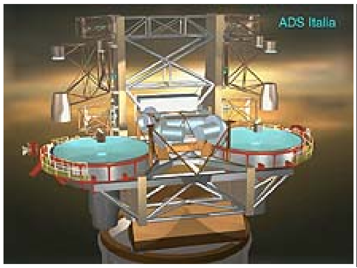

LBT PROJECT 2x8,4m TELESCOPE
Doc.No. : 670s006 Issue : a Date : 15 Mar 2007

LBT PROJECT

2 X 8,4m OPTICAL TELESCOPE

Instrument Rotator and Cable Chain

Systems Analysis

	Signature	Date
Prepared	Dave Ashby, Robert Meeks	15-Mar-07
Reviewed		
Approved		

1. Revision History

Issue	Date	Changes	Responsible
a	15-Mar-07	First draft	Dave Ashby

2. Table Of Contents

3. List of Abbreviations

LBT	Large Binocular Telescope
(L/R)DGR	(Left / Right) Direct Gregorian Rotator
(L/R)(F/C/R)BGR	(Left / Right) (Front / Center / Rear) Bent Gregorian Rotator
RPM	Rotations Per Minute
VAC	Volts Alternating Current
RMS	Root Mean Squared
AGW	Acquisition Guiding and Waterfront unit
ISS	Instrument Support Structure

4. List of Figures

5. List of Tables

6. About This Document

6.1. Purpose

Starting with the LBT instrument rotators and cable chain technical specification [RD1] and requirements [RD2], this document illustrates the development of technical specifications for all major components. The relevant specifications of the selected components are documented along with their specific impact on the design. For the complete design description, see [RD5].

6.2. Reference documents

[RD1]	670s001c	Rotator and Cable Chain Technical Specification
[RD2]	670s003a	Instrument Rotator and Cable Chain Requirements Analysis
[RD3]	670s004a	Instrument Rotator and Cable Chain Conceptual Design Description
[RD4]	670s005a	Instrument Rotator and Cable Chain Conceptual Design Review Record
[RD5]	670s007a	Instrument Rotator and Cable Chain Detailed Design Description
[RD6]	002s004g	Telescope Specification for the Large Binocular Telescope
[RD7]	600s001a	LBT F/15 Direct Gregorian Focus Optical Design
[RD8]	600s003a	LBT F/15 Bent Gregorian Focus Optical Design
[RD9]	670s014a	Instrument Rotator and Cable Chain Cable Chain Torque Estimate
[RD10]	671x002a	Bent Gregorian Rotator Bearing Information
[RD11]	671s010a	Instrument Rotator and Cable Chain Rotator Gallery Analysis
[RD12]	671x005g	Bent Gregorian Gear Drawing
[RD13]	674x004b	Direct Gregorian Gear Drawing
[RD14]	670s012a	Instrument Rotator and Cable Chain Mechanical Design Calculations

7. Introduction

This document fills a gap between the detailed design description presented in [RD5] and the specification and requirements analysis presented in [RD1] and [RD2]. Rather than duplicate the information provided in those documents, it is intended to provide analysis of the requirements to explain how major design decisions and component selections were made and to summarize design analysis performed to validate the compliance of the design with the specifications. In doing so there is inherently some duplication of the description provided in [RD5] and restatement of the requirements and specifications.

This document is intended to be mainly a narrative review of analysis developed in more detail elsewhere. The goal is to provide a brief description of the analysis performed and a summary of the relevant results while referring the interested reader to the more detailed analysis description.

8. Instrument Support

A basic requirement of each rotator is to provide a physical interface for connecting an instrument to the telescope, locating it properly with respect to the focal plane, and allowing the instrument to turn to track the apparent motion of the sky. The motion requirements and specifications are considered elsewhere in this document. This section describes the analysis of the structures and bearings that support each instrument.

8.1. Structures

8.1.1. Bent Gregorian rotator

The Bent Gregorian Rotator attaches to the Rotator Gallery structure. This structure is an existing part of the telescope and cannot be significantly modified as part of the rotator development project. It affects the Bent Gregorian Rotator by potentially deforming in response to changes in the telescope elevation angle.

The rigidity of the rotator structure was established through finite element analysis during the original design of the telescope but documentation of this analysis has not been located. Because the current rotator design uses the same bearings and attachment points to the structure, only a relatively simple analysis was performed for this project to verify that the deformation of the structure was not excessive given the heavier instrument weight. It was concluded that the rotator gallery is sufficiently rigid to support a 3500 kg instrument in all sky orientations based on general information regarding tolerable deflections of astronomical instruments. A summary of this analysis is contained in [RD11]

8.1.2. Direct Gregorian rotator

The Direct Gregorian Rotator attaches to the bottom of the Primary Mirror Cell. It has similar requirements on positioning and rigidity. No new analysis was performed of the mirror cell structure to determine its rigidity. It is believed to be adequately stiff based on previous analysis and it would be infeasible to significantly modify it at this stage.

The bearings and gears attach directly to the bottom of the primary mirror cell. An instrument mount structure then attaches to the bearing. The analysis of this structure is discussed below.

An analysis of the instrument support structure verified that its stiffness was sufficient to limit its differential deflection to approximately 25 microns between zenith and horizon when a full instrument load was distributed uniformly among the 6 cones.

8.2. Bearing stiffness

The stiffness of the Bent Gregorian Rotator bearing was provided by the manufacturer [RD10] and are reproduced in the table below. The table also includes a summary of the estimated deflection of the center of the instrument mounting surface. No information was available regarding the stiffness of the DGR bearing so the values for the BGR bearing were used as a rough estimate.

Table 1: Bearing stiffness and resulting instrument deflection

	BGR	**DGR (est.)**
Radial stiffness	3380 N/ μm	3380 N/ μm
Axial stiffness	4290 N/ μm	4290 N/ μm
Moment stiffness	1270 Nm/ μrad	1270 Nm/ μrad
Decenter	10.2 μm	10.2 μm
Defocus	8.0 μm	8.0 μm
Focal plane tilt	1.5 as	4.2 as

9. Specifications

The performance specification which originate from [RD1] are listed in Table 2. All additional performance assumptions used in the analysis in this document are defined here.

Table 2: Instrument rotator and cable chain performance specifications

	DGR	BGR
Angular velocity (service mode)	5 °/s	
Angular velocity (observing mode)	1.5 °/s	
Angular acceleration	0.3 °/s[2]	
Field diameter	0.5°	0.2°
Short term (5 seconds) on-sky error	10 mas (1)	
Instrument + AGW mass	3500 kg	
Maximum bearing moment	35000 Nm	20000 Nm
Maximum payload imbalance torque (service mode)	3000 Nm	2000 Nm
Maximum payload imbalance Torque (observing mode)	300 Nm	200 Nm
Cable chain relative angle	±5.0 °(2)	
Cable chain tracking accuracy	±60 as	
Cable chain operation rotation range	540 °	
Maximum allowed time for system initialization	5 minutes	5 minutes

9.1. Rotator payload definitions

Much of the analysis appearing in this document requires some assumptions concerning the rotator payload, defined as the AGW plus the instrument. The payload is specified in Table 3 in terms of maximum mass and maximum gravitational bearing moment. However, much of the analysis that follows depends on the payload moment of inertia, which depends on the distribution of mass. For the purpose of systems analysis the distribution of the mass is assumed to be uniform.

Table 3: Rotator payload definition

	DGR	BGR
Mass	3500 kg	3500 kg
Envelope diameter	3 m	2 m
Moment of inertia	3940 kg m^2	1750 kg m^2

[1] Each instrument rotator and guider is allotted 0.015 arcsec rms of error combined. The rotator and guider are assumed to be given equal parts of this budget or 0.010 arcsec.
[2] This specification has not been met by the design.

9.2. Rotator absolute error

The instrument rotator specification [RD1] does not specify the absolute accuracy requirement. A reasonable goal is extrapolated from the telescope specification [RD6] using the following reasoning. The short-term tracking accuracy of the telescope is specified as 30 mas and the absolute pointing accuracy is 300 mas or 10x the short-term tracking accuracy. The same radiometric goal will be set for the instrument rotators. This accuracy goal is not a requirement.

Table 4: Rotator on-sky absolute error

	DGR	**BGR**
On-sky absolute error	100 mas	100 mas

9.3. Cable chain relative angle

The cable chain relative angle specification was not met. The normal range of the relative position is ±1º. The relative position can be sensed over ±2.3º. This performance falls short of the specified ±5º but there is no known reason this limitation will interfere with normal operation.

10. Instrument Rotator Motion

10.1. Performance allocation

The performance allocation begins with the relative allocation of error amongst the subsystem elements. The allocations in this case originate from anticipated performance of existing elements and experience with similar control systems. This said half of the maximum short-term on-sky error is allotted to servo residuals resulting from external disturbances. The remaining tolerable error is equally divided between the encoder and the mechanical bearing run out. All errors are assumed to be uncorrelated.

Table 5: Rotator on-sky short-term error allocations

Mechanical run out	5.00 mas
Encoder accuracy	5.00 mas
Servo residuals	7.07 mas
Total error	10 mas

Both the direct and bent Gregorian rotators foci utilize the f/15 adaptive secondary. The plate scale for both foci is 0.600 mm/as ([RD7] and [RD8]) due to the undersized infrared secondary. The servo and encoder must hold the tolerances defined in Table 6 over the entire FOV. The definition of "short-term" is 5s or 7.5º at the maximum tracking speed of 1.5º/s.

Table 6: Rotational and transitional error allocations

	DGR	**BGR**
Plate scale	0.600 mm/as	
FOV	0.5º	0.2º
Field diameter	1080 mm	432 mm
Mechanical error (7.5º of rotation)	3.0 μm	
Servo residuals	1.62 as	4.05 as
Encoder accuracy (7.5º of rotation)	1.15 as	2.86 as

Servo demand is specified in [RD1] to be synchronized to 22 μs. This level of synchronization will yield 0.12 as of tracking error that must be absorbed in the servo residuals.

10.2. Brakes

The brakes are intended to hold the rotator when the motors are turned off and to be released when the rotator is operating. Ideally the brake capacity would exceed the motor capacity but, since the brake capacity determines the stopping torque exerted under emergency stop conditions, an upper limit on the brake torque was required.

The brakes were also required to be pneumatically released and spring engaged for safety reasons. This is an uncommon configuration for industrial brakes and this limited the selection of available brakes to only a few manufacturers, one of which was the manufacturer that provided the brakes for the main telescope axes. A device from this manufacturer was selected. The table below summarizes the brake selected. The same brakes are used on both rotators.

Table 7: Rotator brake specifications

Make / Model:	MWM Freni-Frizioni SRL
Type	Negative Pneumatic Single Disk Brake
Rated torque	400 N m
Maximum speed	2100 rpm
Diameter	236 mm
Mass	10.5 kg
Inertia (est.)	0.15 kg m^2

The inertia estimates in the table are based on the brake mass and diameter. The brake is mounted so that its body rotates while the rotator is operating and this inertia contributes to the motor inertia.

10.3. Bearing runout

Both the instrument rotators use Kaydon two row angular contact bearings for the main bearing. These bearings contribute axis runout errors in two ways. Any deviation of the races from circularity will cause the entire rotating portion of the bearing to shift laterally and in general it will orbit the mean rotation axis. Similarly, imperfections in the balls will produce a higher frequency (and non repeatable) runout of the rotation axis. The runout of the races is specified to be 0.0006 inches (15.2 microns) and was measured to be about 75% of that value on the Bent Gregorian Rotator bearing.

Both the Bent Gregorian Rotator and the Direct Gregorian Rotator bearings use rolling elements of AFBMA ball class 24. This means individual balls may have random surface errors of 2 microinches (0.05 microns) and diameter errors of 24 microinches (0.61 microns). These ball imperfections produce random runouts because they continuously reorient themselves.

Table 8: Rotator bearing runout summary

	DGR	**BGR**
Runout (full rotation)	15.2 µm	15.2 µm
Runout (7.5º of rotation)	1.27 µm	1.27 µm
Random error	0.61 µm	0.61 µm

The starting torque (static friction) in the Bent Gregorian Rotator bearing is specified to be less than 285 foot-pounds (386 Nm). Measurements of the actual friction torque on the completed bearings verified that this specification was easily met. The actual friction torque was approximately 32.5 Nm. These measurements were made with the bearing loaded with about 11 kN. Similar torque measurements were taken with the inner ring and the outer ring loaded and while rotating both clockwise and counterclockwise. The results were similar in all cases.

The frictional torque in bearings comes from two sources – rolling friction of the balls against the races and sliding friction of the seals against the ring surfaces. The rolling friction depends on the interfacial forces acting within the bearing. Since these are produced by a combination of preload and external loading it is not always possible to conclude a definite relationship between external load and frictional force. It is also the case that in a large bearing such as these, the seal friction often dominates. These factors suggest that the measurements are probably typical of the friction that will actually be experienced on the LBT rotators. These are listed as the expected friction in the table below.

The direct Gregorian instrument rotator also uses a Kaydon two row angular contact bearing but with a pitch diameter of 2925 mm. It is assembled with a similar amount of preload and uses the same type of seal. Because no information was available regarding the specified maximum or the measured actual torque for these bearings, an estimate was made by scaling the BGR bearing information by the square of the pitch diameter. This effectively assumes that the friction is produced uniformly at the pitch diameter of the bearing and is a constant value per linear distance. This is consistent with the mechanics of both rolling and sliding friction.

Table 9: Rotator bearing friction summary

	DGR (est.)	**BGR**
Maximum frictional torque	1170.1 N m	386 N m
Expected frictional torque	98.5 N m	32.5 N m

10.4. Gear Ratios

The reduction ratios for both the direct and bent Gregorian rotators were all but defined by previous design decisions and built into the existing telescope hardware. The gear for the direct Gregorian rotator had already been purchased as had the gears for the cable chains. While it would have been possible to adjust the reduction ratios somewhat, the geometry built into other hardware placed constraints on this. Ultimately, the previously chosen ratios were used.

Table 10: Rotator gear ratios

	DGR	BGR
Rim gear pitch diameter	3300 mm	1880 mm
Pinion pitch diameter	132 mm	80 mm
Rim-pinion reduction ratio	25:1	23.5:1

10.5. Motors and drives

10.5.1. Requirements

10.5.1.1. Torque

Sources of friction include, the main bearing, main bearing seal, motor bearings, motor bearing seals, pinion misalignment and cables. The friction of the main bearing and seal is defined is Table 9. The cable friction and pinion friction are both considered negligible. The designed frictional torque of the DGR bearing is unknown; it is estimated using the known value of the BGR bearing, assuming the same force per unit length. This estimate is considered to be very high. The frictional torque is also assumed to be due to Coulomb friction and thus velocity independent.

The second source of torque is due to acceleration. The acceleration torque depends on the moment of inertia asserted in Table 11 and the required acceleration of $0.3°/s^2$ from [RD1]. The moment of inertia of the rotator itself is considered to be insignificant compared to that of the payload.

The third source of torque is due to payload imbalance. The payload imbalance torque can be particularly high in service mode and relatively low in tracking mode. The imbalance torque is defined in [RD1].

Table 11: Rotator rim gear torque requirements

	DGR	BGR
Acceleration	$0.3°/s^2$	$0.3°/s^2$
Maximum acceleration torque	20 N m	9 N m
Frictional torque	1170 N m	386 N m
Service imbalance torque	3000 N m	2000 N m
Tracking imbalance torque	300 N m	200 N m
Peak service torque	4190 N m	2395 N m
Peak tracking torque	1490 N m	595 N m
Continuous service torque	4190 N m	2386 N m
Continuous tracking torque	1490 N m	586 N m

In computing the required pinion torque, the backlash compensation algorithm is assumed to tolerate torque sign transitions. Because the tracking performance requirements are relaxed in service mode, both motors share the load. However, in tracking mode only one motor delivers the total torque. Additional torque is required to preload the gear and to compensate for internal motor friction. The gear ratio is defined in Section 10.4.

Table 12: Rotator pinion gear torque requirements

	DGR	**BGR**
Gear ratio	25:1	23.5:1
Peak service pinion torque	83.8 N m	51.0 N m
Continuous tracking pinion torque	59.6 N m	24.9 N m

10.5.1.2. Pinion speed

The maximum pinion speed is encountered when the rotator is slewing in service mode.

Table 13: Rotator pinion speed

	DGR	**BGR**
Maximum rotator speed	5°/s	5°/s
Gear ratio	25:1	23.5:1
Pinion speed	20.8 rpm	19.6 rpm

10.5.2. Motor selection

The instrument rotators utilized directly driven pinions. This design decision, which results in an unconventionally low motor inertia relative to the load and very low damping, developed from positive experiences with the Elevation and Azimuth axes of telescope. A large, slow-turning motor generates less vibration and a control system with a low gear ratio relies on active stiffening rather than passive mechanical stiffening. A large motor will also have as low thermal resistance and thus lower surface temperatures for a given input power.

Table 14: Rotator motor specifications

Make / Model:	Kollmorgen Danaher / D103M
Type	Permanent Magnet Brushless AC
Peak torque	501 N m (drive limited to 250 N m peak)
Continuous torque	115 N m / (92 N m at altitude)
Maximum speed	120 rpm (73 rpm at rated torque)[3]
Torque constant	21.4 N m/A
Rotor inertia	0.175 kg m^2
Pole pairs	16
Winding inductance	35 mH
Winding resistance	5.75 Ohms
Thermal resistance	0.253 ºC/Watt
Static friction	3.118 N m
Viscous friction	31.95 N m /krpm

The reflected payload inertia is large compared to that of the motor. It is desirable to keep this ratio to within a factor of 5 or 10. This mismatch is of some concern because of the intent to allow motor torque sign transitions. The additional inertia of the brake will help but software differential damping is necessary to ensure stability during sign transitions. The main encoder is necessary to generate the full service torque unless a flywheel is integrated into the motors to increase the motor inertia. Integration of a flywheel will diminish the achievable bandwidth.

Table 15: Rotator reflected payload / motor inertia ratio

	DGR	**BGR**
Payload inertia	3940 kg m^2	1750 kg m^2
Motor and brake inertia	0.175 kg m^2	0.175 kg m^2
Gear ratio	25:1	23.5:1
Reflected payload/motor inertia ratio	19.4:1	9.75:1

The motor is capable of much high speed than is necessary. Over-speed detection and automatic shutdown electronics must be integrated into the control system.

10.5.3. Thermal analysis

The motor windings are a significant source of heat located very near the optical path. Both motors are insulated but if the winding temperature gets too high, it will become impractical to mask the elevated temperature. The required tracking power is highly dependent on the magnitude of the friction. The expectation is that the friction term is

[3] The motor is capable of running at 120 rpm unloaded. Under load, it is rated for 73 rpm. The integrated encoder and associated interpolator indirectly limit the speed to approximately 30 rpm since the interpolation factor is quite high.

much lower than the estimates, which will drop the DGR motor temperature dramatically. Even with the relatively high friction, the DGR temperature is considered manageable.

Table 16: Rotator motor heat and temperature while tracking

	DGR		BGR	
	Motor 0	**Motor 1**	**Motor 0**	**Motor 1**
Pinion tracking torque	58.8 N m	0 N m	24.9 N m	0 N m
Motor static friction	3.1 N m			
Motor viscous friction	Negligible			
Total motor torque	61.9 N m	3.1 N m	28.0 N m	3.1 N m
Winding current	2.89 A	0.145 A	1.31 A	0.145 A
Thermal losses	48.1 W	0.121 W	9.84 W	0.121 W
Winding temperature rise	12.2 ºC	0.03 ºC	2.49 ºC	0.03 ºC

10.5.4. Acceleration limits

The required motor torque of the instrument rotators is dominated by the imbalance and friction and the inertial acceleration torque is insignificant. The maximum achievable acceleration is therefore very high (about 120 $^{o}/s^{2}$ for the DGR and even higher for the BGR) under drive peaks. The acceleration must be limited by software to 50 $^{o}/s^{2}$ under all conditions except for emergency stops. See Section 11.5 for analysis concerning the emergency stop case.

10.5.5. Line power

The line power requirements are computed by accounting for drive efficiency. The calculation uses the pinion torque requirements defined in Table 12, the pinion speed requirements define in Table 13 and the motor specifications define in Table 14.

Table 17: Rotator line power requirements

	DGR	**BGR**
Maximum pinion torque	83.8 N m	51.0 N m
Maximum motor current	4.06 A	2.53 A
Winding voltage	70.0 V	58.5 V
Continuous power per drive	492 W	256 W
Drive efficiency	80%	80%
Required line power	1.23 kW	641 W

10.6. Motor mounts

10.6.1. Requirements

The stiffness goal is to set the frequency of the first flexible mode to no less than 10 Hz with the full inertial payload. The required locked-rotor stiffness is given in Table 18.

Table 18: Rotator motor mount stiffness requirements

	DGR	**BGR**
First flexible mode frequency	10 Hz	10 Hz
Estimated inertia	3940 kg m^2	1750 kg m^2
Locked rotor stiffness (at rotator bearing)	1.56×10^7 N m/rad	6.91×10^6 N m/rad
Locked rotor stiffness (at motor bearing)	1.24×10^4 N m/rad	6.26×10^3 N m/rad

Motor, motor mount, payload and gear characteristics have been used to estimated the plant response. The anticipated plant response of the DGR is illustrated in Figure 1 and the plant response of BGR is illustrated in Figure 2. The plant response illustrates the effects of cross damping terms.

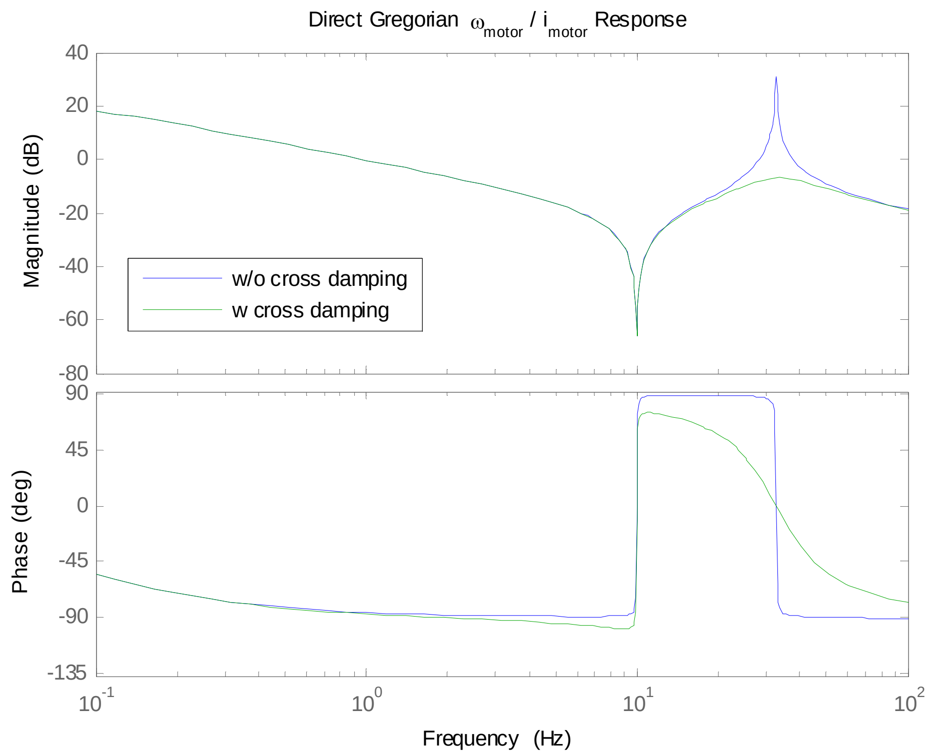

Figure 1: Estimated DGR plant response

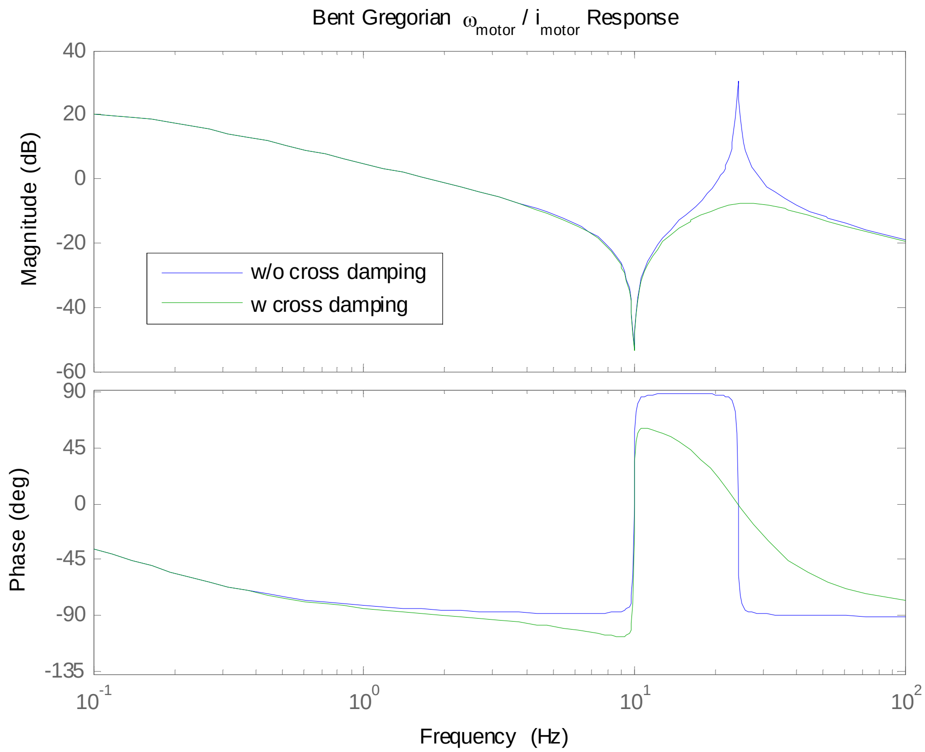

Figure 2: Estimated BGR plant response

10.6.2. Bent Gregorian rotator

The rotator gallery was designed and constructed to mount two small motors for driving the rotator. These are not adequate for mounting the larger motors required to meet the revised specifications. Attempts were made to use the existing interfaces to ensure that the pinions were automatically properly positioned with respect to the large gear.

The servo analysis discussed above determined the required stiffness of the motor mount as seen by the pinion. Besides the mount, there are several other contributors to this stiffness including rigidity of the connection of the mount to the rotator gallery, pinion shaft rotational stiffness, and rigidity of the rotator gallery itself between the motor mount and the rotator gear. Some of these are very difficult to accurately estimate or calculate. To ensure that adequate stiffness is available, a stiffness requirement of 10 times the value determined by the servo analysis was established for the motor mount.

The design of the motor mount is described in [RD5]. A finite element analysis of the motor mount was performed to evaluate the achieved stiffness. The analysis assumed the mount was rigidly attached to the flange of the top beam of the rotator gallery. Arbitrary loads (1000 N) were applied at the location of the pinion contact of the gear at the approximate angle of contact between the gears.

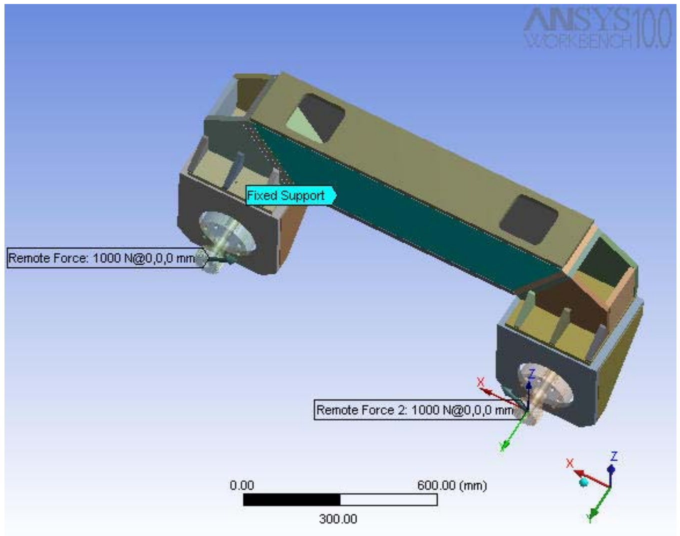

Figure 3: BGR motor mount showing applied loads

The stiffness was then found using the resulting deformation averaged at the motor interface. The stiffness was found to be 167.1 N/micron. This stiffness acts tangentially to the ring gear and can be converted to an effective rotational stiffness at the pinion by multiplying by the square of the ring gear pitch diameter. The effective stiffness obtained this way was approximately 150 MNm/rad or over 3 orders of magnitude greater than required.

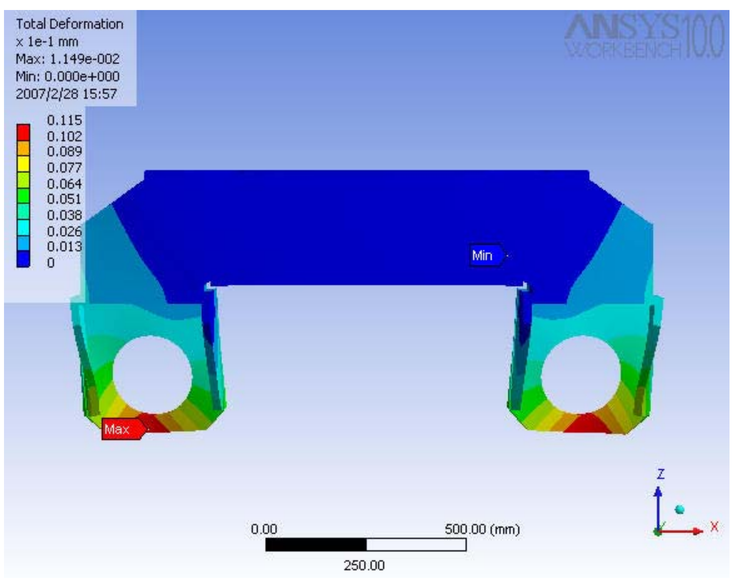

Figure 4: BGR motor mount FEA resulting deformation

The connection between the motor mount and the rotator gallery is through the upper flange of the top beam of the gallery. Because the pinion is loaded approximately 100 mm inboard (away from the mirror side) of the interface, the effect of the loading of the connection will be to open the interface at the top. This joint uses ten M12 structural bolts to preload the joint to provide stiffness. Each bolt develops about 70 kN of compression force in the connection for a total compression of 700 kN. The opening force developed by a 25 kN total interface force between ring gears and pinions corresponding to both motors operating at peak torque is about 12.5 kN, clearly much smaller than the preload.

10.6.3. Direct Gregorian rotator

Analysis of the pinion and shaft summarized later in this document found the pinion stiffness on the motor axis to be approximately 220 kNm/rad when torsion and bending are considered. This is considerably higher than necessary for achieving the required control stiffness. This does not include the flexibility of the mounting ears or mounting plate but these are expected to be several orders of magnitude stiffer than the pinion itself.

10.7. Encoders

10.7.1. Requirements

The encoders must be capable of slewing at the rate of 5º/s and must adhere to the encoder accuracy defined in Table 19. Additionally, the main encoders and motor encoders must have sufficient resolution to compute the rate with adequate accuracy and bandwidth.

As the tracking speed decreases, non-linear features of the plant become evident. The damping appears to increase and the phase and bandwidth of the rate feedback deteriorates which can cause instability and hunting at very low speed. Compensation using gain scheduling or a dead-band is common but effectively decreases the control bandwidth at very low speeds. The minimum tolerable bandwidth of the rate feedback is asserted here to be 10 Hz. If a minimum optimal tracking speed is specified in proportion to the allotted servo error, a reasonable encoder resolution can be defined.

Table 19: Rotator encoder performance requirements

	DGR	**BGR**
Minimum rate feedback bandwidth	10 Hz	10 Hz
Minimum optimal tracking speed	6 as/s	15 as/s
Main encoder resolution	0.0750 as	0.188 as
Main encoder accuracy (7.5º of rotation)	1.15 as	2.86 as
Motor encoder resolution	1.88 as	4.41 as

10.7.2. Main encoder selection

The main encoder selected for all instrument rotators is the Farrand Inductosyn©. A single encoder strip is wrapped around a surface which is integrated into the bull gears for the bent Gregorian rotators and into the instrument mounting assembly for the direct Gregorian rotators. In each case, two read heads are mounted 180º apart to make the main encoder insensitive to bearing run-out. Estimated encoder characteristics are defined in Table 20.

Table 20: Rotator main encoder specifications

	DGR	BGR
Make / Model	Farrand / Inductosyn©	
Type	Linear resolver	
Tape length	9.107 m	4.712 m
Cycle pitch	2.54 mm	2.54 mm
Required resolution	0.527 μm	1.32 μm
Slew frequency (5°/s)	49.8 Hz	25.8 Hz
Interpolation factor	14-bits / head	14-bits / head
Resolution	0.155 μm (22 mas) / head	0.155 μm (43 mas) / head
Max speed[4]	6.5 °/s	12.6 °/s

The short-term accuracy of the Inductosyn© is limited primarily by the accuracy and stability of the electronics.

Table 21: Rotator main encoder interpolation electronics stability requirements

	DGR	BGR
Angular accuracy	1.15 as	2.86 as
Tape diameter	2899 mm	1500 mm
Linear accuracy	2.57 μm	3.41 μm
Electronics offset and gain stability	0.101 %	0.134 %

The electronics stability requirements are sufficiently low that it should not be necessary to trim the gain and offset. Component tolerances should be sufficient for holding the specification.

10.7.3. Main encoder mounting

There are two encoder components that must be mounted on each instrument rotator, the tape containing the serpentine pattern and the read head that moves along the tape and senses position. The tape itself must be mounted on a circular surface machined to

4 The estimates here are computed based on the Analog Devices AD2S83 resolver-to-digital converter which is the core part used in many resolver-to-digital converters.

relatively high precision and with a closely controlled diameter to ensure there is no gap between the ends of the tape where they join and to maintain a constant scale along the tape. The primary requirement of the read heads is that they be accurately positioned with respect to the tape and remain rigidly in place at all times. Both components require electrical cables connected to them.

The diameter on which the encoder tape is mounted was determined somewhat arbitrarily by identifying a convenient surface on the structure where a precision mounting surface could be incorporated. The tape mounting surface for the Bent Gregorian Rotator is a feature machined on the rotator gear. On the direct Gregorian rotator the encoder tape mounts on the instrument support structure near the bearing.

Once an appropriate surface was selected, an exact encoder diameter was determined using the following steps:

1. The tape length corresponding to the approximate diameter was determined by calculating the circumference of the surface.
2. This length was divided by the tape pitch and truncated to find the integer number of periods that fit within the approximate length.
3. The actual tape length was then found by multiplying the integer number of tape periods by the tape pitch. This is the untensioned length of the tape.
4. The exact diameter was then found by dividing by π and subtracting the thickness of the tape. The tape thickness must be subtracted because the length calculation applies to the middle of the tape.

The surface was then toleranced by determining diameter limits that would require the tension in the tape to be half or double the nominal tension recommended by the manufacturer. The manufacturer recommends a nominal tension of 40 pounds (178 N) be applied to the tape during installation. The incremental length and corresponding diameters of the tape for 20 pounds (89 N) and 80 pounds (356 N) of installation tension were calculated. These were added to the exact diameter determined about to establish manufacturing limits on the finished diameter of this surface. The resulting tolerance, while tight, was not identified as unreasonably difficult by any manufacturer consulted nor was it considered a cost driver for this component.

Table 22: Rotator main encoder tape mounting dimensions

	DGR	**BGR**
Nominal tape diameter	2900 mm	1500 mm
Exact tape diameter (untensioned)	2899.103 mm	1499.577 mm
Minimum surface diameter	2899.255 mm	1499.656 mm
Maximum surface diameter	2899.711 mm	1499.892 mm

The encoder read head mounts must satisfy two functions: they must hold the encoder read heads rigidly and provide the means for adjusting their orientation and position relative to the tape. The mounts described in [RD5] provide the required adjustments.

10.7.4. Motor encoder selection

The motor encoders integrated into the Kollmorgen D103M motor are the Heidenhain ECN 113. These encoders are used to by the drive to control commutation. The drive also performs the necessary interpolation up to 256x. The motor encoders allow near-optimal tracking down to speed as slow as 5 as/s.

Table 23: Rotator motor encoder specifications

Make / Model	Heidenhain / ECN 113
Type	optical single-turn absolute
Power supply	5 VDC, 180 mA
Resolution per turn	13-bit absolute / 2048 analog cycles
Number of turns	single turn
Linearity	±20 arcseconds
Interface	EnDat 2.2 / RS 485
Resolution (256 x interpolation)	1.24 as

10.7.5. Absolute encoder selection

The absolute accuracy of the instrument rotator does not appear in the rotator specification [RD1] or the telescope specification [RD6]. A multi-turn absolute encoder is installed on each drive motor. The absolute encoders establish the rotator position at startup.

Table 24: Rotator absolute encoder specification

Make / Model	Pepperl and Fuchs / AHM58-H
Type	optical multi-turn absolute
Power supply	10-30VDC, <180 mA
Resolution per turn	16-bit
Number of turns	14-bit
Linearity	±2 LSB
Interface	SSI / RS 422 Drivers
Maximum readout rate (< 50 m cable)	~10 kHz

Error in the absolute position is due to encoder non-linearity, gear-tooth error and windup of the motor mounts due to payload imbalance. The average of the two absolute encoders is used to compute the initial position, which improves all but the imbalance term. Additional pointing errors exist due to telescope flexure, but these terms are only dependent on the elevation angle and are thus removed by the pointing model.

Table 25: Rotator absolute position accuracy estimation

	DGR	**BGR**
Absolute encoder non-linearity	3.16 as	3.37 as
Rim gear tooth error	4.00 as	7.02 as
Pinion tooth error	4.00 as	7.02 as
Windup under full tracking imbalance	3.97 as	5.97 as
Total rotational error	6.06 as	9.52 as
Total on-sky absolute error	26.4 mas	16.6 mas

11. Cable Chain Motion

11.1. Performance allocations

The cable chain error is equally divided between what is thought to be the three major contributors in Table 26.

Table 26: Cable chain error allocation

Relative position sensor error	34.6 as
Cable chain servo error	34.6 as
Mechanical backlash	34.6 as
Total error	60 as

11.2. Gears and bearings

There are two gear reduction points in the cable chain drive train. On each axis the inner cable tray is driven by a pinion which is in turn driven through a gear box. The reduction ratios between the ring gear and pinion, through the gearbox, and the combined ratios are given in the table below:

Table 27: Cable chain drive train gear ratios

	DGR	**BGR**
Rim gear pitch diameter	3600 mm	2136 mm
Pinion pitch diameter	120 mm	96 mm
Rim-pinion reduction ratio	27.3:1	22.25:1
Gearbox reduction ratio	10:1	10:1
Combined reduction ratio	273:1	222.5:1

The allowable mechanical backlash in each component could be computed by partitioning the allocated backlash to the ring gear, pinion, and gearbox. But the ring gear already exists and the pinion should be manufactured with a similar tooth-to-tooth error. This effectively uses much of the allocated error.

Each ring gear has a single tooth pitch accuracy of 0.025 mm [RD12], [RD13]. This introduces a rotational backlash against a locked pinion determined by the ring gear moving this distance at the pitch radius of the ring gear. A similar effect happens at the pinion except the backlash must be divided by the reduction ratio. The gearbox selected has 4 arcminutes of backlash at its output. This must also be divided by the pinion reduction ratio to obtain the resulting rotational backlash at the cable chain axis. The total backlash is then the algebraic sum of these contributors. Although they are independent, it is more appropriate to add them algebraically rather than in quadrature. The table below summarizes the backlash.

Table 28: Cable chain drive train backlash contributions

	DGR	BGR
Rim gear single tooth pitch accuracy	0.025 mm	0.025 mm
Rim gear rotational backlash	2.9 as	4.8 as
Pinion single tooth pitch accuracy	0.025 mm	0.025 mm
Pinion rotational backlash	2.9 as	4.8 as
Gearbox rotational backlash	4 arcmin	4 arcmin
Gearbox backlash at cable chain axis	8.8 as	10.8 as
Total backlash at cable chain axis	14.5 as	20.4 as

The gears are integrated with Kaydon single-row, 4-point contact ball bearings. No information was available regarding the specified or actual friction torque for these bearings so it was estimated as described above for the DGR bearing. Similarly, the runout and ball precision are unknown but are not expected to have any significant impact on performance of the cable chains or rotators.

11.3. Motors and drives

11.3.1. Requirements

11.3.1.1. Torque

The cable chain torque requirements are derived in an analysis, which is documented separately in [RD9].

Table 29: Cable chain rim gear torque requirements

	DGR	BGR
Total torque	2688 N m	1782 N m

Table 30: Cable chain pinion gear torque requirements

	DGR	BGR
Gear ratio	27.3:1	22.25:1
Pinion torque	98.5 N m	80.1 N m

11.3.1.2. Pinion speed

Table 31: Cable chain pinion speed

	DGR	BGR
Maximum rotator speed	5°/s	5°/s
Gear ratio	27.3:1	22.25:1
Pinion speed	22.8 rpm	18.5 rpm

11.3.2. Motor and gearbox selection

Because the precision of the cable chain is low, compared to the instrument rotators, the cable chain motors are relatively small and must utilize a gearbox to deliver the necessary torque.

Table 32: Cable chain motor specifications

Make / Model:	Kollmorgen Danaher / D103M
Type	Permanent Magnet Brushless AC
Peak torque	60.62 N m
Continuous torque	17.8 N m (14.2 N m at altitude)
Maximum speed	500 rpm (450 rpm at rated torque)
Torque constant	4.431 N m/A
Rotor inertia	0.00864 kg m^2
Pole pairs	8
Winding inductance	18.6 mH
Winding resistance	4.83 Ohms
Thermal resistance	0.502 ºC /Watt
Static friction	1.166 N m
Viscous friction	1.17 N m / krpm

The selection of the gearbox was driven largely by packaging considerations. To eliminate the need for an excessively long shaft connecting the pinion to the gearbox and additional bearings and supports, the gearbox selected for the Bent Gregorian Rotator had to pass through the existing hole in the rotator gallery. The Direct Gregorian Rotator had no similar constraint but it was desirable to use the same (or at least similar) gearbox on both axes.

Several gearboxes were considered. Power transmission gearboxes are widely available and may have been generally suitable. But they usually have relatively large backlash and are designed for high torque or high speed applications. Precision zero-backlash reducers are also common. These, however, are usually restricted to very large reduction ratios or are relatively expensive. And this application did not require zero backlash. A compromise exists with small servo gear heads. There are several manufacturers for these including Parker Bayside. The table below summarizes the gear head selected.

The manufacturer does not provide an estimate of the friction in the gearbox. But the efficiency at full rated torque (97%) is given. This was used to estimate the friction by assuming all losses were from static friction and then using 3% of the rated torque as a frictional torque.

Table 33: Cable chain gear box specifications

	DGR	BGR
Make / Model:	Parker Bayside	Parker Bayside
Type	PS 115	PS 115
Reduction	10:1	10:1
Rated torque	170 Nm	170 Nm
Static friction (est.)	5.1 Nm	5.1 Nm
Viscous friction	Negligible	Negligible

11.3.3. Cable chain moment of inertia

The cable chain moment of inertia is not well known. Because of the gear backlash, the selection of the motor and gearbox places an upper limit on the tolerable moment on inertia of the cable chain. The upper limit of the cable chain moment of inertia bounds the inertia imbalance at 5.0.

Table 34: Cable chain moment of inertia specifications

	DGR	BGR
Motor inertia	0.00864 kg m^2	
Gear ratio	273:1	222.5:1
Maximum cable chain moment of inertia	3200 kg m^2	2100 kg m^2

Only the inner cable trays rotate. Their inertia can be estimated by considering the tray as an L-shaped structure and breaking it into two sections – the inner wall as a thin cylinder and the top/bottom as a thin disk. The cables and energy chain also contribute inertia. For estimation purposes the cable and energy chain mass will be assumed to be located at the edge of the disk. The mass will be estimated at four times the hanging mass used in the torque estimate [RD9]. This is approximately halfway between the inner and out trays and represents a very rough approximation of their mean position.

Table 35: Cable chain inertia estimate

	DGR	BGR
Inner cylinder radius	995 mm	760 mm
Inner cylinder height	310 mm	310 mm
Disk radius	1125 mm	1110 mm
Thickness	6 mm	6 mm
Cylinder mass	90.7 kg	69.3 kg
Cylinder inertia	89.8 kg m^2	40.0 kg m^2
Disk mass	186 kg	181 kg
Disk inertia	235.4 kg m^2	199.1 kg m^2
Cable mass	156.8 kg	268.8 kg
Cable inertia	198.5 kg m^2	331.2 kg m^2
Total mass	433.5 kg	519.1 kg
Total inertia	523.7 kg m^2	570.3 kg m^2

These inertia estimates are well below the maximum allowable values calculated above.

11.3.4. Thermal analysis

The cable chain motor windings are a significant source of heat located very near the optical path. The motors are insulated but if the winding temperature becomes too high, it will become impractical to mask the elevated temperature. The cable chain motor winding temperatures are expected to be quite high. Larger motors could be used to reduce the temperature but, without a better understanding of the load, this seems to be unjustified at this time.

Table 36: Cable chain thermal analysis

	DGR	**BGR**
Pinion torque	98.5 N m	80.1 N m
Motor static friction	1.166 N m	
Motor viscous friction	Negligible	
Gear box static friction	5.1 Nm	
Gear box viscous friction	negligible	
Total motor torque	11.53 N m	9.69 N m
Winding current	2.60 A	2.19 A
Thermal losses	32.7 W	23.1 W
Winding temperature rise	16.4 ºC	11.59 ºC

11.3.5. Line power

The line power requirements are computed accounting for efficiency. The calculation uses the pinion torque requirements defined in Table 30, the pinion speed requirements define in Table 31 and the motor specifications define in Table 32.

Table 37: Cable chain drive line power requirements

	DGR	**BGR**
Maximum pinion torque	98.5 N m	80.1 N m
Maximum motor current	2.60 A	2.28 A
Winding voltage	118 V	88.3 V
Drive power	531 W	349 W
Drive efficiency	80%	80%
Required line power	664 W	436 W

The total 208VAC 3-phase line current computed from the cable chain line power in Table 37 plus the rotator line power in Table 17 is 5.3 A.

11.4. Encoders

11.4.1. Requirements

The relative position sensor requirements are indicated in Table 38.

Table 38: Relative position sensor requirements

	DGR	**BGR**
Sensor range	±5.0 º	±5.0 º
Sensor error	34.6 as	34.6 as
Sensor radius	1.5 m	1.0 m
Linear error	0.252 mm	0.168 mm
Linear range	262 mm	175 mm

The cable chain motor encoder must have sufficient resolution to allow for smooth operation at very low speeds. Therefore, the same constraint defined for the instrument rotator is used for the cable chains.

Table 39: Cable chain motor encoder requirement

	DGR	**BGR**
Minimum rate feedback bandwidth	10 Hz	10 Hz
Minimum optimal tracking speed	6 as/s	15 as/s
Encoder resolution	18.3 as	15.1 as

11.4.2. Relative position sensor selection

The relative position of the cable chain is measured using a non-contact electromagnetic linear position sensor from Pepperl and Fuchs. The device detects the absolute relative position between the cable chain and the instrument rotator.

The relative position sensors selected violate the linear range requirement. As a result, the cable chain will only have ±2.3º of relative motion rather that ±5º as specified. There is no known reason this limitation will interfere with normal operation.

Table 40: Relative position sensor specifications

	DGR	**BGR**
Make	Pepperl and Fuchs	
Model	PMI120-F90-IU-V1	PMI80-F90-IU-V1
Sense range	120 mm (~4.6 degrees)	80 mm (~4.6 degrees)
Repeatability	+/- 0.2 mm (~28 arcsec)	+/- 0.2 mm (~41 arcsec)
Linearity	+/- 0.4 mm (~56 arcsec)	+/- 0.4 mm (~82 arcsec)
Temp. drift -25 to 70ºC	+/-0.5 mm	+/-0.5 mm
Resolution	+/- 0.125 mm (~18 arcsec)	+/- 0.125 mm (~26 arcsec)
Sensing range	1 to 3 mm	

Frequency response	200 Hz
Power supply	18-30 VDC, 35 mA
Output type	4-20 mA and 0 to 10 VDC
Power dissipation	Measure at 864 mW

11.4.3. Cable chain motor encoder selection

Table 41: Cable chain motor encoder specifications

Make / Model	Heidenhain / ECN 113
Type	optical single-turn absolute
Power supply	5 VDC, 180 mA
Resolution per turn	13-bit absolute / 2048 analog cycles
Number of turns	single turn
Linearity	±20 arcseconds
Interface	EnDat 2.2 / RS 485
Resolution (32 x interpolation)	9.92 as

11.5. Emergency stop and limits

Each rotator and cable chain is equipped with four emergency limit switches, one emergency stop circuit and four virtual proximity switches, all of which can interrupt motion. The proximity limit are not physically implemented in hardware but are rather are synthesized from the relative position sensor and the absolute encoders.

Proximity limits bring the rotator and cable chain to a halt under software control. The deceleration is therefore controlled by software which brings the rotator and cable chain to a halt in approximately 100ms or about 0.25º.

The emergency limits and emergency stops to not occur under software control. These circuits interrupt power to the drives and apply the brakes independent of the software. An emergency stop takes about 250 ms before the brakes engage. Once the brakes engage, the stopping torque is very high causing deceleration many times that of normal. However, the linear acceleration experienced by the instrument is similar to that of gravity (see Table 42). The rotator is brought to a halt from an e-stop in less than 1º.

Table 42: Emergency stop deceleration

	DGR	**BGR**
Max deceleration torque	24300 N m	21200 N m
Payload moment of inertia	3940 kg m^2	1750 kg m^2
Envelope diameter	3 m	2 m
Linear Acceleration	0.94 g	1.2 g

11.5.1. Absolute limits

11.5.1.1. Requirements

Table 43: Absolute limit locations

Normal operating range	-90.0º to +450.0º
Proximity limits	≤ -90.1º, ≥ +450.1º
Emergency limits	≤ -91.0º, ≥ +451.0º
Point of damage	≤ -96.0º, ≥ +456.0º

11.5.1.2. Absolute emergency limit switch mounting

A multiturn, gearbox-driven rotary limit switch assembly is used for absolute emergency limit switches for the cable chain. are driven by a gearbox attached to the motor.

11.5.2. Relative limits

11.5.2.1. Requirements

If the rotator and cable chain are moving in opposite directions, it may take 0.5º relative to bring them to a halt once the proximity limit is encountered. This defines the minimum distance between the proximity limits and the emergency limits.

Table 44: Relative limit locations

Relative position sensor range	-2.3º to +2.3º
Normal operating range	-1.0º to +1.0º
Proximity limits	≤ -1.1º, ≥ +1.1º
Emergency limits	≤ -2.0º, ≥ +2.0º

Once engaged, the relative emergency limits must remain engaged for at least 5º. This ensures that the relative recovery direction can be detected.

11.5.2.2. Relative emergency limit switch mounting

The position sensor and emergency limit switches are mounted on a similar radius. The trip points for the proximity limits can be set by pushing a button on the sensor. The trip points of the emergency limits, however, are set by the trip ramps. There is not adjustment provided except by replacing the ramps. This is a deliberate feature to prevent the limits from being adjusted to an unsafe location.

It is desirable to use approximately 40% of the range of the position sensor at the point where the emergency limits are tripped so that the position sensor can be used to determine the relative position after error recovery. The table below lists the relevant parameters for the implementation of this arrangement. The emergency limits are closer than desired for the DGR but are an acceptable distance from the proximity limits.

Table 45: Relative implementation

	DGR	**BGR**
Radius of RPS and limit switches	1600 mm	785 mm
RPS full range	±2.1º	±2.9º
Proximity limits (programmable)	±1.1º	±1.1º
Emergency limits	±2.0º	±2.0º

11.6. Hard stops

The hard stops must prevent excursions beyond the damage point described in Table 43. The must also not begin to engage until the emergency limit switches have been tripped. Table 46 summarizes the action of the hard stops and lists the components selected.

Table 46: Hard stop characteristics

	DGR	**BGR**
Radius of hard stop dampers	1700 mm	800 mm
Contact angle	±2.5º	±2.5º
Stopping distance	1.5º	1.5º
Compression distance	44.5 mm	20.9 mm
Damper make/model	Enidine EM-220	Enidine EM-100
Damper stroke	50.8 mm	25.4 mm

12. Gear and Shaft Stiffness and Stress

12.1. Shaft Stiffness and Stress

The shafts connecting the pinions to the motors had to be relatively long because the existing geometry did not provide adequate space for mounting the large motors near the gears. This raised the possibility that the pinion shafts would be too flexible or experience unacceptably high stresses. However, this was not the case as computed in [RD14] and summarized in the table below:

Table 47: Rotator pinion shaft stiffness and stress

	DGR	**BGR**
Torsional stiffness	220.4 kNm/rad	147.4 kNm/rad
Torsional stress, operating conditions	4.4 MPa	2.6 MPa
Bending stress, operating conditions	7.2 MPa	6.3 MPa
Torsional stress, emergency stop	16.3 MPa	16.3 MPa
Bending stress, emergency stop	26.5 MPa	39.7 MPa

The cable chain shafts are relatively short and are not required to be as stiff as the rotator pinion shafts. However, for the cable chains the radial forces on the pinions must not be beyond the capacity of the gearboxes. The radial loads and gearbox capacities are shown below:

Table 48: Rotator pinion shaft

	DGR	**BGR**
Radial force on pinion	544 N	607 N
Allowable force (approx.)	2900 N	3100 N

12.2. Gear Stresses

The tangential pinion forces that transmit torque also produce bending stresses within the gear teeth. The computation is straightforward and is shown in [RD14]. The stress depends on a form factor based on the geometry of the gear. All of the pinions except the BGR cable chain have enough teeth to make them “normal” gears. The BGR cable chain gear has too few teeth so it must be manufactured with an undercut that weakens it. This is accounted for in the form factor.

There are two situations that must be considered. Under operating conditions the tooth bending stresses must be kept below the endurance limit of the material to prevent fatigue failure over long term operation. Under emergency stop conditions the stress may exceed the endurance limit but must be below the yield strength.

Table 49: Pinion tooth stress

	DGR	DGRCC	BGR	BGRCC
Tangential force, operating	1.8 kN	1.5 kN	1.6 kN	1.7 kN
Tangential force, emergency stop	6.1 kN	N/A	10 kN	N/A
Number of pinion teeth	22	22	20	16
Lewis stress factor	0.31997	0.31997	0.30769	0.27610
Bending stress, operating	21.9 MPa	25.9 MPa	29.9 MPa	33.6 MPa
Bending stress, emergency stop	73.4 MPa	N/A	188.9 MPa	N/A
Gear hardness	248 BHN	248 BHN	248 BHN	248 BHN
Yield strength	434 MPa	434 MPa	434 MPa	434 MPa
Endurance limit	342 MPa	342 MPa	342 MPa	342 MPa

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