

Friction drive and bogies for OWL's main axes, technological step backwards or cost effective alternative?

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ABSTRACT

The drive and bearing technologies have a major impact on the static and dynamic performances of a steerable telescope. The costs related to the complexity of the design and its Reliability, Availability, Maintainability and Safety (RAMS) are not negligible. The design constraints of Extremely Large Telescopes (ELT) depart from those applicable to the current generation of 8 to 10 meter class telescopes, thus suggesting that alternative solutions should be investigated. This paper discusses the feasibility of implementing a design based on friction drives and bogies, tailored to OWL's azimuthal and altitude degrees of freedom. The estimated static and dynamic performance of the mechanical structure, the achievable angular resolution, the optimal distribution of loads and stresses, the RAMS performance and finally its cost efficiency, make this solution particularly attractive.

Keywords: OWL, mechanical design, costs analyses, performance analyses.

1. INTRODUCTION

An essential function of the telescope mechanical structure, is to provide a smooth and stiff transfer of loads to the site soil. The OWL telescope has its tracks embedded into the concrete foundation. Thus the well established, traditional Fork / Tube concept standing on a pier, has evolved into a Azimuth Ring / Altitude Structure concept embedded into crater-like foundation. The smooth and even transfer of the loads from the optomechanical structures to the soil via the altitude structure, azimuth ring and concrete foundation can only be achieved using a large number of mechanical interfaces which shall also provide the kinematics of the telescope. The telescope is constrained in a hyper-static mode, which requires implementing accurate and reliable drive and bearing systems, thus avoiding concentration of stresses and deformation of the structural steel element and of the concrete foundation.

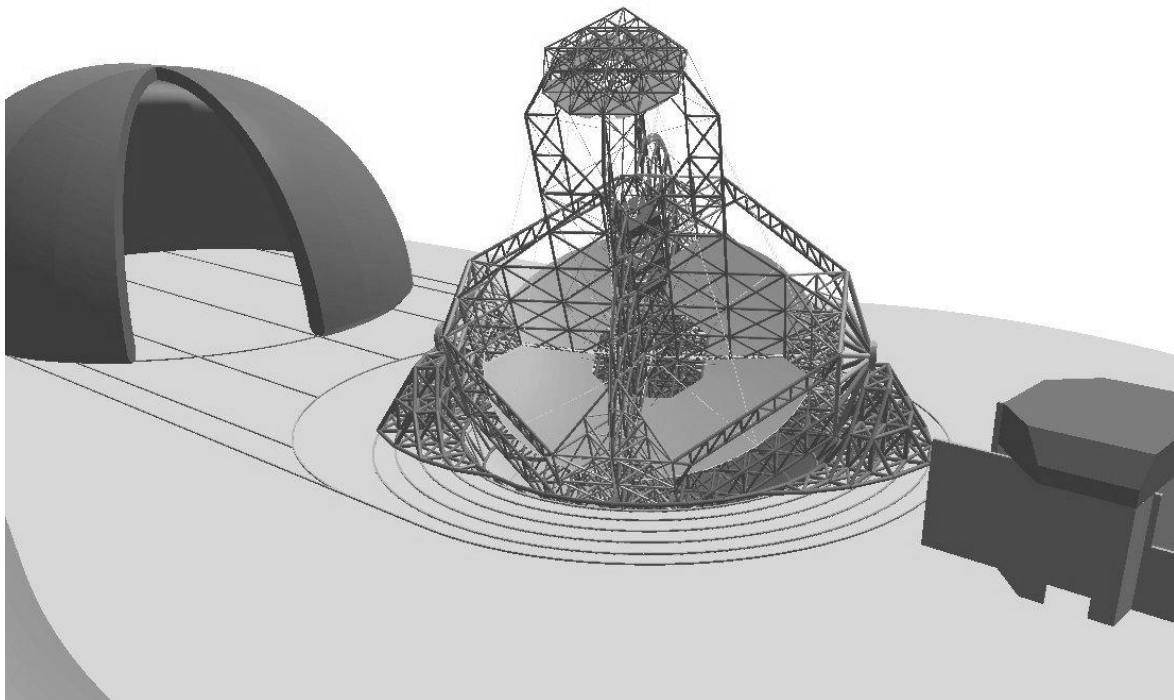


Fig. 1: Layout of the OWL telescope and facilities.

2. SHORT SUMMARY OF OWL MECHANICS.

Azimuth ring

The azimuth ring, shown in, is a 5000 tons steel structure embedded into the foundation. It rotates over 11 axial and 3 radial annular tracks, with the largest having a diameter of about 170 meters. The mass of these annular tracks is 4500 tons.

Different types of drives and bearings systems can be implemented between the azimuth ring and tracks. However for a typical angular accuracy of 1 arcsec, equivalent to a tangential displacement of about 0,3 mm over a mean 60 meters radius, must be well within the performance of the adopted solution, while interfaces and kinematics must ensure a homogeneous load transfer to the foundation.

The azimuth ring incorporates a 60 meters radius radial track "cradle" for the rotation of the altitude structure, along with two large roller bearings, which define unambiguously the altitude axis. The azimuth ring also incorporates sub-systems like cable wraps and encoders.

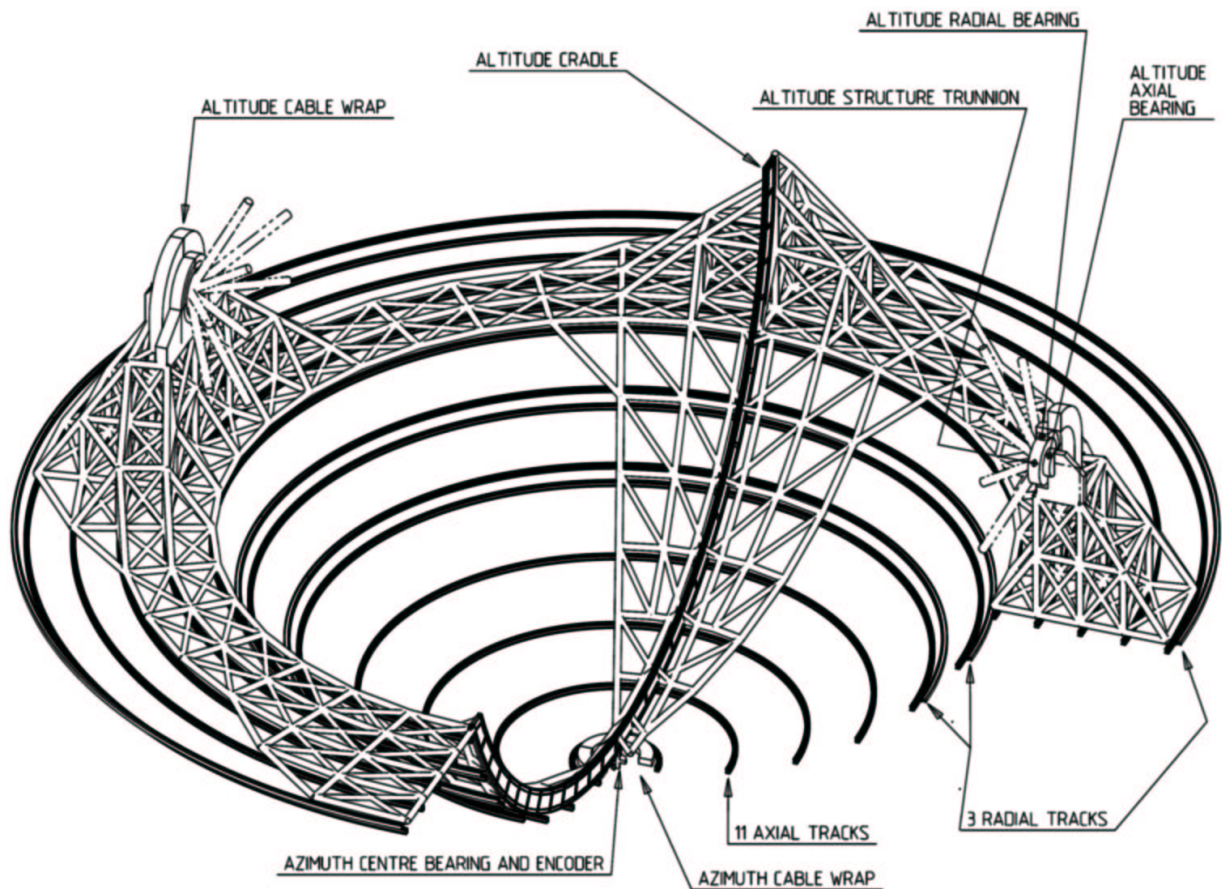


Fig. 2: Azimuth ring structure.

Altitude structure

The altitude structure is a 5600 tons steel structure, which rotates over the azimuth ring around its trunnions and will use the same drive concept as per the azimuth ring. Thus the altitude structure is also constrained hyper-statically by the azimuth ring.

The altitude structure supports the primary and secondary mirrors and the corrector units, which add other 3000 tons to the telescope rotating mass. It has provisions for their safe and efficient maintenance.

The altitude structure has a rotation range from zenith to horizon, 0° to 60° operational range. Rotation to $z = 90^\circ$ is used for maintenance purpose and parking position only.

3. CONSIDERATION ABOUT DRIVE AND BEARING SYSTEMS FOR THE 8-10 M AND 100 M CLASS TELESCOPE.

State of the art kinematics of 8-10 m class telescopes suggests that direct drives and hydrostatic bearings are a most convenient technological solution in term of performance versus realization and operational costs.

Feasibility and costs associated to critical aspects like:

- Tracks alignment tolerances, cleanness and protection.
- Number of bearings.
- Oil re-circulation, cleanness and temperature control.
- Drive alignment tolerances, cleanness and protection
- Inspection and maintenance operation.
- Stress concentration.

and so on, are acceptable on a 8 to 10 m scale. The excellent performance which can be obtained justifies the investments and the operational costs. However the design and operational constraints of an ELT and in particular OWL differs from the previous generation of telescopes, thus the design guidelines of a 8-10 m class telescope cannot be scaled up to a 100 m telescope. For instance, the developed length of the azimuth tracks can help us to better grasp the jump between these generations of telescopes. While for one VLT unit telescope, about 84 m of azimuth tracks could provide an excellent support for the 400 tons of its rotating mass, OWL needs 4500 m of azimuth tracks for about 13100 tons of rotating mass. Obvious considerations related to the aspects listed above, show that not only the procurement and Assembly Integration and Verification (AIV) but also RAMS can generate unacceptable costs and complexity. The following sections will discuss in detail the pros and cons of different drive and bearing technologies.

4. TELESCOPE DRIVES AND BEARINGS.

OWL performance and cost constraints require that different solutions be assessed. Several design iteration and trade off, indicate that merging drive and bearing functions into friction type mechanical devices ("bogies", Fig. 3), reduces the complexity and cost of the design and allows to constrain the telescope in hyper-static mode, without jeopardizing the telescope operation its-self. The large number of bogies, about 250 installed on the azimuth ring and 60 on the altitude structure, assures a smooth and homogeneous transfer of the loads to the site soil. Each bogie has 4 spherical wheels for the azimuth and altitude (cradle) flat tracks. Each wheel, with a diameter of 630 mm, is independently driven by a commercially available brushless ring torque motor equipped with an angular encoder. The required mean angular accuracy is 3 arcmin. at the wheel-motor axis.

In order to obtain homogeneous reaction forces on each bogie and consequently on the foundations, a "hydraulic whiffle tree" system is implemented, which consists of communicating hydraulic cylinders, their displacement being monitored by linear encoders. This whiffle tree system can be locked during observation to attain high dynamic performance.

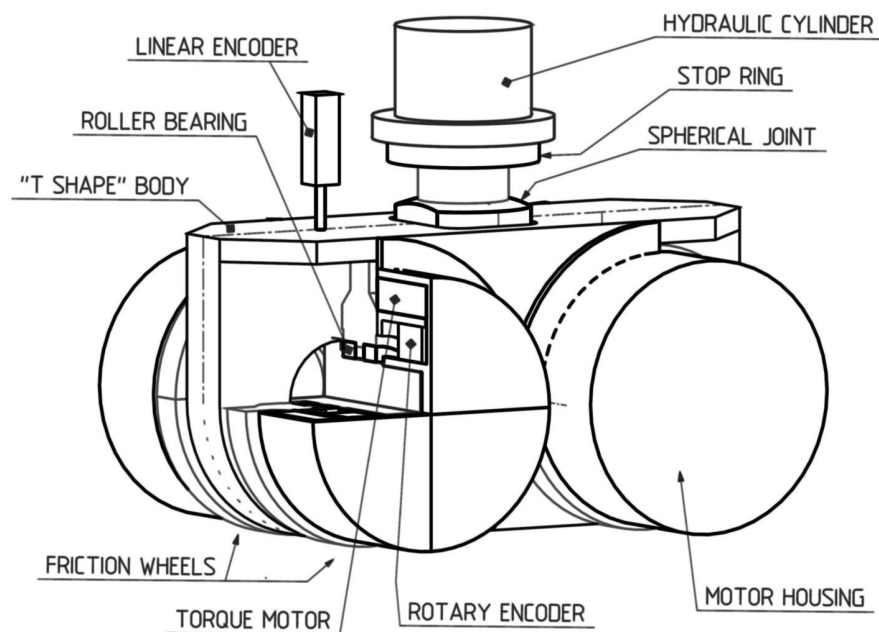


Fig. 3: Bogie system.

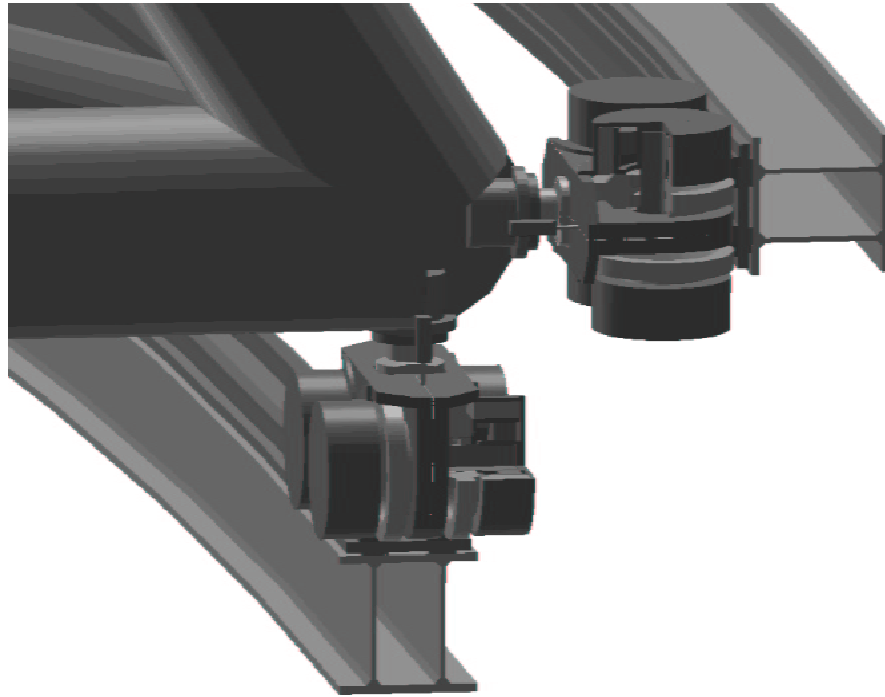


Fig. 4: Axial and radial bogies of the azimuth axis.

5. HERZ PRESSURE AND FRICTION COEFFICIENT

The maximum allowable load capacity and resulting friction force have been calculated on the basis of the current bogie design and wheel dimensions. This allows us to estimate the required number of bogies.

According to the theory of Hertz for a sphere-plane contact (spherical wheel and flat track) the maximum Hertz pressure can be calculated on the basis of the material hardness of the contacting parts. In order to be conservative, the dynamic load assumptions for Hertz pressure have been applied. According to the appropriate formulas given in [1], the maximum allowed Hertz pressure is 2640 MPa. The corresponding normal force per wheel results in 736 kN and the radius of circular contact 11.5 mm. The maximum tensile and shear stresses in the contact region become 352 MPa and 818 MPa, respectively. This allows to apply to each bogie a maximum load of about **300 tons**.

The total friction depends on the rolling resistance of the wheel and on the bearing friction. To analyze the rolling resistance a “friction lever arm” can be assumed according to [1]. For railway-type bogie wheels on the wheel diameter. For the maximum normal wheel force the tangential friction force is about 760 N. The bearing friction force results in 400 N according to the bearing manufacturer. Hence, the total friction force is about 1200 N per wheel. This corresponds to an equivalent friction coefficient of 0.0016. With a total normal force of 34 MN transmitted by the altitude bogies the total friction torque for the altitude axis becomes 3.2 MNm. With the same assumptions as for the altitude axis, the total friction torque for the azimuth axis under gravity results in 13.3 MNm.

6. TORQUE REQUIREMENT

The torque required to the drive systems during observation is shown on Table 1

	Azimuth axis	Altitude axis
Acceleration	59,3 MNm	18,5 MNm
Friction	13,3 MNm	3,2 MNm
Wind (10 m s ⁻¹)	5 MNm	5 MNm

Table 1: Main axes torque requirements.

The main axes drive systems are dimensioned to provide the following characteristics:

Blind angle at zenith	$\leq \pm 0,5$ degree.
Acceleration	0,1 degree s ⁻² .
Maximum velocity	0,5 degree s ⁻¹ .

Acceleration: Altitude mass moment of inertia of $10,6 * 10^9 \text{ kg m}^2$. Azimuth mass moment of inertia of $33,96 * 10^9 \text{ kg m}^2$.

Friction: the friction assumptions are quite conservative, improved surface qualities and accurate alignment, can reduce considerably the friction. Measurements on other friction drive systems of the VLT, show that the final performance, of a friction drive system, is much better than that calculated during the design phase. In some instances we notice a factor of ten between as designed and as measured values. In order to determine the correct value of the friction, prototypes of the bogie will be manufactured and tested, in "phase B".

Wind: the maximum wind torque disturbance during operation, is estimated to be 5 MNm, as generated by a uniform pressure over the telescope. However the values reported [2], show that a torque disturbance of 5 MNm is very conservative, due to the non correlated space and time nature of the wind disturbance.

7. STATIC AND DYNAMIC BEHAVIOR

In comparison with hydrostatic bearing systems, the bogies system offers a further advantage of better dynamic performance. In the static case, the 4 friction spherical drive wheels of each bogie does settle over the flat tracks in the position where the axial forces are minimum or close to zero. Thus this system does not over-constrain the structure of the telescope, which is free to adjust itself according to change in temperature or loads. This behavior is similar to that of the hydrostatic bearing systems. However in the dynamic case, due to the friction between the wheel and track, 2 supplementary degrees of freedom are constrained¹.

For instance the azimuth axial bogie shown in the picture below is constrained as follow:

- Vertical Degree of Freedom (DoF), constrained by the track.
- Radial DoF, constrained by the wheel-track friction (via roller bearings of the bogie)
- Tangential DoF, constrained by the wheel-track friction (via torque motor of the bogie)

The sum of both cases results in better dynamic performance, without introducing high local stresses typical of hyper-static structure.

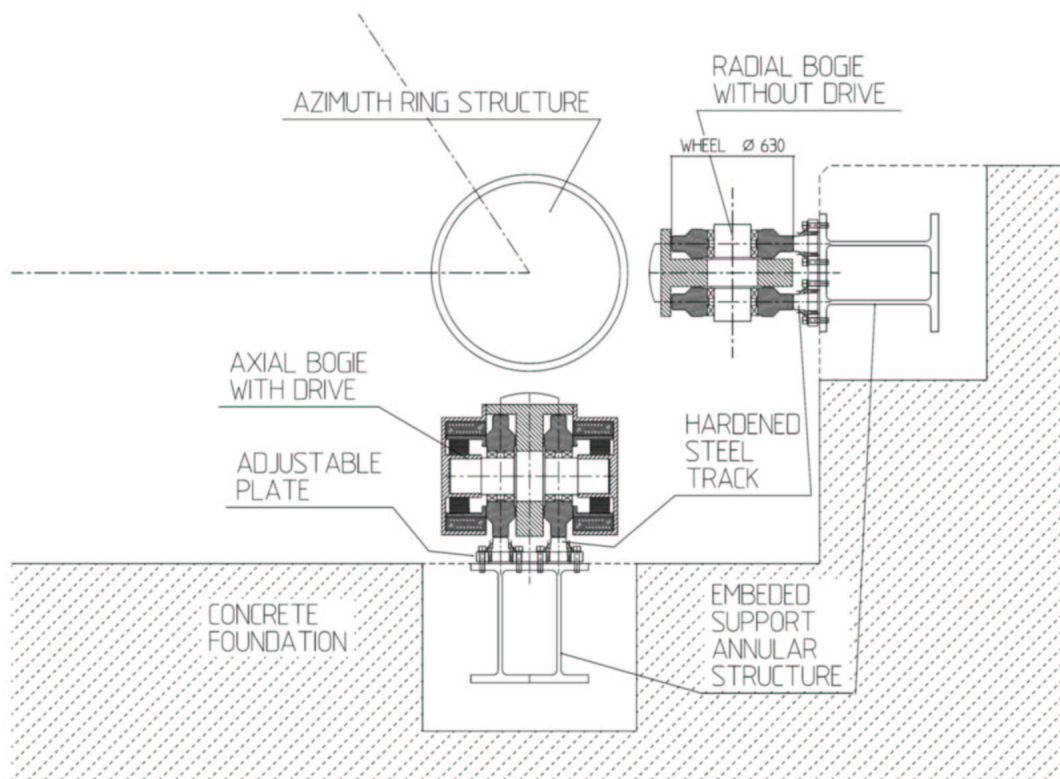


Fig. 5: Axial and radial bogies

¹ Conversely the hydrostatic bearing only constrain the vertical DOF.

8. TRACK ALIGNMENT

The large number of annular tracks, which are necessary to support and constrain the 13100 tons of rotating mass and needs to be aligned and monitored during the operational lifetime, are a major cost. The concept adopted for OWL foresees that for each track only one degree of freedom has to be aligned. Measurement systems which allow real time continuous measurements, similar to the one shown in Fig. 6 and based on communicating vessels principle, provide rapid converging towards the required tolerances. It can also be installed permanently on board of the telescope azimuth ring, to monitor the horizontal level of the telescope.



Fig. 6: Azimuth axial track alignment operation (Fogale level system)

Track alignment tolerance.

The alignment tolerances of the telescope tracks have a major impact on the cost of the telescope. These tolerances are defined according to the range of loads at which the bearing systems can operate. The OWL's bogies can operate with a load ranging from 50 tons, which is the minimum load required to transmit the maximum torque delivered by the drive, to 300 tons, as described in section 5. This large range of operational loads allows relaxing of the tracks alignment tolerances. Again this is a clear advantage in comparison with the tolerance required to hydrostatic bearing systems.

9. IMPACT OF THE FRICTION DRIVE AND BOGIES ON THE TELESCOPE PERFORMANCE

1. Dynamic performance

One of the most important design drivers was to optimize the lowest locked rotor frequencies in order to minimize the impact of the dynamic wind load to the main axis controllability. The higher the locked rotor frequency, the higher the bandwidth of the control loop can be. Therefore, a design solution has been elaborated with a hyperstatic support structure for the azimuth as well as for the altitude bearings. As a result of this design approach also the lowest vertical "pumping" mode of the structure has been increased significantly. This facilitates the control of the individual primary mirror segments.

An adequate FE model has been built to investigate the dynamic and static behavior of the present

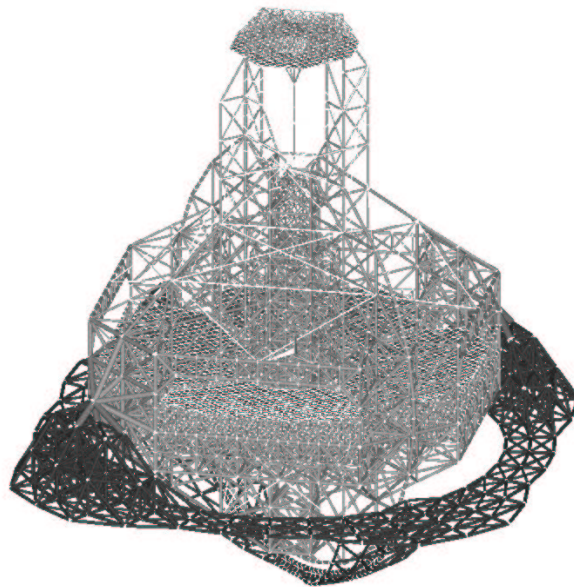


Fig. 7: FE Model of OWL.

baseline design. The FE model shown in Fig. 7 consists of about 28000 elements and 6200 nodes.

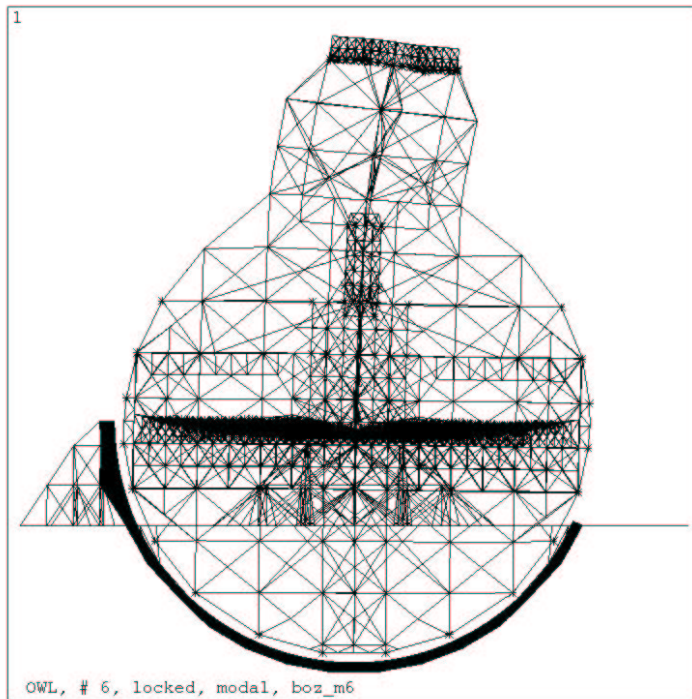


Fig. 8: Locked rotor mode at 2.1 Hz.

The numerous roller bearings and drives are simulated in the model by appropriate boundary conditions at the base of the azimuth structure and by nodal couplings between the azimuth and the altitude structure. The stiffness of the soil is not represented in this model. Both the locked and the free altitude rotor modal analyses have been carried out.

The lowest locked rotor eigenfrequency of the telescope occurs at **2.08 Hz**, and the corresponding free rotor frequency at **2.46 Hz**. As shown in Fig 8, the locked rotor mode shape corresponds to a bending of the M2 tower about the altitude axis. The deflection affects mainly the upper part of the altitude structure. The first dominant piston mode of the telescope corresponds to a vertical vibration of the altitude structure at **3.7 Hz**. The lowest azimuth rotor mode occurs at 2.7 Hz and corresponds to a local rotation of the M2 tower about its vertical axis. The first dominant locked rotor frequency about the azimuth axis becomes 4.4 Hz; in this mode a large part of the altitude and azimuth structure is involved.

Table 2 shows the improvement of the dynamic performance of the present baseline design compared to previous iterations. Due to the new hyper-static support concept with the friction drive and bogies, also the piston mode could be considerably increased. All the listed iterations are based on primary and secondary mirror segments Zerodur. It is obvious, that significant improvement of the performance can be achieved with much lighter SiC mirror segments. This option is at the moment still under investigation.

Baseline Version	Support	Total Mass [ton]	Altitude inertia [$\text{kgm}^2 \cdot 10^6$]	Eigenfrequencies in [Hz]		
				locked	free	Piston
1 [3]	hyper-static	16800	20000	1.37	-	-
2 [4]	iso-static	13700	13200	1.50	1.68	1.79
3	iso-static	12000	9900	1.69	1.88	1.56
4	hyper-static (hydrostatic bearing and direct drive)	16000	9600	1.82	2.33	2.97
6	hyper-static (friction drive and bogies)	13100	10600	2.08	2.46	3.68

Table 2: Dynamic performance comparison.

To investigate the energy content of the various eigenfrequencies, a harmonic response spectrum analysis has been performed. Based on the free rotor model the open loop transfer function was analyzed between the M2 unit displacement (along y-direction) and the uniform altitude motor torque of 1 kNm. The transfer functions shown in Fig. 9 demonstrate the considerable improvement of the present design in terms of stiffness and open loop bandwidth compared to previous ones. The first dominant amplitude peaks represent the lowest free rotor eigenfrequencies listed in Table 2

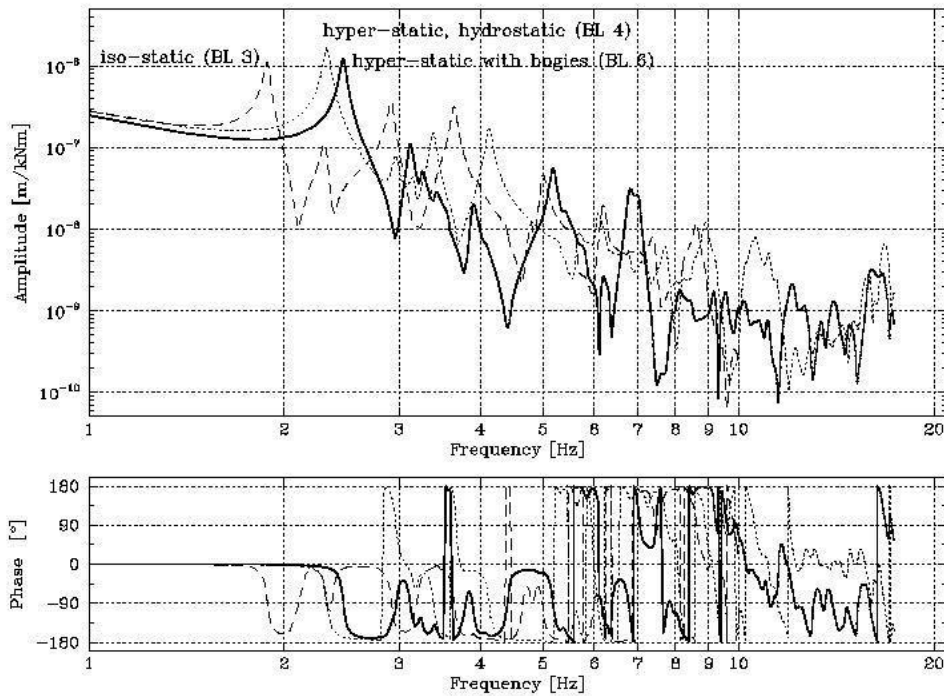


Fig. 9: Open loop transfer functions between M2 unit and altitude motor torque.

2. Static performance

The behavior of the primary and secondary mirror units under gravity load has been investigated to assess wavefront (including tilt) control requirements. Table 3 represents the mean displacements and tilt of the main mirror units for a differential gravity load between 0° and 60° , i.e. the telescope moving from zenith (0°) to 60° from zenith. The differential values between M1 and M2 are indicated in the last row. While the mean differential piston between M1 and M2 is calculated to be about 5 mm, the maximum PTV piston is about 33 mm for M1 and 13 mm for M2, respectively. Since the primary and secondary mirror units rotate in opposite directions, the mean global tilt correction is about 100 arcsec. The mean decenter between the two mirror units becomes about 33 mm. However, due to the flat secondary mirror segments, this error needs not to be corrected. Compared to previous designs, the maximum stress in the steel structure could be reduced by about 50 % down to 200 MPa under gravity conditions. This level of stress ensures now sufficient safety margin to the material's yield limit. The risk of failure of the steel structure due to buckling is being under investigation. The total radial reaction force to be transmitted by the altitude bogies under gravity is 34 MN, whereas the distribution of the forces is non-linear, as in radial roller bearings. The maximum reaction force occurs close to the azimuth axis and the minimum at both ends of the cradle. The impact of quasi-static wind loading on the main mirror units has been investigated too. The

Mirror	Piston (uz) [mm]	Tilt (rotx) [arcsec]	Decenter (uy) [mm]
M1	-4.3	-48	1.1
M2	-8.9	45	-31.9
M2 - M1	-4.6	93	-33.0

Table 3: Mean displacements under differential gravity load ($0^\circ - 60^\circ$).

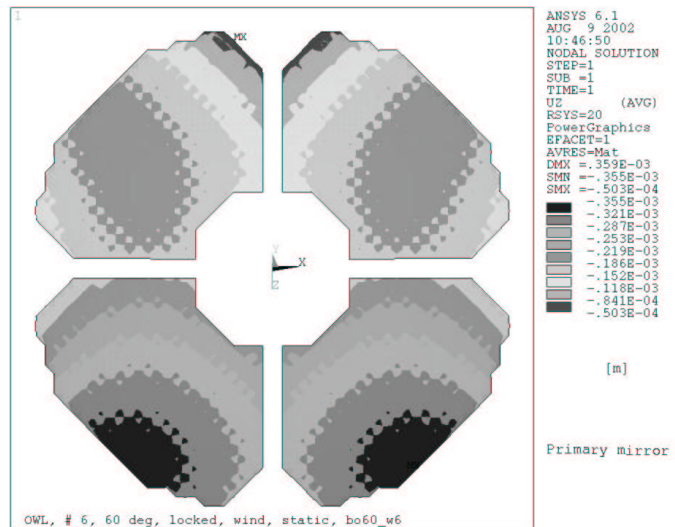


Fig. 10: M1 deformation along Z-axis for 2nd wind load case.

following two wind load cases have been evaluated (more are planned):

1. Telescope in zenith configuration, wind load applied on full telescope, wind speed of 10 m/s, conservative drag coefficient values used, which generates maximum tilt and decenter.
2. Telescope in 60° from zenith position, wind load applied only on M1 and M2, wind speed of 10 m/s, conservative drag coefficient values used, which generates the maximum piston.

Mirror	Piston (uz) [mm]	Tilt (rotx) [arcsec]	Decenter (uy) [mm]
M1	-0.216	0.420	-0.129
M2	-0.336	1.680	-1.132

Table 4: Maximum mean displacements out of both wind load cases.

The resulting mean displacements and tilts of the main mirror units are summarized in Table. 4. Only the worst values out of the two wind load cases have been indicated. The mean piston displacements under worst case operational wind load conditions become for M1 about $220 \mu\text{m}$ and for M2 about $340 \mu\text{m}$, respectively. The mean global tilt of the primary mirror is about 0.5 arcsec and 1.7 arcsec for the secondary. These values are very small for a structure of this size. This confirms again the high stiffness and wind resistance of the telescope.

The primary mirror deformation along the optical Z-axis is shown in Fig. 10 for the 2nd wind load case with the assumption of maximum pressure on the mirror. The maximum relative sag on each of the four mirror quarters results in about $200 \mu\text{m}$. This type of spherical-like deformation is caused by the design of the altitude structure, which consists of four vertical beam walls that provide high vertical stiffness in this region.

10. TELESCOPE MAIN AXES CONTROL FOR BOGIES

A drive system similar to the proposed one has been developed for the VLT coude rotating platforms and the VLTI Delay Lines. In these cases, however, the low stick friction helps significantly to reach a very good tracking performance. The relatively high amount of friction generated by the bogies has a direct impact on the performances. In particular this friction is highly non-linear (slip-stick) and can be only partly modeled as simple viscous force. Theory supported by simulations and previous experience with systems (VLTI Auxiliary Telescopes) having comparable friction, shows that limit cycles are very likely to occur when a traditional proportional-integral position control strategy is implemented.

Two main sources of friction are generated in the bogies: the roller bearings at the shaft of each wheel and the rolling friction of each wheel on the track. Both contribute to the overall friction, although they have different characteristics: the rolling friction is about twice as large as the bearing friction but it has a much lower stiffness at rest.

The friction on the main axes can be modeled by the Maxwell model shown in the following Fig. 11. Each slider (loaded with F_{Ni}) represents a roller bearing friction. The springs (k_i) represent the stiffness at rest. This model describes the hysteresis behavior (Dahl curve) at direction reversal, the distributed breakaway of different bogies and the stiffness at rest. For the friction of the individual sliders the Coulomb model is used. In such a model the friction in the region close to zero velocity is represented by a discontinuity. This simple model on one side is not accurate in proximity of the transition, on the other side is prone to numerical instability in the simulations. More sophisticated models can be used if further experiments and simulations show that the model used is not suitable. A model which has demonstrated to be more reliable is proposed in [5]

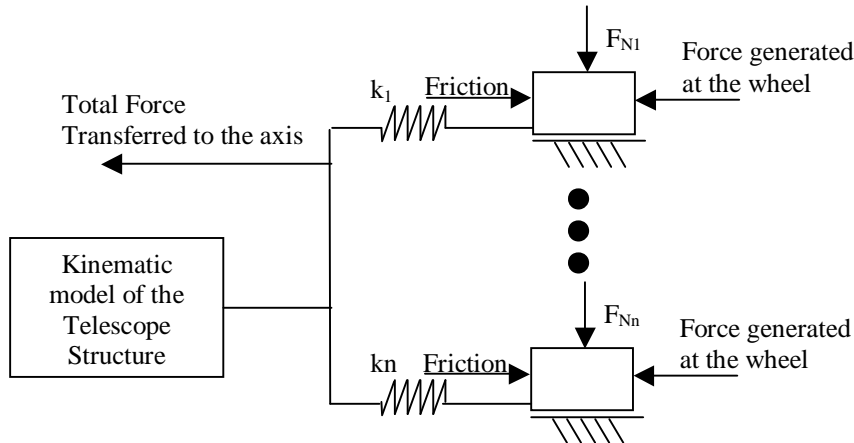


Fig. 11: Friction Maxwell model

Sharing the overall friction among several bogies with similar but slightly different characteristics, smoothing out the sudden change in friction force at direction reversal, is expected to reduce the slip/stick effect and hence reduce the risk of limit cycles. A suitable control system has to be designed to allow attaining the required positioning accuracy in spite of the disturbances, with particular reference to the friction.

From the control point of view, the ideal solution would be to implement a feedback force control at each actuator. The force transferred to the structure could be measured at the support of the bogie. This would virtually compensate for the whole friction and allow the driving system to behave almost ideally. In practice such a solution would be extremely complex requiring a large amount of sensors to be installed, calibrated, maintained and processed, and it would have an unbearable impact on the operation.

A technically simpler solution is proposed, as depicted in Fig. 12. This has a traditional cascaded structure with an outer position loop and an inner velocity loop. The velocity loop supplies the torque reference to all the actuators. Note that the torque is controlled in open loop, while the loop is actually closed on the current of the actuator. A torque equalization is also needed to distribute the torque reference among the torque controllers. The velocity feedback is obtained as an average of the encoder measurement at each wheel. In principle the rotary encoders are not essential (although useful for monitoring and reliability) as the average velocity can be obtained by one single sensor.

The more intuitive structure in which the overall velocity loop is replaced by several velocity loops, locally closed at each wheel, could lead to instability or uneven distribution of the forces among the actuators. This is because the local loop would not have the possibility to detect whether the actuator is actively generating torque or is passively pulled by the structure.

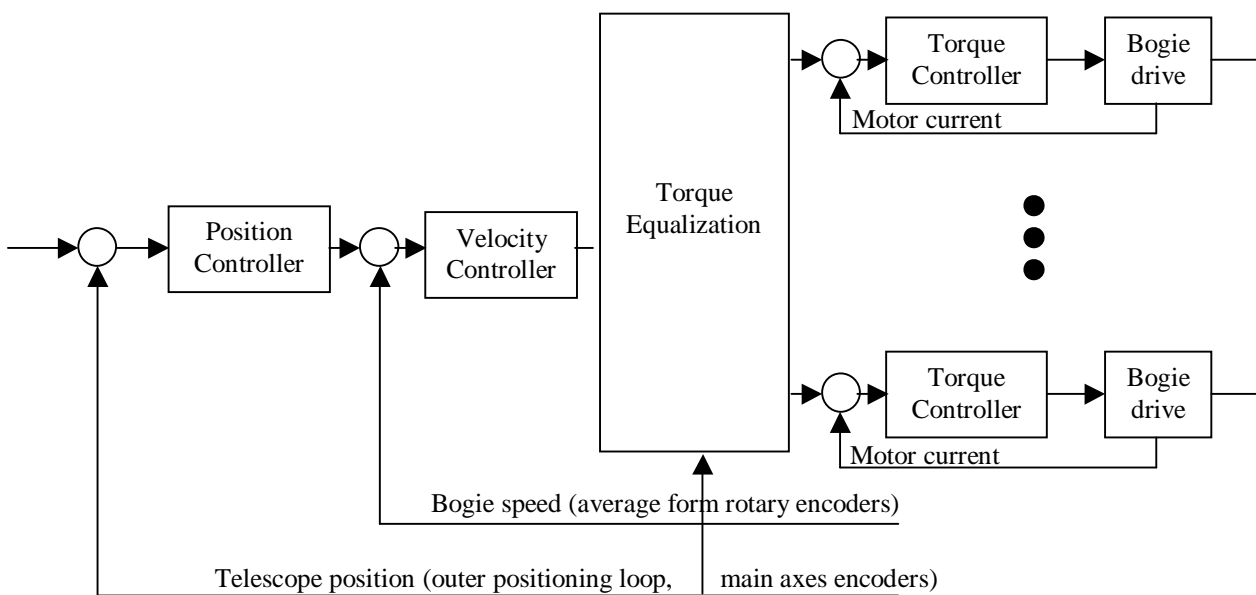


Fig. 12: Main axes controller

For friction compensation two methods should be investigated:

- 1) Introducing a derivative part in the position controller.
- 2) Adaptive feed-forward friction compensation based on LuGre friction mode [5].

The first approach, although simple, has given satisfactory results in the VLTI Auxiliary telescopes where the friction is relatively high. The second method is very promising and can be considered in case the first one cannot lead to satisfactory performances.

11. DRIVE & BEARING SYSTEMS COSTS.

The following Table 5 compares the cost estimates associated to OWL's different drive and bearing systems, delivered ex works. Costs related to transport and integration on site are not shown.

	Direct drive & Hydrostatic bearing	Friction Drive & Bearing (bogies)
Azimuth tracks	87,3	12
Azimuth base plate	Not applicable	11
Azimuth understructure	15	12,8
Azimuth bearing	17,5	Not applicable
Azimuth drive	11,5	Not applicable
Azimuth bogies	Not applicable	21
Altitude cradle tracks	3,7	0,7
Altitude bearing	10	Not applicable
Altitude drive	4	Not applicable
Altitude bogies	Not applicable	4,5
Total	149 MEuro	62 MEuro

Table 5: Cost breakdown of OWL's drive and bearing systems..

Table 5 shows that the adopted solution of the telescope drive and bearing systems has a major impact on the total cost of the project. The difference in cost persists also for transport, integration on site and during the operational lifetime of the telescope in term of RAMS.

The main source of cost difference is the type of annular tracks, which have to be manufactured and aligned within the severe tolerance necessary to the hydrostatic bearings and direct drive systems. The values reported on the table are based on extrapolation of the existing VLT systems

12. CONCLUSION

The bogie solution for OWL's drive and bearing systems can be considered the most suitable solution to bring the performance and costs to an acceptable level. This solution has the potential to guarantee that the telescope can fulfill its requirements with high reliability at a very minimum realization costs and also during the most severe operational and survival load cases. However the friction introduced into the system is not negligible; this requires that further analyses and simulation be performed in order to precisely quantify the "rejected value" at the OWL's main axes. The generated friction be accurately determined by measurement; this will require extensive tests on prototypes. This test campaign is planned within "phase B" of the project. These measurements will clarify the uncertainty related to friction coefficients and stiffness. They will allow dimensioning of the torque motors and improve the definition of telescope control model and strategies, as well as the dynamic performance.

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