ASTRI PROJECT

CTA - CHERENKOV TELESCOPE PROTOTYPES

MECHANICAL DESIGN DESCRIPTION

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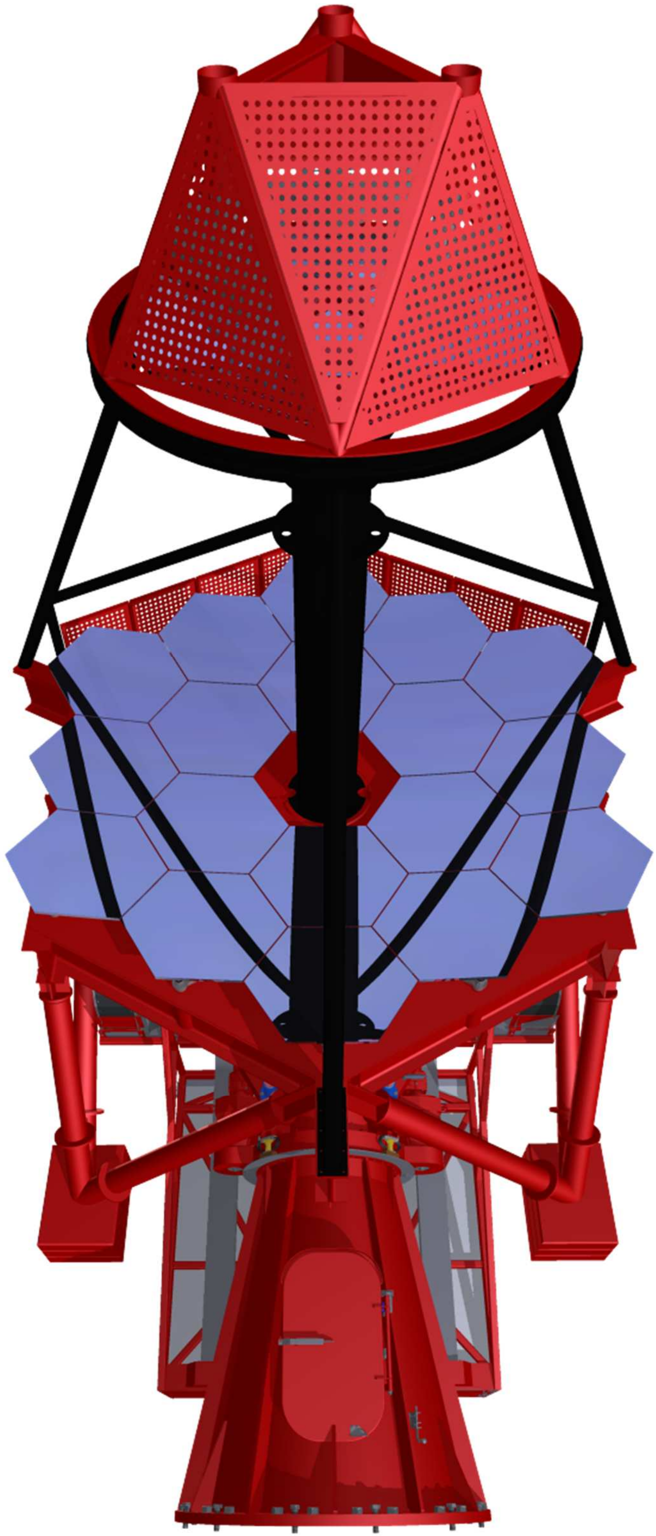

Istituto Nazionale di Astrofisica
OSSERVATORIO ASTRONOMICO DI BRERA

CHANGE RECORDS						
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TABLE OF CONTENTS

1. INTRODUCTION

ASTRI (Astrofisica con Specchi a Tecnologia Replicante Italiana) is a flagship project of the Italian Ministry of Education, University and Research (MIUR) and it is strictly linked to the manufacturing of the first real precursor of the CTA (Cherenkov Telescope Array) international project.

CTA plans the construction of many tens of telescopes for Gamma Ray observation, divided in three configurations, in order to cover the energy range from a tens of GeV (Large Size Telescope, LST), to a tens of TeV (Medium Size Telescope, MST), and up to 100 TeV (SST).

Within this framework, GEC has in charge the manufacturing of the ASTRI CTA small-size telescope prototypes, whose mount exploits the classical altitude-azimuth configuration, with a dual-mirror Schwarzschild-Couder optical design. The proposed telescope layout is fully compliant with the CTA requirements for the SST array. The telescope design is compact, with a 4.3 m diameter segmented primary mirror, a 1.8 m diameter monolithic secondary mirror and a primary-to-secondary distance of 3 m. The major parts of each telescope are the mount, composed by the base and the fork, and the optical supporting structure, which includes the primary mirror support, the Mast, the secondary mirror support and the balancing counterweights.

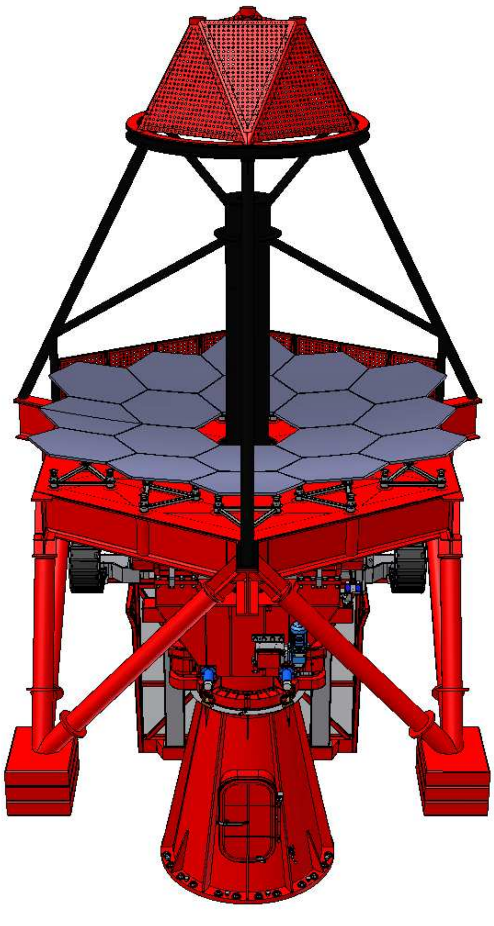


Figure 1-1 ASTRI CTA-SST front view

The construction of the prototypes has been scheduled to follow three subsequent phases: The design and supply of the electrical subsystem, factory pre-assembly and testing of all structural, electrical and electro-mechanical subsystems, disassembly into pre-assembled main parts, transportation and installation on the final site, commissioning and acceptance activities are in the first phase scope. The second phase will foresee a reassessment and design study: GEC will perform a revision of the design based on the lessons learnt with the first prototype activities. The last phase will lead to the construction, factory pre-assembly, tests, disassembly, and packing of the mini-array telescopes, in order to be ready for the delivery on site.

The ASTRI first prototype has been installed and tested successfully at Serra La Nave, 1735 m a.s.l. on the Etna Mountain near Catania (Sicily-Italy), at the INAF observing station “M. G. Fracastoro”, while the second series of telescopes will constitute the SST mini-array of INAF, to be placed at final CTA Southern Site.

This document outlines the entire design of the telescope mount based on the experience gathered on the prototype. The description is divided in two main parts which respectively describes the following:

- structural parts,
- mechanisms.

In the second part, the analyses and verifications have been included.

2. APPLICABLE AND REFERENCE DOCUMENTS

2.1. APPLICABLE DOCUMENTS

AD1	ASTRI-SPEC-GEC-3100-000	Technical Requirements
AD2	ASTRI-SPEC-IASFMI-3000-007	Requirement document on Structure (Addendum)
AD3	ASTRI-ICD-IASFMI-3000-008	Interface Control Document
AD4	ASTRI-PLA-OAPD-3000-003	ASTRI quality plan

2.2. REFERENCE DOCUMENTS

RD1	OAB.010.004.4	Executive Design Description
RD2	ASTRI_BCV_Spec_01_3	Telescope Structure - Technical Specification
RD3	ASTRI-DES-IASFMI-1000-001	DM-SST Design Concept

2.3. LIST OF ACRONYMS

AD	Applicable Document
EIE	EIE Group
RD	Reference Document
AIP	Assembly Inspection Point
AMCU	Active Mirror Control Unit
ASTRI	Astrofisica con Specchi a Tecnologia Replicante Italiana
AZ	Azimuth axis
BUS	Back-Up Structure
CG	Center of Gravity
COTS	Common Off The Shelf
CTA	Cherenkov Telescope Array
EL	Elevation axis
FEA	Finite Element Analysis
GEC	Galbiati EIE Consortium
ICD	Interface Control Document
INAF	Istituto Nazionale di Astrofisica
M1	Primary Mirror
M2	Secondary Mirror
MAC	Motor Axis Controller
MIP	Manufacturing Inspection Point
PSU	Power Supply Unit

SST	Small Size Telescope
TBD	To Be Defined
TCH	Telescope Control System
TCU	Telescope Control Unit
THCU	Telescope Health Control Unit
TM	Telescope Mount
TMA	Telescope Mount Assembly
UPS	Uninterrupted Power Supply

3. STRUCTURAL ITEMS DESIGN DESCRIPTION

3.1. MOUNT ASSEMBLY

Mount Assembly provides the support for the optical support structure. It includes two main structural elements:

- The Base
- The Azimuth fork structure

The Base is fixed to the foundation and the Azimuth fork is installed on base top through the Azimuth bearing, by virtue of which, the Azimuth relative rotation between the two elements is possible.

Material used is common S355J0 painted steel; only shields are made of aluminium for weight purposes.

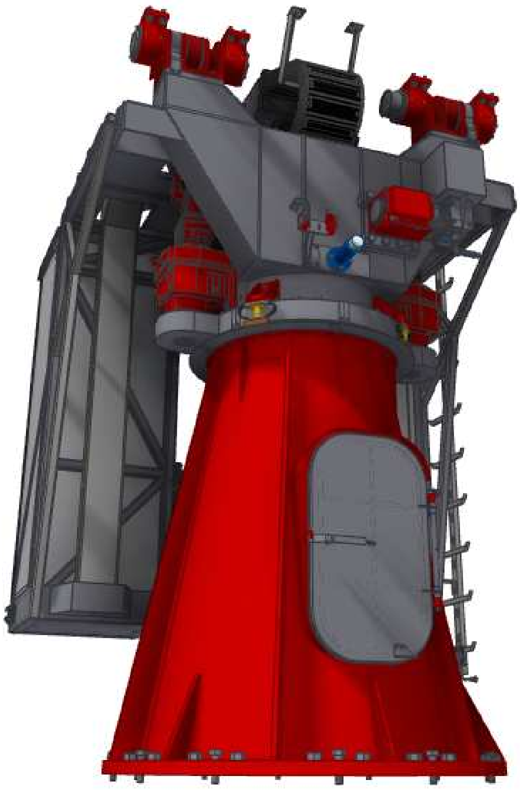


Figure 3-1 Mount Assembly view

The base coordinate system (fixed) foresees:

- X axis is horizontal and parallel to EL axis (towards East),
- Y axis is horizontal and oriented opposite to the base access door side (towards North),
- Z axis oriented vertically towards up, coincident with AZ axis.

The fork coordinate system (AZ axis) foresees:

- X axis is horizontal and parallel to EL axis oriented towards EL encoder side,
- Y axis is horizontal and oriented towards the electrical cabinets side,
- Z axis oriented vertically towards up, coincident with AZ axis.

3.1.1. Base

The Base main structure basically consists in a cone which links the foundation to the Azimuth bearing. For these reasons the interfaces imposed by the foundation anchor bolts and the bolts required by Azimuth bearing are essential to establish the correct shape of this item. The cone shape guarantees the best compromise between proper transmission of the loads of the telescope to the foundation and the lowest mass necessary. Ribs are implemented to distribute loads in the desired way and at the same time to improve deformation stability and stiffness where required.

Figure 3-2 Base view

Furthermore, it guarantees to provide access to the volume contained inside; this aspect is essential for providing access to the items installed inside of the base for their sensitivity to the weather agents (for example: azimuth encoder, cables, azimuth switches etc.). Thanks to that, alignment and maintenance procedures are possible and at the same time can be carried out safely.

Figure 3-3 Mount Assembly section view

The access door is the key for safe operations, as it is connected to the interlock chain by means of a proper safety switch. To be sure that, the key can be brought back to its stowing place only with door closed, there will be a safety sign close to the door with written: “Do not remove locker key if door is open – No remover la llave si la puerta està abierta”.

Figure 3-4 Base door detail

The structure is studied in details in order to provide the alignment of every single item it hosts. In particular for the azimuth encoder the interfaces provide to install simple aligning devices which simplify drastically the mounting sequence and time.

The bush of the stow pin is located in Azimuth position angle of -90° and it consists in a sheet of me tal able to bear the survival conditions imposed.

3.1.2. Azimuth fork main structure

The Azimuth fork main structure is one of the key structural element; it provides support and interfaces for these following essential subsystems:

- Azimuth bearing;
- Elevation bearings;
- Azimuth ratio-motor drives;
- Elevation actuator drive;
- Azimuth and Elevation stow pins;
- Azimuth and Elevation switches;
- Azimuth cable wrap and Elevation cable wraps;
- Elevation bumpers;
- Support for the electrical cabinets;

Figure 3-5 Fork structure

These items grant the motion of the telescope mount main axes as well as the telescope safety during survival conditions under personnel and hardware points of view. These are the reasons that make this item so critical for the entire telescope structure.

Also for the Azimuth fork main structure are present all the alignment devices described in previous section and presented in the pictures here below:

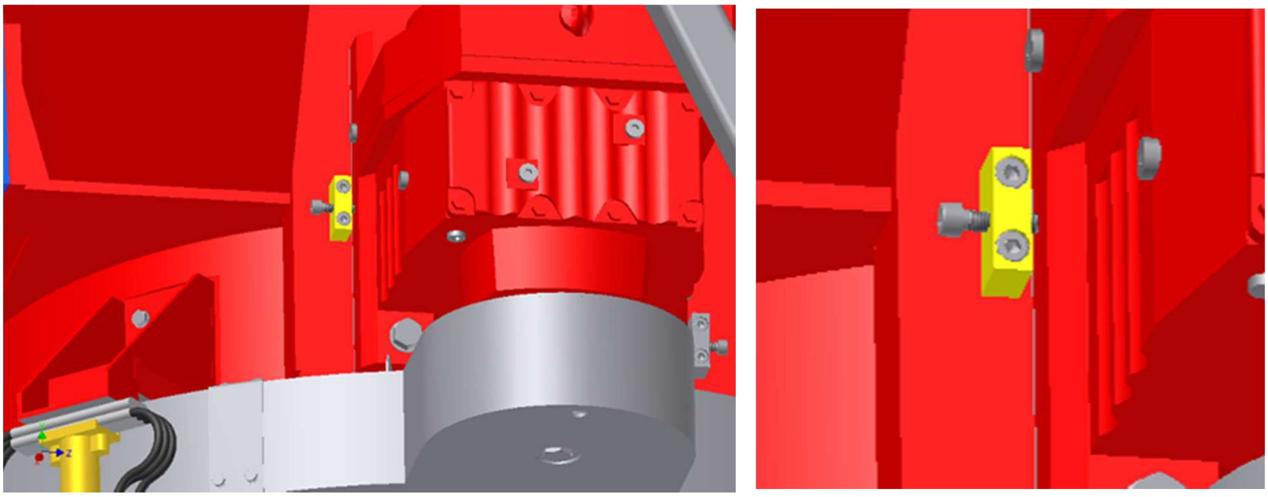


Figure 3-6 Alignment devices

Elevation cable wraps are the key to provide power and communication for all telescope subsystems; in this contest, the azimuth fork main structure has a key role also in the cable routing as its shape has been defined in order to host cable trays and ways. The cabinets support frame has the role to provide interface for the electrical cabinets and at the same time grant the support for the cable trays and Elevation cable wraps.

Figure 3-7 Cabinets support view

3.2. OPTICAL SUPPORT STRUCTURE

The optical support structure includes: M1 dish, Mast, Central tube and the M2 BUS. The design concept of these items has been heavily changed with respect to the design of the prototype.

The design work performed for these item regards the completion of the assemblies with all the standard parts, such as bolts and nuts, bearings, springs etc. Manufacturing details were included in the drawings and the aspects of mounting and maintenance were added in order to grant proper means to carry out these operations (e.g. additional flanges for handling and maintenance).

The OSS coordinate system foresees:

- X axis is horizontal parallel to EL axis,
- Y axis is horizontal and oriented in order to point towards up when telescope is at EL=0°,
- Z axis oriented coincident with optical axis towards up when telescope is at EL=90°.

Figure 3-8 Optical Support Structure view

3.2.1. M1 Dish

The primary mirror structure concept has been reviewed entirely with respect to the prototype one, in order to:

- Reduce weight.
- Simplify its construction using standard steel profiles.
- Allow actuator supports which follow a radial symmetry.

The Dish is split in two halves to grant its transportability within the standard sizes.

The hexagon shape of the dish has been maintained although it has been rotated of 30 degrees in order to reduce the number of mast from four to three and consequently simplify the M2 BUS.

The structure main beams are three: one vertical and the other two are at 60 degrees from the vertical one, to provide support to the mast while actuators interfaces have reinforcing ribs to give stiff response and guarantee the adequate stability to keep the mirror segments in their position. Interfaces to elevation bearings and elevation actuator are very similar to the prototype ones.

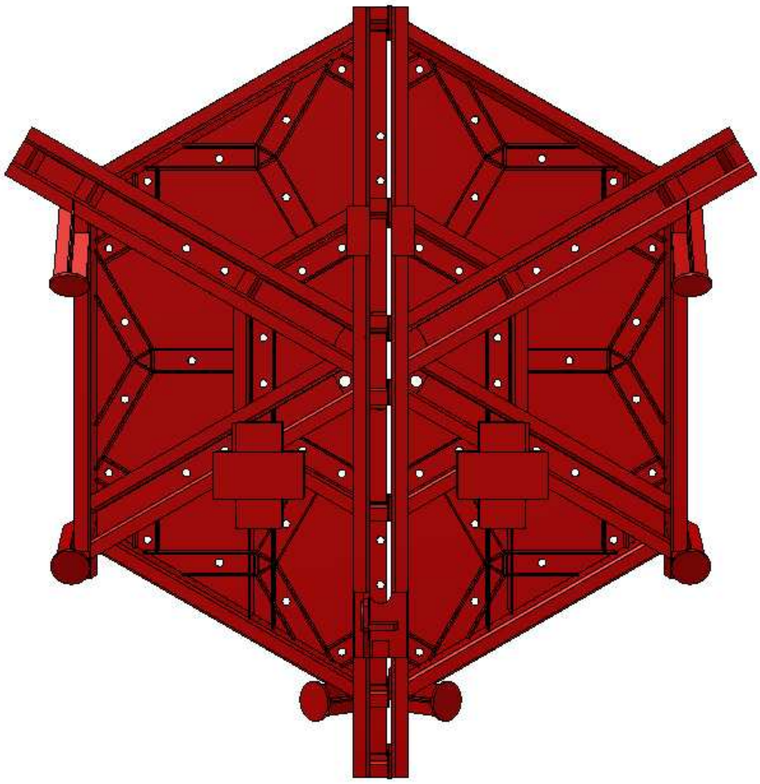


Figure 3-9 M1 dish structure rear view

The central tube interface remain unchanged even if two holes are provided for the electrical and fluids supply to the camera; fluids are necessary for the CHEC camera only.

The elevation stow pin now engage the dish only in the elevation angle position of 0 degree; for this purpose a dedicated welded structure is fixed by means of bolts and nuts. The concept here is to have a stiffer behavior as well as an easier manufacturability.

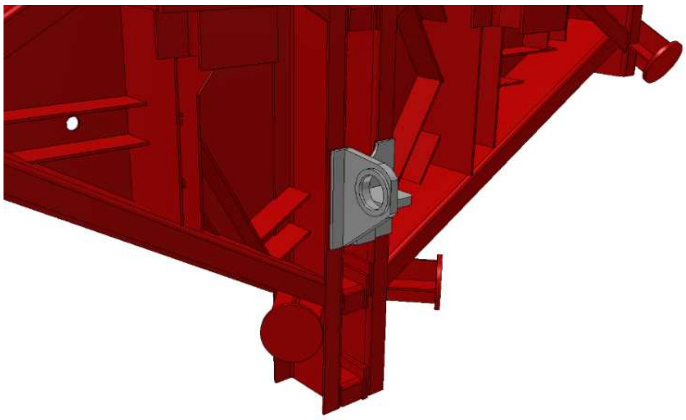


Figure 3-10 Elevation stow pin bush

Finally, a group of shields made of perforated aluminium sheets are located on the upper two sides of the hexagon providing snow and ice protection to the M1 supports when the telescope is in its parking position. The sides are naturally sloped to ease snow and ice evacuation.

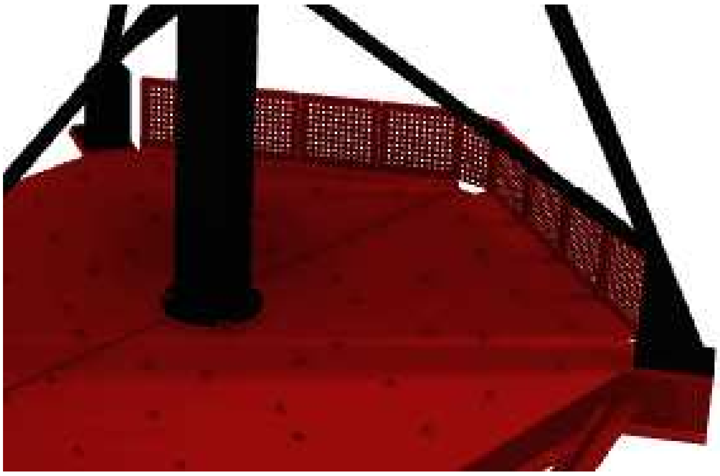


Figure 3-11 M1 shields

3.2.2. Counterweights

The Counterweights supporting beams have been reviewed in order to construct them with a single plane angle instead of double angle flanges. This eases the manufacturing process and improve at the same time the operations during the installation phases. The principle of installation of the masses remains unchanged with some aligning pins that are made to ease installation. Threaded rods allow to install fine balancing masses.

Figure 3-12 Counterweights

3.2.3. OSS Upper Structure

As outlined in the previous sections, the mast is provided with three pipes support systems spread at 120 degrees to each other. The real reason to this design choice is to provide a much simpler M2 BUS and a better stability of its interface. The top ring guarantees to have a single stiff structure that improves repeatability during M2 BUS removal and/or installation. In this way, alignment is also easier. Finally, the top ring provides also some sort of baffle for the M2 even if its main function has been described earlier.

Figure 3-13 OSS Upper structure with mast and central tube

3.2.4. M2 Back-Up Structure

The structure designed to host the support suitable for M2 is simplified with respect of the prototype, with a triangular layout which tries to avoid as much as possible complication during manufacturing.

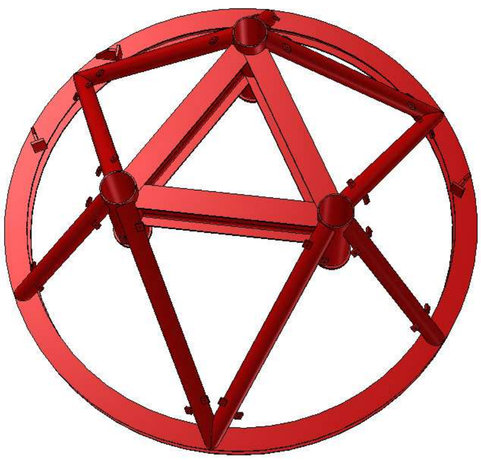


Figure 3-14 M2 Back-Up Structure

Basically, there is a M2 BUS ring which mates with OSS upper structure top ring and an upper triangle which is designed to support the M2 actuators. Some structural pipes are positioned in order to grant the best load distribution configuration from the mast to the actuators interfaces.

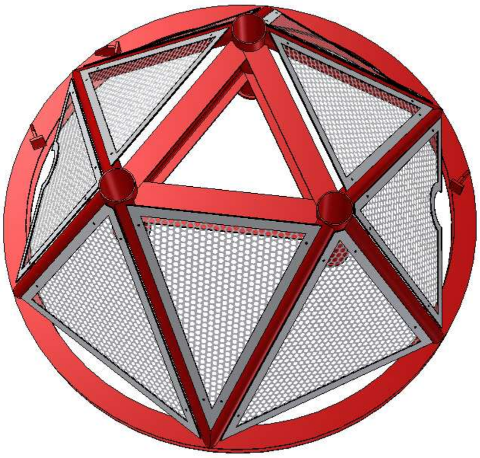


Figure 3-15 M2 shields

To protect the M2 actuators and load-spreaders from wind and snow some shields made of perforated aluminium sheets are provided.

4. MECHANISMS DESIGN DESCRIPTION

The description of the mechanisms present in the ASTRI Telescope is subdivided following the product tree. Having that in mind, the description will outline Mount Assembly (3110-000) including all sub-systems and Optical support structure (3120-000).

4.1. MOUNT ASSEMBLY

Mount assembly represent that part of the Telescope which hosts all motion and safety systems. Here below is reported an excerpt of the product tree which helps to understand the location of the sub-systems:

Level 1		Level 2		Level 3	
3110-000	Mount Assembly	3111-000	Base structure		
				3111-100	Base
				3111-200	AZ encoder
		3112-000	AZ bearing		
		3113-000	AZ fork	3113-100	AZ fork structure
				3113-200	EL axis bearing
				3113-300	AZ & EL Drives
				3113-400	EL encoder
				3113-500	AZ & EL switches
				3113-600	AZ & EL cable wraps
				3113-700	AZ & EL Stow pins
				3113-800	EL bumpers

The following picture provides the locations and the volumes occupied by all sub-systems.

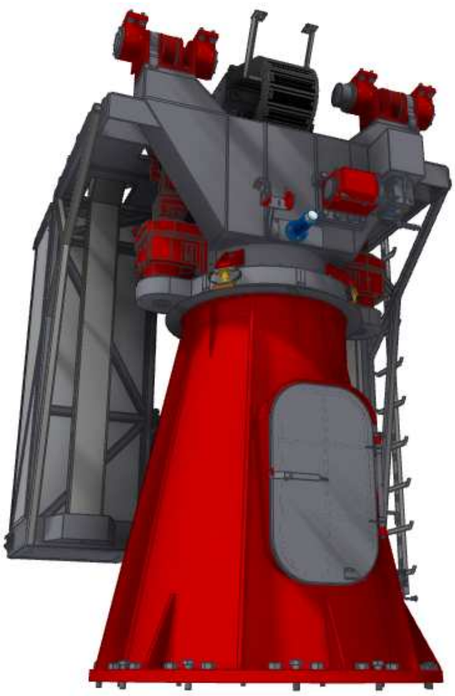


Figure 4-1 Populated Mount Assembly

Pictures highlight that Azimuth fork hosts the largest number of subsystem, that to optimize the cable length and routing. The Base, hosts only the Azimuth encoder and the witnesses necessary for the Azimuth switches to minimize the cable going through the cable wraps, whether in Azimuth or Elevation.

4.1.1. Azimuth encoder

The Azimuth encoder includes two main sub-systems: the steel scale tape and a group of scanning heads. In this case, scanning heads are 4 and are displaced along a circle at 90° one from the others. The tape slot is machined in the Azimuth bearing and the scanning heads are installed in proper supports which grant a complete adjustability along all 3 axes.

Under the reliability of reading point of view, the large scanning field for these type of Heidenhain encoders additionally reduces sensitivity to contamination. In many cases this can prevent encoder failure. Even if the contamination from printer's ink, PCB dust, water or oil is up to 3 mm in diameter, the encoders continue to provide high-quality signals.

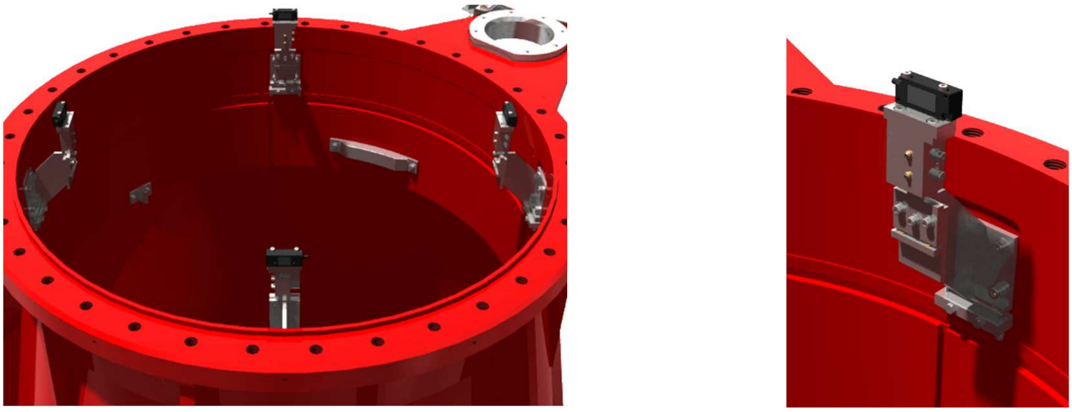


Figure 4-2 Azimuth encoder scanning heads installation and support detail

4.1.1.1. Azimuth encoder calculation

The scanning heads number comes out from the error budget carried out by Heidenhain which takes into account the geometry of the system, the run-out of the Azimuth bearing and other aspects which are intrinsic of this application such as mounting errors etc.

A first trade-off study has been carried out by Heidenhain between configuration with 2 and 4 scanning heads. The configuration with 2 scanning heads seemed to be insufficient to guarantee the requested accuracy for the application.

The Error budget has been provided by Heidenhain for the configuration with 4 scanning head and the results are outlined in the following figure:

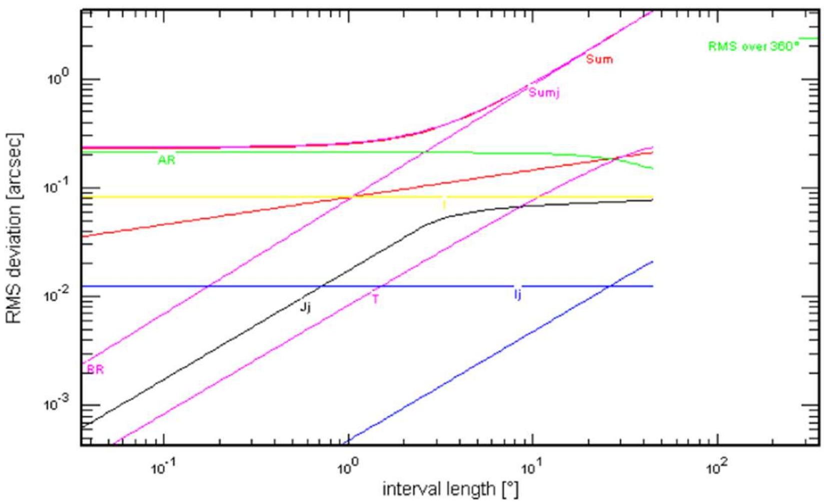


Figure 4-3 Accuracy without online interpolation error compensation

As reported on this graph the system RMS accuracy over 360° (marked in green) is equal to about 2.3 to 2.4arcsec. It improves to 1arcsec in a stroke of about 10°. The tape slot diameter foreseen for this a pplication is 713.24mm.

4.1.2. Azimuth bearing

Azimuth bearing is a custom slewing bearing made by SKF (RKS) on the basis of a standard one. The elements which permit the rotation are cylindrical crossed rollers. This choice was made in order to satisfy the stiffness requests by the application.

The outer face of the ring fixed on the Azimuth base upper flange is provided with a rack in order to permit Azimuth axis rotation by means of two pinion driven by the Azimuth drives.

The inner face, the one fixed on the Azimuth fork lower flange is provided with a proper slot in order to allow encoder tape installation.

4.1.2.1. Azimuth bearing calculation

SKF has provided a series of analyses which gave the following essential characteristics:

- Lifetime > 800.000 revolutions
- Static friction torque 1577Nm (@ 40℃), 2360Nm (@ -5℃)
- Dynamic friction torque 1027Nm (@ 40℃) and 1536Nm (@ -5℃)
- Rack nominal diameter 984mm, module 8mm, teeth number 123,

With the lifetime provided by SKF, it is possible to verify if that is compatible with Telescope requests.

Considering that:

- A typical Telescope Mount nightly survey operation consists of approximately 5 sources observations.
- Each source observation is performed with 20 minutes of tracking followed by a 0.5° slew and settle.
- These operations are repeated typically 4 times before the source change.
- The source change may consist in the large pointing movement.
- On average, the Telescope Mount will operate 8 hours a night, every night. In general, the duty cycle during daytime maintenance will be negligible.
- Lifetime is equal to 30 years

And assuming that:

- Every pointing slew is equal to the worst case (180°)
- Every 20 nights only one is not suitable for observations (e.g. cloudy, raining etc.)
- Elevation angle during observation is in its worst position for this case (89.2°) so to have the maxi mum Azimuth axis speed

The calculation provides:

	formula	value	unit	note/assumptions	
Number of sources	N=	5			
Repositioning per source	n=	4			
Slewing angle per repositioning	α=	0.5	°		
Time tracking per repositioning	t_t=	20	min		
Average elevation angle	EL=	89.2	°	worst case	

Tracking speed	ω_t=360/24/60/60/cos(EL)=	0.30	°/s =	1074.33	arcsec/s
Tracking angle per repositioning	β=ω_t*t_t*60=	358.11	°		
Slewing per pointing	s=	180	°	worst case	
Total angle per night	δ=N*s+N*n*(α+β)=	8072.21	°		
Percentage of observation night	p=	0.95	%	1 night every 20 is cloudy	
Total angle per year	δ_y=365*p*δ=	2799037	°		
Total revolutions per year	rpy=δ_y/360=	7775.10			
Lifetime	L=	30	years	imposed 30yrs	
Total revolutions per lifetime	δ_L=	233,253			
Bearing lifetime revolutions	L_b=	800,000			
Bearing lifetime safety factor	b=L_b/δ_y=	3.43		Verified	

Which guarantee lifetime compliance of this item for this application bearing in mind that proper maintenance need to be performed for the overall lifetime length.

4.1.3. Elevation axis bearings

The minor changes performed on this item provided only the details necessary for maintainability, assembly and feasibility. The concept of this sub-assembly remains very similar to the prototype as there are two couples of conical preloaded bearings. The size of the bearings are big compared to the loads involved for this application. Moreover, these bearings are used in the Elevation actuator hinges in order to improve availability and reduce at the same time type of COTS used in the telescope.

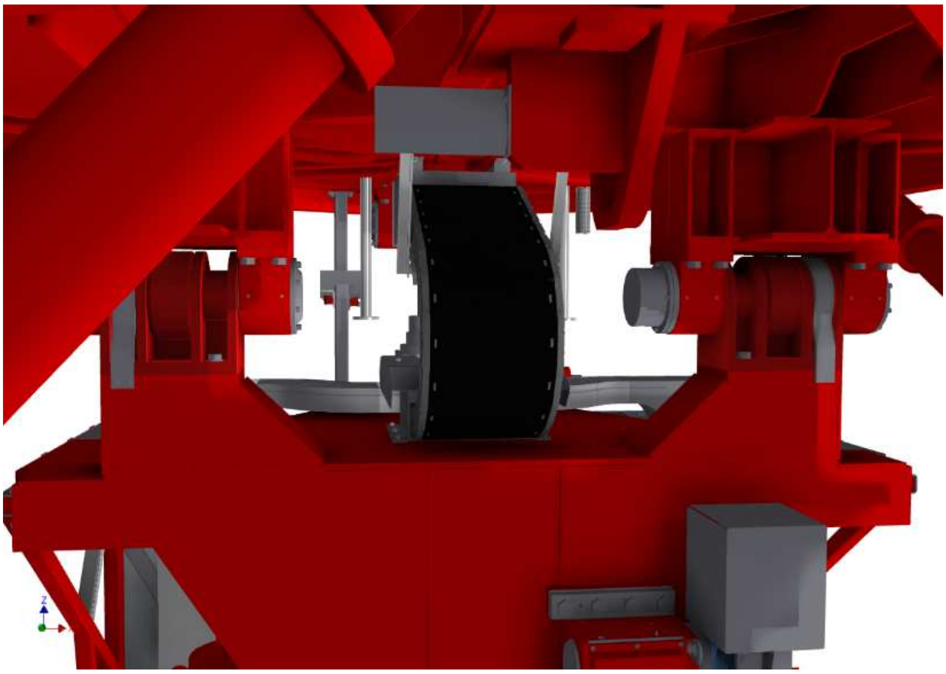


Figure 4-4 Elevation bearings view

4.1.3.1. Elevation bearing calculation

To assess the loads for the elevation bearings the wind load pressure at 170km/h is calculated and the gravity is taken into account. To have margins these loads will be applied to one single bearing, while in reality the bearings are 8 in 4 preloaded couples. The calculations are outlined here below:

	formula	value	unit	note
Wind load				
Temperature	T=	-15	℃	From survival conditions
Altitude over sea level	h=	2150	m	
Reduced performance wind speed	v=	47.25	m/s =	170.1 km/h
Gas constant	R=	0.0821	l atm/mol K	
Air molecular mass	m=	28.6	g/mol	
Pressure at altitude	$P=(1-(0.0065/288.15*h)^{5.25588}=$	0.77	atm	
Density at T and P	$\rho=P*m/R/(273.15+T)=$	1.04	kg/m^3	
Dynamic pressure due to wind	$q=\rho/2*v^2=$	1159.97	N/m^2	
Shape load coefficient	$c_p=$	1.5		pg57 EuroCode1
M1 Area	$A_{up}=$	12.57	m^2	estimation at EL=0°
EL wind torque	$T=q*c_p*A_{up}=$	21864.87	N	
Loads				
Mass	M=	15000	kg	Mass of telescope without base
Gravity load	G=M*g=	147150	N	
Load	P=(G^2+T^2)^0.5=	148765.57	N	
Bearing lifetime Verification				
Bearing speed	n=	1	min^{-1}	
Axial Load	$F_a=T=$	21864.87	N	A safe assumption is to impose axial bearing load equal to frontal wind load
Radial load	$F_r=P=$	148765.57	N	
Axial/Radial loads ratio	$F_a/F_r=$	0.15		
Dynamic load coefficient	e=	0.43		From SKF catalogue (SKF 32216T78J2/QDBC110)
Coefficient Y_1	$Y_1=$	1.60		From SKF catalogue (SKF 32216T78J2/QDBC110)
Coefficient Y_2	$Y_2=$	2.30		From SKF catalogue (SKF 32216T78J2/QDBC110)
Equivalent dynamic load for 4 points bearing	P=	183749.37	N	From SKF catalogue (Page 612 SKF catalogue for load formulas)
Dynamic load	C=	391000	N	From SKF catalogue (SKF 32216T78J2/QDBC110)
Static load	$C_0=$	490000	N	From SKF catalogue (SKF 32216T78J2/QDBC110)
Type of bearing exponent	p=	3.33		Exponent 3 for ball bearing (roller 10/3)
Reliability factor	$a_1=$	0.21		From SKF catalogue (Table 1 page 53)
Fatigue load limit	$P_u=$	57000	N	From SKF catalogue (SKF 32216T78J2/QDBC110)

	formula	value	unit	note
Contamination factor	η_c=	0.5		From SKF catalogue (Table 4 page 62)
Ratio to obtain correction factor	η_c*P_u/P=	0.15510		
Outer bearing diameter	D=	140	mm	From SKF catalogue (SKF 32216T78J2/QDBC110)
Inner bearing diameter	d=	80	mm	From SKF catalogue (SKF 32216T78J2/QDBC110)
Medium bearing diameter	$d_m=(D+d)*0.5$=	110	mm	
Requested lubrication viscosity	v_1=	6000	mm^2/s	From SKF catalogue (Diagram 5 - Page 60)
Operational temperature	T=	10	℃	From Environmental condition
Lubricant viscosity	v=	460	mm^2/s	From SKF catalogue (Diagram 6 - Page 61). Lubricant: ISO VG680
Viscosity ratio	$\kappa=v/v_1$=	0.0767		
Correction factor	a_{SKF}=	0.1		From SKF catalogue (Diagram 2 - Page 55)
Lifetime	$L_{nm}=a_1*a_{SKF}*(C/P)^p$=	0.26		Millions of rounds
Maximum turn per motion	tpm=	1.14		Actuator motion
Observable nights percentage	nig=	0.95		5 nights every 100 cloudy
Turns per year	T_y=tpm*nig*365=	394.91		
Lifetime in years	$L_y=L_{nm}/T_y$=	659		
Lifetime safety factor	Lsf=Ly/30=	22	>1	Verified
Minumum preload for functioning	$F_{am}=0.017*C$=	6647.00	N	
Verification of minimum working conditions	F_r/F_{am}=	22.38	>1	Verified
Stiffness of a single bearing				
Estimated rolling elements	d_w=	15	mm	
Estimated rolling element ultimate stress	σ_u=	1500	Mpa	
Estimated rolling elements yielding stress	σ_y=	1000	MPa	
Young modulus	E=	210000	MPa	
Yielding strain (σ0.2)	$\varepsilon=\sigma_y/E$=	0.00476		
Yielding deformation	$f=d_w*\varepsilon$=	0.0714	mm	
Safety factor for elastic behaviour	sf_e=	1.5		
Estimated stiffness	$k_b=C_0/f/sf_e/1000$=	4573	MN/m	

4.1.4. Azimuth drive

The Azimuth axis motion is granted by two ratio-gear motors located at 180° one from the other. Moreov er there is a differential torque between them in order to provide a preload which aim to guarantee precision during motion during all operational conditions.

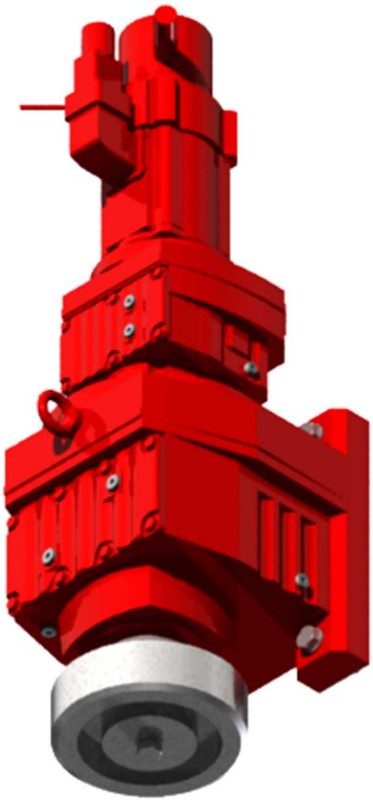


Figure 4-5 Azimuth motors group view

The ratio-gear used for this specific item is SEW R97R57 with a motor SEW CM71S equipped with brake (braking torque equal to 5Nm), manual brake release and emergency shaft in order to be driven with a battery drill in case of power failure.

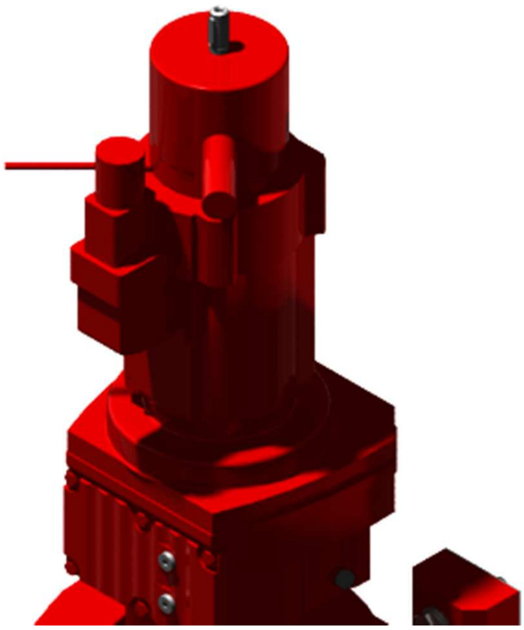


Figure 4-6 Detail of Azimuth motor brake, hand release and emergency shaft

4.1.4.1. Azimuth drive calculation

To proceed with selection of standard items SEW Catalogue was used in order to obtain all the characteristics of the ratio motor gear.

Considering:

- The worst peak load with inertia during acceleration transient, maximum wind speed of 50km/h, static friction torque, preload torque. This condition applies when motors need to reach stow position in the worst possible scenario. (It must be noted that, during motion, the friction torque will be lower and that preload increase the stress in one motor but decrease it in the other)
- Lifetime is equal to 30 years.
- Azimuth axis speed equal to 4.5deg/s.
- Azimuth axis acceleration equal to 2deg/s^2 in operational conditions and 1deg/s^2 in transition conditions.

Assuming:

- Every 20 nights only one is not suitable for observations (e.g. cloudy, raining etc.)

	formula	value	unit	note	
Torque on AZ axis					
Rotating Inertia	$J=$	66640.5	kgm^2	from 3D @ EL:0°+ 50%	
Azimuth bearing rack diameter	$D_r=$	984	mm		
Pinion diameter	$D_w=$	256	mm		
Diameters ratio	$i_w=$	3.84			
Tracking minimum angular speed	$\omega_{trk}=$	4.17E-03	deg/s =	0.0001	rad/s
Max. angular speed	$\omega_{max}=$	4.75	deg/s =	0.0785	rad/s
Proximity angular speed	$\omega_{prox}=$	1	deg/s =	0.0175	rad/s
Max angular acceleration	$\alpha_{max}=$	1	deg/s^2 =	0.0175	rad/s^2
Pinion speed	$n_2=\omega_{max}*i_w*60/360=$	2.88	min^{-1}		
Peak inertia Torque	$T_i=J*\alpha_{max}=$	1163	Nm		
Operational wind torque (36km/h)	$T_{wo}=$	1535	Nm	From OAB.010.004.4	
Reduced performance Torque (60km/h)	$T_{wr}=$	4264	Nm	From OAB.010.004.4	
Static friction torque	$T_{fs}=$	2447	Nm	Extrapolated from SKF (@-10°C)	
Dynamic friction torque	$T_{fd}=$	1593	Nm	Extrapolated from SKF (@-10°C)	
AZ axis maximum reduced performance Torque	$T_{dw}=T_i+T_{wr}+T_{fs}=$	7874	Nm		
Ratiogear selection					
Nr of units	$n=$	2			
Nr of units used in emergency	$n_e=$	1			
Efficiency of Rack & Pinion	$\eta_m=$	0.950		(Direct motion)	
Differential torque to preload	$T_p=1.2*(T_{fs+}T_{wo})/n/\eta_m/i_w=$	654	Nm	20% of margin introduced	
Torque on wheel axis with preload	$M_{dw}=T_{dw}/i_w/\eta_m/n+T_p=$	1732	Nm		

	formula	value	unit	note	
Preload percentage wrt to T_{dw}	Preload%=	60.69	%		
Max motor speed	n_{max}=	2000	min^{-1}		
Approximate speed ratio	$I_a=n_{max}/n_2$=	657			
Max Trasmissible Torque	T_t=	3000	Nm	SEW R97R57	
Speed ratio	i_{rg}=	549			
Ratio gear efficiency	η_r=	0.902		(Direct motion)	
Reversibility verification	$\eta_{rinv}=2-1/\eta_r$=	0.892		(Inverted motion)	
Minimum motor torque	$m_{min}=M_{dw}/i_{rg}$=	3.156	Nm		
Stiffness of ratiogear					
Stiffness 1st stage ratiogear	k1=	155	Nm/ arcmin	532850.7495	Nm/ rad
Ratio 1st stage ratiogear	i1=	25.03			
Stiffness 2nd stage ratiogear	k2=	25	Nm/ arcmin	85943.66927	Nm/ rad
Ratio 2nd stage ratiogear	i2=	21.93			
Total stiffness at ratiogear slow shaft	$k_{tot}=(1/k_1+1/(k_2*i_1^2))^{-1}$=	527629.21	Nm/rad		
Stiffness at slewing bearing	$k_{sb}=k_{tot}*n*i_w^2$=	15590825	Nm/rad		
Motor selection					
Nominal motor torque	m_o=	5.0	Nm	SEW CM71S/BR/HR/KTY/RH3L/SB50	
Motor inertia (brake included)	J_{motb}=	0.000672	kgm^2	1.4181	kgm^2
Motor inertia (without brake)	J_{mot}=	0.000499	kgm^2	1.0530	kgm^3
Motor efficiency	η_{mo}=	0.980			
Standstill torque	M_0=	7.300	Nm		
Standstill current	I_0=	3.200	A		
Motor speed	$n_{mot}=n_2*i_{rg}$=	1670.59	min^{-1}		
De-rating factor	D_f=	0.61	%		
Max power during pointing	$P=m_{min}*2*\pi*n_{mot}/60/\eta_{mo}/D_f$=	923.49	W =	0.79	kW
Motor torque threshold for ratio gear safety	$T_{limited}=T_t/i_{rg}/\eta_m/\eta_r/\eta_{mo}$=	6.50	Nm	An overcurrent threshold is NOT needed	
Motor differential current	$I_p=I_0/M_0*T_p/i_{rg}/\eta_{mo}$=	0.53	A	BIAS of current to apply to motors with opposite sign to obtain preload	
Brakes					
Brake Torque limited to	T_b=	5.00	Nm		
Maximum energy per brake operation	E_b=	22.00	kJ		
Response time for braking	t_{res}=	0.006	s		
Energy to be stopped in emergency	E=	229.01	J		
Energy brake safety factor in emergency	$a_E=E_b*1000/E/n_e$=	96.07	>1	Verified	

	formula	value	unit	note	
Time for braking	$t_{br}=t_{res}+J*\omega_{max}/(n_e*T_b*i_w*i_{rg})=$	0.53	s		
Angle of braking	$\beta=(J*\omega_{max}^2/(2*T_b*i_w*i_{rg})*180/\pi)+\omega_{max}*t_{res}=$	1.27	°		
Braking deceleration	$\alpha_{br}=\omega_{max}*(t_{br}-t_{res})=$	9.07	°/s^2		
Distance to M2	$d_{M2}=$	4127.40	mm		
Braking acceleration to M2	$\alpha_{M2}=\alpha_{br}*\pi/180*d_{M2}/9.81=$	0.07	g		
Average proximity speed during braking	$\omega_{pavg}=\omega_{prox}/2=$	0.50	°/s =	0.0087	rad/s
Time for proximity braking	$t_{pbr}=t_{res}+J*\omega_{pavg}/(n_e*T_b*i_w*i_{rg})=$	0.06	s		
Angle of proximity braking	$\alpha_{pbr}=\omega_{prox}*(t_{pbr}-t_{res})=$	0.03	°		
Ratiogear verification					
Average mass acceleration factor	$M_a=((J)*(i_{rg}*i_w)^{-2})/(J_{mot}+J_{motb})=$	12.780	(<20, goal 10)	Verified	
Daily operating time	$t_{day}=$	8	hours/ day		
Starting frequency	$Z=$	3		number of start-stop/hour	
Service factor	$f_b=$	1.400		Severe shock load (curve III) due to M_a (Page 32 SEW catalogue)	
Gear torque during motion	$T_{gdw}=f_b*M_{dw}=$	2425	Nm		
Ratiogear torque safety factor	$a_{rg}=T_t/T_{gdw}=$	1.24	>1	Verified	
Overhung load	$F_R=$	13534.86	N		
Overhung load on gear unit output	$F_{Ra}=$	16900	N	from SEW catalogue (page 38)	
Length a	$a=$	255.5	mm	from SEW catalogue (page 38)	
Length b	$b=$	195.5	mm	from SEW catalogue (page 38)	
Admissible bending moment	$c=$	1190000	Nmm	from SEW catalogue (page 38)	
Length f	$f=$	0	mm	from SEW catalogue (page 38)	
Diameter d	$d=$	60	mm	from SEW catalogue (page 38)	
Length l	$l=$	120	mm	from SEW catalogue (page 38)	
Overhung load application point wrt shaft	$x=$	86.5	mm		
Admissible load according to service life	$F_{xL}=F_{Ra}*a/(b+x)=$	15311.88	N		
Admissible load according to shaft strength	$F_{xW}=c/(f+x)=$	13757.23	N		
Overhung load safety factor	$a_{Ra}=\min(F_{xL}, F_{xW})/F_R=$	1.02	>1	Verified	

It must be said that with 36km/h of operational wind a_{rg}=1.55 and a_{Ra}=1.27. In that case, the required torque for the motor will be 2.523Nm which is half of its nominal torque (safety factor of 2).

4.1.5. Elevation drive

The motion in Elevation axis is possible with a preloaded ball screw jack driven by a brushless motor.

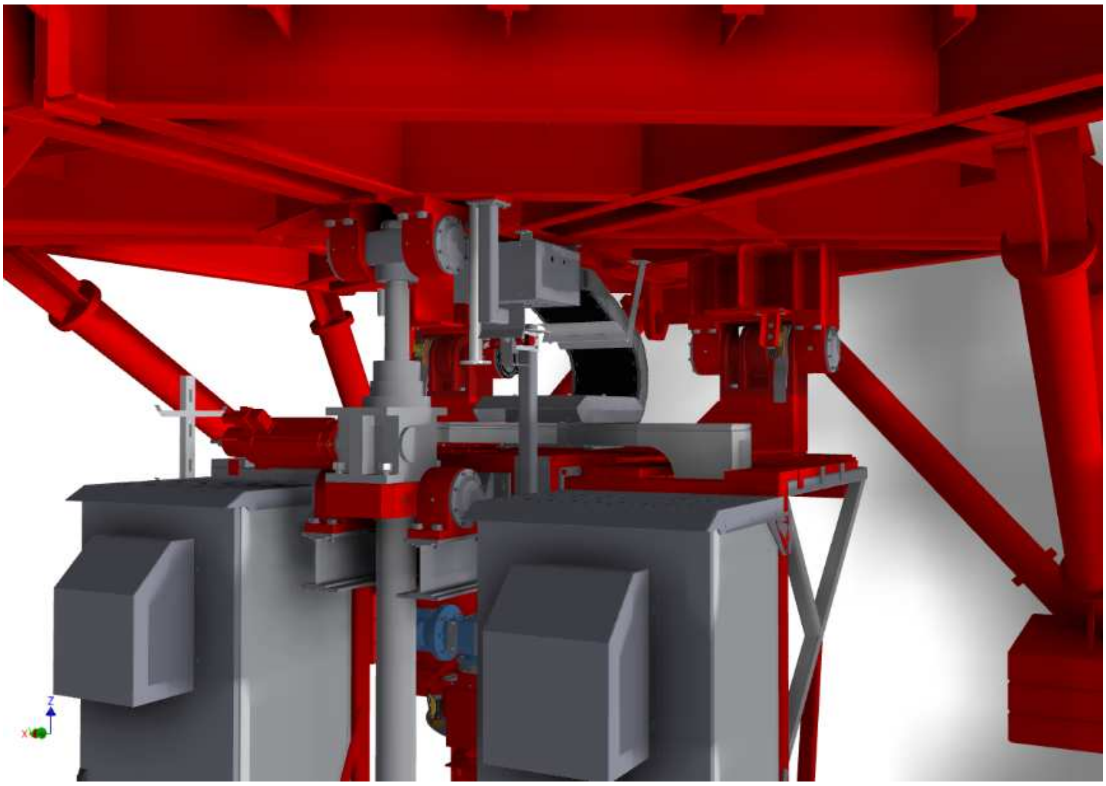


Figure 4-7 Elevation drive view

The preloaded ball screw jack is fixed by means of hinges one located on the top of the ball screw and the other installed on the ball screw jack body. This allows to provide motion of the M1 dish and thus, all Optical support structure. The ball screw is fully deployed with Elevation angles close to horizon and completely retracted when Elevation angles are close to zenith position. To protect the ball screw from weather agents when it is deployed, a bellow is provided; since the screw is passing through the jack, a protection tube hosts it when the ball screw is fully retracted.

The servomotor used for this application is a SEW CM90M equipped with brake and hand release in order to provide safe operation and no damages for people and hardware. The brake stops any motion, the hand release, used with the 2° emergency shaft, permits to move the ball screw jack manually with the aid of a portable electrical drill.

4.1.5.1. Elevation drive calculation

Considering that:

- A typical Telescope Mount nightly survey operation consists of approximately 5 sources observations.
- Each source observation is performed with 20 minutes of tracking followed by a 0.5° slew and settle.
- These operations are repeated typically 4 times before the source change.
- The source change may consist in the large pointing movement.
- On average, the Telescope Mount will operate 8 hours a night, every night. In general, the duty cycle during daytime maintenance will be negligible.
- Lifetime is equal to 30 years.

And assuming that:

- Every pointing slew is equal to the worst case (60°)
- Every 20 nights only one is not suitable for observations (e.g. cloudy, raining etc.).

Stroke of 1936mm for Elevation motion from 90° to - 1°.

	formula	value	unit	note	
Number of sources	N=	5			
Repositioning per source	n=	4			
Slewing EL angle per repositioning	α=	0.5	°		
Time tracking per repositioning	t_t=	20	min		
Tracking speed	ω_t=	0.0042	°/s	15 arcsec/s	
Tracking EL angle per repositioning	$\beta=\omega_t*t_t*60$=	5	°		
EL slewing per pointing	s=	60	°	worst case	
Total EL angle per night	δ=N*s+N*n*(α+β)=	410	°		
Percentage of observation night	p=	0.95	%	1 night every 20 is cloudy	
Total angle per year	$\delta_y=365*p*\delta$=	142167.5	°		
Total stroke per year	$spy=\delta_y/360$=	1579.64			
Stroke	s=	1936	mm		
Lifetime	δ_L=	30	years		
Total ball screw jack stroke per lifetime	$L_b=spy*\delta_L*s/1000$=	91,745	m	92	km
Ball screw jack lifetime stroke	L_{screw}=	10000	km		
Bearing lifetime safety factor	$b=L_b/\delta_y$=	109	>1	Verified	

The estimation of the wind torque on EL axis is carried out as per following:

	formula	value	unit	note	
Temperature	T=	-15	°C		
Altitude over sea level	h=	2150	m		
Reduced performance wind speed	v=	16.67	m/s =	60.012	km/h
Gas constant	R=	0.0821	l atm/mol K		
Air molecular mass	m=	28.6	g/mol		
Pressure at altitude	$P=(1-(0.0065/288.15*h)^{5.25588}$=	0.77	atm		
Density at T and P	ρ=P*m/R/(273.15+T)=	1.04	kg/m^3		
Dynamic pressure due to wind	$q=\rho/2*v^2$=	144.38	N/m^2		
Shape load coefficient	c_p=	1.5		pg57 EuroCode1	
CoP above EL axis	CoP_{up}=	2000.00	mm	estimated	
Area above EL axis	A_{up}=	18	m^2	estimation at EL=30°	
Moment of area wrt EL axis	$M_{ALT}=(A_{up}*CoP_{up}/1000-A_{dw}*CoP_{dw}/1000)$=	36.00	m^2m		
EL wind torque	$T_{ALT}=q*c_p*M_{ALT}$=	7796.64	Nm		

The calculation and verifications necessary for the items included in this application regard the ball screw jack (SERVOMECH Catalogue) and the brushless motor (SEW Catalogue). Finally a verification of the hinge bearings (SKF Catalogue) is provided to provide confidence of good functioning all over telescope lifetime.

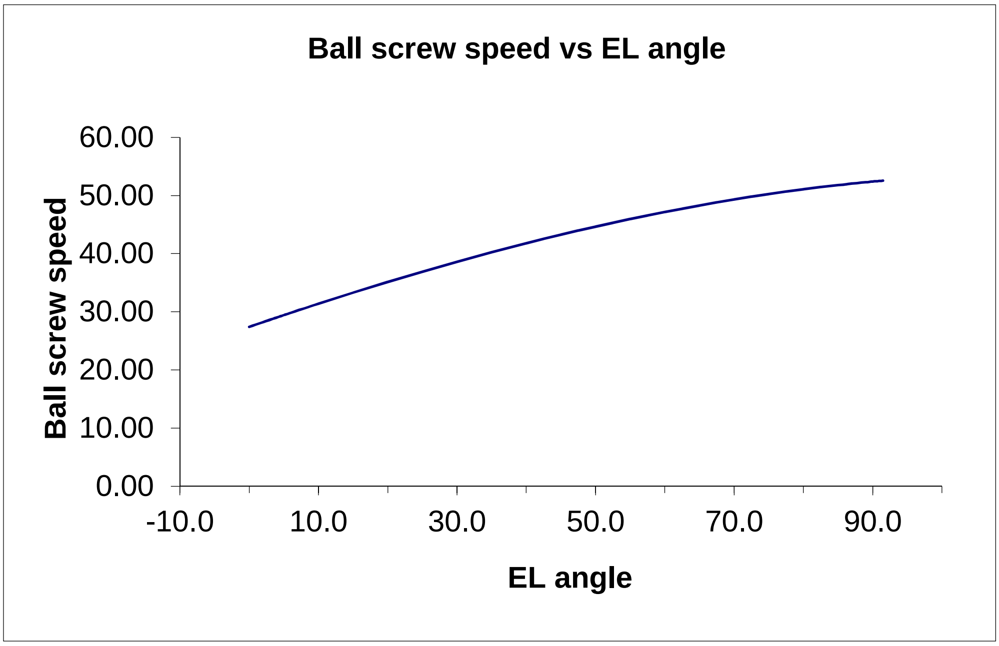
	formula	value	unit	note	
Torque on EL axis					
Rotating Inertia	$J=$	52864.5	kgm^2	from 3D + 50% of margins	
Tracking angular speed	$\omega_{trk}=$	4.17E-03	deg/s =	0.0001	rad /s
Max. angular speed	$\omega_{max}=$	2	deg/s	0.0349	rad /s
Max angular acceleration	$\alpha_{max}=$	1	deg/s^2	0.0175	rad/s^2
Peak intertia Torque	$T_i=J*\alpha_{max}=$	923	Nm		
Operational wind torque	$T_{wo}=$	2807	Nm	from Wind loads (36km/h)	
Reduce performace wind Torque	$T_{wr}=$	7796	Nm	from Wind loads (60km/h)	
Static friction torque	$T_{fs}=$	62	Nm		
Kinetic friction torque	$T_{fk}=$	0	Nm		
Unbalancing torque	$T_u=$	6481	Nm	50% of margin added to wind (@36km/h) extrapolated SPEC @ EL=60° (/cos(30))	
Maximum Torque against unbalancing	$T_{dw}=T_i+T_{wr}+T_{fs}+T_u=$	15078	Nm		
Maximum Torque helped by unbalancing	$T_{dw}=T_i+T_{wr}+T_{fs}-T_u=$	2115	Nm		
Jack selection					
Jack stroke	$s=$	1939	mm		
Jack arm @EL=0°	$b_0=$	785	mm		
Jack arm @EL=90°	$b_{90}=$	1546	mm		
Jack load @EL=0°	$F_0=T_{dw}/b_0*1000=$	19207.13	N		
Jack load @EL=90°	$F_{90}=T_{dw}/b_{90}*1000=$	9752.65	N		
Nr of units	$u=$	1			
Distance from EL axis and lower actuator hinge	$a_1=$	1589.51	mm	from EL axis study	
Distance from EL axis and upper actuator hinge	$a_2=$	1525.09	mm	from EL axis study	
Angle between EL axis and hinge pos. @EL=90°	$\theta=$	29.41	°	from EL axis study	
Jack speed @EL=0°	$v_0=$	27.41	mm/s	See EL act. speed sheet (deriving pos. Equation $(4852430.993-4848281.61*\cos(29.41+EL))^{0.5}$)	
Jack speed @EL=90°	$v_{90}=$	52.55	mm/s	See EL act. speed sheet (deriving pos. Equation $(4852430.993-4848281.61*\cos(29.41+EL))^{0.5}$)	
Minimum screw rpm @EL=0°	$n_{j0}=v_0/p*60=$	164.46	min^{-1}		
Maximum screw rpm @EL=90°	$n_{j90}=v_{90}/p*60=$	315.31	min^{-1}		
Ball screw turn EL angle ratio @EL=0°	$i_0=n_{j0}/\omega_{max}*360/60=$	493.37			

	formula	value	unit	note	
Ball screw turn EL angle ratio @EL=90°	$i_{90}=n_{j90}/\omega_{max}*360/60=$	945.93			
Ball screw diameter	D=	80		SERVOMECH MS 200 BS 80x10 Mod.A 8 RV Vers.4 U-RH C2537 G-TG T B (MSA spinta?)	
Jack ratiogear ratio	i=	8.00		SERVOMECH Reduction RV	
Ball screw pitch	p=	10	mm	From SERVOMECH (10mm or 20mm)	
Jack efficiency	$\eta_j=$	0.506		SERVOMECH reports 0.92*0.55 (page 59)	
Torque on ball screw @EL=0°	$M_{dw0}=(F_0*p/(2*\pi)+M_{fs0})/\eta_j=$	60.41	Nm		
Torque on ball screw @EL=90°	$M_{dw90}=(F_{90}*p/(2*\pi)+M_{fs90})/\eta_{j=}=$	30.68	Nm		
Motor selection					
Nominal motor torque	$m_o=$	14.5	Nm	SEW CM90M/BR/HR/KTY/RH3L/SB50	
Motor inertia	$J_{mot}=$	0.00272	kgm^2	SEW catalogue page 300	
Motor efficiency	$\eta_{mo}=$	0.980			
Reversibility verification	$\eta_{rinv}=2-1/\eta_j=$	0.024		(Inverted motion)	
Motor speed @EL=0°	$n_0=n_{j0}/i=$	1315.65	min^{-1}		
Motor speed @EL=90°	$n_{90}=n_{j90}/i=$	2522.47	min^{-1}		
Motor tracking speed @EL=0°	$n_{trk0}=n_0*\omega_{trk}/\omega_{min}=$	2.74	min^{-1}		
Motor tracking speed @EL=90°	$n_{trk90}=n_{90}*\omega_{trk}/\omega_{max}=$	5.26	min^{-1}		
Motor torque @EL=0°	$T_{m0}=M_{dw0}/i/\eta_{mo}=$	7.706	Nm		
Motor torque @EL=90°	$T_{m90}=M_{dw90}/i/\eta_{mo}=$	3.913	Nm		
Motor power	P=	2077.043	W		
De-rating factor	dr=	0.61	%		
Effective motor power	$P_{eff}=P/dr=$	3404.99	W =	3.04	kW
Mass acceleration factor @EL=0°	$M_{a0}=((J/u)*i_0^2*(1/i)^2)/J_{mot}=$	1.248	(<20, goal 10)	Verified	
Mass acceleration factor @EL=90°	$M_{a90}=((J/u)*i_{90}^2*(1/i)^2)/J_{mot}=$	0.339	(<20, goal 10)	Verified	
Brakes					
Brake Torque limited to	$T_b=$	20.00	Nm	SEW catalogue page 300	
Maximum energy per brake operation	Eb=	15.00	kJ	SEW catalogue page 300	
Response time for braking	$t_{res}=$	0.035	s	SEW catalogue page 331	
Energy to be stopped	E=	32.21	J		
Energy brake safety factor	$a_E=E_b*1000/E=$	465.74		Verified	
Time for braking @EL=0°	$t_{br0}=t_{res}+J*\omega_{max}/(u*T_b*i_0*i)=$	0.06	s		
Time for braking @EL=90°	$t_{br90}=t_{res}+J*\omega_{max}/(u*T_b*i_{90}*i)=$	0.05	s		
Angle for braking operation @EL=0°	$\beta_0=(J*\omega_{max}^2/(2*T_b*i*i_0)*180/\pi)+\omega_{max}*t_{res}=$	0.09	°		

	formula	value	unit	note	
Angle for braking operation @EL=90°	β_{90}=(J*ω_{max}^2/(2*T_b*i*i_{90}))*180/π) +ω_{max}*t_{res}=	0.08	°		
Braking deceleration @EL=0°	α_{br0}=ω_{max}*(t_{br0}-t_{res})=	85.56	°/s ^{2}	@EL=-1°	
Braking deceleration @EL=90°	α_{br90}=ω_{max}*(t_{br90}-t_{res})=	164.04	°/s ^{2}	@EL=89°	
M2 distance	a_{M2}=	4127.4	mm		
Deceleration load @EL=0°	G $_0$=α_{br0}*π/180*a_{M2}/1000/9.81=	0.63	g		
Deceleration load @EL=90°	G_{90}=α_{br90}*π/180*a_{M2}/1000/9.81 =	1.20	g		
Jack verification					
Tolerable buckling load	F_{buck}=	90	kN	SERVOMECH Catalogue page 14	
Backling safety factor	s_b=F_{buck}*1000/F_{max}=	4.63	>1	Verified	
Jack lifetime travel length	L_{travel}=	92	km		
Jack lifetime with max load	La=	10000	km	Table page 46 SERVOMECH	
Lifetime safety factor	s_L=	109.00	>1	Verified	
Hinge bearing lifetime Verification					
Bearing speed	n=	1	min^{-1}		
Axial Load	F_a=	0.00	N		
Radial load	F_r=	19207.13	N		
Axial/Radial loads ratio	F_a/F_r=	0.00			
Dynamic load coefficient	e=	0.43		From SKF catalogue (SKF 32216T78J2 /QDBC110)	
Coefficient Y_1	Y_1=	1.60		From SKF catalogue (SKF 32216T78J2 /QDBC110)	
Coefficient Y_2	Y_2=	2.30		From SKF catalogue (SKF 32216T78J2 /QDBC110)	
Equivalent dynamic load for 4 points bearing	P=	19207.13	N	From SKF catalogue (Page 612 SKF catalogue for load formulas)	
Dynamic load	C=	319000	N	From SKF catalogue (SKF 32216T78J2 /QDBC110)	
Static load	C_0=	490000	N	From SKF catalogue (SKF 32216T78J2 /QDBC110)	
Type of bearing exponent	p=	3.33		Exponent 3 for ball bearing (roller 10/3)	
Reliability factor	a_1=	0.21		From SKF catalogue (Table 1 page 53)	
Fatigue load limit	P_u=	57000	N	From SKF catalogue (SKF 32216T78J2 /QDBC110)	
Contamination factor	η_c=	0.5		From SKF catalogue (Table 4 page 62)	
Ratio to obtain correction factor	η_c*P_u/P=	1.48382			
Outer bearing diameter	D=	140	mm	From SKF catalogue (SKF 32216T78J2 /QDBC110)	

	formula	value	unit	note	
Inner bearing diameter	d=	80	mm	From SKF catalogue (SKF 32216T78J2 /QDBC110)	
Medium bearing diameter	d_m=(D+d)*0.5=	110	mm		
Requested lubrication viscosity	v_1=	6000	mm^2/s	From SKF catalogue (Diagram 5 - Page 60)	
Operational temperature	T=	10	℃	From Environmental condition	
Lubricant viscosity	v=	460	mm^2/s	From SKF catalogue (Diagram 6 - Page 61). Lubricant: ISO VG680	
Viscosity ratio	$\kappa=v/v_1$=	0.0767			
Correction factor	a_{SKF}=	0.1		From SKF catalogue (Diagram 2 - Page 55)	
Lifetime	$L_{nm}=a_1*a_{SKF}*(C/P)^p$=	483.73		Millions of rounds	
Maximum turn per motion	tpm=	1.14		Actuator motion	
Observable nights percentage	nig=	0.95		5 nights every 100 cloudy	
Turns per year	T_y=tpm*nig*365=	394.91			
Lifetime in years	$L_y=L_{nm}/T_y$=	1224904			
Lifetime safety factor	Lsf=Ly/30=	40830	>1	Verified	
Minumum preload for functioning	F_{am}=0.017*C=	6647.00	N		
Verification of minimum working conditions	F_r/F_{am}=	2.89	>1	Verified	

The study of the system geometry permitted to observe the behaviour of the ball screw speed in function of Elevation angle. This study result is outlined here below:



For what regard the stiffness of the transmission it can be studied in different position in function of the ball screw length (considering negligible bearings effect as their stiffness are an order of magnitude greater):

		@EL=90°	@EL=60°	@EL=30°	@EL=0°		
Young modulus	E=	210000	210000	210000	210000	Mpa	
Poisson module	v=	0.3	0.3	0.3	0.3		
Shaft diameter	d=	43	43	43	43	mm	
Arm (D3113-321-00-00)	l=	39	39	39	39	mm	
Shaft bending stiffness	$K_{sb}=(3*E/l^3)*(\pi*d^4/64)=$	1782340	1782340	1782340	1782340	N/mm	
Shear modulus	G=E/(2*(1+v))=	80769	80769	80769	80769	Mpa	
Shaft shear stiffness	$K_{sh}=G*\pi*d^2/4=$	117293174	117293174	117293174	117293174	N/mm	
Shaft total stiffness	$K_s=(K_{sb}^{-1}+K_{sh}^{-1})^{-1}=$	1755661	1755661	1755661	1755661	N/mm	
Ball screw diameter	d'=	72.856	72.856	72.856	72.856		
Ball screw length	l'=	626.5	1378	2025	2522	mm	
Ball screw stiffness	$K_{sh}'=E*\pi*d'^2/(4*l')=$	1397394	635317	432329	347132	N/mm	
Ball nut stiffness	$K_{sb}'=$	2590	2590	2590	2590	N/µm	(Shuton pg58)
		2590000	2590000	2590000	2590000	N/mm	
Ratiogear wheel diameter	r=	95	95	95	95	mm	
Screw pitch	p=	20	20	20	20	mm	
Ratio gear stiffness at slow axis	$K_{wg}=$	200	200	200	200	Nm/arc min	(Rossi pag76)
		687549354	687549354	687549354	687549354	Nmm/rad	
Ratio gear stiffness at screw	$K_w=K_{wg}*(2*\pi/p)^2=$	67858401	67858401	67858401	67858401	N/mm	
Total stiffness	$k=((2*K_s)^{-1}+K_{sh}'^{-1}+K_{sb}'^{-1}+K_w^{-1}+(2*K_s)^{-1})^{-1}=$	593105	393014	304555	259661	N/mm	

In this case, it is possible to state that stiffness of the actuator can be evaluated in function of the stroke:

$$K_{actuator} = \left(\frac{1}{1030478} + \frac{(l_0 + s)}{875467086}\right)^{-1} \ [N/mm]$$

Where l_0 is equal to 626.5mm and s corresponds to the stroke between 0 and 1895.5mm.

4.1.6. Elevation encoder

The design has been adapted in function of the encoder shape. The encoder is an Heidenhain RCN2580 as per INAF request. The accuracy of this item is 2.5arcsec within a range of temperature of 0℃ to 50℃. Since the encoder is covered, its functioning for the prototype is accepted despite the imposed environmental conditions foresee an operational temperature range of -10℃ to 40℃.

Now, the encoder is located in the inner side of the EL bearing unit in order to be easier to be accessed from the central platform on top of the fork structure.

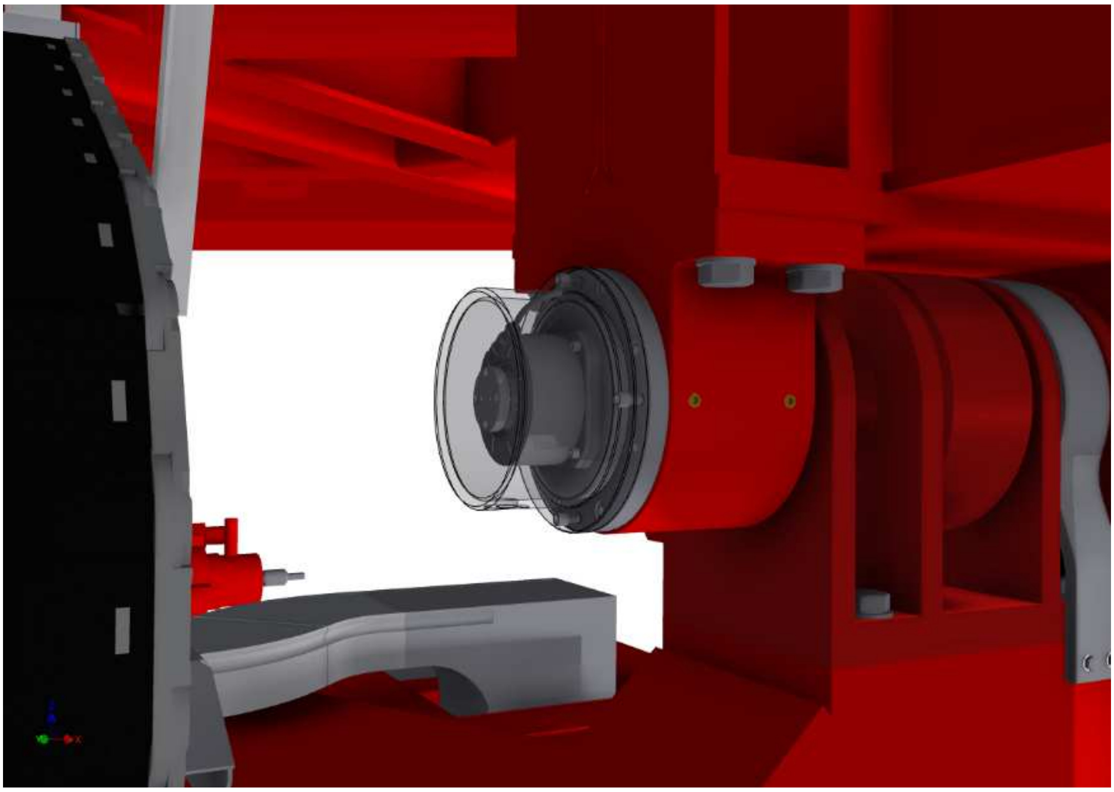


Figure 4-8 Elevation encoder installation detail

4.1.7. Azimuth switches

The Azimuth switches are located in a proper bracket which offers interface for 7 switches. In detail the type of switches are four and listed here below:

- 2 inductive proximity switches located at Azimuth angle ±260° in order to provide a deceleration when telescope is in "proximity" of the Azimuth motion bounds. The speed of Azimuth axis when these switches are engaged is reduced to 1°/s. That is essential to guarantee safety operations for hardware (Azimuth cable wrap) in the range of angles close to the outer limits of Azimuth axis motion.
- 2 directional safety limit switches located at Azimuth angle ±270° which do not allow the drive to continue motion to angle wider than 270°. The motion in the opposite direction is still allowed. These switches are not part of the interlock chain.
- 2 power-off safety limit switches located at Azimuth angle ±272.5° which cut off drives power in order to preserve hardware from damages. Without these cautions the Azimuth cable wrap could suffer damages for excessive twisting. These switches are part of the interlock chain.
- 2 rotation limit switches (the redundancy is necessary for safety reasons) with witness located at 0° which have the function to allow a rotation of more than 360°. This switch has the function of deactivate the positive or negative angle limit switches in order to by pass and ignore their engagement. In particular, when this switch is in "positive" position (positive Azimuth angles) the switches necessary for lower bound (-260°, -270° and -272.5°) are deactivated; when it is "negative" position (negative Azimuth angles) the switches necessary for upper bound (+260°, +270° and +272.5°) are deactivated.

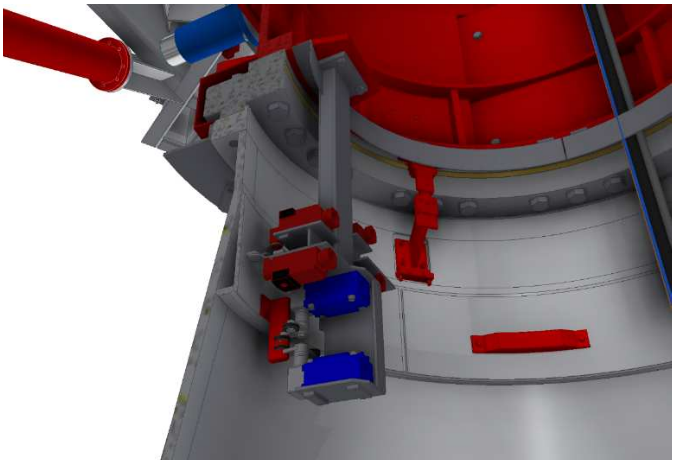


Figure 4-9 Azimuth switches details

All switches supports are provided with a series of systems to adjust their position adequately.

4.1.8. Elevation switches

The Elevation switches are located in a proper blade support which offers interface for 6 switches. In detail the type of switches are three and listed here below:

- 2 inductive proximity switches (PROX) located respectively at Elevation angle +91° and +0.5° in order to provide a deceleration when telescope is in "proximity" of the Elevation motion bounds. The speed of Elevation axis when these switches are engaged is reduced to 1°/s. That is essential to guarantee safety operations for hardware and thus, to avoid unwanted dynamic loads for the optical elements in the range of angles close to the outer limits of Elevation axis motion.
- 2 directional limit switches (DIR) located respectively at Elevation angle +91.25° and -0.1° which do not allow the drive to continue motion to angle wider respectively over +91° and 0°. The motion in the opposite direction is still allowed. The switch at 0° will be adjusted in order to allow stow pin insertion without being engaged. That, to avoid unwanted behaviour with motion to reach correctly stow pin position.
- 2 power-off limit switches (EM) located respectively at Elevation angle +91.5° and -0.35° which cut off drives power in order to preserve hardware from damages. If the motion continues over -1° Elevation bumper start to be engaged.

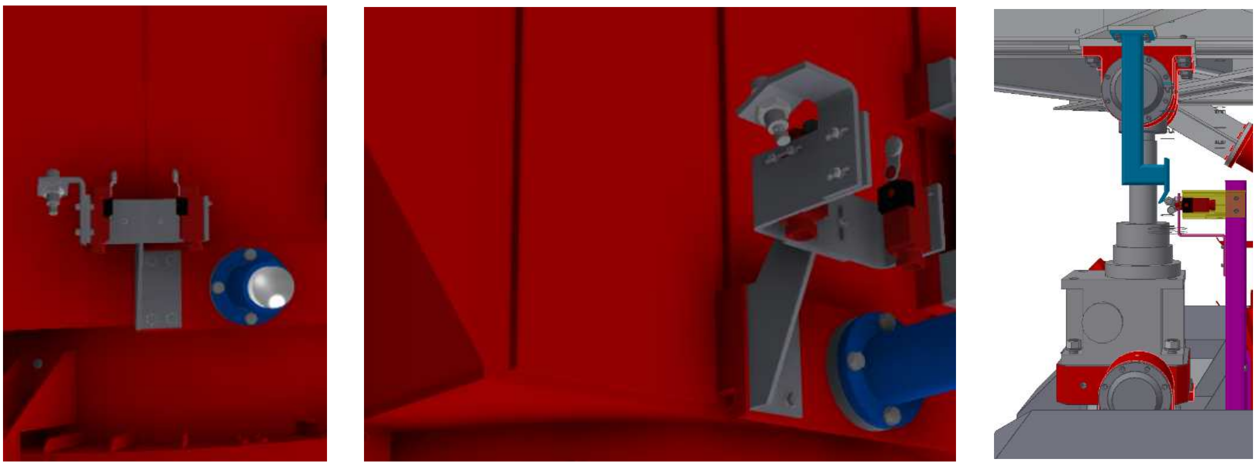


Figure 4-10 Elevation switches details

Here below is reported a summary of the Elevation limit switches plus the stow and bumper positions:

	BMP END	BMP ENG	EM 00	DIR 00	STOW	PROX 00	RANGE	PROX 90	DIR 90	EM 90
Angle	-0.8	-0.4	-0.35	-0.1	0	0.5		91	91.25	91.5
Speed (°/s)		0	0.1	0.1	0.1	0.1	2	0.1	0.1	
Acceler. (°/s ^{2})				0	0	-8	1	-8	0	
Interlock			Yes							Yes

4.1.9. Azimuth cable wrap

The “transport” of the power and communication cables from the Base to the Azimuth fork is guaranteed by a festoon hanged in the Azimuth fork and fixed in the Base with a surplus of length in order to allow the motion without damages due to excessive twisting.

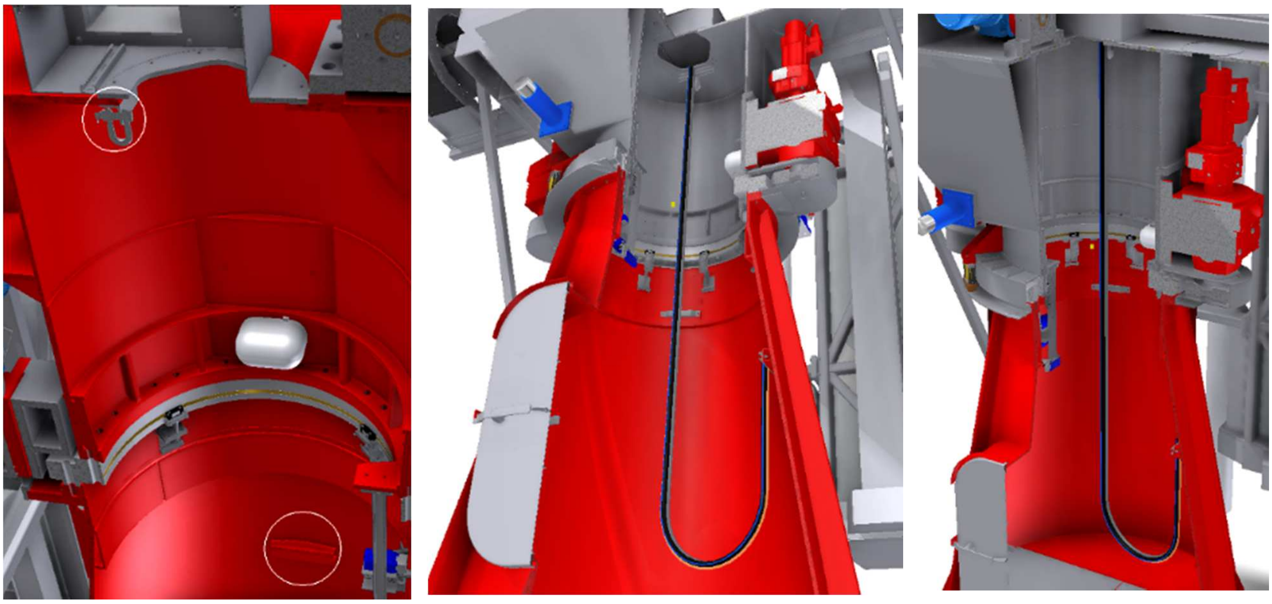


Figure 4-11 Azimuth cable wrap fixing points (see white circles), bottom and side views

The hoses which supply the cooling to the camera can become part of the same cable drape (total length of cooling supply hose for camera equal to about 15050mm and radius of curvature of about 150mm).

4.1.10. Elevation cable wrap

Cable ways from Azimuth fork and Optical support structure is allowed by the use of a unique, central Elevation cable wrap. This cable wrap is standard IGUS R4.56.30.150 which grants protection for the cables as well as the correct guide from Azimuth fork till M1 dish. Its motion is provided passively following Elevation axis one. Spare is reserved for eventual camera cooling hoses.

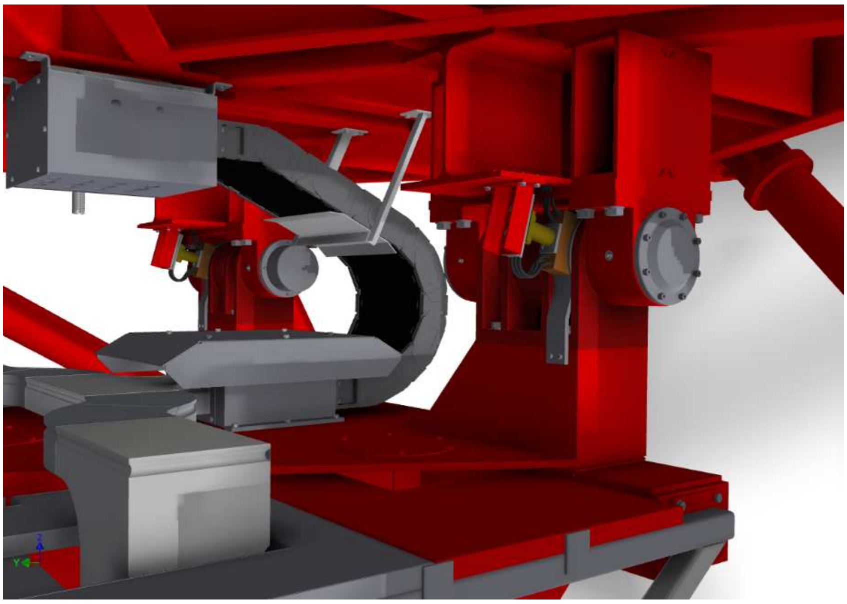


Figure 4-12 Elevation Cable Wrap view

4.1.11. Azimuth stow pin

The Azimuth stow pin guarantees no motion for Azimuth axis and it is very similar to the concept of the Elevation stow pin. The loads for this unit (shear load of about 95kN), is lower than the Elevation one (see following paragraph); for this reason the concept has been kept in order to speed up production of parts.

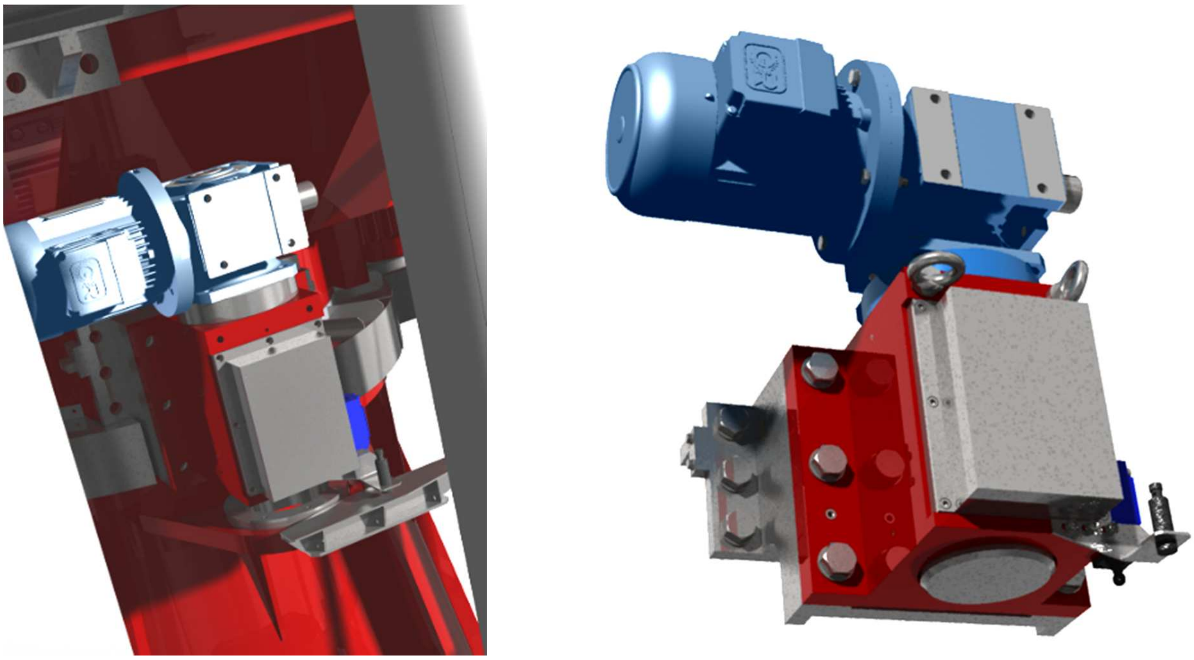


Figure 4-13 Azimuth stow pin installation and detail view

The Azimuth stow pin has 3 switches which give status stow pin fully deployed, not deployed, and proximity (which has the function to slow Azimuth axis motion).

Ratio motor installed consists in a standard ROSSI Riduttori coupled with a standard three phase asynchronous motor.

Witnesses for the switches installed on Azimuth stow pin are located on the Base.

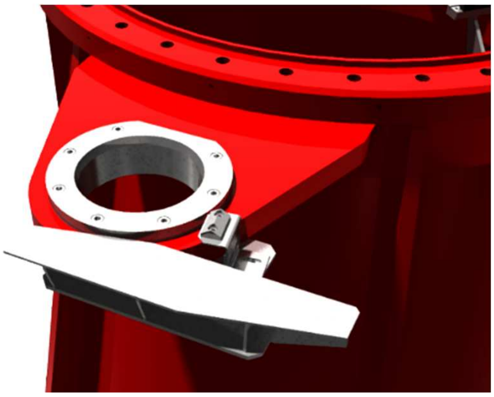


Figure 4-14 Azimuth stow pin witness details

The calculation for Elevation Stow Pin are applicable also to Azimuth one. Since loads are heavier in Elevation case, the calculations are presented in elevation section only.

4.1.12. Elevation stow pin

The Elevation stow pin guarantees no motion during maintenance and Telescope parking position. The position is only one, precisely at Elevation angles 3.5°. It must be mentioned that, parking position is the same of the maintenance.

The Elevation stow pin has 3 switches which give status stow pin fully deployed, not deployed, proximity to engage (which has the function to slow Elevation axis motion).

Ratio motor installed consists in a standard ROSSI Riduttori coupled with a standard three phase asynchronous motor.

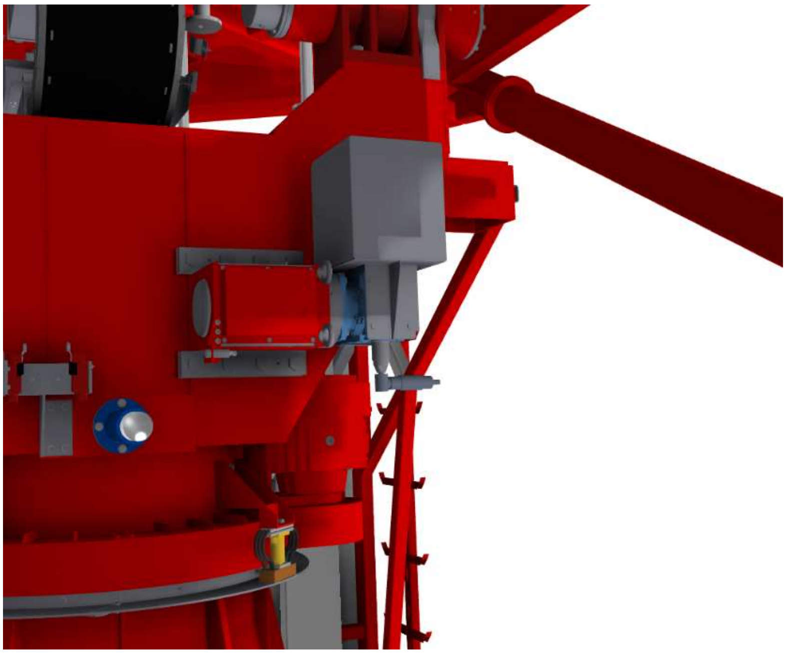


Figure 4-15 Elevation stow pin installation detail view

4.1.12.1. Elevation stow pin calculation

Survival wind load has been estimated in a similar way seen for operational conditions in section § 4.1.5.1. Here are reported the results:

	formula	value	unit	note	
Temperature	T=	-15	℃		
Altitude over sea level	h=	2150	m		
Reduced performance wind speed	v=	47.25	m/s =	170.1	km/h
Gas constant	R=	0.0821	l atm/mol K		
Air molecular mass	m=	28.6	g/mol		
Pressure at altitude	$P=(1-(0.0065/288.15*h)^{5.25588}=$	0.77	atm		
Density at T and P	$\rho=P*m/R/(273.15+T)=$	1.04	kg/m^3		
Dynamic pressure due to wind	$q=\rho/2*v^2=$	1159.97	N/m^2		
Shape load coefficient	$c_p=$	1.5		pg57 EuroCode1	
CoP above EL axis	$CoP_{up}=$	2000.00	mm	estimated	
Area above EL axis	$A_{up}=$	18	m^2	estimation at EL=30°	
EL wind torque with upper area only	$T_{ALTmax}=q*c_p*(A_{up}*CoP_{up}/1000)=$	62638.25	Nm		

Consequently, the calculations relative to the locking pin components sizing are provided in the following:

	formula	value	unit	note
Pin Resistance Verification				
Survival shear load	$F_{su}=$	76148.19005	N	62638Nm wind @60km/h+ 9000Nm unbalancing @0.884m
Operative shear load	$F_{op}=$	14110.86	N	7796.64Nm wind @60km/h+ 9000Nm unbalancing @0.884m
Stow pin outer diameter	D=	125	mm	
Stow pin inner diameter	d=	80	mm	
Pin resistant area	$A=\pi/4*(D^2-d^2)=$	7245.30	mm^2	
Yelding stress	$\sigma_y=$	400	Mpa	2C40 UNI EN 10083/1
Bush distance	b=	33.5	mm	
Bending moment	$M_b=F*b=$	2550964	Nmm	
Second moment of area	$J=\pi/64*(D^4-d^4)=$	9973606		
Perpendicular distance to neutral axis	x=D/2=	63	mm	
Bending moment stress	$\sigma_b=M_b*x/J=$	15.99	MPa	
Shear stress	$\tau_{eff}=F_{su}/A=$	10.51	MPa	
Von Mises Stress	$\sigma_{eq}=(\sigma_b^2+3*\tau_{eff}^2)=$	24.23	Mpa	
Pin safety factor	$a=\sigma_y/\sigma_{eq}=$	9.53		
Stow pin shaft verification during extraction				
Friction coefficient	f=	0.5		
Pin extraction load	$P=f*F_{op}=$	7055.43	N	

	formula	value	unit	note
Stow pin screw diameter	ds=	60	mm	
Stow pin screw pitch	p=	9	mm	
Angle of screw helix	α=arctan(p/πds)=	2.73	°	
Screw friction coefficient	μ=	0.2		
Screw efficiency	η_v=sen(α)/(sen(α)+μ*cos(α))=	0.1925		
Screw torque	M=P*p/(2*π*η_v)=	52.49	Nm	
Shaft diameter	d=	32	mm	
Second moment of area	I=	102943.71	mm^3	
Perpendicular distance to neutral axis	y=	16	mm	
Yelding stress	σy=	400	Mpa	2C40 UNI EN 10083/1
Shear yelding stress	$\tau_y=\sigma_y/(3^{0.5})$=	231	MPa	
Torque stress on shaft	τ_{sh}=M*1000*y/I=	8.16	MPa	
Shaft safety factor	a=τ_y/τ_{sh}=	28.31		
Ratiogear Selection				
Expected manouvre time	t_e=	20	s	
Pin stroke	L=	100	mm	
Screw rounds per stroke	rps=L/p=	11.11		
Preliminary ratio gear slow shaft speed	n_s=rps*60/t_e=	33.33	rpm	
Power at slow shaft	P_s=M*2π*n_s/60=	183.21	W	
De-rating	dr=	0.61	%	
Derated power at slow shaft	$P_{sdr}=P_s$/dr=	300.35	W	
Service factor	f_s=	0.85		Rossi Motorid. Catal. VITE A04 dicembre 2011 Pag.13 (h/d<2 e "b")
Ratio gear slow shaft torque	M_2=M*f_s=	4.46	daNm	
Motor power	P_m=	0.75	kW	Rossi Motoriduttori MR V 63 UO3D/40 80B 4 230.400 B8/35
Power provided at slow shaft	P_{slow}=	0.57	kW	From Rossi Motoriduttori catalogue
Efficiency	ηr=P_{slow}/P_m=	0.760		(Direct motion)
Reversibility verification	η_{rinv}=2-1/ηr=	0.684		System is reversible
Trasmission ratio	i=	40		From Rossi Motoriduttori catalogue (chapter 9)
Motor speed (rpm)	n_m=	1400	rpm	From Rossi Motoriduttori catalogue (chapter 9)
Ratiomotor slow shaft speed (rpm)	n_{slow}=	35	rpm	From Rossi Motoriduttori catalogue (chapter 9)
Real manouvre time	t=rps/n_{slow}*60=	19.05	s	
Bearing lifetime Verification				

	formula	value	unit	note
Bearing speed	n=	35	min^{-1}	
Axial Load	F_a=	7055.43	N	
Radial load	F_r=	0.00	N	
Axial/Radial loads ratio	F_a/F_r=	7.E+19		
Equivalent dynamic load for 4 points bearing	P=	7549.31	N	From SKF catalogue (SKF QJ208) (Page 455 SKF catalogue for load formulas)
Dynamic load	C=	56000	N	From SKF catalogue (SKF QJ208)
Static load	C0=	49000	N	From SKF catalogue (SKF QJ208)
Type of bearing exponent	p=	3		Exponent for roller bearing
Reliability factor	a_1=	0.21		From SKF catalogue (Table 1)
Fatigue load limit	P_u=	1900		From SKF catalogue (SKF QJ208)
Contamination factor	η_c=	0.5		From SKF catalogue (Table 5)
Ratio to obtain correction factor	η_c*P_u/P=	0.12584		
Outer bearing diameter	D=	80	mm	
Inner bearing diameter	d=	40	mm	
Medium bearing diameter	$d_m=(D+d)*0.5$=	60	mm	
Minumum preload for functioning	$F_{am}=K_a*C_0/1000*(n*d_m/100000)^2$=	0.02	N	
Requested lubrication viscosity	v_1=	350	mm^2/s	From SKF catalogue (Diagram 5 - Page 60)
Operational temperature	T=	10	℃	From Environmental condition
Lubricant viscosity	v=	100	mm^2/s	From SKF catalogue (Diagram 6 - Page 61). Lubricant: ISO VG100
Viscosity ratio	$\kappa=v/v_1$=	0.2857		
Correction factor	a_{SKF}=	0.18		From SKF catalogue (Diagram 1 - Page 54)
Lifetime	$L_{nm}=a_1*a_{SKF}*(C/P)^p$=	15.43		Millions of rounds
Maximum turn per motion	tpm=	25		Engage and Disengage
Motion per day	mot=	3		2 standard plus 1 of emergency
Observable nights percentage	nig=	0.95		5 nights every 100 cloudy
Turns per year	T_y=tpm*mot*nig*365=	26006.25		
Lifetime in years	$L_y=L_{nm}/T_y$=	593.28		
Requested lifetime	Ltot=	50		
Duration safety factor	$a_L=L_y/L_{tot}$=	11.87		

4.1.13. Elevation bumper

The Elevation axis design imposed a certain unbalance in order to provide no backlash during operation. The unbalance introduces a torque which would bring the Optical support structure to move spontaneously from zenith to horizon position. For this reason a safety issue rise. In case the Elevation actuator (mechanical jack), for whatever reason, is not able to stop the rotation of the Optical support structure from zenith to horizon there must a be a safety device able to stop without damages this undesired motion. This device is a bumper system composed by two shock absorbers as per the following figure:

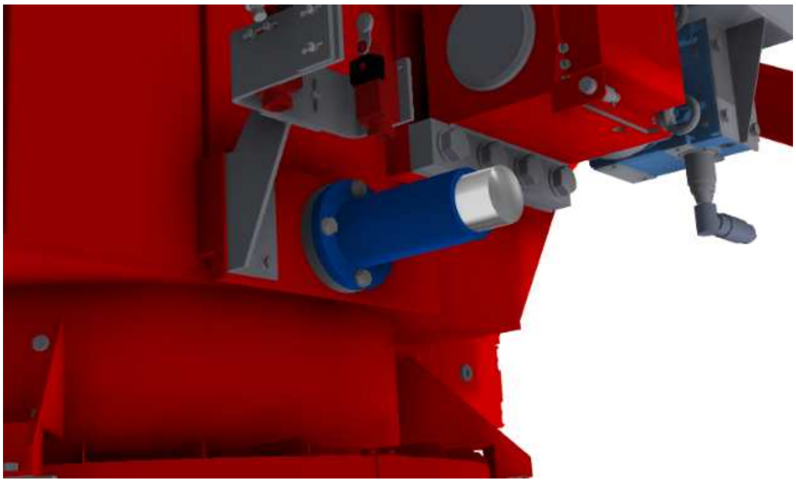


Figure 4-16 Elevation bumper view

4.1.13.1. Elevation bumper calculation

The shock absorbers used for this application are manufactured by JARRET. For this reason ENIDINE Catalogue was used for choice and verification. Here below are reported the calculations performed:

	formula	value	unit	note	
Torque on EL axis					
Rotating Inertia	J=	52864.5	kgm^2	from 3D + 50% of margins	
Tracking angular speed	ωtrk=	4.17E-03	deg/s =	0.0001	rad/s
Max. angular speed	ωmax=	2	deg/s =	0.0349	rad/s
Max angular acceleration	αmax=	1	deg/sec^2 =	0.0175	rad/s^2
Wind torque at 60km/h	Twr=	7796	Nm	see EL Actuator calculation	
Static friction torque	Tfs=	62	Nm		
Unbalancing torque	Tu=	4766	Nm		
Maximum Torque	T_{dw}=Ti+Twr+Tfs+Tu=	12500	Nm		
EL bumper sizing					
Motor inertia	J_{mot}=	0.00272	kgm^2	see EL Actuator calculation	
Ratio gear ratio	i_{rid}=	493.37		see EL Actuator calculation	
Worm gear ratio	i_{wgr}=	8		see EL Actuator calculation	
EL axis angle from EL=90°to -1°	α=	0.4	°=	0.007	rad
Rotational work due to torque	$W=T_{dw}*\alpha$=	87.27	J		
Kinetic energy before failure	$E_k=J*\omega_{max}^2$=	64.41	J		
Total energy	$E=W+E_k$=	151.68	J		
Maximum speed before impact	$\omega_{impact}=(2*E/((I+J_{mot}*i_{rid}*i_{wgr})))^{0.5}$=	0.08	rad/s	4.3397	°/s
Bumper location radius	r=	1377	mm	1.377	m

	formula	value	unit	note	
Impact speed	v=	104.30	mm/s	0.10	m/s
Equivalent mass	$m=(I+J*i_{rid}*i_{wgr})/r^2=$	27881.62	kg		
Number of bumpers		1			
Maximum energy capacity	$E_b=$	3400	J	Enidine Jarret BC1EN	
Stroke	C=	45	mm	Page 95 Enidine Catalogue	
Standard reaction force	$R_{dy0}=$	45000	N	Page 95 Enidine Catalogue	
Maximum reaction force	$R_{dymax}=$	100000	N	Page 95 Enidine Catalogue	
Allowable impact frequency	F=	896.62			
Effective stroke	$C_e=$	3.26	mm	<C	
Effective reaction	$R_{dye}=$	39695	N	$<R_{dymax}$	
Bumper angle stroke	β=	0.14	°		
Tempo di decelerazione		0.06	s		
Decelerazione		1.67	$m/s^2=$	0.17	g
EL bumper verification					
Motors at full torque plus imbalance	Tm=	32277	Nm	(included 14.5Nm of motor efficiencies of 0.98 and 0.506) see Actuator calculation	
EL axis angle from EL=90°to -1°	θ=	0.4	°=	0.0070	rad
Number of bumpers	n=	1			
Maximum motor speed	ω=	2.00	°/s	0.0349	rad/s
Energy of system	$W=0.5*J*\omega^2+Tm*\theta=$	225.90	J		
Bumper energy dissipation	E=W/n=	112.95	J		
Bumper location radius	r=	1377	mm	1.377	m
Impact speed	v=ω/r=	48.07	mm/s	0.05	m/s
Allowable impact frequency	$F=20*E_b/(E*n)=$	241.18			
Effective stroke	$C_e=C*(((E)/(E_b*(0.03*v+0.24)+1.36))^{0.5}-1.17)=$	11.73	mm	<C	
Effective reaction	$R_{dye}=(((R_{dyemax}-R_{dy0})/C)*C_e+R_{dy0})*(0.1*v+0.8)=$	47756	N	$<R_{dymax}$	
Bumper angle stroke	$\beta=C_e/r*180/\pi=$	0.49	°		
Deceleration	$a=R_{dye}*r^{2}*n/J=$	1.71	$m/s^2=$	0.17	g

4.2. OPTICAL SUPPORT STRUCTURE

The so-called Optical Support Structure represents that part of the Telescope which hosts the support of the optics. Here below is reported an excerpt of the product tree which helps to understand the location of the sub-systems:

Level 1		Level 2		Level 3	
3120-000	Optical Support Structure	3121-000	M1 dish	3121-100	M1 Shields
		3122-000	Counterweights		
		3123-000	OSS Upper Structure	3123-100	Mast
				3123-200	Central Tube

				3124-300	Top ring
		3124-000	M2 Back Up Structure	3124-100	Optical Baffle & M2 Shield
		3125-000	M1 Segment Support Assembly	3125-100	M1 segment support
				3125-200	M1 segment calibration
				3125-300	M1 segment handling
		3126-000	M2 Support	3126-100	M2 Actuator
				3126-200	M2 actuator driving unit
				3126-300	M2 Lateral fixed point
				3126-400	M2 Loadspreader
		3127-000	Pointing Monitor Camera		

The following picture provides the locations and the volumes occupied by all sub-systems.

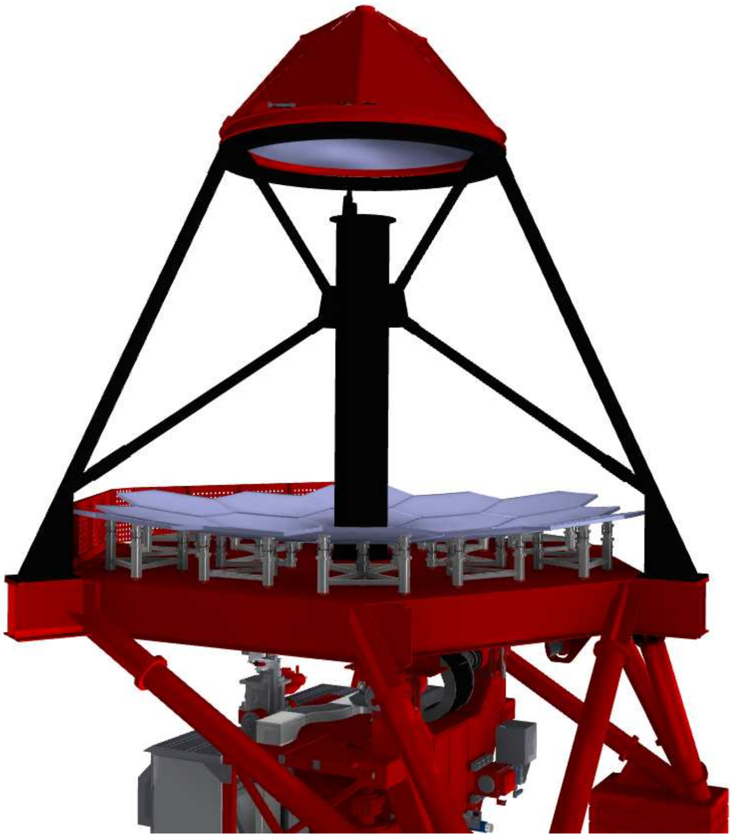


Figure 4-17 Populated OSS

Pictures highlight that M1 Dish and M2 BUS host all the subsystems, that it is obvious since the optics here are the primary and the secondary mirror. The central tube only hosts the camera necessary for the observations.

4.2.1. M1 segment Support Assembly

The M1 segment assembly layout is drastically changed, as the Cartesian coordinate system has been replaced by a radial, easier system. In this way it was possible to design only three type of units: the inner, the intermediate and the outer ones. These units are repeated six time with an “hexagonal symmetry”.

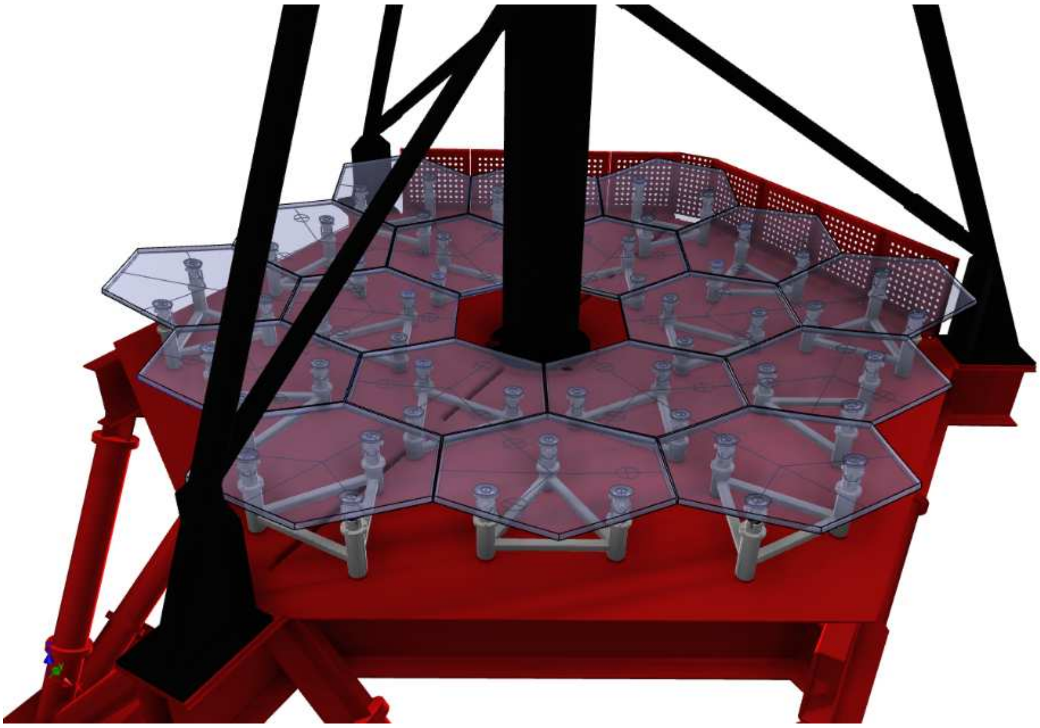


Figure 4-18 M1 segments support assembly view with transparent M1

Each segment is provided with three passive actuators preloaded by means of springs with a stroke of ±5mm and fixed in their positions with tapered locking devices which works with friction.

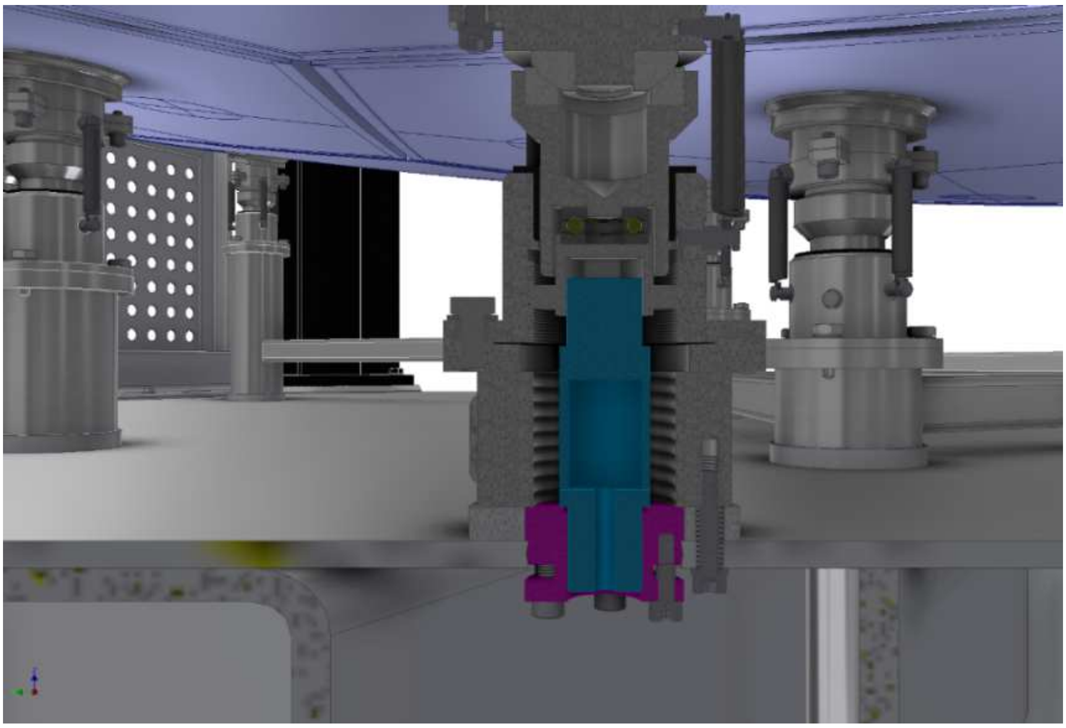


Figure 4-19 M1 segment support actuator with locking device

The handling is possible by removing on a single actuator per time, the locking device the fixing screws and installing, the sliding bush and the relative pipe. It is essential to do these operation on a single actuator per time in order to avoid damages. For this reason there will be a sign on the bottom of the M1 cell in correspondence with each segment indicating:

“remove locking device and install the sliding system before removing the remaining locking devices of the actuators belonging to the same segment – remover el acoplamiento e instalar el sistema de deslizamiento antes de remover los acoplamientos restantes que pertenecen a los actuadores del mismo segmento”.

When the sliding devices are installed on all three actuators, the segment can be slid out to permit the mirror removal.

Mirror can be detached from the actuators removing three fixing screws per pad. An eyebolt will be sufficient to lift the mirror segment up; springs will remain in their places.

It must be taken into account that a cherry picker is sufficient for the mirror segment removal as the operators can hold them by virtue of its light weight.

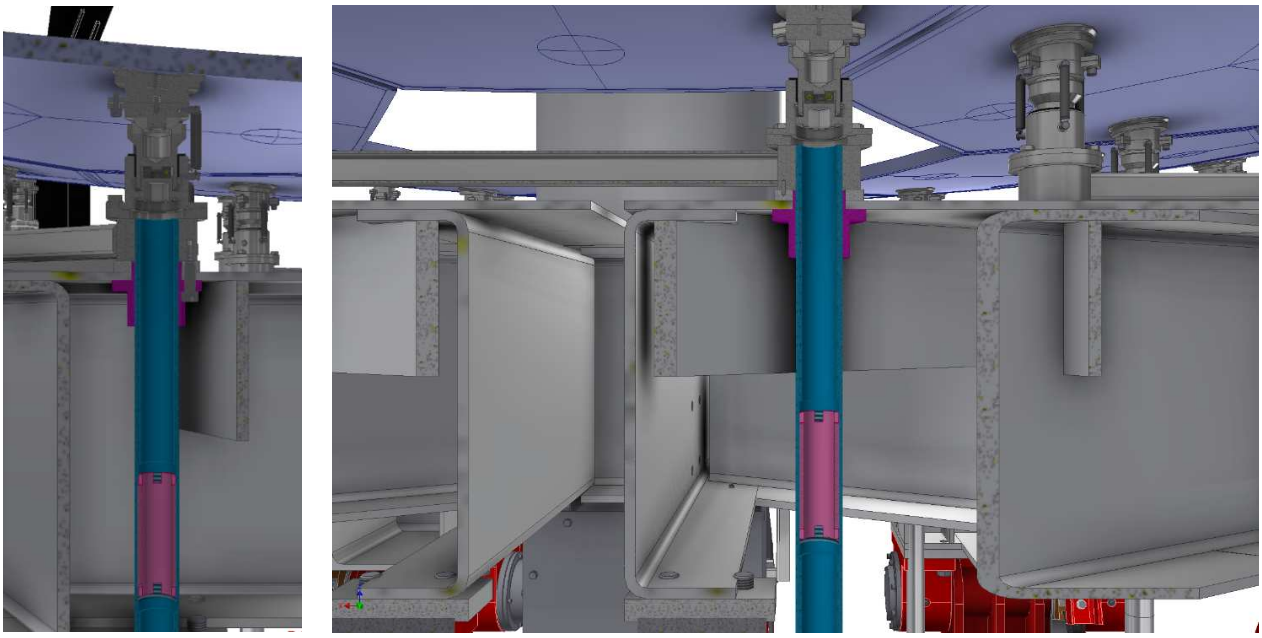


Figure 4-20 M1 segment sliding tool

4.2.1.1. M1 Actuator springs calculation

To avoid backlash an the actuator supporting thread during normal operation within all Elevation configurations and to obtain a precise and stable motion during calibration, springs are adopted, to apply a preload able to permit the functioning till 60km/h (which can be the gust of 36km/h wind speed averaged over 10 minute). It is important to assess also the survival loads in play and gravity one.

Wind load				
Temperature	T=	-15	-5	°C
Altitude over sea level	h=	2150	2150	m
Reduced performance wind speed	v=	47.25	16.67	m/s =
Gas constant	R=	0.0821	0.0821	l atm/mol K
Air molecular mass	m=	28.6	28.6	g/mol
Pressure at altitude	$P=(1-(0.0065/288.15*h)^{5.25588}=$	0.77	0.77	atm
Density at T and P	$\rho=P*m/R/(273.15+T)=$	1.04	1.00	kg/m^3

Wind load				
Dynamic pressure due to wind	$q=\rho/2*v^2=$	1159.97	139.00	N/m^2
Shape load coefficient	$c_p=$	0.8	0.8	pg57 EuroCode1
Mirror area		0.62	0.62	m^2
Survival load per mirror		575.34	68.94	N
Gravity load				
M1 segment mass	$m_{M1}=$	7.5	kg	
Gravity load	$G=m_{M1}*9.81=$	73.58	N	
Spring sizing				
Spring stiffness	c=	5.91	N/mm	DIM T42460
Undeformed length	lo=	54.4	mm	
Nominal length	l =	62	mm	
Stroke	s=	5	mm	
Minimum length on actuator	lmin=l-s=	57	mm	
Maximum length on actuator	lmax=l+s=	67	mm	
Number of actuators	n=	3		
Minimum force per actuator	Fmin=(lmin-l)*c*n=	46.098	N	All segment actuators 138.294 N
Nominal force per actuator	Fnom=(lo-l)*c*n=	134.748	N	All segment actuators 404.244 N
Maximum force actuator	Fmax=(lmax-l)*c*n=	223.398	N	All segment actuators 670.194 N

For M1 segments, gravity is applied axially to the actuators when telescope elevation angle is equal to 90 degree and in those conditions wind can rather give a lifting effect which counteracts with gravity. At elevation angles close to 0 degree the gravity act tangentially so the springs are not affected.

With these considerations, it is possible to state that springs guarantee no backlash at least till a wind speed of 60km/h.

4.2.1.2. M1 actuator axial stiffness assessment

The estimation of the axial stiffness has been done with a simplification (with safety margins) of the actuators in 3 elements based on geometry as reported here below:

Elements		1	2	3	
Young modulus	E=	200000	200000	200000	Mpa
Element outer diameter	D=	38	29	42	mm
Element inner diameter	d=	25	23	30	mm
Area of element section	A=	643.24	245.04	678.58	mm^2
Element length	l=	73	15	18	mm^2
Element stiffness	k=EA/l=	1762304.372	3267256	7539822	N/mm
Total axial stiffness	K=(1/k1+1/k2+1/k3)^(-1)=	993902.22	N/mm		
		993.90	MN/m		

So, following these assumptions, the total axial stiffness of a single actuator is 993.90 MN/m.

4.2.2. M2 Support

M2 is supported by a system which includes an active tripod system which allow M2 precise positioning along Z axis and tip-tilt (rotation around X and Y axes), and a lateral support which grants safety support and safe load transmission. Tripods consist in three loadspreaders provided with 3 flexures each to transmit axial loads only.

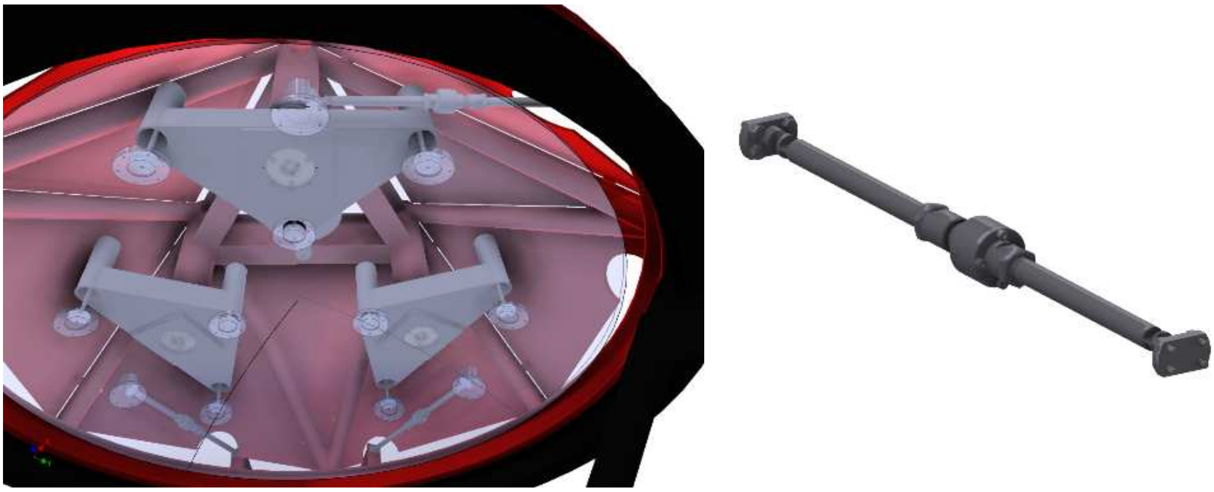


Figure 4-21 M2 Support view and isometric view of lateral fixed point

The motion is possible with a motor and a ratio gear which permits the actuators to move with an axial stroke of ±7.5mm. each driving unit is equipped with an absolute encoder and two electrical limit switches to have full feedback of position and safety of motion. The joint between the flexures structure and the actuator consists in a double-tilting system which grants the M2 to be tilted and displaced along the optical axis without constraints.

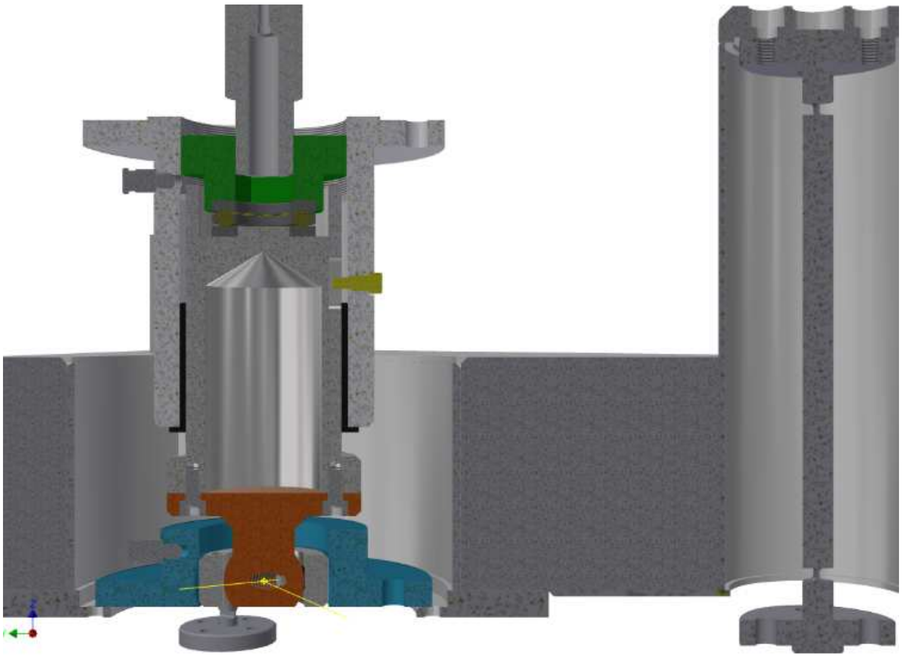


Figure 4-22 Section of M2 actuator and Loadspreader flexure

Maintenance or replacement of the lateral fixed points can be done when the telescope is in its parking position by means of threaded rod which support temporarily the mirror loads during substitution.

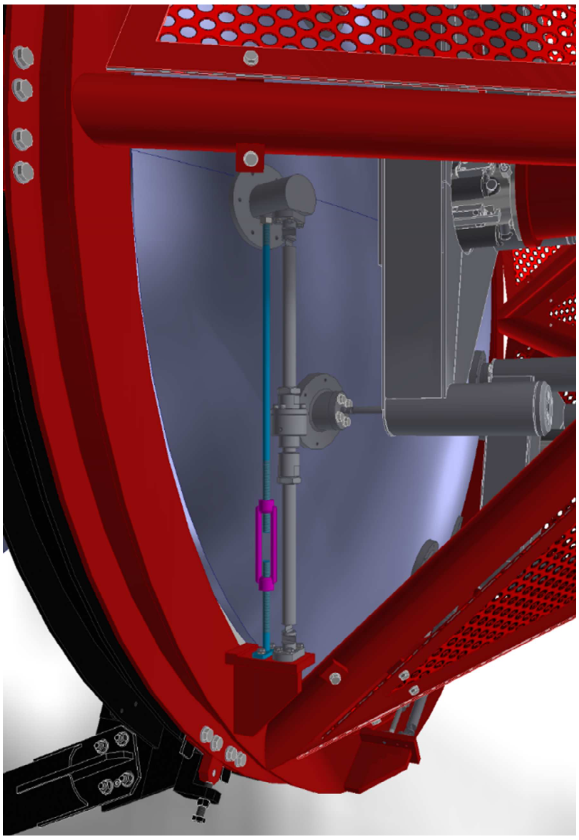


Figure 4-23 Maintenance threaded rod for lateral fixed points

M2 removal can be done only removing the M2 BUS.

4.2.2.1. M2 actuator spring calculation

To avoid backlash an the actuator supporting thread during normal operation within all Elevation configurations and to obtain a precise and stable motion during calibration, springs are adopted, to apply a preload able to permit the functioning till 60km/h (which can be the gust of 36km/h wind speed averaged over 10 minute). It is important to assess also the survival loads in play and gravity one.

Wind load				
Temperature	T=	-15	-5	℃
Altitude over sea level	h=	2150	2150	m
Wind speed	v=	47.25	16.67	m/s
Gas constant	R=	0.0821	0.0821	l atm/mol K
Air molecular mass	m=	28.6	28.6	g/mol
Pressure at altitude	$P=(1-(0.0065/288.15*h)^{5.25588}=$	0.77	0.77	atm
Density at T and P	$\rho=P*m/R/(273.15+T)=$	1.04	1.00	kg/m3
Dynamic pressure due to wind	$q=\rho/2*v^2=$	1159.97	139.00	N/m^2
Shape load coefficient	$c_p=$	0.8	0.8	pg57 EuroCode1
Mirror projected area	A=	2.73	2.73	m^2

Survival load per mirror	$W=q*A*c_p=$	2535.03	303.77	N
Gravity load				
M2 mass	$m_{M2}=$	180	kg	
Gravity load	$G=m_{M2}*9.81=$	1765.8	N	
Spring sizing				
Spring stiffness	c=	10.05	N/mm	DIM T43180
Undeformed length	lo=	113	mm	
Nominal length	l =	143	mm	
Stroke	s=	7.5	mm	
Minimum length on actuator	lmin=l-s=	135.5	mm	
Maximum length on actuator	lmax=l+s=	150.5	mm	
Number of actuators	n=	3		
Minimum force per actuator	Fmin=(lmin-l)*c*n=	678.375	N	All actuators 2035.125N
Nominal force per actuator	Fnom=(lo-l)*c*n=	904.5	N	All actuators 2713.5 N
Maximum force actuator	Fmax=(lmax-l)*c*n=	1130.625	N	All actuators 3391.875 N

For the secondary mirror, gravity is applied axially to the actuators when telescope elevation angle is equal to 90 degree and in those conditions wind can rather give a pulling effect which acts concurrent with gravity. At elevation angles close to 0 degree the gravity act tangentially so the springs are not affected.

With these considerations, it is possible to state that springs guarantee no backlash with a wind speed gusts till 60km/h even if, at the minimum length at 90 degrees of elevation the load on the M2 mirror can be close to the spring preload (only if wind causes a pulling effect). This can be stated considering that it is very unlikely that, the followings happen at the same time:

- M2 actuators should be all simultaneously at their minimum height (a piston motion of -7.5mm is surely greater of the one foreseen in the budget as otherwise there will be no enough stroke for tilting M2);
- The wind should have a pulling effect (thus, with a depressurization on the aluminized side);
- The real shape coefficient is at least equal or greater than 0.8;
- A wind of 36km/h should have not less than 60km/h gusts;
- The elevation angle should be greater than 80 degree;
- The temperature should be at -5℃ (as it affects d ensity);

In these simultaneous conditions the difference between the loads and the spring preload will be of about 35N (1766+304=2070 N against 2035N). The first, of the previous points, is sufficient to state the compliance of the system. The other points permit to increase confidence on this statement.

4.2.2.2. M2 actuator axial stiffness assessment

The estimation of the axial stiffness for a single M2 actuator, has been done with a simplification (with safety margins) of the actuators in 3 elements based on geometry as reported here below:

Elements		1	2	3	
Young modulus	E=	200000	200000	200000	Mpa
Element outer diameter	D=	60	24	46	mm

Element inner diameter	d=	45	0	22	mm
Area of element section	A=	1237	452.39	1012	mm^2
Element length	l=	108	35	35	mm^2
Element stiffness	=EA/l=	2290744.64	2585082	5782857	N/mm
Total axial stiffness	K=(1/k1+1/k2+1/k3)^(-1)=	1256201.26	N/mm		
		1256.20	MN/m		

So, following these assumptions, the total axial stiffness of a single actuator is 1256.20 MN/m.

4.2.2.3. M2 driving unit calculation

The motors are sized to move the actuators till 60km/h wind speed (which represents the gust speed of a wind of 36km/h speed averaged over 10 minutes). Under these condition the calculations are reported here below:

	formula	value	unit	note
Wind load				
Temperature	T=	-5	℃	
Altitude over sea level	h=	2150	m	
Reduced performance wind speed	v=	16.67	m/s =	60.012 km/h
Gas constant	R=	0.0821	l atm/mol K	
Air molecular mass	m=	28.6	g/mol	
Pressure at altitude	$P=(1-(0.0065/288.15*h)^{5.25588}=$	0.77	atm	
Density at T and P	ρ=P*m/R/(273.15+T)=	1.00	kg/m^3	
Dynamic pressure due to wind	$q=\rho/2*v^2=$	139	N/m^2	
Shape load coefficient	c_p=	1.5		pg57 EuroCode1
Torque on screw				
Screw diameter	d=	64	mm	M64x1
Screw pitch	p=	2	mm	
M2 mass	m_{M2}=	180	kg	
Torque due to mass	$Tm=m_{M2}*9.81*p/(2*\pi*1000)=$	0.562	Nm	
Spring max load	F=	3391	N	
Torque due to spring max load	Ts=F*p/(2*π*1000)=	1.079	Nm	
M2 diameter	dM2=	1.865	m	
M2 area	A=π*dM2^2/4=	2.732	m2	
Load due to wind	W=q*cp*A=	569.569	N	
Torque due to wind	Tw=W*p/(2*π*1000)=	0.181	Nm	
Friction coefficient on screw	f=	0.2		
Torque due to friction	Tf=(F+W)*f*d/1000/2=	25.348	Nm	Worst case @EL=0°(no load reduction due to M2 mass)
Total torque	T=Ts+Tw+Tf=	26.608	Nm	
Number of actuators	a=	3		
Max torque per actuator	Ta=T/a=	8.869	Nm	

	formula	value	unit	note
Ratiogear selection				
Ratiogear nominal torque	Trg=	16	Nm	Kollmorgen Xtrue XT060-010
Ratiogear ratio	i=	10		
ratiogear efficiency	ηrg=	0.88		
Motor torque	Tm=Ta/ηrg/i=	1.008	Nm	
Motor selection				
Motor efficiency	ηm=	0.85		
Torque necessary for motor	Tmot=Tm/ηm=	1.186	Nm	
Actuator speed	v=	0.750	mm/s	
Motor speed	n=v/p*60*i=	225	min^{-1}	
Nominal torque	Tnom=	1.800	Nm	@250min^{-1} Kollmorgen PMX3410-A10
Safety ratio	s=Tnom/Tmot=	1.518		

After this calculation it is possible to state that motion is still feasible also with a wind speed of 36km/h averaged over 10 minutes with gusts till 60km/h) considering that there is a further margin due to influence of gravity at EL angles greater than 0 degree (e.g. for EL=20deg safety ratio is 1.776)

4.2.2.4. M2 lateral fixed point calculation

The lateral device must be able to bear till a maximum load of 2900N and than for safety reason must interrupt load transmission. The load foreseen for this application is 1010N. To grant this performance a mechanical fuse is introduced with this characteristics:

	formula	value	unit	note
Fuse diameter calculation				
Yielding stress	σ_y=	235	MPa	Steel SJ235
Shear yielding stress	$\tau_y=\sigma_y/3^{0.5}$=	135.68	MPa	
Min failure stress	σ_{rmin}=	360	Mpa	
Shear min failure stress	$\tau_{rmin}=\sigma_{rmin}/3^{0.5}$=	207.85	MPa	
Max failure stress	σ_{rmax}=	510	Mpa	
Shear max failure stress	$\tau_{rmax}=\sigma_{rmax}/3^{0.5}$=	294.45	MPa	
Maximum survival load	S=	2900	N	R.103
Maximum operational load	F=	1010	N	R.104
Minimum resistence area	$A=S/\tau_{rmax}$=	9.85	mm^2	
Number of resistant section	n=	2.00		
Fuse diameter	$d=((4*A/\pi)^{0.5})/n$=	2.50	mm	
Survival load with Min failure stress	$S_{min}=A*\tau_{rmin}$=	2047.06	N	
Fuse diameter	d_F=	2	mm	
Minimum bearable load	$F_{min}=(\pi*d_F^2)/4*n*\tau_{rmin}$=	1305.94	N	
Max bearable load	$F_{max}=(\pi*d_F^2)/4*n*\tau_{rmax}$=	1850.08	N	
Fuse length calculation				
Young modulus	E=	210000	MPa	
Poisson modulus	ν=	0.3		

	formula	value	unit	note
Shear young modulus	$G=E/(2*(1+v))=$	80769	MPa	
Rod length	$L=$	518	mm	
Rod diameter	$D=$	20	mm	
Rod stiffness	$K_r=E*\pi*D^2/(4*L)=$	127362	N/mm	
Flexional flexure width	$w=$	20	mm	
Flexional flexure thickness	$t=$	1	mm	
Flexional flexure length	$l=$	11	mm	
Flexional flexure stiffness	$K_f=E*w*t/(4*l)=$	381818	N/mm	
Number of flexures	$m=$	4		
Total stiffness without fuse	$K_{wf}=(m/K_f+1/K_r)^{-1}=$	54562	N/mm	
Fuse gap due to coupling tolerances	$x=$	0.10	mm	
Fuse stiffness	$K_{fuse}=n*G*A/x=$	15909788	N/mm	2 shear areas works as two parallel springs
Fuse minimum length	$l_f=E*A/K_r=$	0.13	mm	
Total stiffness with fuse	$K=(n/K_f+1/K_r+1/K_{fuse})^{-1}=$	54375	N/mm =	54.375 MN/m
Stiffness loss caused by fuse	$K_{loss}=$	0.34	%	

A pin with diameter of 2mm will be used with a gap between the two arm elements of 0.1mm. This will allow a stiffness loss of only 0.34% with respect to a continuous rod. This pin will fail with a load within the range of 1306N and 1850N which fulfils the requirements.

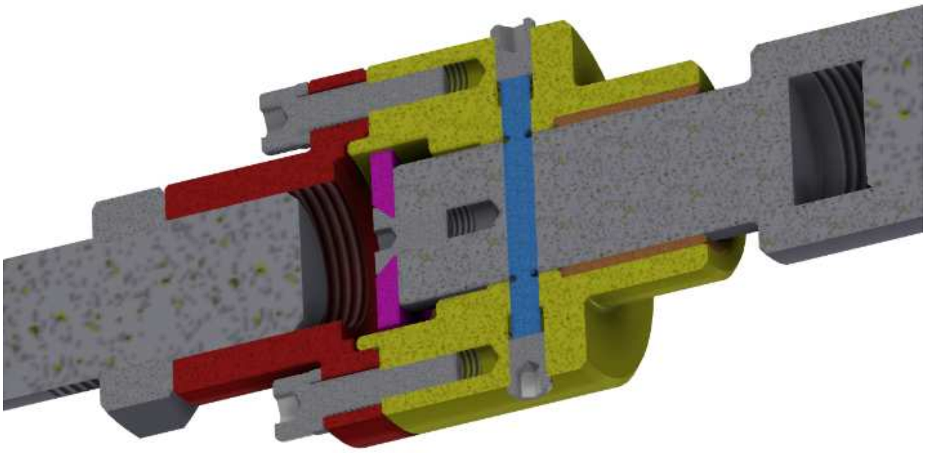


Figure 4-24 Section of lateral fixed point with mechanical fuse

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