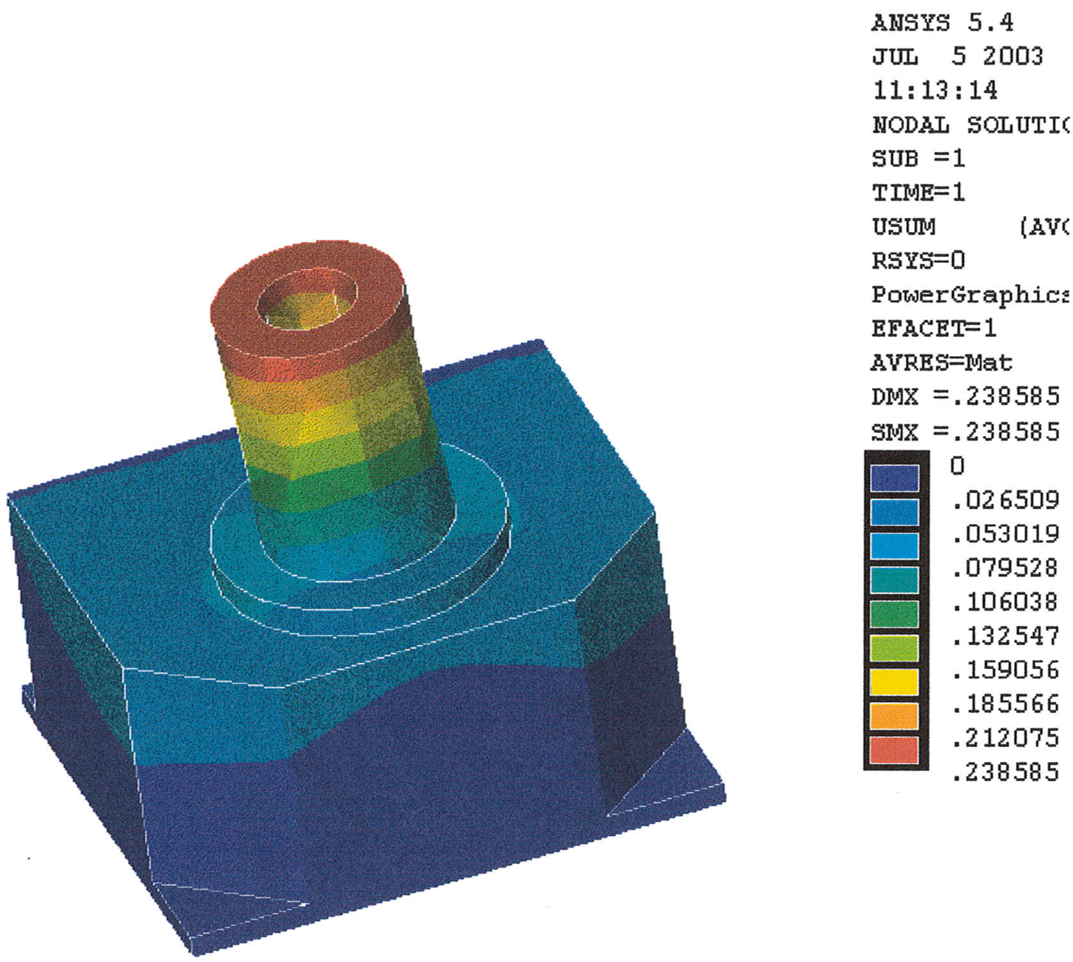
MATERIAL PROPERTY:

MODULUS OF ELASTICITY : 2.1×10^4 kg/mm^2
POISSONS RATIO : 0.3

STRUCTURAL ANLYSIS OF STOVELOCK:

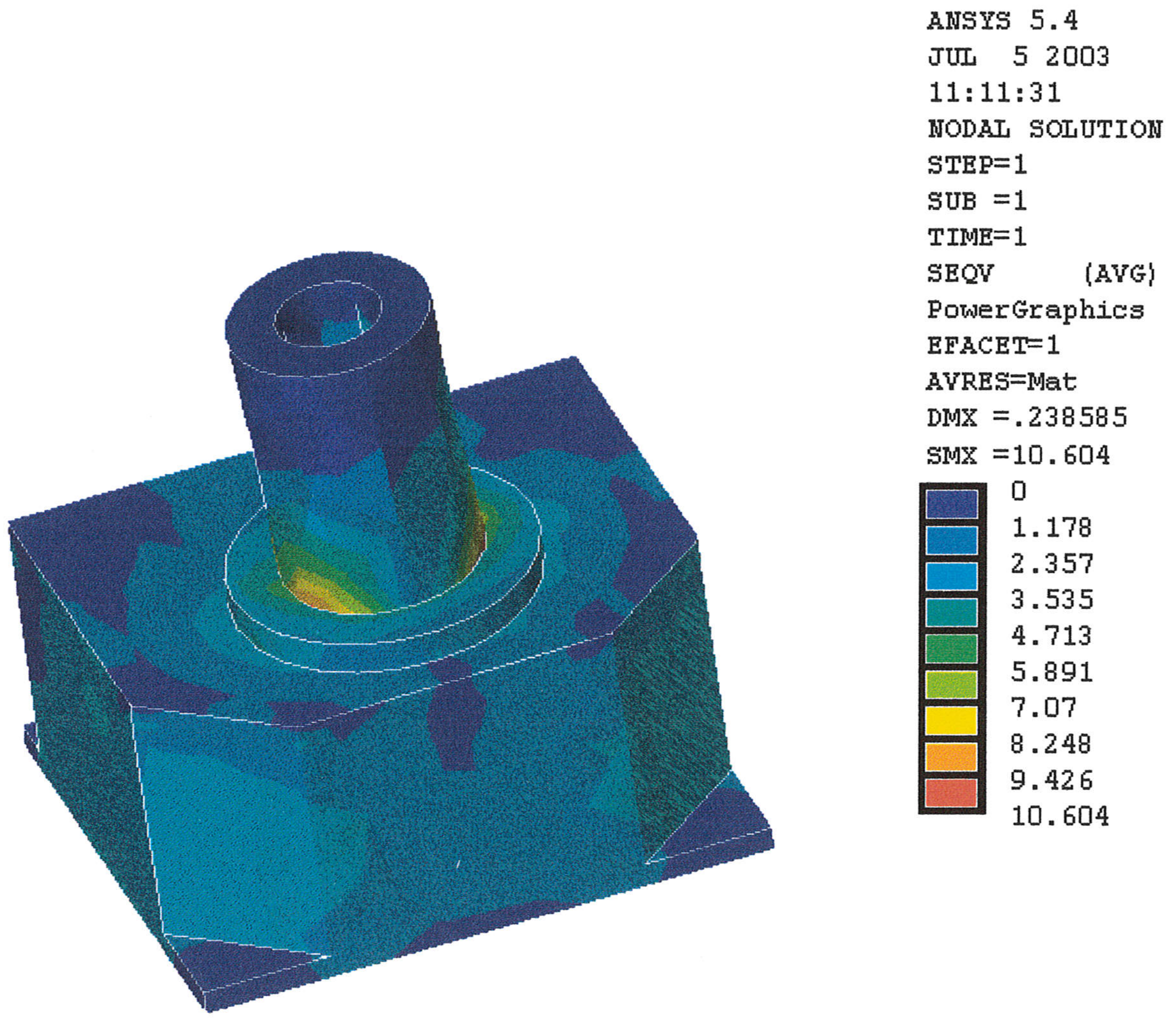


STRUCTURAL ANLYSIS OF STOVELOCK:



RESULT: **DEFLECTION = 0.238 mm**

STRUCTURAL ANLYSIS OF STOVELOCK:



RESULT: **STRESS = 10.604 kg/mm^2**

STRUCTURAL ANLYSIS OF STOVELOCK:

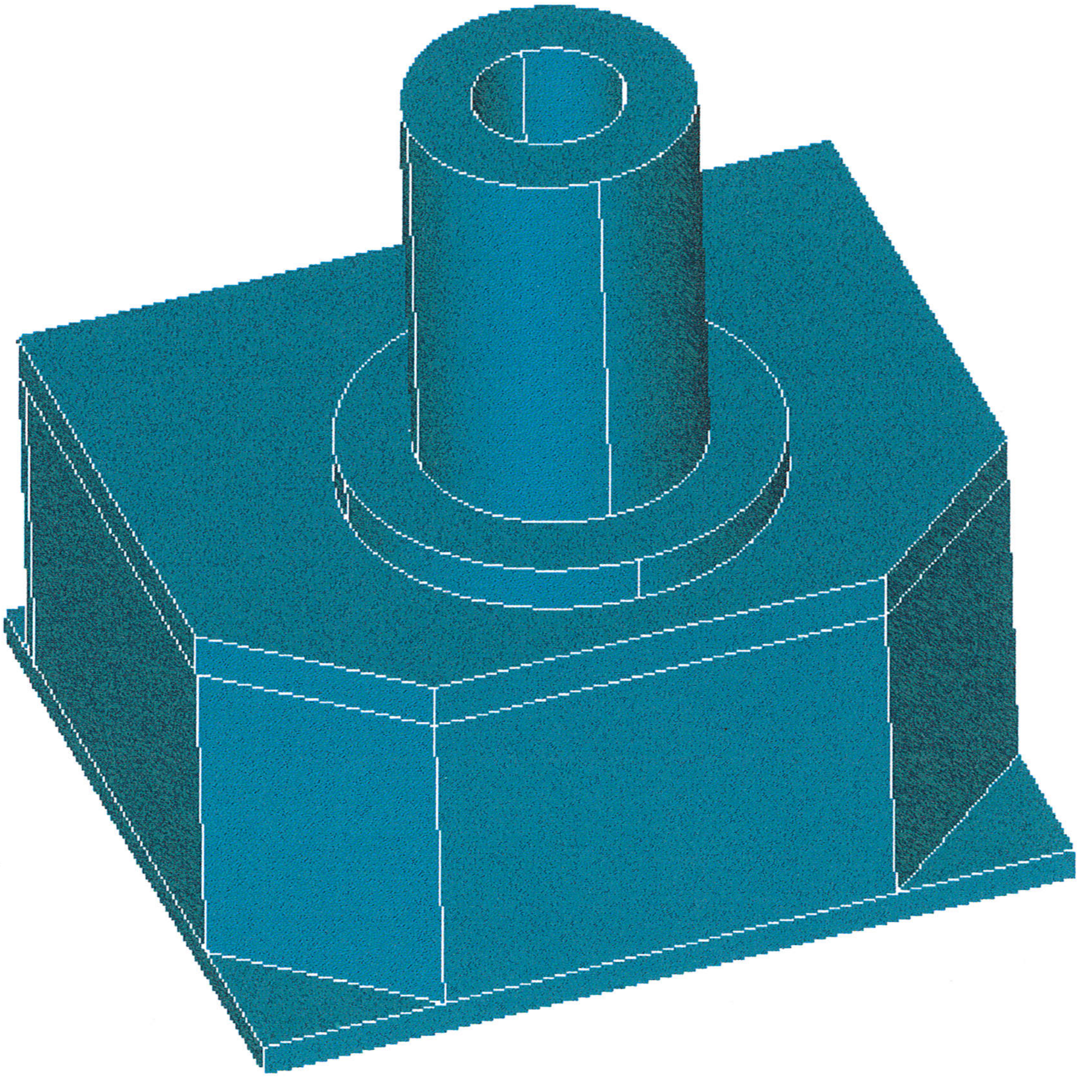


Fig. 1 *STOVELOCK ASSEMBLY*

STRUCTURAL ANLYSIS OF AZIMUTH BRACKET:

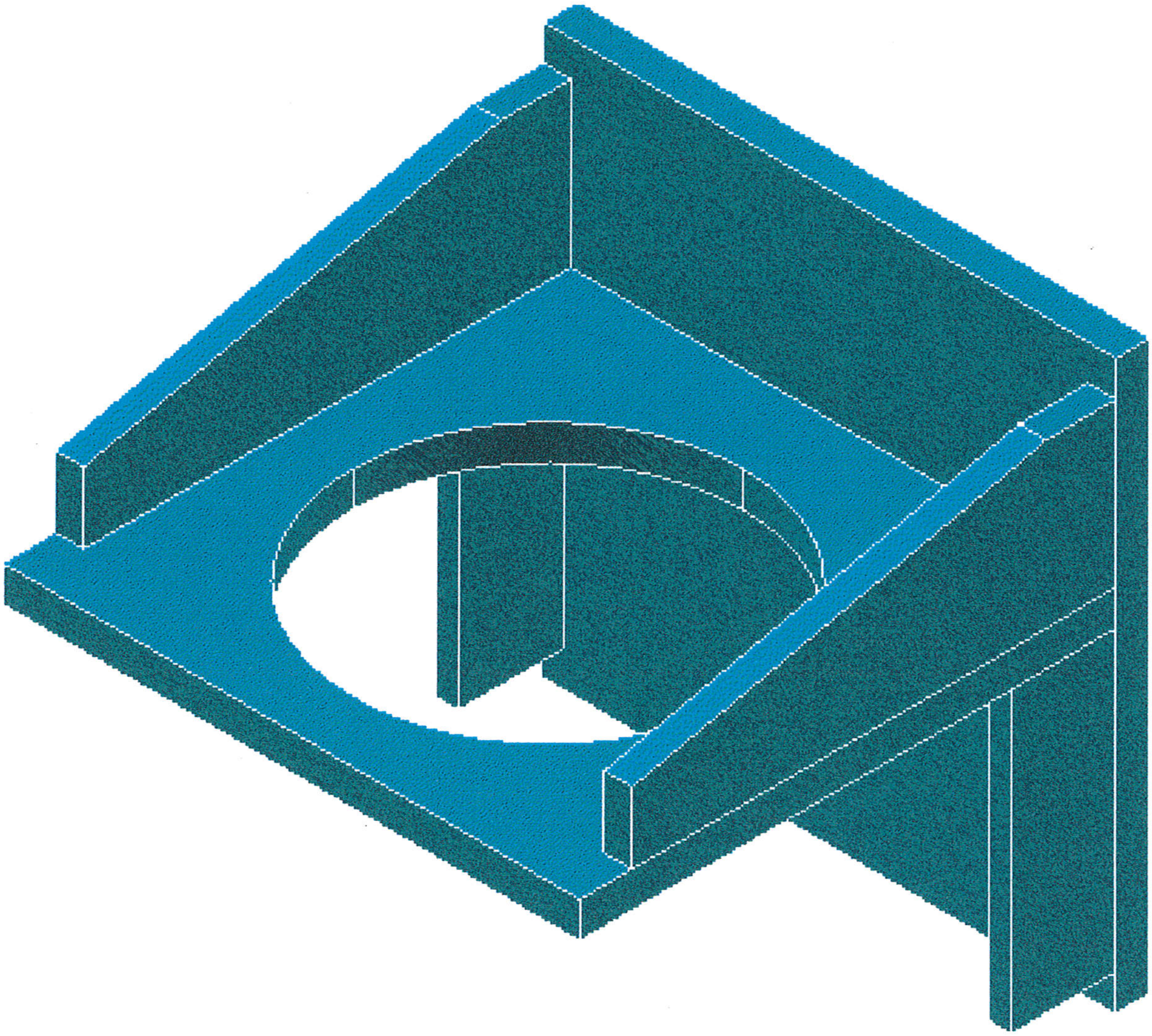


Fig. 1 ***AZIMUTH BRACKET MODEL***

STRUCTURAL ANLYSIS OF AZIMUTH BRACKET:

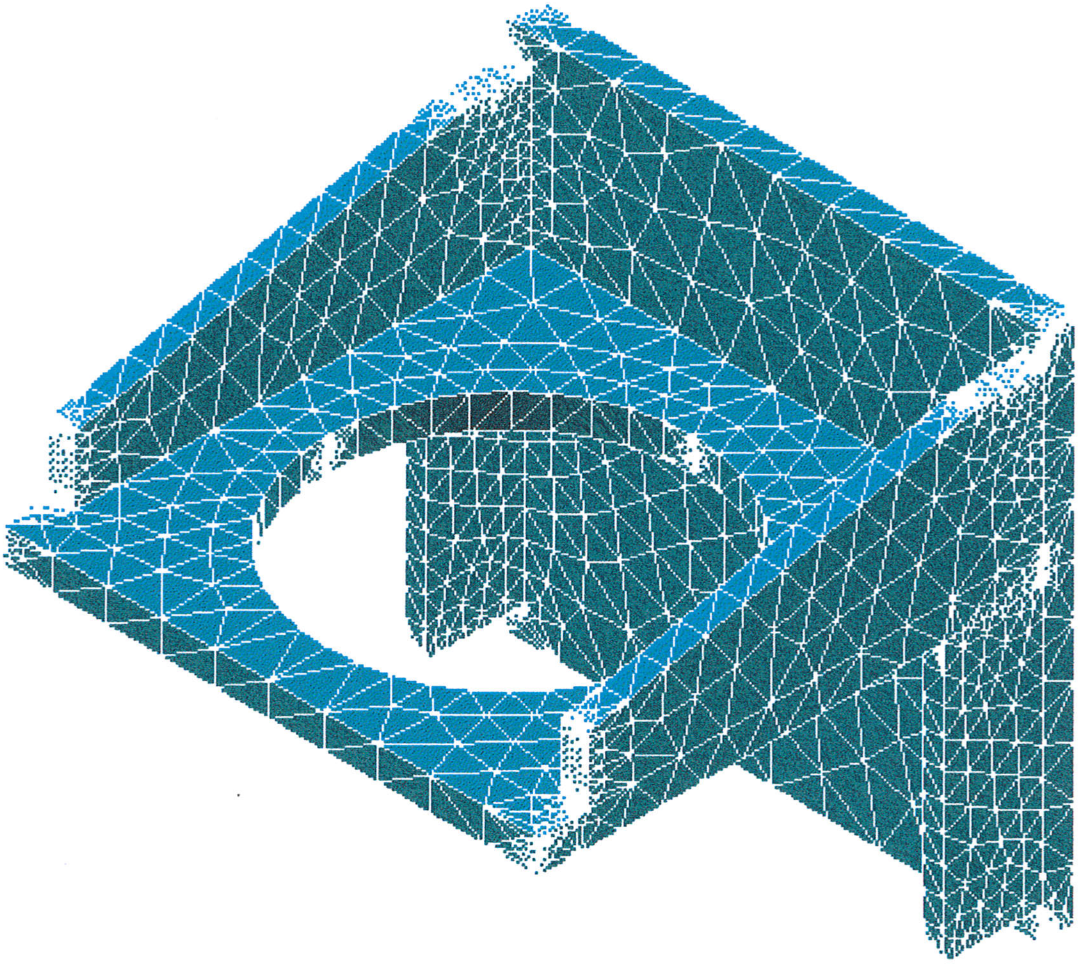


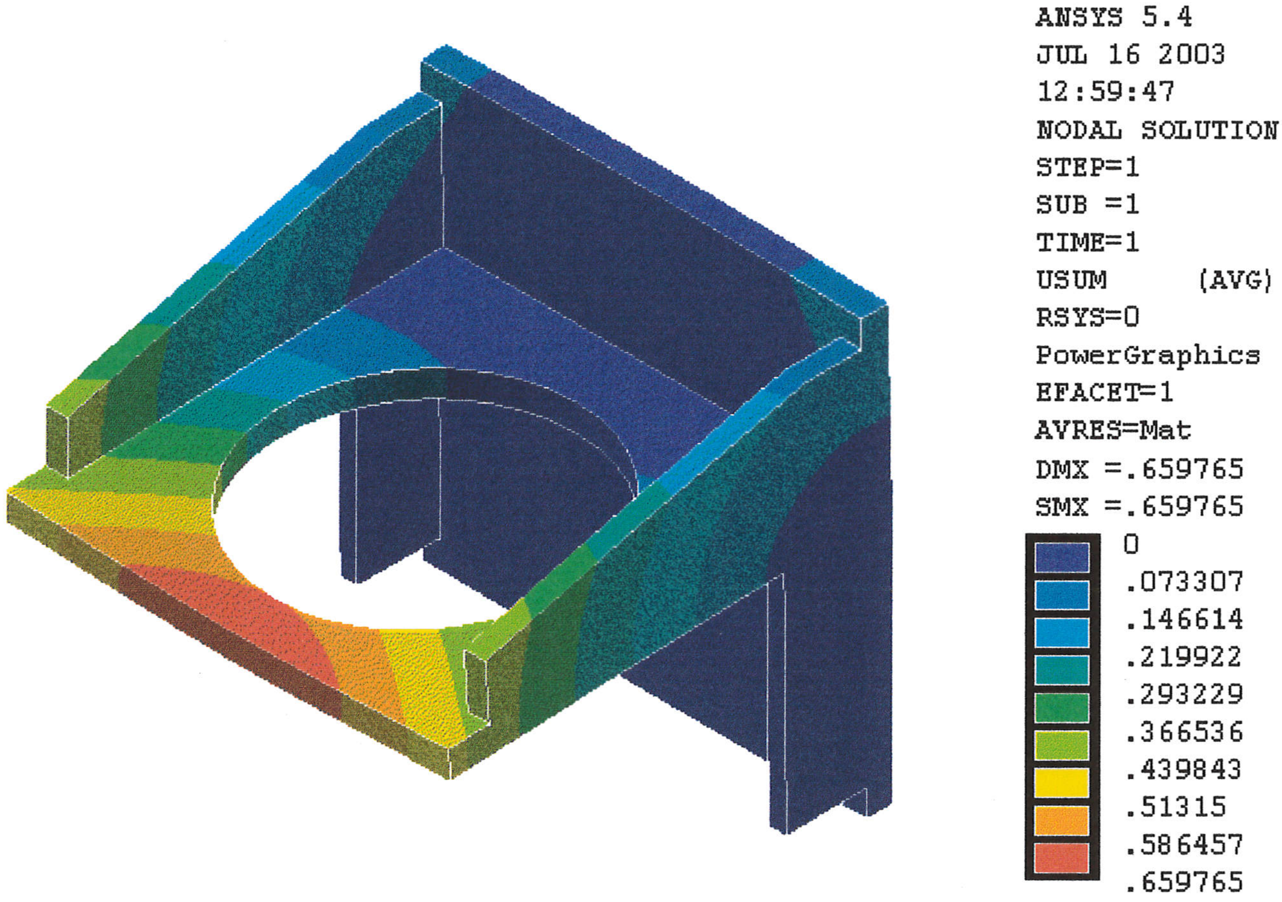
Fig. 2 ***AZIMUTH BRACKET MESH MODEL***

MATERIAL: St – 42

MATERIAL PROPERTY:

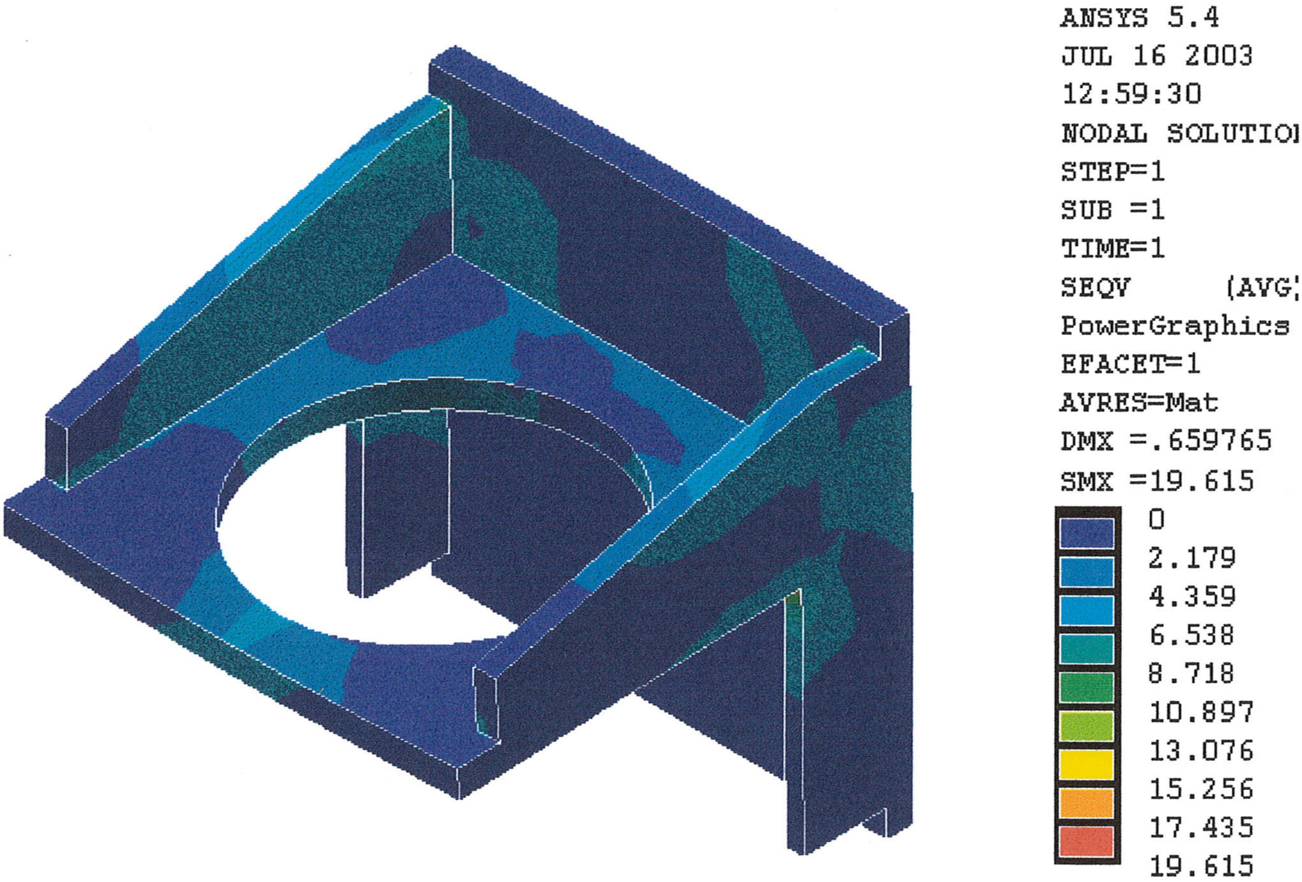
MODULUS OF ELASTICITY	**:**	**2.1×10^4 kg/mm^2**
POISSONS RATIO	**:**	**0.3**

STRUCTURAL ANLYSIS OF AZIMUTH BRACKET:



RESULT: DEFLECTION = **0.659 mm**

<u>STRUCTURAL ANLYSIS OF AZIMUTH BRACKET:</u>



RESULT: STRESS = **19.615 kg/mm^2**