



Cúpula

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Preliminary Design Report

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GTC Dome Shutters Upgrade

PHASE II: PRELIMINARY DESIGN DESIGN REPORT

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ABBREVIATIONS

AD	Applicable Document
CRFP	Carbon Reinforced Fiber Plastics
GTC	Gran Telescopio de Canarias
ORM	Observatorio de Roque de los Muchachos
RD	Reference Document
ROM	Rough Order of Magnitude
TBD	To Be Defined
TBC	To Be Confirmed

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1 SCOPE OF THE DOCUMENT

This document describes the Preliminary Design developed during the Phase II of the Definition of the GTC Dome Shutters Upgrade project. It explains the rationale behind the design conceived and includes the corresponding calculations (Appendix A) and engineering drawings (Appendix B) .

Together with the Preliminary Cost Estimation (18242-LIS-014-02), Reliability Analysis (18242-PLA-012-02) and Implementation Plan (18242-TRE-013-02), this report constitutes the Final Delivery Package for the project.

2 APPLICABLE AND REFERENCE DOCUMENTS

2.1 APPLICABLE DOCUMENTS

	Title	Reference	Iss.	Date
AD 1	Statement of Work for the definition of GTC Dome Shutters Upgrade	ESP/EDIF/0200-R	1.A	20/03/2013
AD 2	Current Status of the Dome Shutters	RPT/EDIF/0386-R	1.A	18/03/2013
AD 3	Informe de pruebas de la Compuerta Superior de la Cúpula	RPT/EDIF/nnnn-R	1.A DRAFT-2	17/01/2014
AD 4	Informe del Control Dimensional de las Vigas Arco de la Cúpula	RPT/EDIF/0371-R	1.A	06/09/2012

2.2 REFERENCE DOCUMENTS

	Title	Reference	Iss.	Date
RD 1	1 st Progress Meeting Presentation	18242-PRE-001	1	11/10/2013
RD 2	1 st Progress Report	18242-TRE-002	1	9/10/2013
RD 3	1 st Progress Meeting Minutes	18242-MIN-003	1	16/10/2013
RD 4	Phase I: Analysis of Alternatives Report	18242-TRE-004	1	25/11/2013
RD 5	End of Phase I Meeting Presentation	18242-PRE-005	1	29/11/2013
RD 6	End of Phase I Meeting Minutes	18242-MIN-006	2	02/12/2013
RD 7	Phase II: Preliminary Cost Estimation	18242-LIS-014	2	11/04/2014
RD 8	Phase II: Implementation Plan	18242-PLA-013	2	11/04/2014
RD 9	Phase II: Reliability Analysis	18242-TRE-012	2	11/04/2014
RD 10	Phase II: Addendum to Preliminary Design Report	18242-TRE-018	1	11/04/2014
RD 11	The Complete Guide to Chain (http://chain-guide.com)			
RD 12	Michael Hess – Replacing the K1 Dome Shutter drive system	Proc. SPIE 6273-17		

3 INTRODUCTION

3.1 BACKGROUND

The current shutter system of the GTC dome does not fully comply with the expected performance. The problems detected are essentially: a) a restricted range on main shutter opening, which limits the telescope observation angle to 70 deg. above horizon; b) premature damage on shutter bogies; c) the unfeasibility of deploying the windscreen, which has not been commissioned yet; d) damages (cracks, corrosion, deformations) in the dome arch beams. These problems are described by GRANTECAN in document AD2.

GRANTECAN has launched this preliminary design consisting of: a) an analysis of alternatives aimed to finding solutions to the current problems of the GTC dome shutters; b) selecting the best solution in terms of performance, cost and implementation time and developing it up to a preliminary design.

During the Phase I of this project the diagnostics developed by GRANTECAN was analysed and completed and individual actions were identified aimed to solve each problem detected. In order to evaluate the feasibility of these actions, they were combined into 3 particular packages, intended to configure 3 potential solutions that were physically developed up to a conceptual design stage to check and illustrate their applicability. A final trade-off analysis was developed to evaluate the pros and cons of each solution. All this work is compiled in RD4.

At the end of Phase I, GRANTECAN reviewed the delivered documentation and decided on the solution out of the 3 proposed options which will be further developed up to a preliminary design stage. The solution selected was the first alternative proposed (**Option A**), **but modified to include some additional actions**.

Option A consists in basically keeping the existing drive system typology but introducing a set of incremental actions aimed to improve the current performance and solve the main problems. The safety and reliability of each of the components (chains, motors, gearboxes) is increased as their demanding loads will be lower than the allowable values. The actions include:

- Replace current upper shutter chains by others with higher strength
- Implement a counterweight system to reduce load at the last part of upper shutter travel
- Improve load sharing between drives by a proper control system strategy.
- Install a take-up system in chains to reduce the required preload
- Install axial supplements in the guiding rails to reduce the rail runout
- Replace motors and gearboxes of Lower shutter by others with higher power
- Move the location of windscreen latching system to avoid overlapping
- Reduce the windscreen weight
- Replace current windscreen rollers cam followers for friction reduction
- Modify the bottom profile of the windscreen guides
- Monitor crack propagation, register a crack inventory and take actions according to the propagation speed (grind-repair, crack arrest drilling, structural reinforcement...)

The additional actions:

- **Bogies design upgrade**
- **Possibility of excluding the counterweight system**
- **Prevision of emergency systems for the lower shutter operation**

3.2 OVERVIEW OF THE PROPOSED DESIGN

The GTC Dome Shutters proposed modifications affect to:

- The drive system, control system and guiding system of both upper shutter and lower shutter.
- Enhancement of the Windscreen
- Proposed actions for the defects detected on the dome arch beams and shutters structure

Upper & Lower shutters:

- Regarding the drive system, the upgraded design includes:
 - Reduction of the chain pre-load of the upper shutter chains by means of installing a Lower take-up mechanism in chains M4 & M6 and an upper take-up mechanism in chains M5 & M7 (not required for lower shutter)

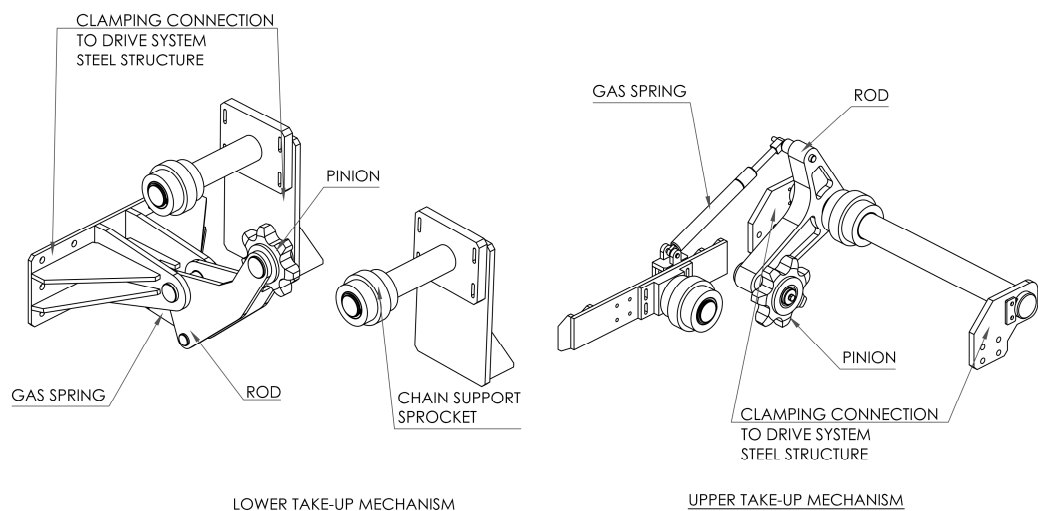


Fig. 3-1 Upper shutter Take-up system

- Replacement of current roller chains by higher capacity chains, in both upper and lower shutter
- In addition, for the lower shutter:
 - replacement of main motors for higher power motors
 - replacement of current gearboxes for double input gearboxes to include an auxiliary gearmotor for redundancy or a double input $i=1:1$ ratio gearbox at the current gearbox entrance, which will enable emergency operation in case of failure of the main motor.

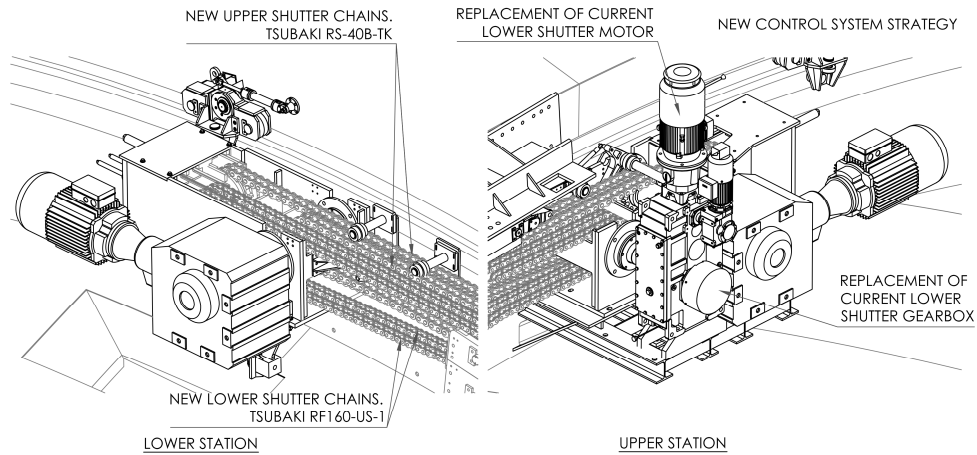


Fig. 3-2 Actions in shutters drive systems

The proposed redesign also includes an optional counterweight system, which would contribute to reduce the loads in the last stage of the upper shutter travel.

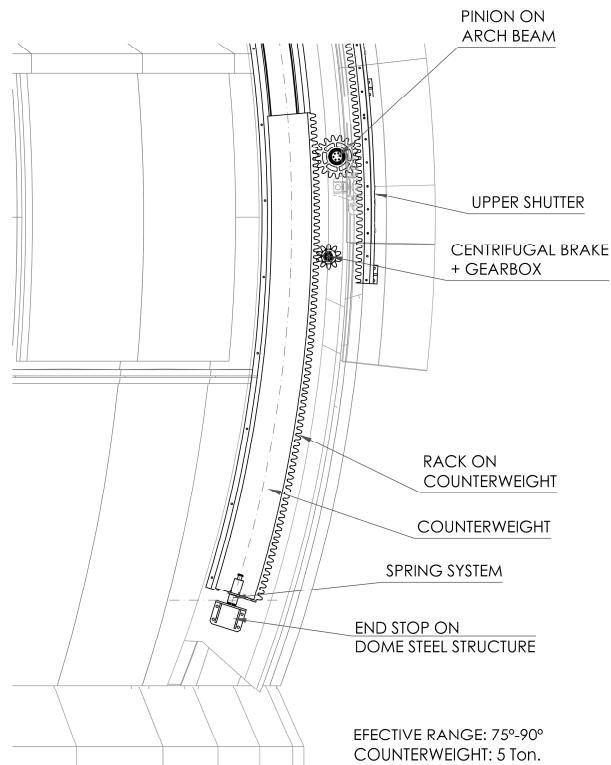


Fig. 3-3 Counterweight system

- For the guiding system, an improved bogie design is proposed with capability to absorb the rail radial misalignment by means of a hydraulic equalization system. Regarding the equalization system different strategies have been analysed which will be explained in the following sections.

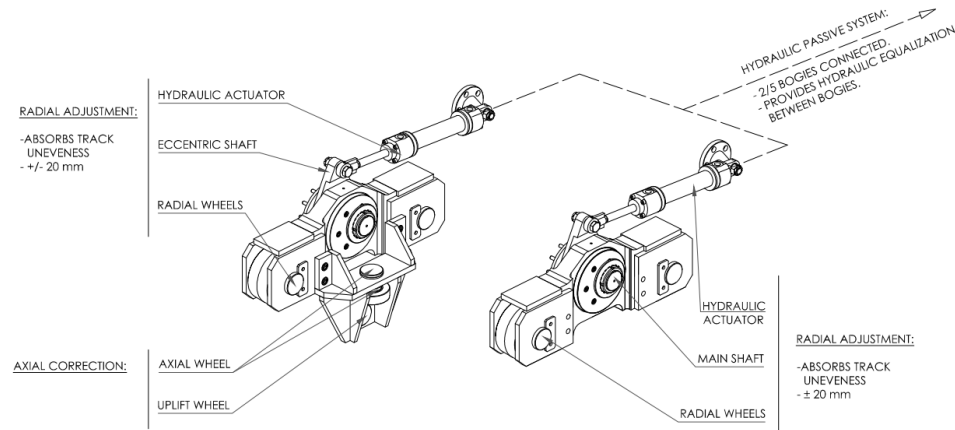


Fig. 3-4 Guiding system upgrade

Apart from the hydraulic actuator and the eccentric shaft for radial adjustment, the new bogie design includes two radial wheels instead of one, for additional strength under the acting loads.

The shutters lateral guidance is also improved by means of axial correction supplements welded to the rails.

- For the control system, an optimized control strategy (“Speed Trim Follower”) is proposed aimed to provide a better load sharing between drives. This configuration consists of:
 - the master drive is operated in speed regulation
 - the slave drives are operated in speed regulation with a speed trim that is function of the difference between the master drive and each slave drive torque.

The equal load distribution between drives achieved with this new control strategy, together with the replacement of drive chains and the improvement on shutter lateral guidance provided by the axial correction supplements, can enable the operation of the upper shutter with only three out of the four drives in emergency conditions.

Enhancement of the Windscreen:

After a diagnostic of the Windscreen deficiencies was carried out the following modifications are proposed in order to improve the current Windscreen performance.

- Latching system redesign
- Modification of Windscreen guides profiles
- Replacement of the current rollers for cam followers

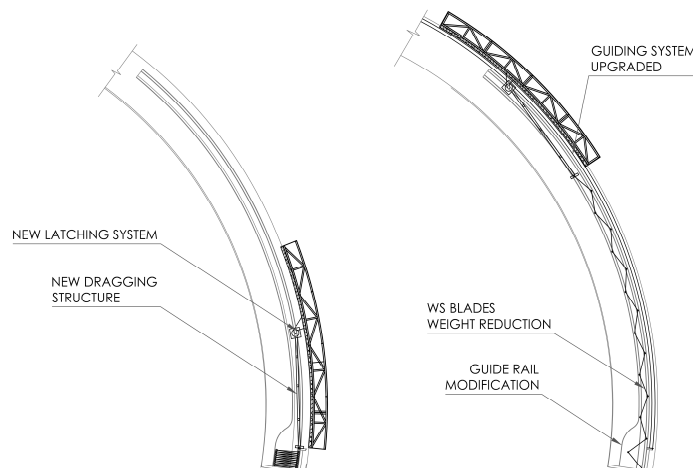


Fig. 3-5 Modification in the Windscreen

Actions for the damages on the dome arch beams and shutters structure:

An action plan has been outlined to monitor and tackle the potential propagation of existent cracks and repair the corrosion effects.

3.3 DOCUMENT STRUCTURE

The present report is structured as follows:

- Next Chapter 4 describes the preliminary design for the Upper Shutter Upgrade, including the actions dedicated to the drive system improvement (including control strategy optimization) and the enhancement of the guiding system.
- Chapter 5 explains the upgrade of the Lower Shutter (drive mechanisms and guiding system).
- Chapter 6 is dedicated to the actions in the Windscreen System, including:
 - ✓ Replacement of current rollers for cam followers
 - ✓ Modification of windscreen guides profile
 - ✓ Latching system redesign
 - ✓ Weight reduction
- Chapter 7 defines the Action Plan for the damages on the Dome Arch Girders and Shutters Structure
- Finally, Chapter 8 compiles the Final Conclusions related to the Preliminary Design developed.

This document includes also two appendices attached at the end of the report:

- Appendix A: Related Analyses and Calculations

Appendix B: Engineering Drawings for definition of the Preliminary Design

Regarding the bogie redesign requested by GRANTECAN a new addendum document has been issued (see RD 10).

4 UPPER SHUTTER UPGRADE

4.1 INTRODUCTION

In order to guarantee the total aperture of the Upper Shutter a combination of actions over the Drive and Guide System are considered.

- 1) Drive System
 - a. Chain replacement in order to guarantee a $SF=4$.
 - b. Preload reduction system for chain friction reduction by adding a take-up system
 - c. A new Control Strategy for a good load share between the driving motors
 - d. Safety shutter operation in Emergency Conditions
 - e. Proposal of a counterweight system (optional)
- 2) Guiding system
 - a. Proposal of a new support strategy
 - b. Design of the new bogies
 - c. Axial correction of the rails

The following sections describe the actions listed above along with the required structural modifications for their implementation.

It is also studied the possibility of discarding the installation of the counterweight

4.2 DRIVE SYSTEM

The scope of the changes over the Drive System consists on:

- Reduction of the system loads
- Enhancement of the load sharing between drives
- Increase of the chain strength
- Proposal of an emergency operational strategy
- **Theoretical Load Budget**

A theoretical load budget was developed, quantifying the values of the different loads and computing the resulting margin available (if any) till reaching the chains, motors and gearboxes capacity.

The load budget for the upper shutter drive system is shown in the next plot. For the chains, a working safety factor of 4 has been considered to obtain the resulting margin. Also, the motor altitude de-rating factor and gearbox efficiency have been considered.

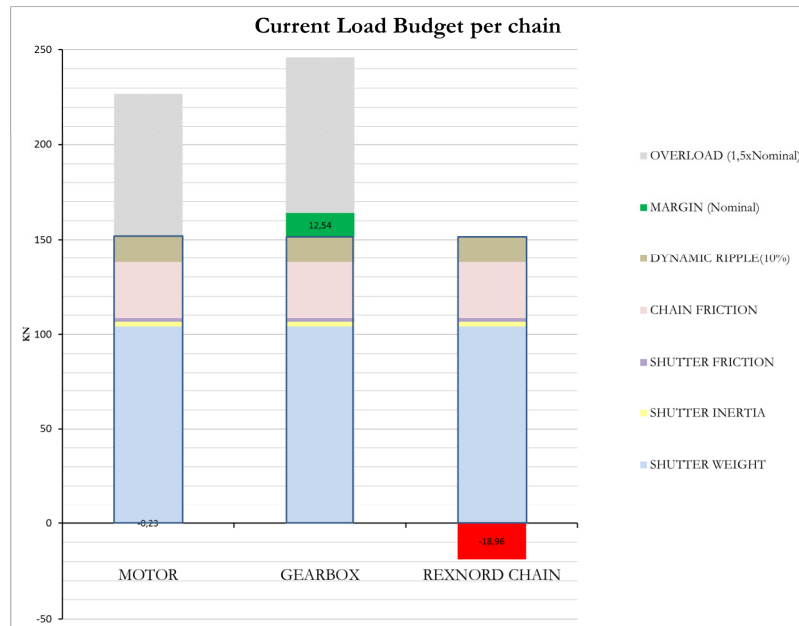


Fig. 4-1. Load budget for the current upper shutter drive system

According to this load budget:

- The strength of the current chains is not enough to withstand the current loads with the design safety factor (4).

The proposed changes, described in the following sections, show the following theoretical load budget for the new scenario.

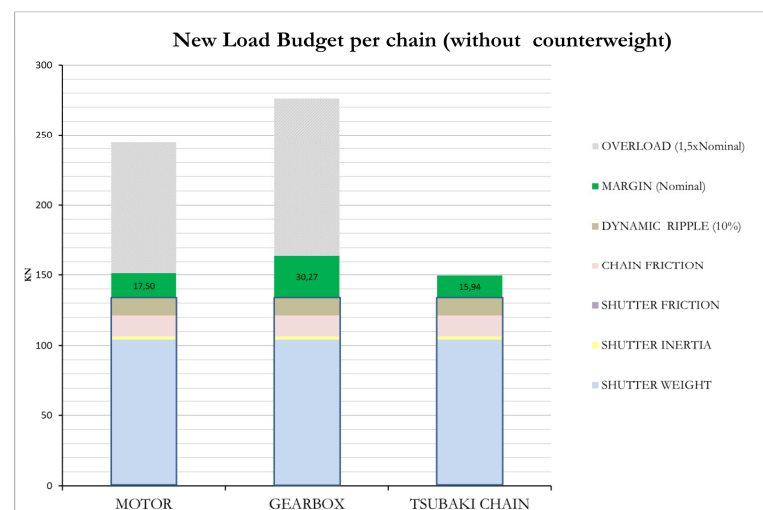


Fig. 4-2. Theoretical new load budget with the upgrade of the drive system

The Load Budget study takes us to the following conclusions:

- Due to the take-up mechanism, the chain friction has been reduced
- The increase in the chain capacity and the reduction in the chain load guarantees a Safety factor of 4.

- **Measured operational chain load**

During October 2013, the complete 90° Upper Shutter opening/closing cycle was carried out. The description of the execution and the data acquired is presented by GRANTECAN in AD 3. The chain load values at the gearbox end position, are shown in the following graph.

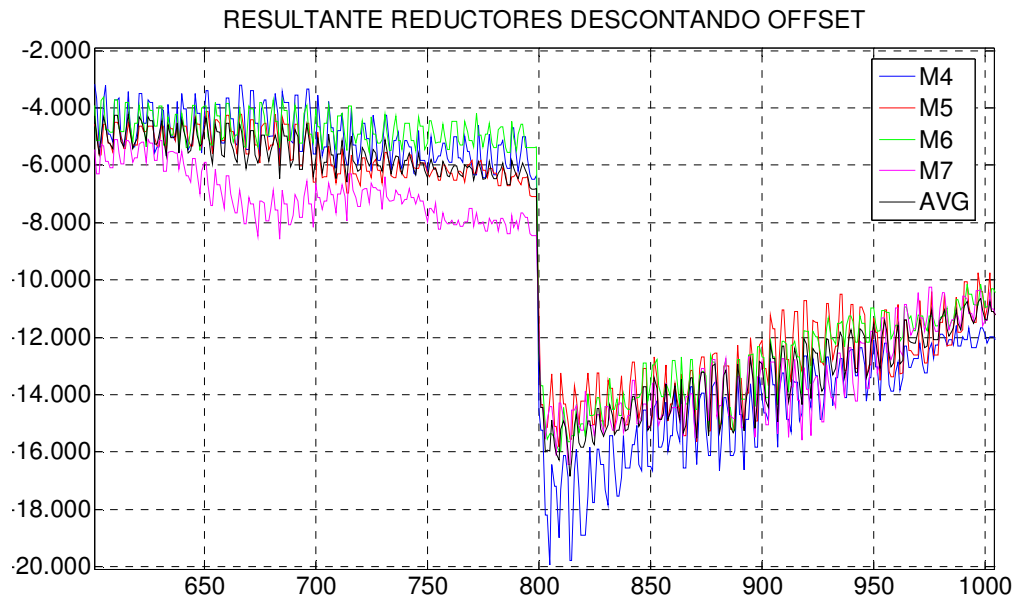


Fig. 4-3 Chain load (N) at gearbox end (Figure extracted from document AD 3)

The conclusion of this study is that if a good load sharing is obtained by means of an appropriate control system strategy, then the maximum chain load in the current chain configuration (without applying any upgrade) is 160-165KN (equal to the average load shown in black in the figure above).

Therefore the measured chain load is 10KN higher than the theoretically estimated value. In order to ensure that the changes match correctly with the real requirements, the measured values have been used for chain selection.

4.2.1 Preload Reduction System

In order to ensure the engagement at driving/driven sprockets, the current chain system must be properly tensioned.

From the preload reduction systems proposed in RD 4, GRANTECAN selected the passive Take-up system as the most suitable solution. The best value solution agreed with GRANTECAN was to install one Take-up system per chain.

Two different designs are proposed, one for the upper motor chain, in which the slack chain is lifted up and the other for the lower motor chain, in which the slack part is pulled inside the chain loop. In both configurations, the Take-up systems are installed next to the driving sprocket.

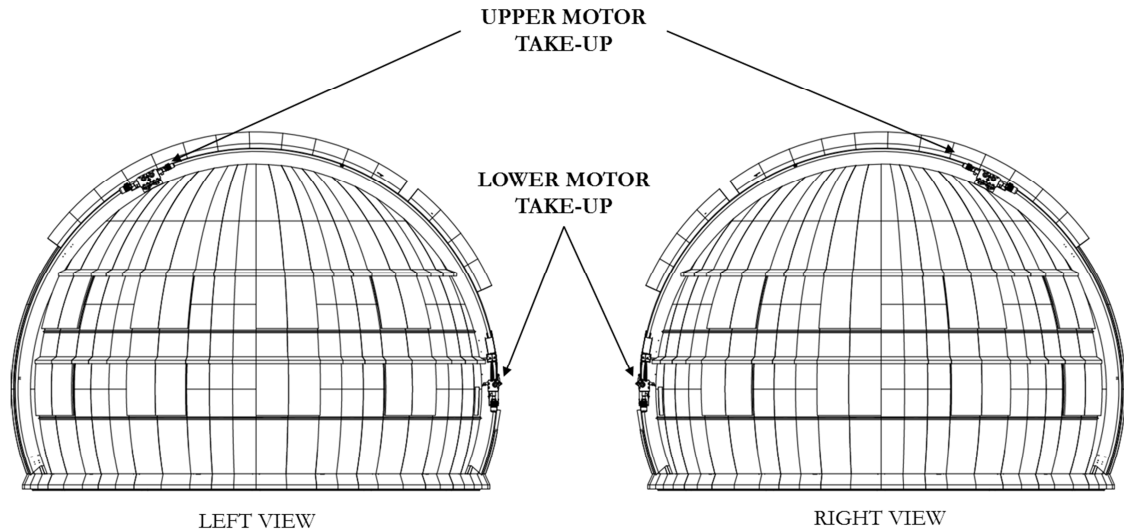


Fig. 4-4 Upper motor and lower motor Take-up overall position view

Upper motor chain Take-up:

Regarding the M4 and M6 chains, the most demanding situation is when the Upper Shutter is positioned at 90°. Considering that the elastic stiffness of the chain is 35KN/m per chain metre and being 153KN the chain load at this position. Therefore, the slack length of the chain is around 200mm, being 150kg the chain weight that must be lifted.

The proposed design is based on a lever pinned in a horizontal bar, fit with an idler sprocket at one side and being pushed by means of a gas spring in the other. A nearly constant force of 1500N is applied in the chain, which allows lifting-up around 200-250mm of slack chain.

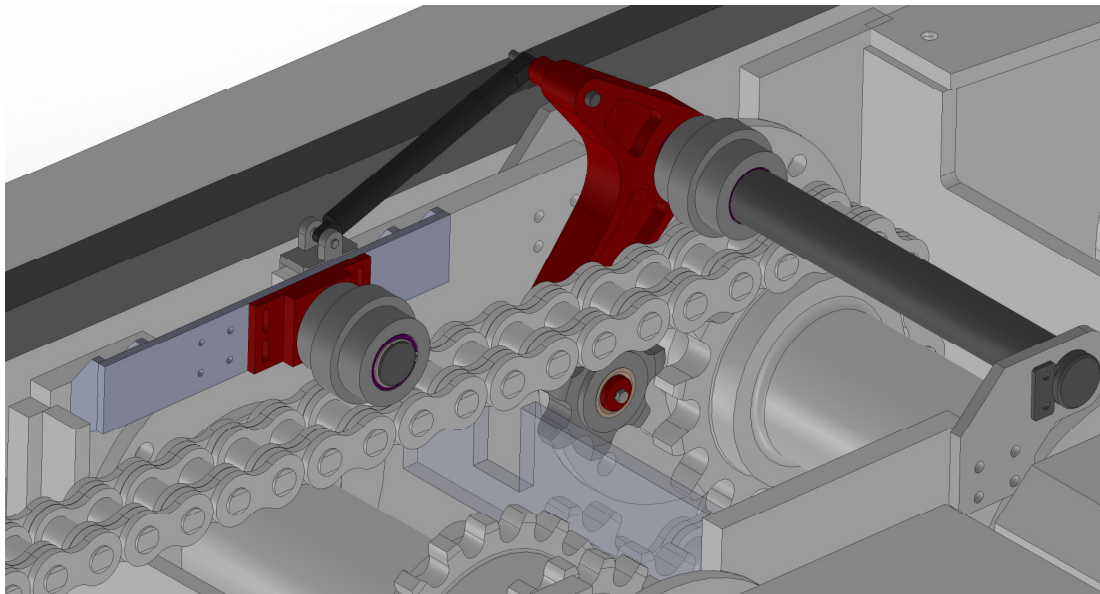


Fig. 4-5 Upper motor Take-up 3D view

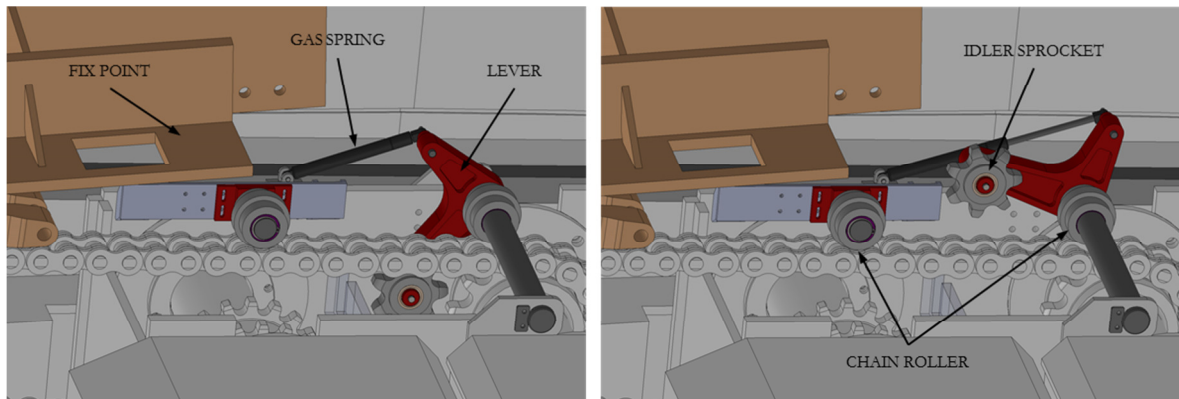


Fig. 4-6 Upper motor Take-up; at the initial position on the left and at the final position on the right

The working principle of the take-up is shown below,

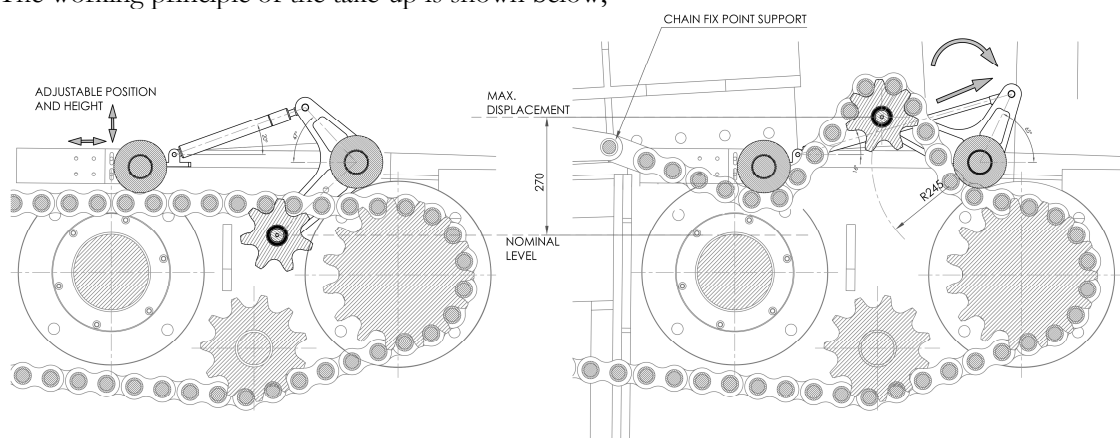


Fig. 4-7 Upper motor Take-up functionality

Lower motor chain Take-up:

Regarding the M5 and M7 chains, the most demanding situation is when the Upper Shutter is positioned at 0° . The slack length of this chain is close to 100mm and around 500N force is needed to move the slack part.

The proposed design is based on a lever pinned in a holding frame, fit with an idler sprocket at one side and being pushed by means of a gas spring in the other. In this design the chain is also pulled outwards by way of the idler sprocket. Overhanging from the arch beam two chain rollers are placed, in order to guarantee to take-up up to 120mm of slack chain.

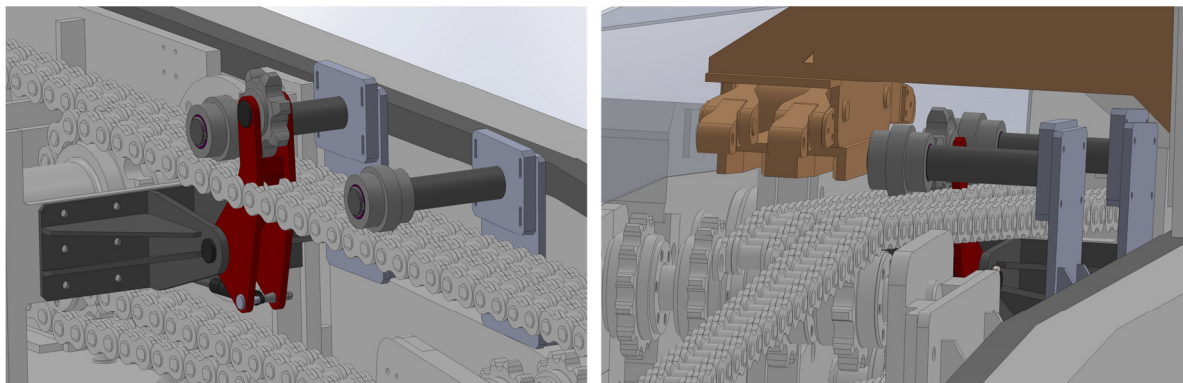


Fig. 4-8 Lower motor Take-up 3D view

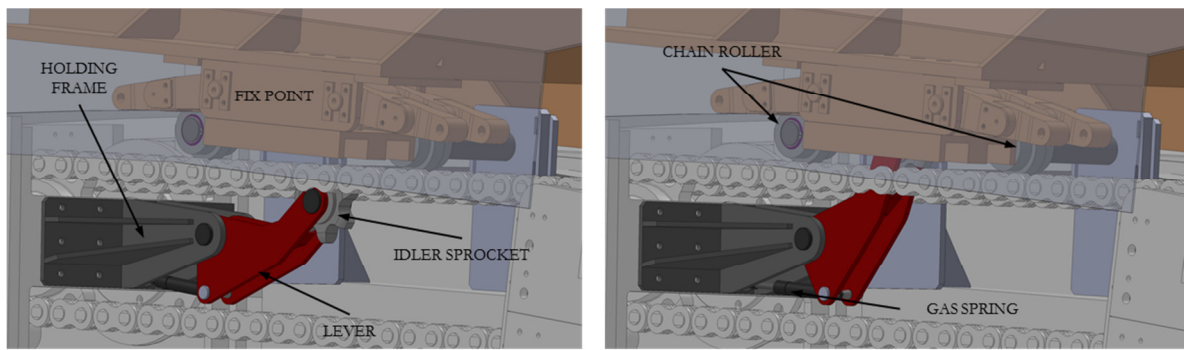


Fig. 4-9 Lower motor Take-up image; at the initial position on the left and at the final position on the right

The working principle of the take-up is shown below,

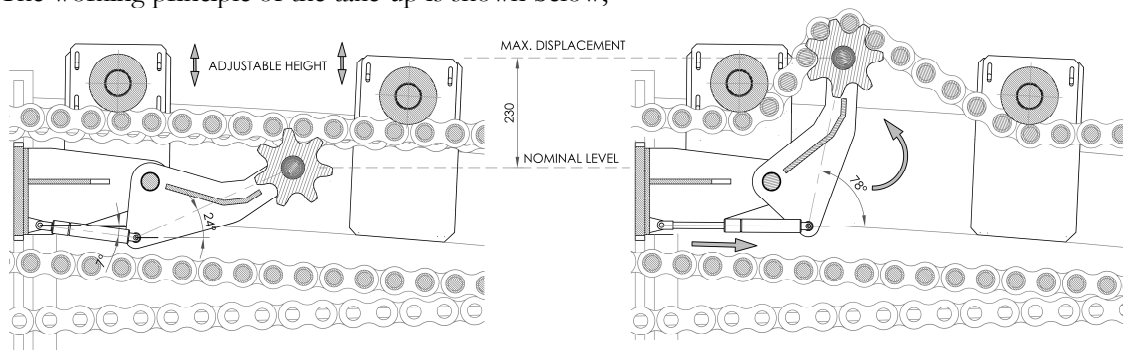


Fig. 4-10 Lower motor Take-up functionality

The main characteristic and assembly of the Take-up systems are shown in Appendix B. The relevant drawings are:

- 18242-DWG-03-000
- 18242-DWG-03-001
- 18242-DWG-03-002

4.2.2 Control System

Regarding the Control System, considering the limitations of the control strategies tested for the shutter drive system, an alternative control configuration (Speed Trim Follower) is proposed, which is suitable for flexible systems and can provide a proper load sharing. It is usually applied in systems as chain conveyors.

A Speed Trim Follower strategy consists of:

- the master drive is operated in speed regulation
- the slave drive is operated in speed regulation with a speed trim that is function of the difference between the torques of both drives.

This control strategy can be implemented either in the PLC or in the drives control unit. The VSD currently used on the GTC (SIEMENS SIMODRIVE) do not allow implementing this strategy and the distributor recommends the drives to be replaced by more modern SINAMICS (as the SIEMENS SIMODRIVE production will be soon discontinued). Considering all this, IDOM also recommends to upgrade the VSDs as their enhancement will increase the reliability of the system and guarantee obtaining spare components during the next 5-10 years.

Alternatively, the strategy can be implemented in the PLC, as shown in the figure below.

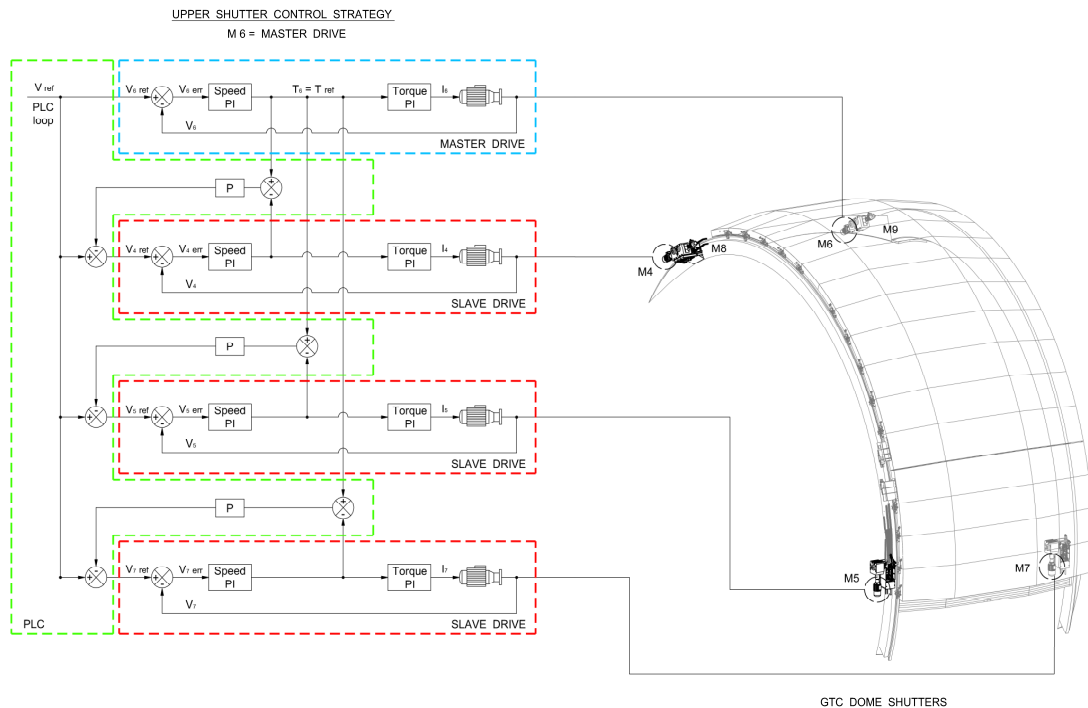


Fig. 4-11. Diagram for Speed Trim Follower configuration for the Upper Shutter.

This control strategy has been already implemented (in the PLC) and successfully tested by GRANTECAN during January 2014 as shown in AD 3.

4.2.3 Chain Replacement

The under capacity of the current chains to work in the complete shutter travel with the required design safety factor demands the substitution of the current chains.

As it is pointed out in the document AD 2 section 3.2 Drives, according to the EU Directive 2006/42/EC on Machinery, which is reflected in the Spanish regulation RD 1644/2008, the working coefficient required for lifting chains as general rule shall be equal to 4.

In order to fulfill this requirement for the normal operational conditions two chain options are selected:

- 1) A customized TSUBAKI RS-40B-TK
- 2) The REXNORD Laschenkette 64-1 SH

From the design point of view, these chains are different:

The customized TSUBAKI RS-40B-TK has been developed based on DIN8187 (transmission chain). These type of chains are made up of bearing inner/outer plates (1&2), pins (4), bushings (6) and rollers (7), with a typical chain elongation of 1,5%. The bushings reduce the friction and wear in the rotational area. The customized version has thicker plates in order to increase the ultimate strength value.

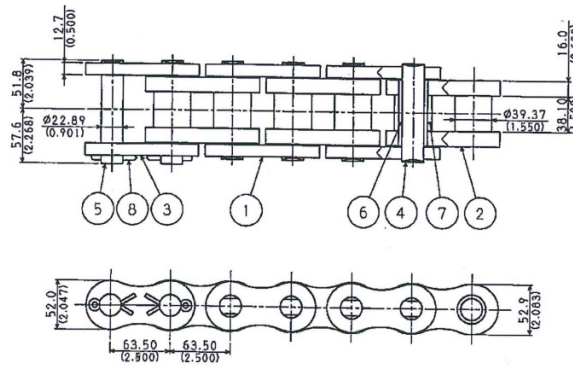


Fig. 4-12 TSUBAKI RS-40B-TK chain representation

Instead on the other hand the REXNORD Laschenkette 64-1 SH is an elevation chain composed by bearing pins, rollers and inner/outer plates with a typical chain elongation of 2%. This type of chain configuration is used to lift vertical loads as the absence of a bushing allows increasing the chain ultimate strength by increasing the pin diameter. However the drawback is an increase on the chain friction and chain wear in the rotational area, which is relevant in our application as the chain arrangement is circular rather than vertical.

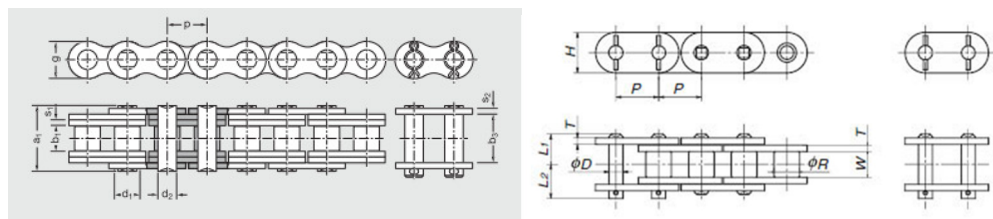


Fig. 4-13 REXNORD Laschenkette 64-1 SH chain representation

4.2.3.1 Operational chain loads

As it is shown in section 4.2, the maximum chain load in the current chain configuration (without applying any upgrade) is 160-165KN. The good load sharing is guaranteed with the Speed Trim Follower strategy proposed and explained in section 4.2.2.

By means of the Take-up system solution proposed in section 4.2.1 Preload Reduction System, a 1-1.5KN reduction in the chain load is estimated. Hence, **during normal operational conditions it is expected a maximum load of 147.5-152.5KN.**

This means that **600KN is the required minimum ultimate tensile strength** in order to satisfy the EU Directive 2006/42/EC on Machinery (SF=4).

- 1) The customized **TSUBAKI RS-40B-TK** has a **minimum ultimate tensile strength of 610KN** which satisfies the requirement. This means that a working load up to 152.5KN is available guaranteeing a safety factor of 4.

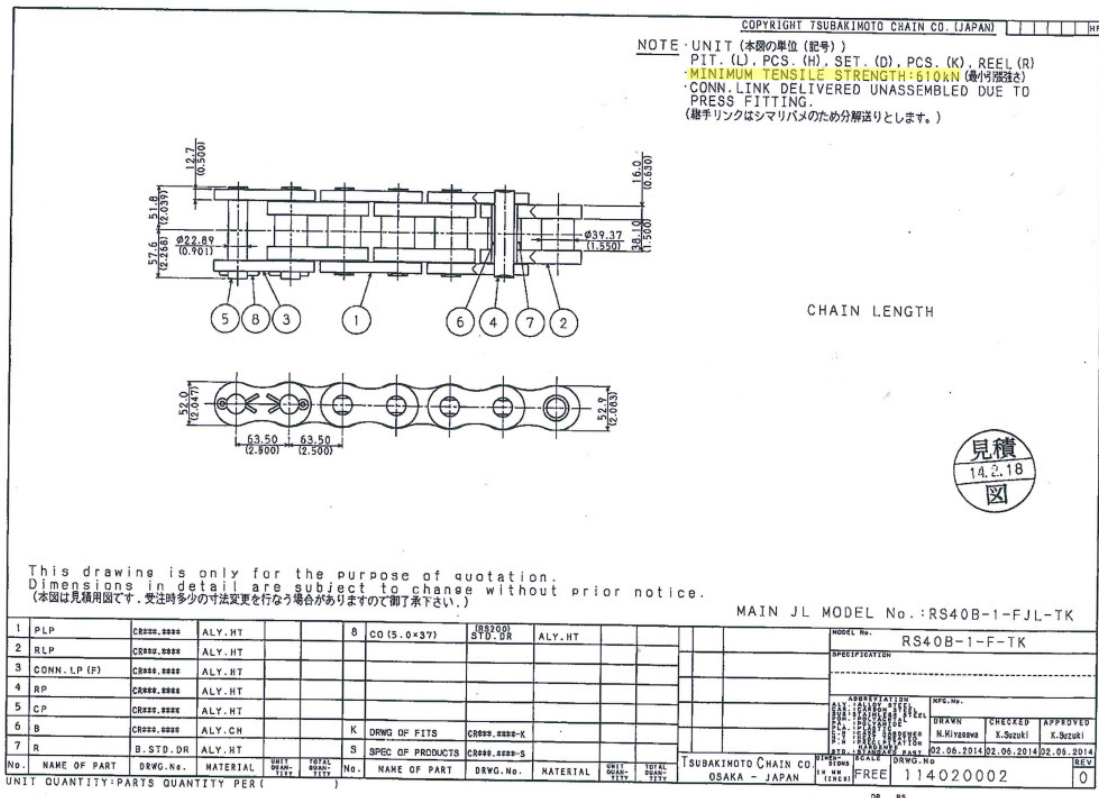
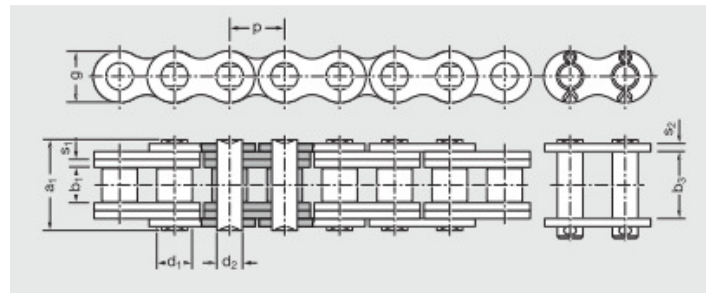


Fig. 4-14 TSUBAKI RS-40B-TK chain main data

- 2) The **minimum ultimate tensile strength** of the **REXNORD Laschenkette 64-1 SH** is **600kN** and therefore a working load of 150kN is feasible with a safety factor of 4. The main data of the chain is shown in the figure below.



Chain No.	Pitch p		Roller Width	Roller Diameter	Pin Diameter	Outer Width	Transverse	Linkplate Height	Overall Width	Plate Thickness		Bearing Area	Ultimate Strength ca.	Weight
	inch	mm	b_1 min. mm	d_1 max. mm	d_2 max. mm	b_2 min. mm	e mm	g max. mm	a_1 max. mm	s_1 mm	s_2 mm	A cm ²	F_B N	$\approx q$ kg/m
50-1 SL/A	2.00	50.8	31.55	28.58	20.35	57.5	—	47.8	75.5	6.3	6.3	5.1	270 000	14.9
50-2 SL/A	2.00	50.8	31.55	28.58	20.35	128.0	70.5	47.8	146.0	6.3	6.3	10.2	540 000	29.4
64-1 S	2.50	63.5	38.1	39.38	30.0	77.0	—	59.5	104.0	9.5	9.5	11.4	530 000	28.0
64-1 SBG	2.50	63.5	39.1	39.38	30.0	59.0	—	59.5	83.0	9.5	8.0	5.7	450 000	27.9
64-1 SH	2.50	63.5	38.1	39.38	30.0	71.5	—	59.5	101.0	8.0	11.0	9.6	600 000	25.5
64-1 SHBK	2.50	63.5	27.6	39.38	30.0	67.0	—	59.5	95.0	9.5	11.0	11.4	600 000	27.5

Fig. 4-15 REXNORD Laschenkette 64-1 SH chain main data

The proposed RS 40B-TK chain, developed for this particular application by Tsubaki, reaches the expectations from both the specification and design point of view.

- A minimum tensile ultimate strength of 610kN in order to satisfy a safety factor of 4.

- Regarding the design, as it was previously mentioned, being the chain fit by a bushing improves the chain's functionality, reducing the chain friction and wear. The chain width (103.6mm) and height (52mm) fulfill the geometrical limitations. In addition, the design based on a DIN8187 chain, the main dimensions for sprocket engagements are guaranteed (roller diameter and width).
- The working load of this chain is shown in Fig. 4-16 being 153KN the working load of the chain with a SF=4 (fatigue life = 10⁸cycles)

Therefore, the **RS 40B-TK is the proposed chain replacement.**

4.2.4 Operation in Emergency Conditions

It is considered an Emergency Condition when the shutter need to be closed just using 3 out of the 4 drives (one drive is inoperative).

In this scenario, the control system will share the load among the operative 3 drives. The load will be distributed equally among them, thus the shutter will be pulled with twice the load in one side than in the other.

During the previously mentioned tests made by GRANTECAN in October 2013, the upper shutter was moved by means of 3 motors for different angle ranges. The processed data, presented in the document AD3, showed a total additional resistance load of 40-80KN when compared with the normal operational conditions, being 20KN the dragging force of the inoperative gearbox and 20KN-60KN the new friction forces of the shutter and chains.

Assuming that the axial corrections will provide a better axial guiding of the shutter, it is expected to reduce the friction component down to half its value (30KN).

So in Emergency Conditions the maximum chain load is expected to be:

$$F_{\text{CHAIN}} = \frac{(150\text{KN} \times 4) + 30\text{KN}}{3} = 210\text{KN}$$

The motor and gearbox capacities are above this load requirement and thus, they do not need to be replaced.

Regarding the chain, the load in these conditions (210KN) does not comply with the safety factor of 4. However, it must be considered that this would be an emergency condition and therefore an exceptional scenario is considered and studied as so.

Therefore, a fatigue criteria is used to assess the feasibility of using the previously proposed TSUBAKI RS 40B-TK chain. This will define the chain expected life under a certain number of cycles with emergency loads, guaranteeing a safety condition before the chain has to be replaced. The following figure provided by TSUBAKI shows the F-N curve of the RS 40B-TK.

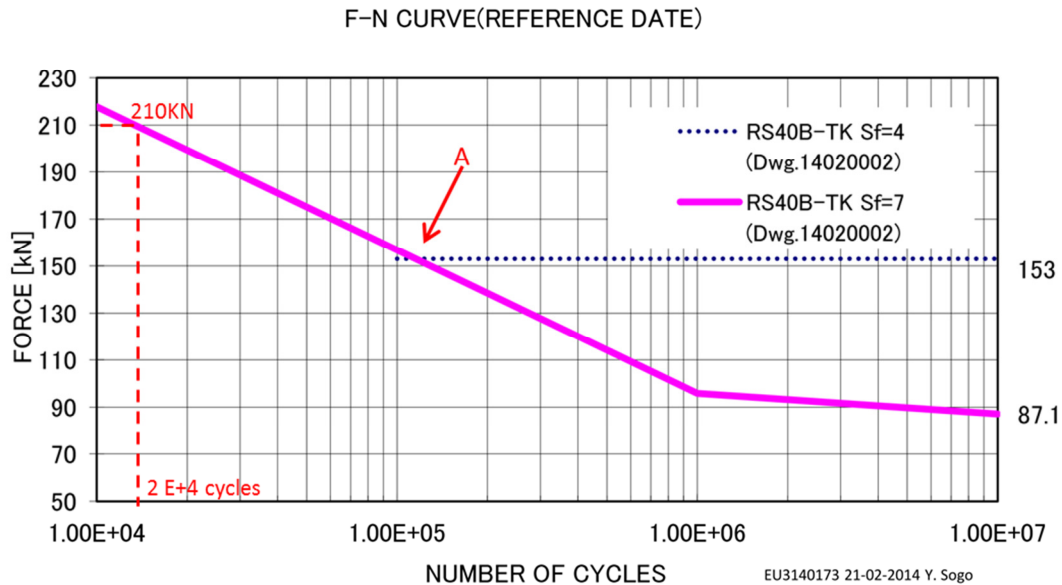


Fig. 4-16 F-N curve of the TSUBAKI RS 40B-TK

From the F-N it is concluded that the chain would withstand 210kN during $2 \cdot 10^4$ cycles with SF=7.

There are $2 \times 2 = 4$ chain cycles during the shutter Open-Close cycle and therefore, considering the fatigue analysis as the design criteria, the chain would be capable of withstanding emergency loads (210kN) for 5000 emergency conditions.

Therefore, a safety scenario is guaranteed under Emergency Operational conditions without the installation of a counterweight.

4.2.5 Counterweight System (optional)

In accordance with the decision taken by GRANTECAN in the meeting to close out the Phase I of this project a trade-off between the two options proposed during Phase I (pulley and cable driven mechanism and rack and pinion based solution) has been made based on the results from a risk analysis and a reliability assessment. The conclusion is that the rack and pinion option is more robust and reliable, so this solution has been selected for further development.

The main challenge of this mechanism is to ensure a correct engagement between the racks and the pinion, due to the radial misalignment of the shutters guiding rails. To minimize this risk, the shutter rack position should be adjusted by means of steel shims and a high module ($m=30$) has been selected for the gear teeth, so that they will be able to absorb variations around $\pm 15\text{mm}$ in the theoretical engagement distance.

Regarding the impact force at the first teeth, it is proposed on one hand to install a first easily replaceable short length (4-5 teeth) rack segment at the shutter rack and on the other hand to reduce the speed of the shutter in a controlled way in order to guarantee the contact at a certain speed.

During the Detailed Design phase this limiting speed shall be computed. This is a function of the longitudinal stiffness of the shutter rack holding structure, as this stiffness along with the shutter speed will determine the acceleration of the counterweight. The total force (weight and inertial force) and the allowable impact force at the teeth will determine the limiting shutter speed.

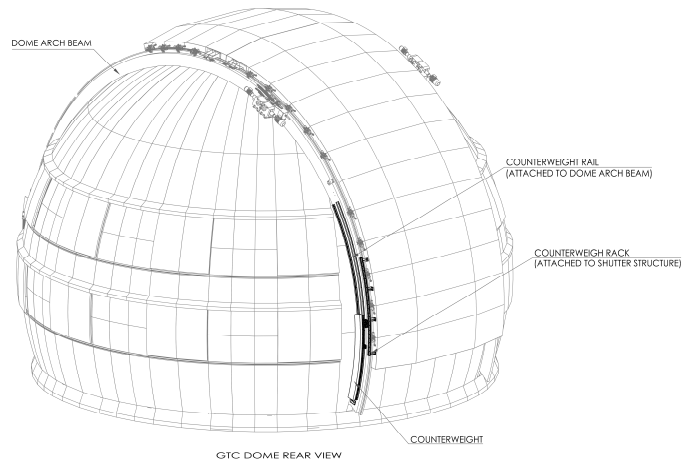


Fig. 4-17 GTC Dome rear view – Counterweight system position

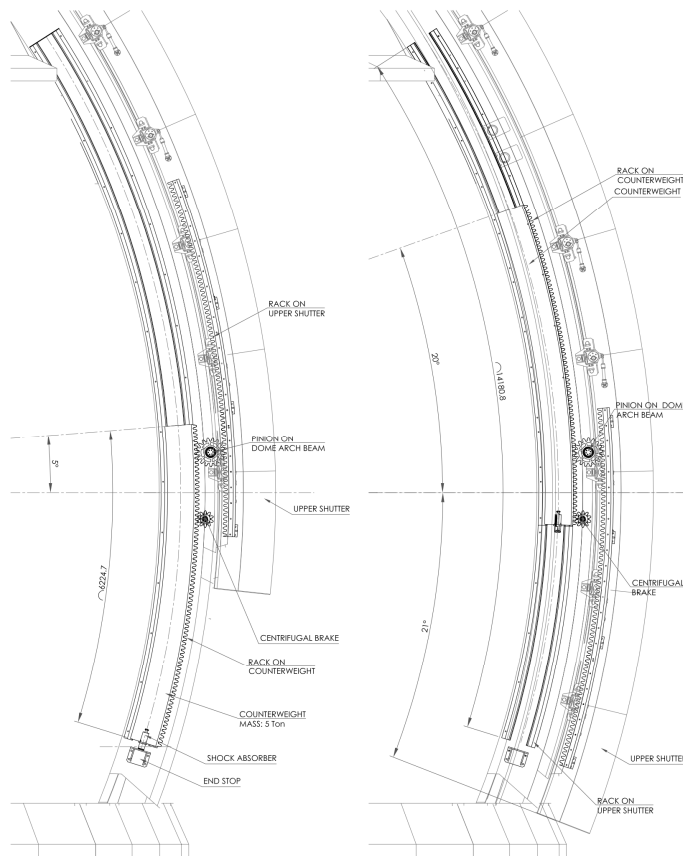


Fig. 4-18 Counterweight extreme positions

The main characteristics of the Counterweight:

- The weight: 5tonnes.
- The operation range: 75°-90°.
- Safety system based on a centrifugal brake with a gearbox.

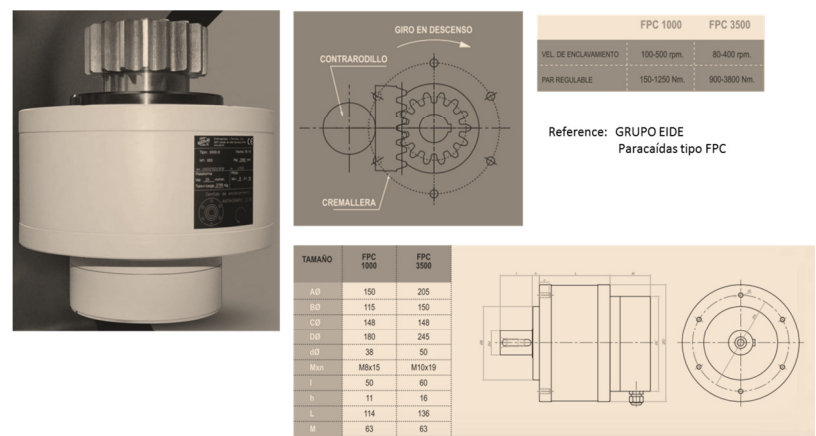


Fig. 4-19 Centrifugal brake. Ref. EIDE

As reference, the centrifugal brake EIDE FPC 3500 has been selected, connected to a gearbox ($i=1:10$), to adapt to the shutter speed and increase the output torque to the required value, and an auxiliary pinion engaging the counterweight rack. The limit speed of the centrifugal brake can be set around 250 rpm (0,5 m/s in the counterweight side).

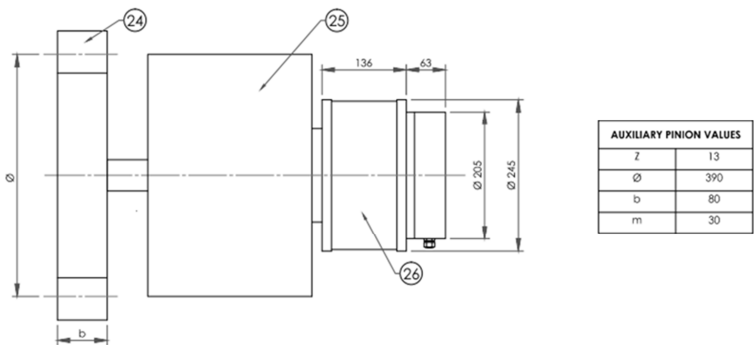


Fig. 4-20 Representation of the centrifugal brake assembly with an auxiliary pinion and a gearbox

- Installation of a spring system at the bottom to ensure a progressive load engagement.

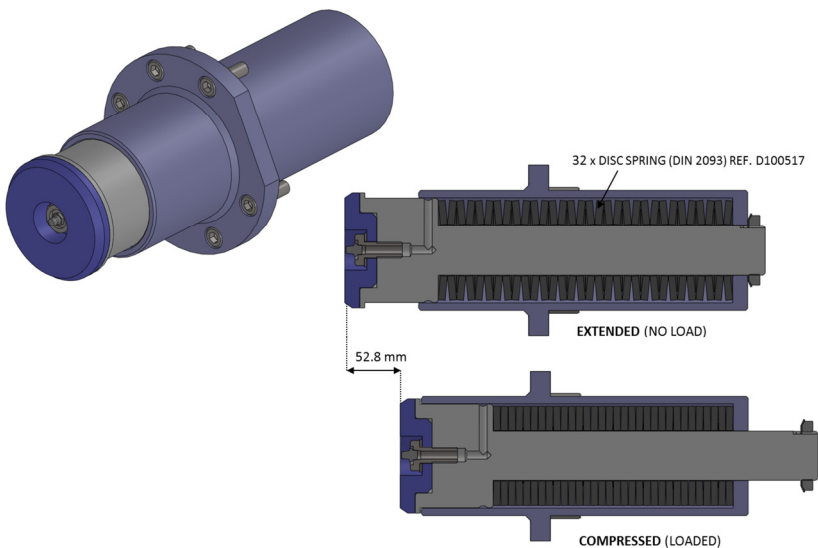


Fig. 4-21 Proposed spring system build-up by Belleville washers

Below is the analysis of the Belleville washers (DIN 2093) for the counterweight shock absorber:

Selection: D100517 spring washers

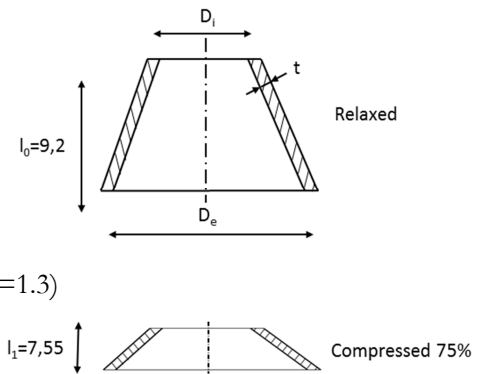
- $D_e=1000\text{mm}$
- $D_i=51\text{mm}$
- $t=7\text{mm}$
- $l_0=0.2\text{mm}$
- 75% deflection
 - $f=63793\text{N}$
 - $\delta=1.65\text{mm} \rightarrow l_1=l_0+\delta=7.55\text{mm}$

Total load = $F = 5 \text{ tonnes} \cdot 9.8 = 49\text{KN} < 63.8\text{KN}$ (SF=1.3)

Serial mounting (32 washers):

$F=f=63.8\text{KN}$

$\Delta = n \cdot \delta = 32 \cdot 1.65 = 52.8\text{mm}$



All these changes are presented in the 18242-DWG-04-000 and 18242-DWG-04-001 drawings.

As it is mentioned in the chapter 4.2.4 *Operation in Emergency Conditions*, the chain capacity of working with a safety coefficient of 4 during normal operational conditions and safe life fatigue criteria under Emergency conditions, enables the possibility of managing without the installation of the counterweight system.

In case it is decided to install the counterweight a detailed design of the system are and modifications in the dome rear stairs and pass ways are required.

The Load budget considering the counterweight system would be as shown below:

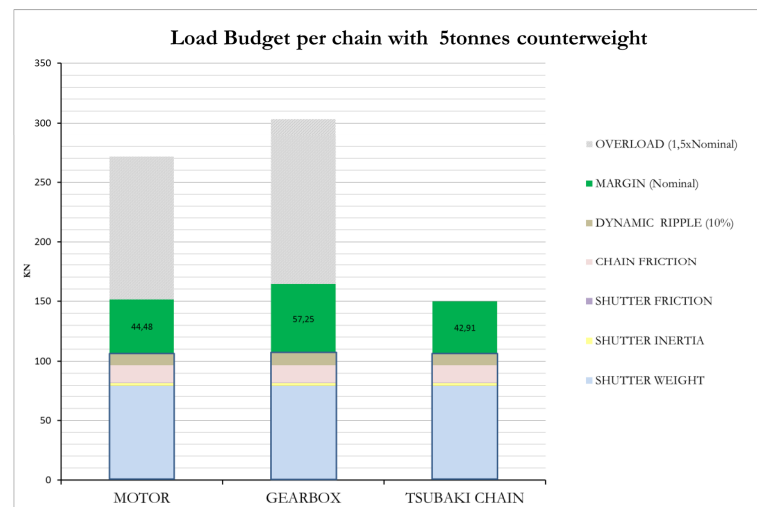


Fig. 4-22 Load budget per chain with 5 tonnes counterweight

The support of the pinion has been modified in order to be able to weld it to the dome arch. A stress analysis is shown below, where a 80KN force has been applied at the pinion shaft.

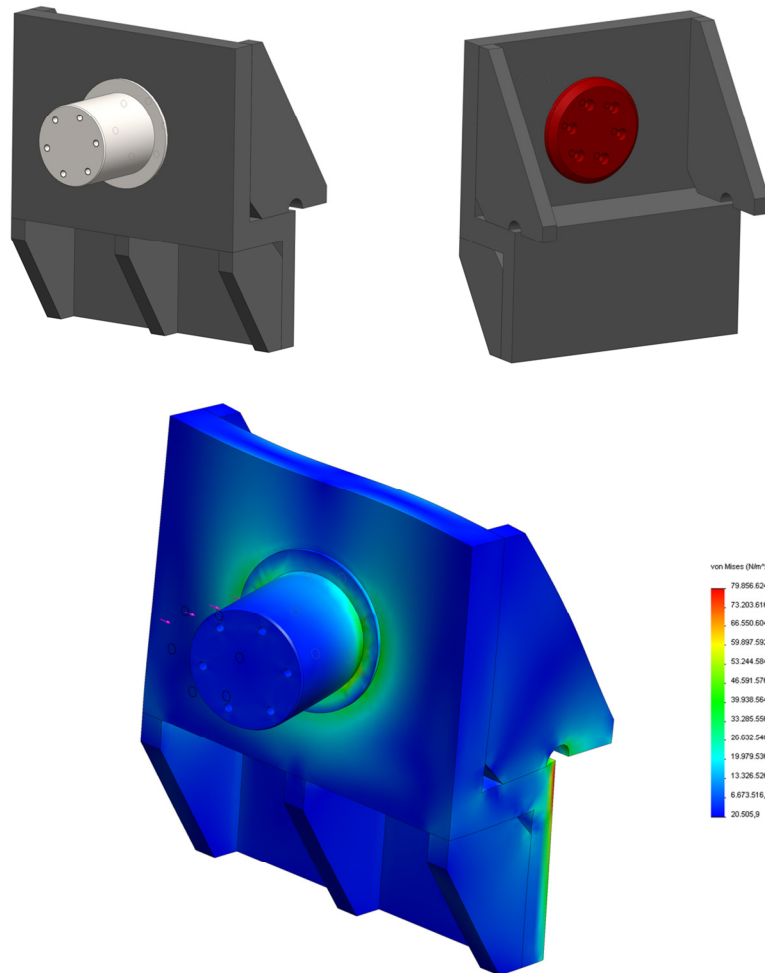


Fig. 4-23 Pinion support design and stress analysis

4.3 GUIDING SYSTEM

4.3.1 Introduction

Currently, the shutter moves on two radial rails, fixed on the dome arch beams, by means of 26 bogies, 13 at each side, which support its weight and which guide laterally the shutter during its motion. These bogies are fixed to the shutter by means of an elastomeric block to absorb the unevenness of the rails during the shutter movement. Currently, the capacity of these elastomeric blocks is insufficient to absorb the radial irregularities.

In order to assess those irregularities GRANTECAN measured the radial and axial positions of the dome arch beams rails during the summer of 2012 and the processed data is shown in document AD4.

The results of those measurements showed high irregularities of the rails both in radial and axial direction. These high radial unevenness together with the lack of capability to absorb them by the bogies, produces peak loads on the radial wheels causing their premature damage and an inadequate performance of the current supporting system. The axial irregularities cause the axial movement and rotation of the shutter.

It was agreed with GRANTECAN to maintain the radial runout, redesigning a new supporting mechanism capable of adapting to the rails irregularities, and correct the axial guiding by adding supplements

4.3.2 New Supporting Mechanism

The proposed solution distributes the radial reactions in the interface between the supports arranged in different sectors by means of a **hydraulic equalization system**.

4.3.2.1 Supporting Strategy

The best value supporting strategy will be a compromise between:

- 1) Better load distribution
- 2) Minimization of radial displacements at bogie positions (due to gravity load equalization and the subsequent shutter structure accommodation & rail radial unevenness)

The best load distribution will be achieved with the minimum number of hydraulic systems, 4 in this case (2 at each side), corresponding to an isostatic supporting strategy.

However, by increasing the number of hydraulic systems the radial displacements of the supports are also reduced and the movement of the shutter as a rigid body is also reduced.

In the following section, it is presented the proposed solution for which the bogie reactions and their radial deviations have been checked. A FE analysis has been made for checking the radial reactions and the deviations due to the shutter accommodation to whiffle tree load equalization, which have been added to the deviations due to the rail radial unevenness.

In order to select the best value solution, two strategies, the first one with 4 hydraulic systems (two at each side) of 5 bogies each, and the second with 6 hydraulic systems (three at each side) of 3 bogies each, have been assessed.

The results are shown in Appendix A and during the Detailed Engineering phase the most convenient solution shall be selected. At this state the second solution, with 6 hydraulic systems, (three at each side) appears to be the best option since the maximum radial bogie load has just been increased from 54KN to 72KN while the maximum shutter displacement (due to the shutter flexibility) has been reduced from 32,4mm to 11,78mm.

4.3.2.1.1 Mechanism Layout and reaction Equalization system

In order to describe the load equalization system the proposed supporting mechanism is composed of 18 bogies, 9 at each side, which transmit the radial reactions. Bogies are arranged in six groups of 3 bogies each, being the reactions hydraulically equalized among the bogies in the same group.

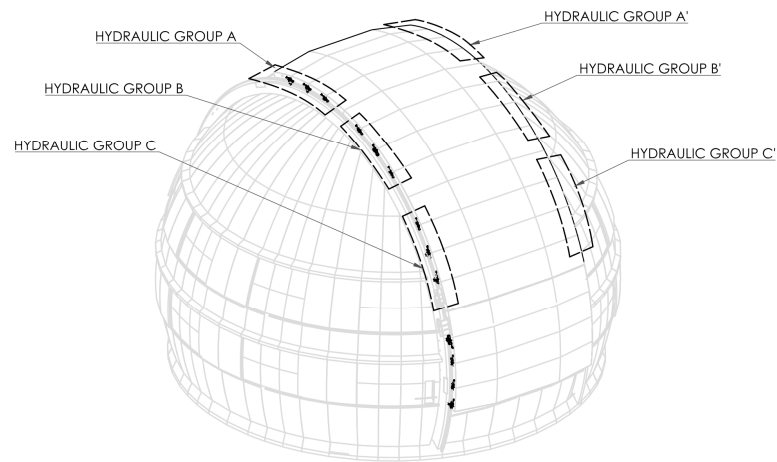


Fig. 4-24 Upper Shutter hydraulic systems

The equalization is achieved by hydraulically connecting the three cylinders which supports the reactions on each sector.

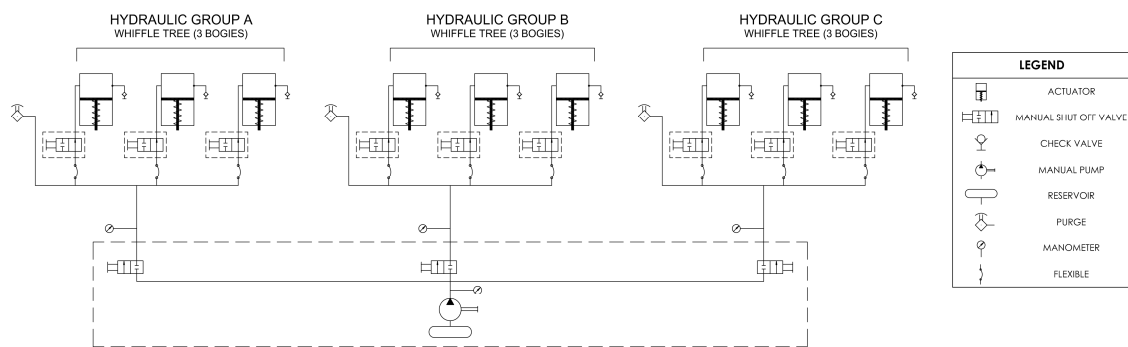


Fig. 4-25 Hydraulic diagram of the Group A, Group B and Group C

The hydraulic group distribution and diagram are presented in the drawing 18242-DWG-01-001.

In the Detail Engineering phase it will be studied the necessity of installing limit switches at actuator locations and pressure sensors in order to monitor the status of the system.

4.3.2.2 New bogie design

The new bogies have to be capable of absorbing the rail radial runout. There is a radial runout of 29mm in Rail A and 41mm in Rail B [RD 4]

A new bogie design has been conceived, with capability for radial adjustment (+/- 20 mm) to the track unevenness.

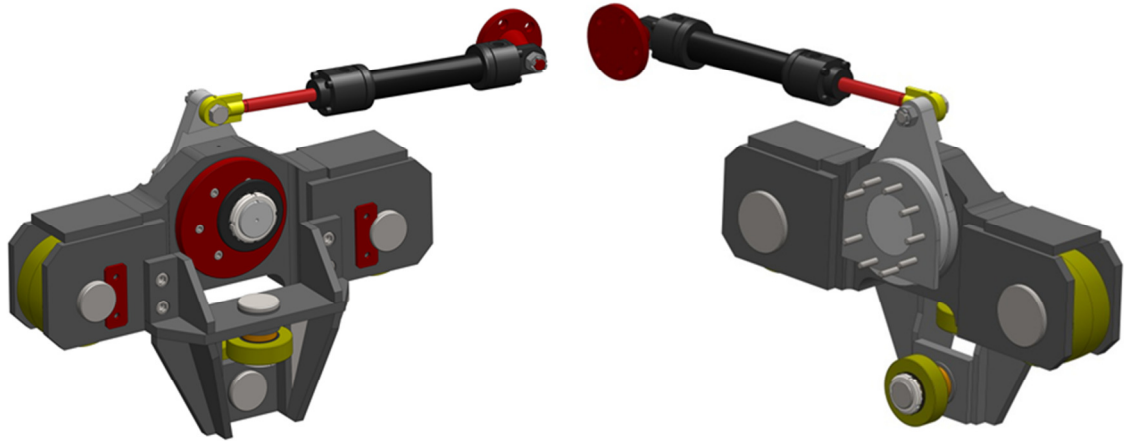


Fig. 4-26 Front and Back view of the new bogies

The following figure shows the main components of the bogie.

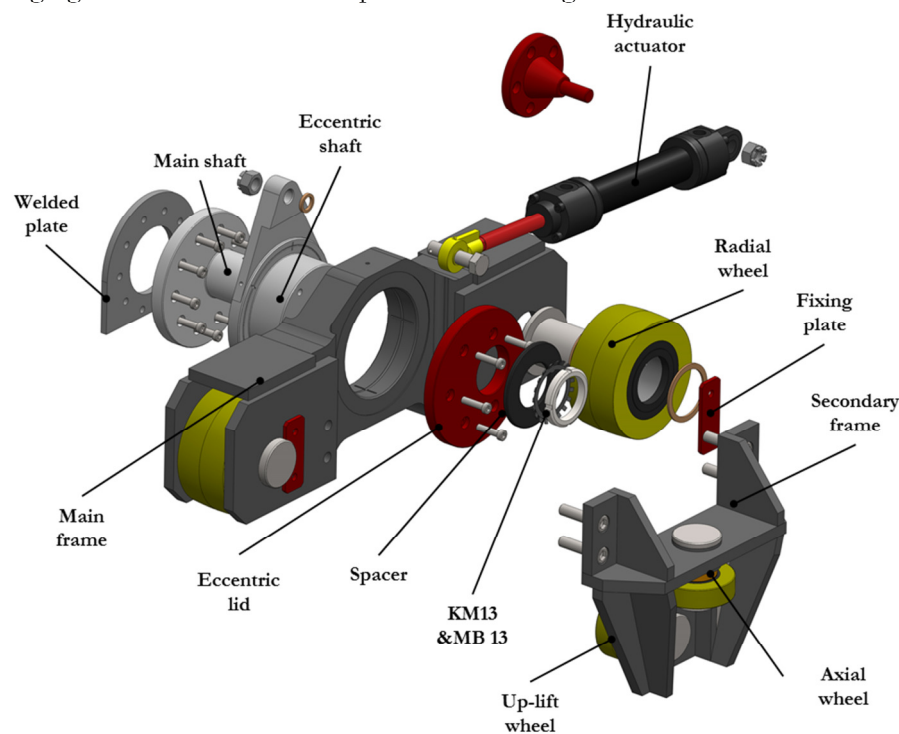


Fig. 4-27 Main bogie parts representation

The base design of the bogie includes two radial wheels. There would be another configuration including also an axial and uplift wheel by means of the secondary frame. The positions of each bogie type are specified in section 4.3.2.2.2.

Due to the irregularities of the shutter wall, some alignment (axial and vertical) and rotating (yaw) capability is needed in the bogies. The vertical alignment is obtained either by welding the welding plate in a higher or lower position or by rotating the main shaft hole pattern an small angle, as this allows to move upwards and downwards the main shaft. Regarding the axial alignment and rotating capability, this shall be obtained by introducing shims between the main shaft and the welded plate.

In order to obtain a major axial alignment ($\pm 10\text{mm}$) and rotating capability, during the Detailed Engineering phase the possibility of fitting smaller radial wheels (NNTR55x140x70) in the current design could be studied as this will allow major gap between the main frame and the shutter wall.

Introducing shims between the main frame and the secondary frame will allow axial positioning adjustment of the axial and up-lift wheels. The axial and up-lift shafts have been modified in order to replace the KM & MB blocking system by a cir-clip. Thus, any impact of the up-lift wheel shaft with the arch beam due to the lateral movements of the shutter is avoided.

A detailed assembly view of the bogies is shown in Appendix B drawing 18242-DWG-01-003.

4.3.2.2.1 Working principle

The following figure shows the vertical movement of the bogies, which is shown in more detail in Appendix B the drawing 18242-DWG-01-002.

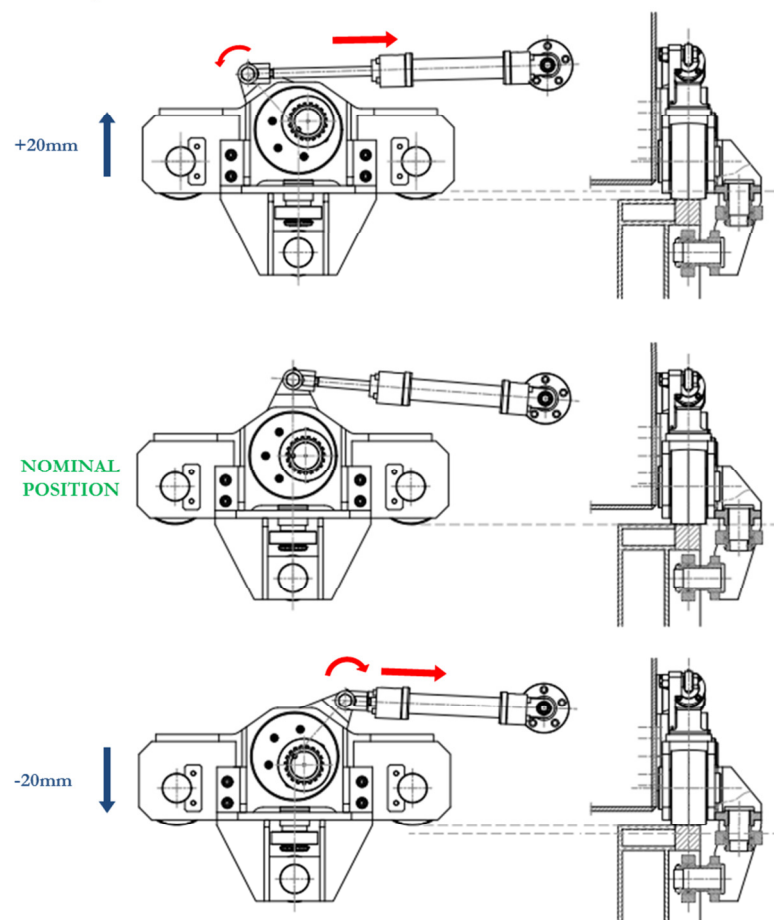


Fig. 4-28 Bogie working principle

The required compliance is provided by:

- Eccentric shaft
 - Allows vertical displacements
 - Allows main frame rotation for radial reactions equalization (pinned joint)
- Eccentric lid
 - Restrains axial displacement of the bogie
- Hydraulic actuator
 - Compensates the vertical movement and the radial loads transmitted by means of the eccentric shaft
 - Fed by the previously explained hydraulic passive system, connecting several bogies, and providing load equalization.

4.3.2.2.2 Bogie types and positions

Due to the asymmetry of the proposed bogies, a bogie version A is needed for those placed on side A of the shutter while its symmetrical version, the version B, is needed for the others located in side B.

In addition to that, the two bogies placed at each of the extreme locations of the shutter have a secondary frame with an axial and an up-lift wheel.

The following figure shows the different bogie types.

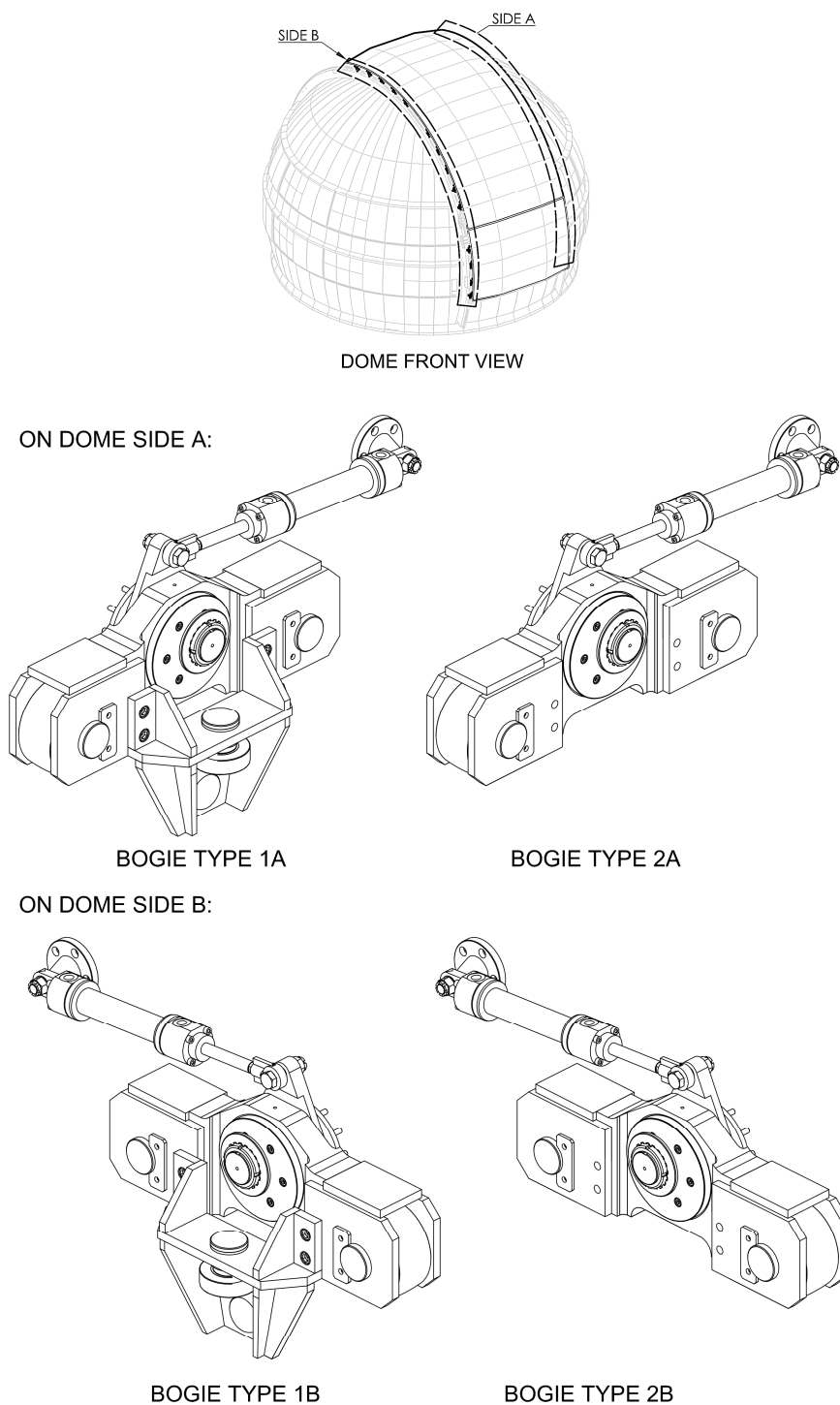


Fig. 4-29 Bogie types representation (Type 1A & 2A for side A and Type 1B & 2B for side B)

The position of the different bogie types is specified in the 18242-DWG-01-000.

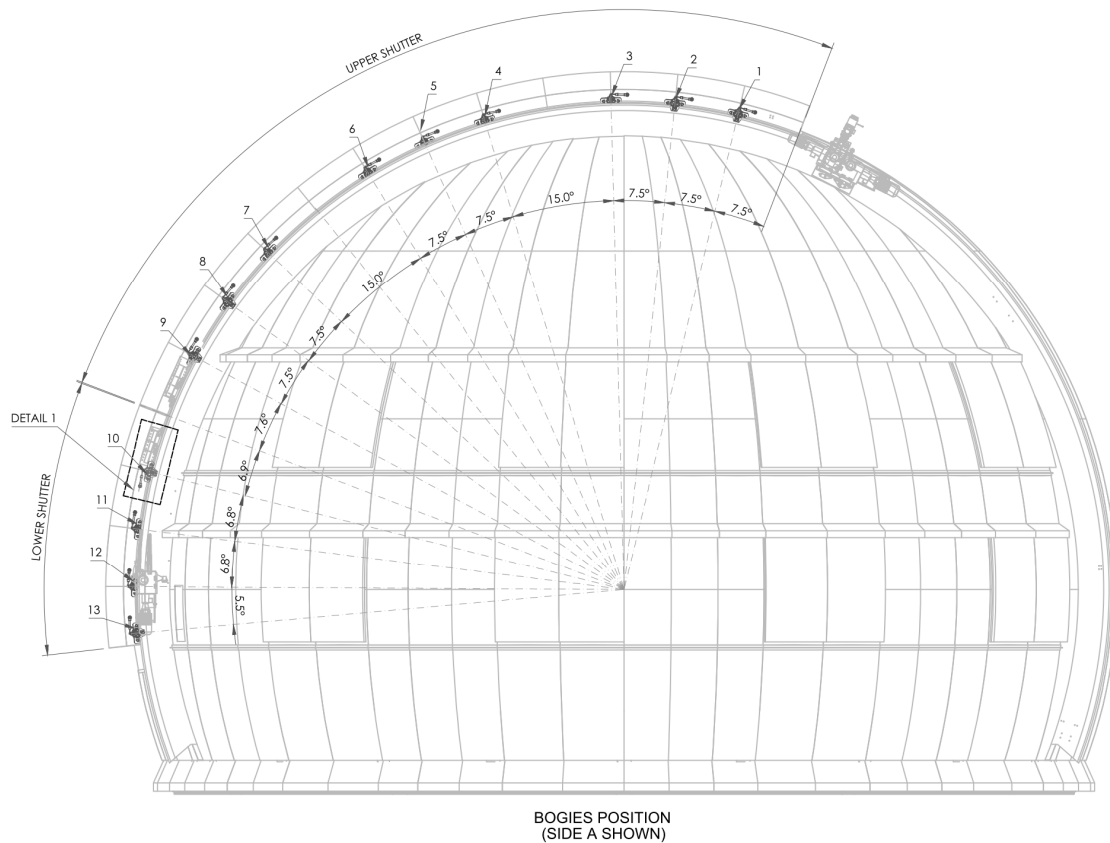


Fig. 4-30 upper shutter bogie layout

4.3.2.2.3 Structural analysis of the bogies

A preliminary stress analysis has been carried out in order to verify the new bogie design in extreme and fatigue load conditions. The related analysis and calculations are presented in the Appendix A.

4.3.3 Correction of Axial Misalignment

The axial misalignment of the rails can be corrected by means of welded supplementary plates as initially proposed by GRANTECAN. This would provide major control in the radial movement of the shutters.

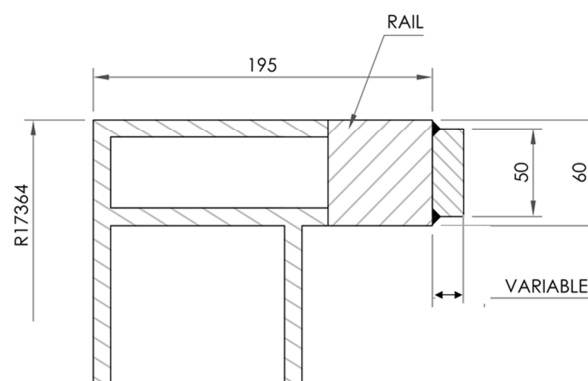


Fig. 4-31 Schematic representation of the rail with the welded supplements

The corrections will take place in both rails, over 2 zones in side A and over 3 zones in side B. This is illustrated in the following figure.

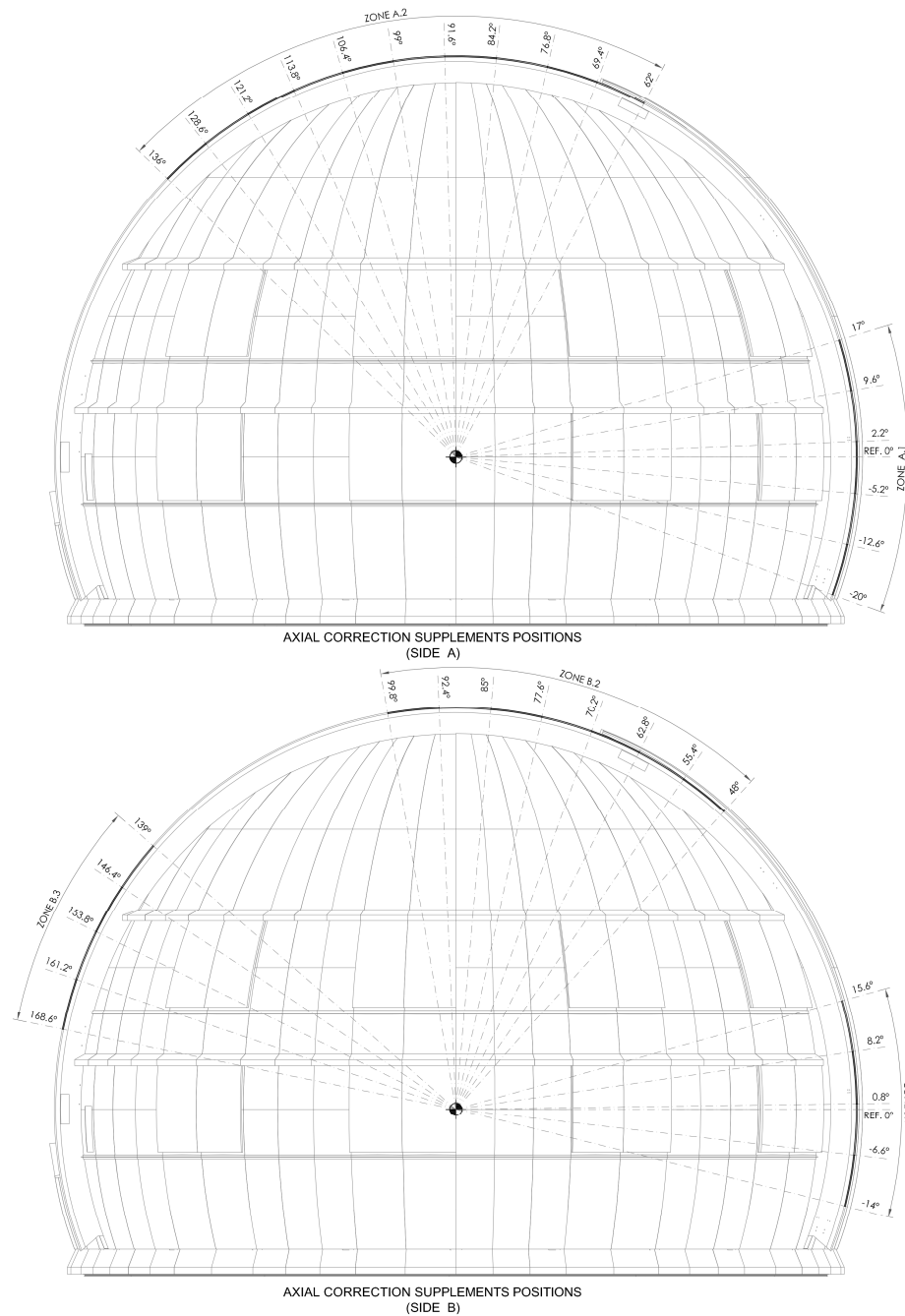


Fig. 4-32 Axial supplements installation zones

Each supplement will cover 7.40° (2232mm arc length) with 50mm height and a variable thickness from “a” to “b”. They can be manufactured from carbon steel S355.

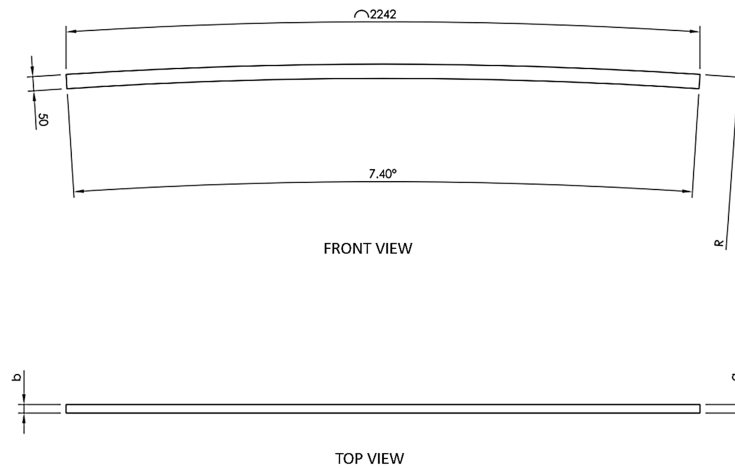


Fig. 4-33 Axial supplements

The dimensions and location of each of the supplements is tabulated in Appendix B drawing 18242-DWG-02-000.

An assessment has been made considering the current axial misalignment of the rails and the remaining misalignment after the axial correction proposed, to calculate the clearance of the bogie axial wheels with the rails.

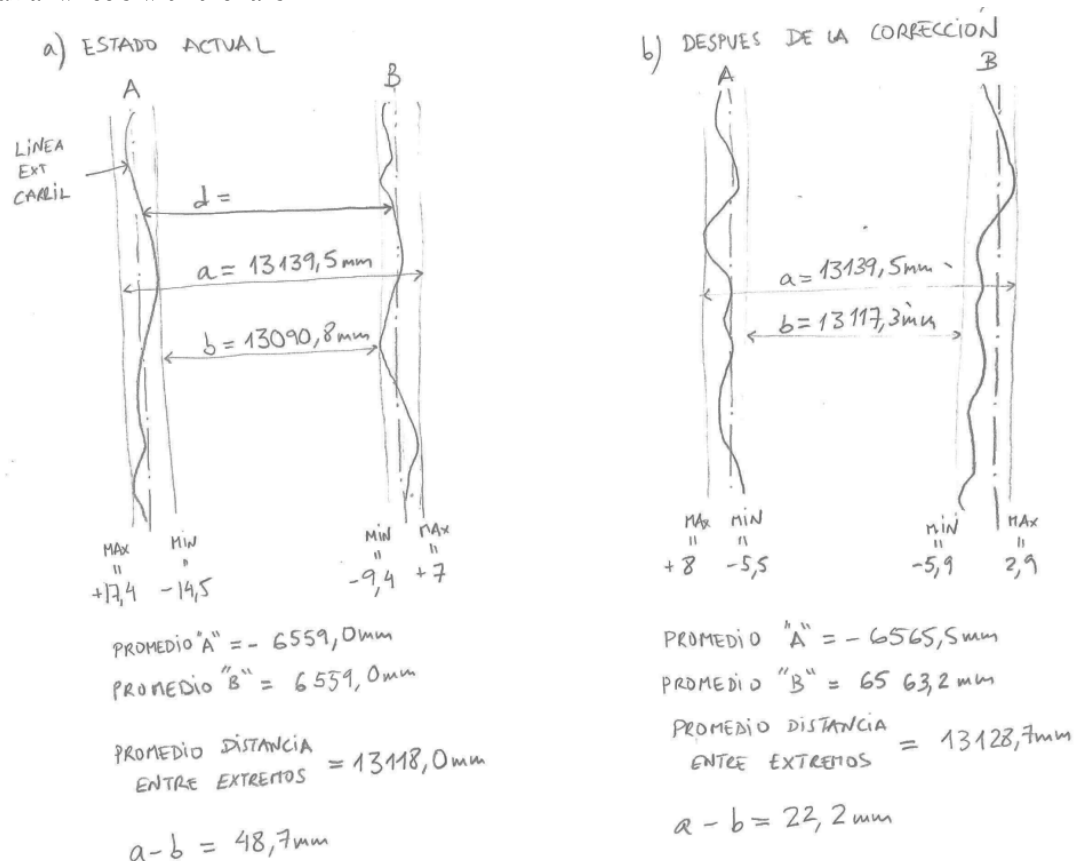


Fig. 4-34 Maximum and minimum distance between the rails. Current status on the left and status after the corrections on the right.

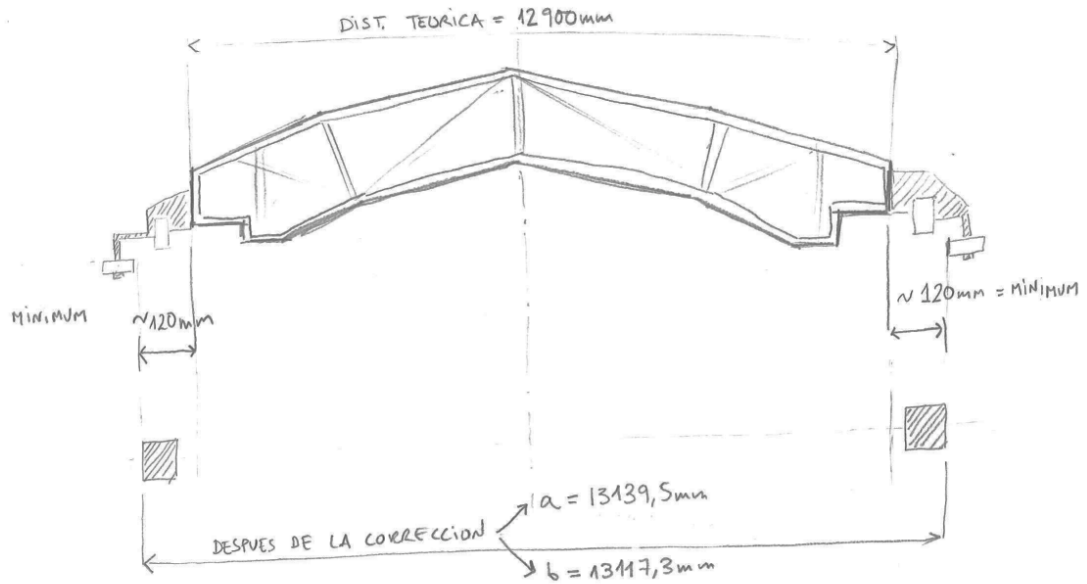


Fig. 4-35 Clearance representation of the bogie axial wheels

A clearance of 125mm is needed from the shutter lateral wall to the axial wheel. The following figure illustrates this distance.

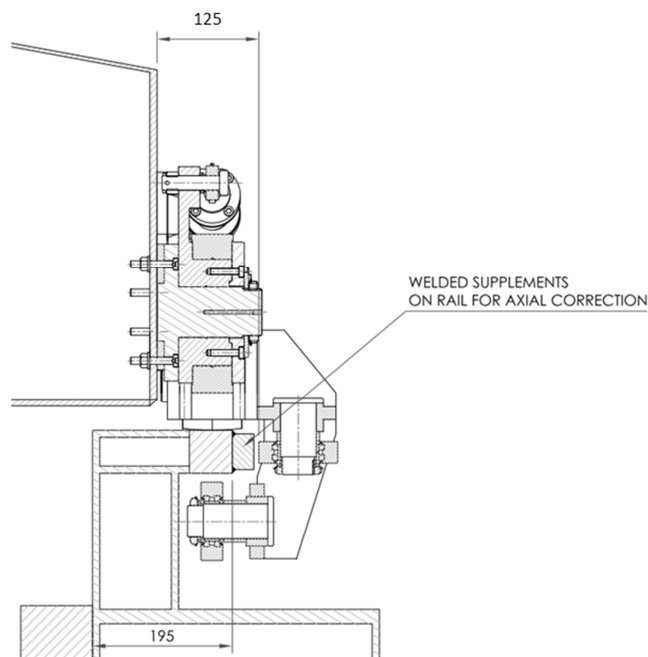


Fig. 4-36 Bogie cross section and 125mm for axial clearance

4.4 CONTROL DATA TRANSFER

4.4.1 Introduction

In order to transfer the data from the movable part (shutter) to the Dome a radio signal transfer is proposed.

RCoax radiating cables operate as antennae from an access point that provides reliable radio links in areas in which control data cannot be transmitted by conventional contact cables.

The IWLAN radio links would be installed in the shutter and the radiating cable is routed alongside the travelling path of the arc beam and due to its special radiation characteristic, it is an antenna that can be laid as cable.

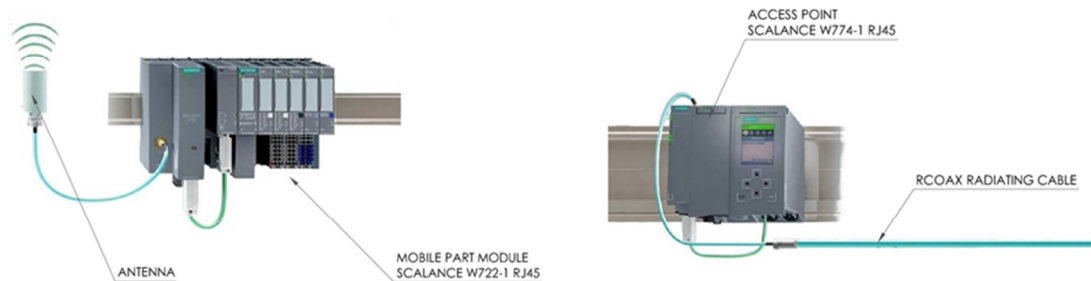


Fig. 4-37 Antenna on the left and the Radiating cable on the right

This would ensure a reliable radio link all along the shutter movement, so by means of this compact system the data transmission will be contact-free.

4.4.2 Installation

The radiation cable must be installed in all the Shutter travel length. So as two different lines are required, one per shutter, the Upper Shutter “RCoax” cable will be installed along the A side arc beam (see drawing 18242-DWG-06-001). The cable range shall be at least as long as the shutter travel, however two meter of extra cable will be installed in the extremes.

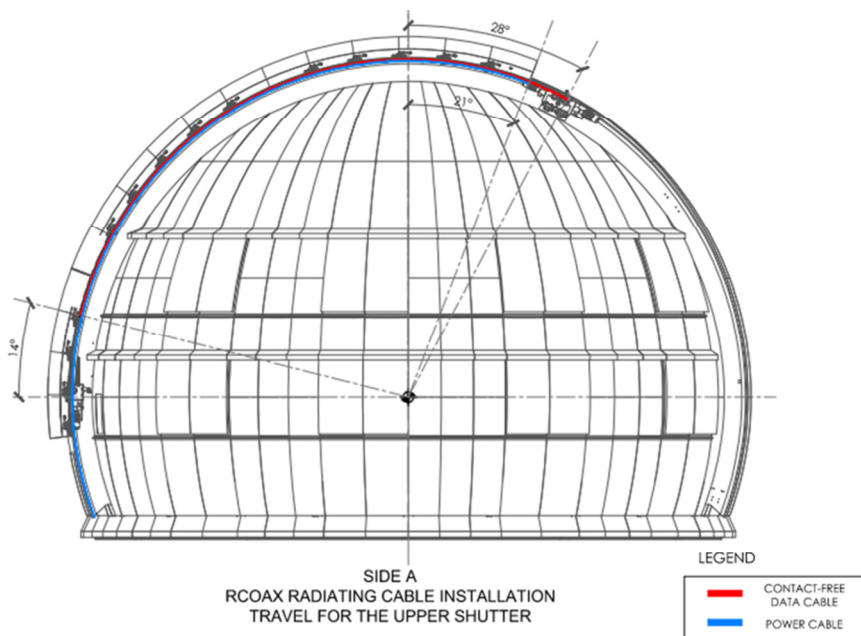


Fig. 4-38 RCoax cable installation

On the other hand, the installation of antenna must be done on the mobile part, so that it will move together with the shutter. In the Upper Shutter case, the antenna will be installed in the lowest part,

allowing the transmission of the data all along the shutter travel range. This is shown in the figure below.

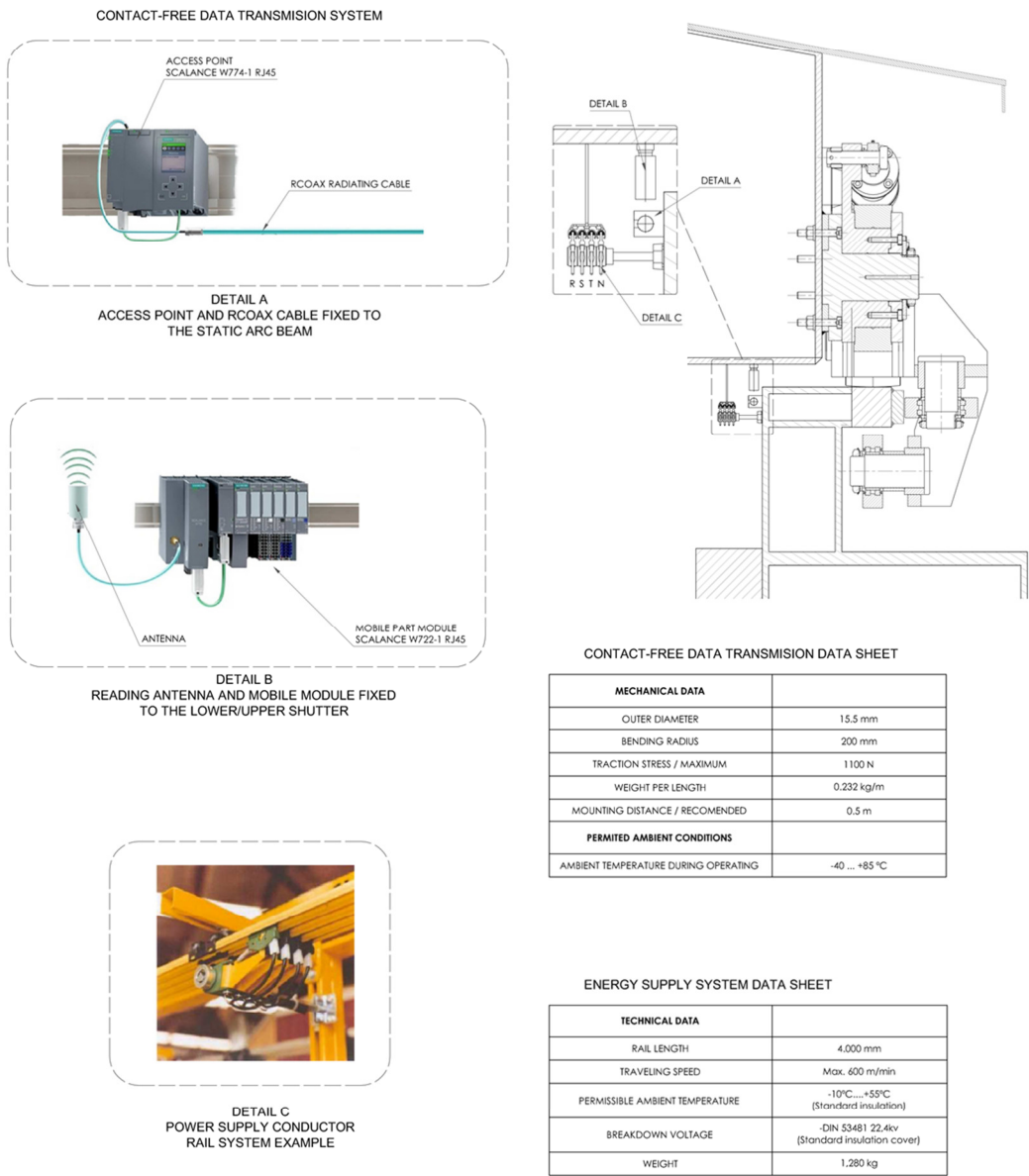


Fig. 4-39 Control data transfer system installation

The following data should be measured and transfer:

- 8 chain load cells data
- 1 hydraulic actuator pressure sensor per bogie (20 in total)
- 2 hydraulic actuator Limit switches per bogie (40 in total)
- Proximity sensors for hydraulic actuators (optional - 20 in total)

Those analogic and digital signals will be transferred to an I-O block installed in the Shutter. This I-O block will be connected to the mobile module through Profibus connection.

4.4.3 Slip Ring Power Supply

Power supply must be transferred to the reading antenna and the mobile module. This is obtained by a conductor rail system using single-pole insulated conductor rails, which provides the electric energy supply for the mobile parts. These conductor rails have been selected because of their compact installation, 20 mm spacing, and the capability to be installed in curved systems.

For the current application galvanized steel conductor rails have been selected covered with 10 amps galvanized conductor and standard insulation. The installation details can be seen in Appendix B, drawing 18242-DWG-06-001.

There are other alternatives that could be considered instead of a standard slip ring system, such as:

1. An improved slip ring, with less maintenance costs
2. Contactless systems, successfully implemented in other systems as cranes, such as:
 - a. *Inductive Power Transfer* (IPT ®) by Wampfler.
 - b. *Contactless Power System* (CPS ®) by Demag
3. A Battery Charging Dock, such as the *Battery Charging System* proposed by Demag for AGVs (Automated Guided Vehicles). The system could be installed in the shutters parking position.

5 LOWER SHUTTER UPGRADE

5.1 INTRODUCTION

According to the specifications given by GRANTECAN, the Lower Shutter shall be moved on a range between -15° and $+90^\circ$ with respect to the stow position as well as being able to deploy the windscreen.

GRANTECAN observed that in the current configuration the shutter is able to complete its opening, but without being capable of deploying the windscreen. The causes of this underperformance are a limited capacity of the shutter motors and the transmission chains together with a high friction of the windscreen.

Regarding the Lower Shutter some actions are taken on:

- 1) Drive System
 - a. Chain replacement in order to guarantee a $SF=4$.
 - b. Motor and gearbox replacement; increase of the reliability with an auxiliary motor input (operation in emergency conditions) and substitution of current main motor by another with higher power.
 - c. Change of the control system strategy
- 2) Guiding system: Same changes as for the Upper Shutter

5.2 DRIVE SYSTEM

Distinction shall be made between the independent operation of the shutter and the movement together with the windscreen. The windscreen adds a considerable load to the shutter drive, not only because of its self-weight but also because of the high friction inherent to the blades deployment and guiding process.

The current lower shutter drive system capacity has been assessed considering two scenarios:

- a) Movement of the Lower Shutter standalone (without windscreen)
- b) Movement of the Lower Shutter dragging the windscreen in current conditions.

In this load budget, factors as the de-rating of motor torque with altitude, the efficiency of the gearbox and the design safety margin for the chains have been considered.

For the lower shutter standalone movement, the more demanding position corresponds to the center of gravity placed at the horizon (rotation angle of -9° relative to the stowed position). The load decreases when the shutter moves to higher positions.

The windscreen load, conversely, increases with the rotating angle (deployment span). As a consequence, the more demanding position for the lower shutter moving together with the windscreen is at 40° rotation angle (maximum of the sum resulting from the decreasing curve for the shutter load and the increasing curve for the windscreen load).

The load budget plots indicate in green colour the margin available with respect to the capacity of the current drive system. Negative values of this margin indicate that the capacity is not sufficient to cover the load requirements. The final margins for the three scenarios are collected in Fig. 5-2.

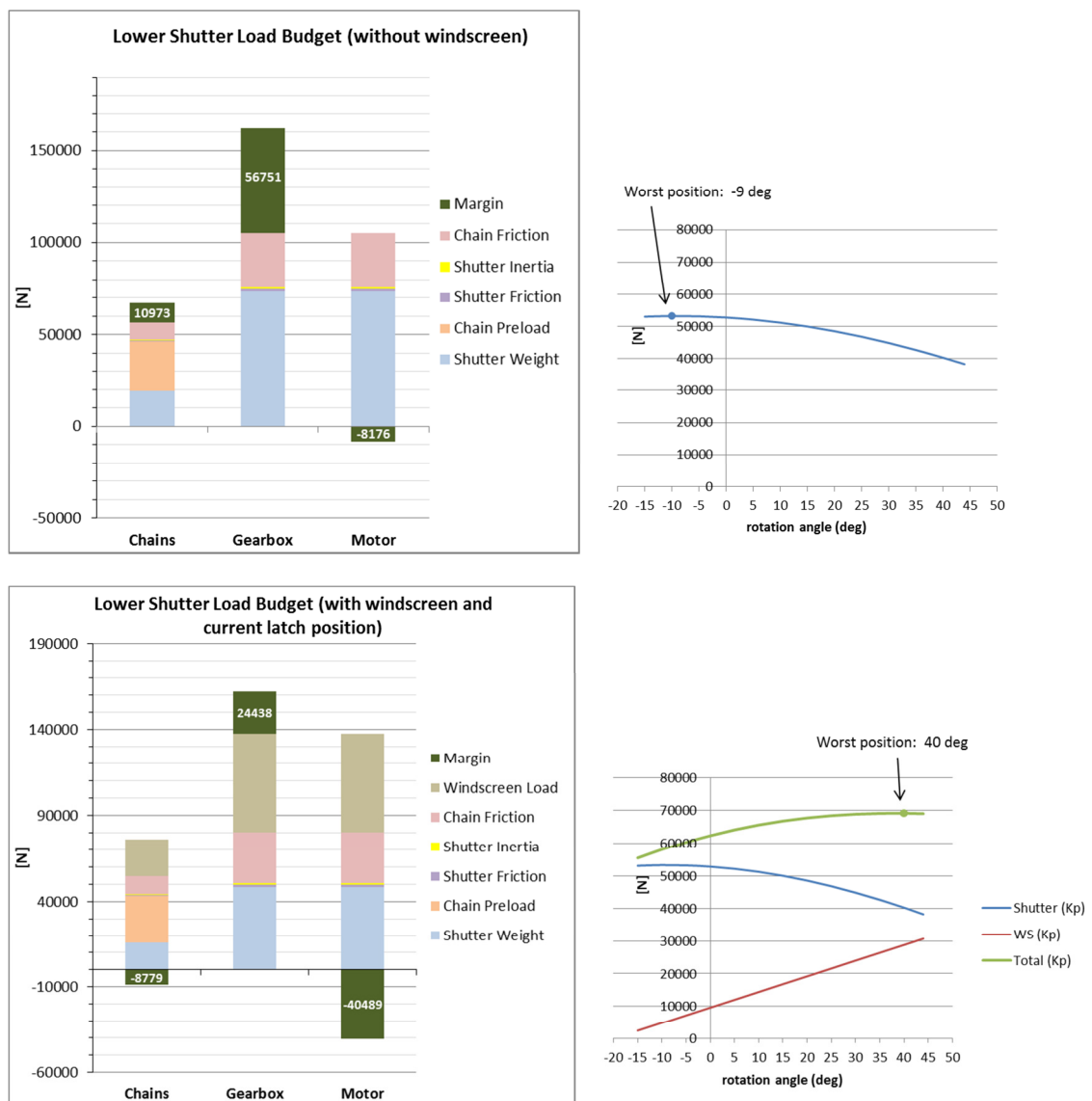


Fig. 5-1 Lower Shutter Load Budget: a) w/o windscreen; b) with windscreen and current latch

SUMMARY TABLE	w/o windscreen		w. windscreen current latch		Current system capacity	
	Total	Margin	Total	Margin		
Required chain capacity [N]	56527	10973	76279	-8779	67500	(safety margin = 4)
Required gearbox torque [N m]	16243	8757	21229	3771	25000	
Required motor torque [N m]	78	-6	102	-30	72	

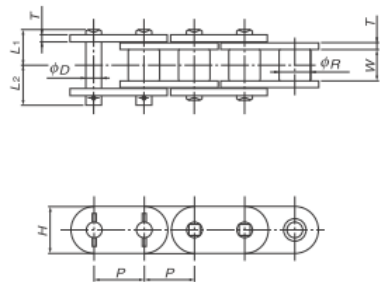
Fig. 5-2. Required values and existing margin for operation of lower shutter

Conclusion: The capacity of the current motors and chains is sufficient to move the lower shutter alone, but not to deploy the windscreen, even if the latching point position is modified.

5.2.1 Chain Replacement

As in the case of the upper shutter, it is also necessary to change the current roller chains by new ones with higher capacity as it was initially proposed by GRANTECAN on page 64 of the applicable document AD 2.

The proposed chain is a TSUBAKI RF160-US-1 with the main characteristics shown below,



TSUBAKI Chain Number	Pitch P	Roller Diameter R	Inner Width of Inner Link W	Plates		Pins				Minimum Tensile Strength kN(kgf)	Average Tensile Strength kN(kgf)	Maximum Allowable Load Tension kN(kgf)	Approximate Mass kg/m
				Thickness T	Width H	Diameter D	L1 + L2	L1	L2				
RF100-US-1	31.75	19.05	19.05	4.8	30.1	10.32	47.7	22.35	25.35	149{15200}	172 {17500}	39.2{4000}	5.07
RF120-US-1	38.10	22.23	25.40	5.6	36.2	12.28	59.1	27.55	31.55	213{21700}	245 {25000}	53.9{5500}	7.22
RF140-US-1	44.45	25.40	25.40	6.4	42.2	13.97	63.7	29.5	34.2	273{27800}	314 {32000}	63.7{6500}	9.24
RF160-US-1	50.80	28.58	31.75	7.1	48.2	15.62	74.7	34.5	40.2	341{34800}	392 {40000}	85.3{8700}	12.19
RF200-US-1	63.50	39.68	38.10	9.5	60.3	20.41	93.9	42.95	50.95	580{59100}	667 {68000}	108{11000}	20.47
RF240-US-1	76.20	47.63	47.63	12.7	72.4	24.73	119.7	54.8	64.9	853{87000}	981{100000}	151{15400}	31.69

Fig. 5-3 Specifications of the TSUBAKI RF160-US-1

According to the estimations made after the design of the proposed actions for the windscreen enhancement (see Chapter 6), a chain load of 53KN is necessary in order to deploy the Lower Shutter and the Windscreen in the most demanding position. The minimum ultimate tensile strength for the proposed new chain is 341KN which ensures a safety factor of 4 while deploying the Windscreen. The following figure shows the load budget for the chain.

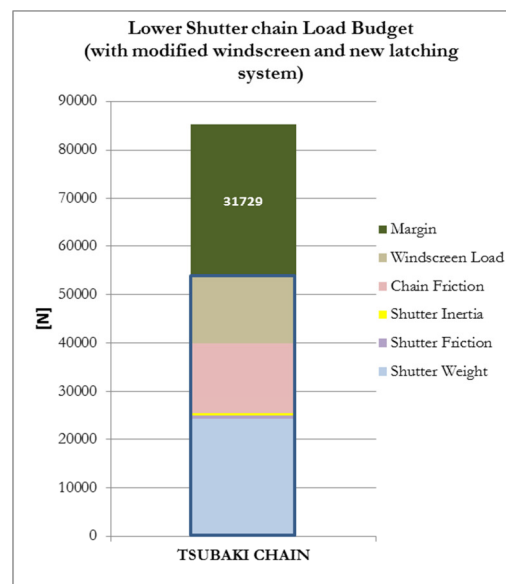


Fig. 5-4 Lower Shutter load budget for the TSUBAKI RF160-US-1¹

¹ The modified windscreen and new latching system is explained in Section 6 Windscreen Enhancement

5.2.2 Motors Upgrade

The under capacity of the current motors to move the Lower Shutter with the Windscreen (see **¡Error! No se encuentra el origen de la referencia.**) demands the substitution of the motors. Two alternative motors are considered: a Demag ZBA 160B4 and an equivalent SIEMENS motor (1LE1001-1DB43-4GB4-Z) in order to guarantee the total aperture.

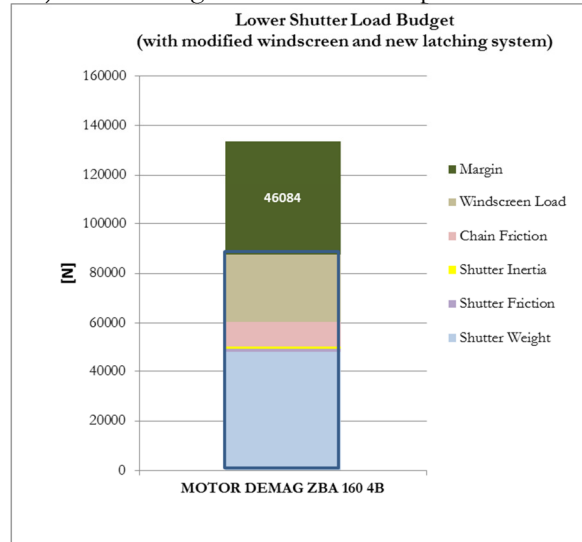


Fig. 5-5 Lower Shutter load budget for the DEMAG ZBA 160 4B¹

5.2.3 Control System

The same Control Strategy as in the Upper Shutter, “Speed Trim Follower” (see Section 4.2.2), is proposed for the Lower Shutter. The control diagram is shown in the figure below.

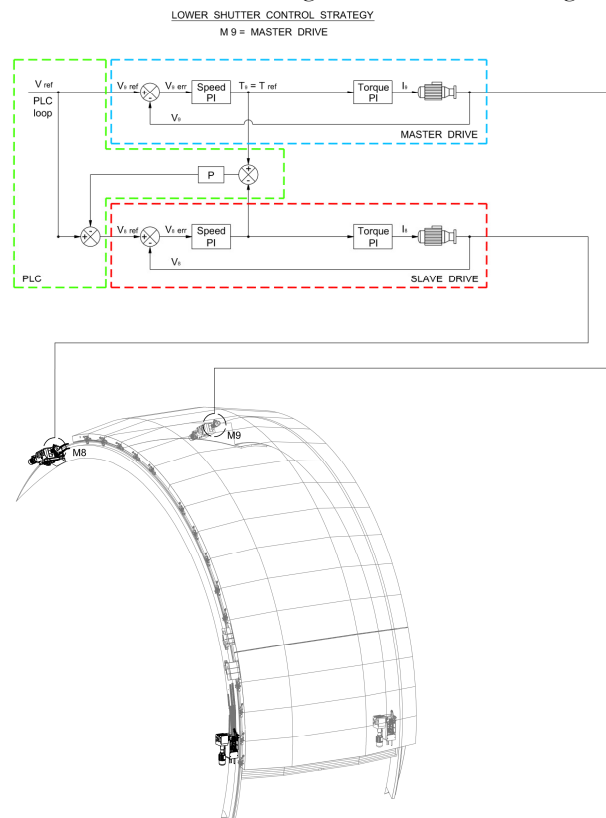


Fig. 5-6. Diagram for Speed Trim Follower configuration for the Lower Shutter.

5.2.4 Operation in Emergency Conditions

It is considered an Emergency Condition when one of the motors of the shutter is inoperative. Considering the necessity of introducing an Emergency System, two alternatives have been studied:

- a) Motor-gearbox solution with an auxiliary input
- b) Double input $i=1:1$ ratio gearbox at the current gearbox entrance

a) Motor-gearbox solution with an auxiliary input:

It was studied the possibility of introducing an auxiliary input to the current gearbox, however this solution was discarded after a meeting with FLENDER, as it is not feasible to introduce an additional input in the required reduction stage of the current gearbox. Therefore an alternative solution which implicates replacing the gearbox is proposed. The increase in the reliability of the proposed solution with respect to the current system is shown in RD9

The selected gearbox must consider the following aspects:

- Mechanical requirement:
 - Output torque. 25000Nm
 - Reduction ratio: $i=272$
 - Possibility of installing an auxiliary input
- Geometrical limitation:
 - The 315mm is the maximum distance between the output shaft and the lower shutter gearbox housing.

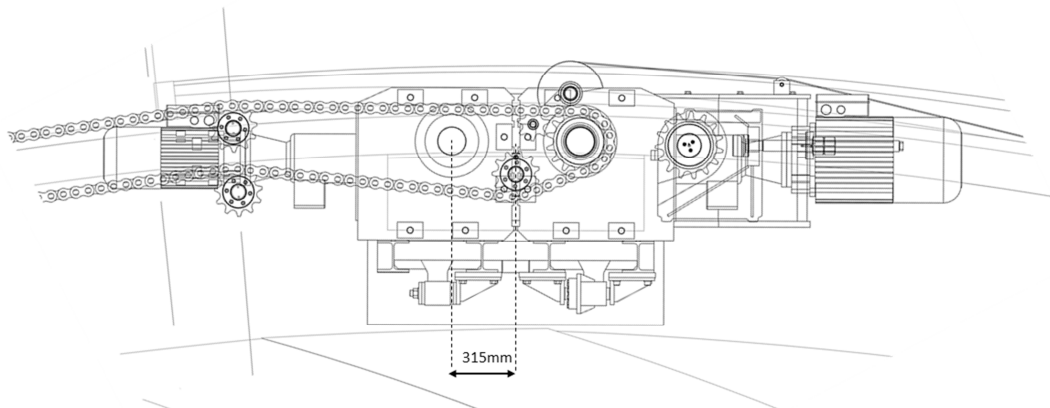


Fig. 5-7 Geometrical limitation in the upper station

Within the commercial solutions from the FLENDER catalogue the gearboxes that fulfill the mechanical requirements do not comply with the geometrical requirement if it is placed in horizontal position. Therefore it has been decided to install the gearbox in vertical position

In order to satisfy these requirements the new FLENDER B4DH08B280 gearbox must be oriented vertically. Thus the main motor-gearbox block is in a vertical position being the motor pointing downwards. The auxiliary input is in the outer face of the gearbox in a horizontal axis.

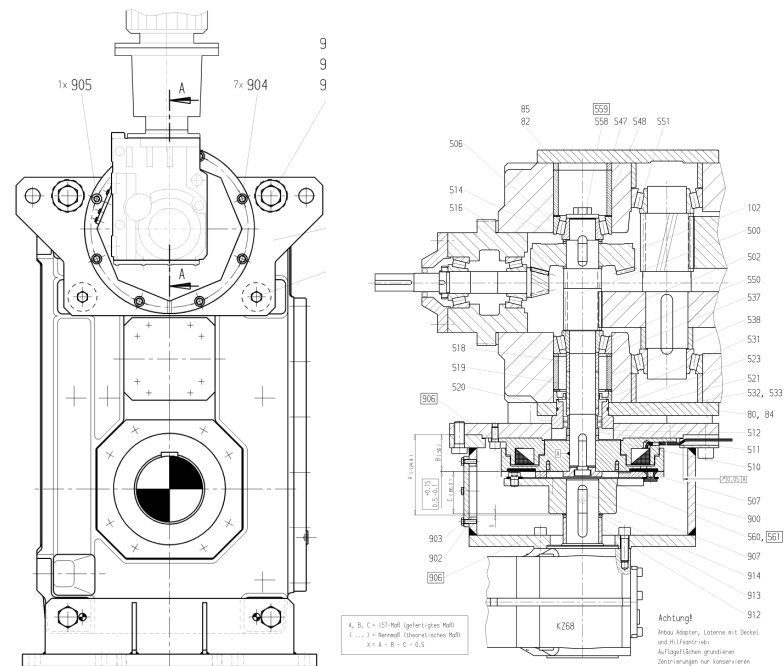


Fig. 5-8 New FLENDER B4DH08B280 gearbox with the auxiliary input on the left and Section A-A on the right (auxiliary input)

The following image shows the conceptual idea of the proposed solution, the final selection of the components is under study by FLENDER.

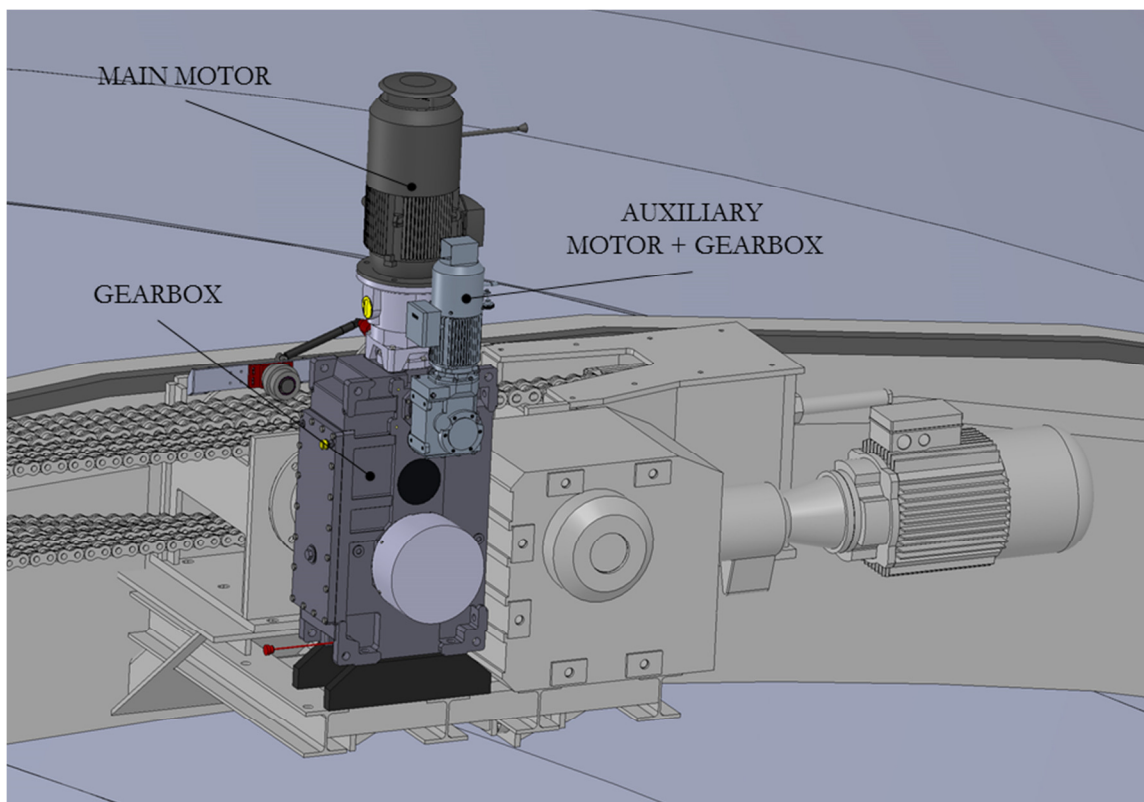


Fig. 5-9 3D view of the solution

In case of a failure of one of the main motors, both main motors will be disconnected and their brake relieved. The auxiliary motors will then be mechanically connected by means of the

engagement of a clutch and driven either in a direct way or controlled through VSD-s. In the former, the absence of a control system requires a manual movement of the shutter. However, in case it is decided to implement an automatic control to the auxiliary motors new VSD-s need to be installed and a control strategy for the Emergency Condition is needed. This should be defined in the Detail engineering phase.

b) Double input $i=1:1$ ratio gearbox at the current gearbox entrance:

In order to minimize the impact it has also been studied the possibility of installing a $i=1:1$ ratio gearbox at the current gearbox entrance.

The proposed gearbox is the Graessner P110, which has the characteristics shown below.

		Abbr.	Unit	P54	P75	P90	P110	P140	P170	P210	P240	P280
Output torque	i=1:1	T _{2N}	Nm	15	45	78	150	360	585	1300	2150	3200
		T _{2B}	Nm	23	68	117	225	540	878	1950	3225	4800
		T _{2Not}	Nm	30	90	156	300	720	1170	2600	4300	6400
	i=1.5:1	T _{2N}	Nm	15	45	78	150	360	585	1300	2150	3200
		T _{2B}	Nm	23	68	117	225	540	878	1950	3225	4800
		T _{2Not}	Nm	30	90	156	300	720	1170	2600	4300	6400
	i=2:1	T _{2N}	Nm	12	42	68	150	330	544	1220	2010	3050
		T _{2B}	Nm	18	63	102	225	495	816	1830	3015	4575
		T _{2Not}	Nm	24	84	136	300	660	1088	2440	4020	6100
	i=3:1	T _{2N}	Nm	12	33	54	120	270	450	1020	1650	2850
		T _{2B}	Nm	18	50	81	180	405	675	1530	2475	4275
		T _{2Not}	Nm	24	66	108	240	540	900	2040	3300	5700
	i=4:1	T _{2N}	Nm	–	28	52	100	224	376	860	1410	2300
		T _{2B}	Nm	–	42	78	150	336	564	1290	2115	3450
		T _{2Not}	Nm	–	56	104	200	448	752	1720	2820	4600
	i=5:1	T _{2N}	Nm	–	25	40	85	196	320	740	1210	2000
T _{2B}		Nm	–	38	60	128	294	480	1110	1815	3000	
T _{2Not}		Nm	–	50	80	170	392	640	1480	2420	4000	
Input speed	i= 1:1	n ₁ Standard	min ⁻¹	2500	2000	1700	1400	1100	1000	800	700	650
	i= 1.5:1 ; 2:1	n ₁ Standard	min ⁻¹	3000	2500	2000	1600	1400	1300	1050	950	850
	i= 3:1 ; 4:1 ; 5:1	n ₁ Standard	min ⁻¹	3500	3000	2500	2100	2000	1800	1600	1350	1200
		n ₁ max **	min ⁻¹	7500	6500	5500	4500	3500	3000	2200	2000	1700
	**On request, special measures required											

Fig. 5-10 Table of characteristics

The new gearbox is fitted with two input shaft and a single output shaft. The previously mentioned SIEMENS motors (97Nm) are installed in both inputs and the output shaft is connected to the current gearbox entrance by way of a coupling.

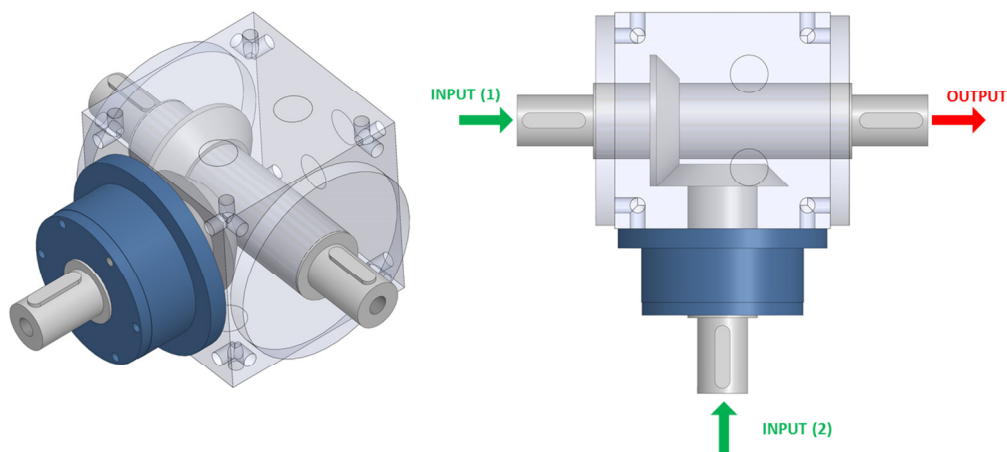


Fig. 5-11 3D and top-view of the gearbox

In normal operational conditions, the system shall be driven by both motors, equally sharing the required torque between both of them.

Regarding the Control System in this scenario, a control configuration suitable for rigid systems is proposed which can provide a proper load sharing.

The strategy consists of:

- the master drive is operated in speed regulation (input 1)
- the slave drive is operated in torque regulation which is directly measured at the master drive

Between the two master drives from side A and side B a Speed Trim Follower strategy shall be configured in order to guarantee a good load share between the motor of both sides. This would be as explained in chapter 5.2.3 Control System.

5.3 GUIDING SYSTEM

5.3.1 New Supporting Mechanism

(See Section 4.3.2)

5.3.1.1 Supporting Strategy

(See Section 4.3.2.1)

5.3.1.1.1 Mechanism Layout and reaction Equalization system

The proposed supporting mechanism is composed of 8 bogies, 4 at each side, which transmit the radial reactions. Bogies are arranged in four groups of 2 bogies each, being the reactions hydraulically equalized among the bogies in the same group.

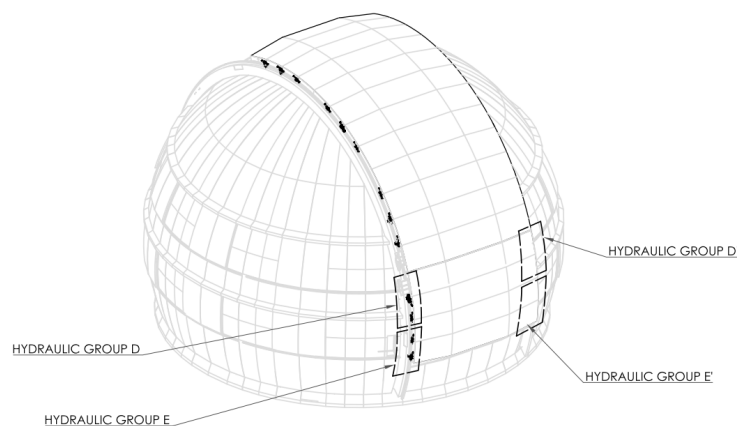


Fig. 5-12 Lower Shutter hydraulic groups

The equalization is achieved by hydraulically connecting the two cylinders which supports the reactions on each sector.

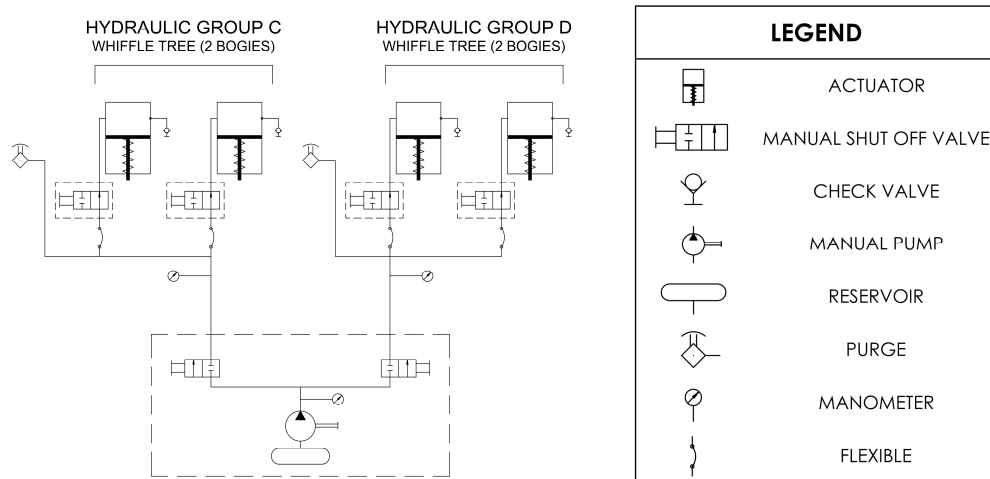


Fig. 5-13 Hydraulic diagram of the Group C and Group D

The hydraulic group distribution and diagram are presented in Appendix A drawing 18242-DWG-01-001.

As it was mentioned for the Upper Shutter, during the Detailed Engineering phase it should be studied the necessity of installing bladder accumulators in order to reduce the stiffness of the system. It is also advisable to install limit switches at actuator locations and pressure sensors in order to monitor the status of the system.

5.3.2 New Bogies Design

(See Section 4.3.2.2)

5.3.2.1.1 Working principle

(See Section 4.3.2.2.1)

5.3.2.1.2 Bogie types and positions

(See section 4.3.2.2.2)

The position of the different bogie types are specified in the 18242-DWG-01-000.

5.3.3 Correction of Axial Misalignment

(See Section 4.3.1)

5.4 CONTROL DATA TRANSFER

5.4.1 Introduction

(See section 4.4.1)

5.4.2 Installation

For the Lower Shutter the installation of the free-contact data transmission system will be similar to the Lower Shutter. The “RCoax” Cable will be installed in side B of the arc beam avoiding any

interference with the Upper Shutter “RCoax” cable. Additionally the cable travel range will be different, see drawing 18242-DWG-06-001 to see the differences between both installations.

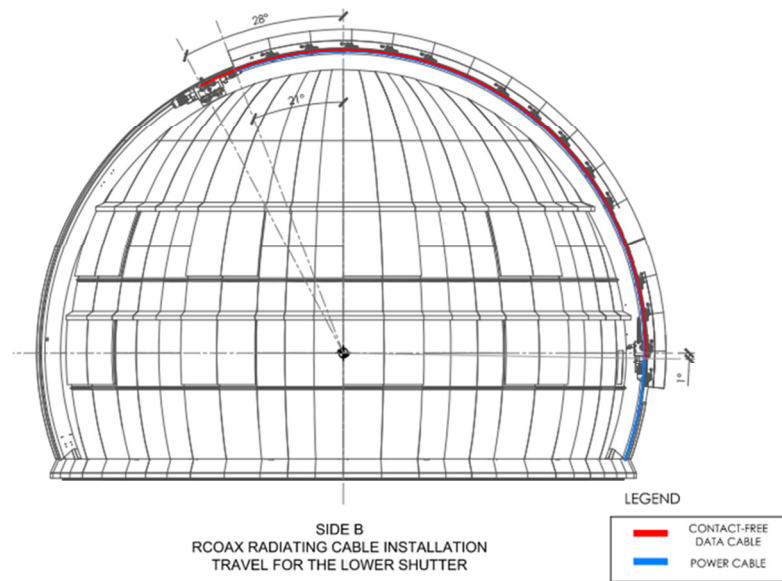


Fig. 5-14 RCoax cable installation (red) Lower shutter

On the other hand the installation of antenna must be done on the mobile part, thus it will move with the shutter. See Fig. 4-39.

The following data should be measured and transferred:

- 8 chain load cells data
- 1 hydraulic actuator pressure sensor per bogie (8 in total)
- 2 hydraulic actuator Limit switches per bogie (16 in total)
- Proximity sensors for hydraulic actuators (optional, 8 in total)

Those analogic and digital signals will be transferred to an I-O block installed in the Shutter. This I-O block will be connected to the mobile module through Profibus connection.

5.4.3 Slip Ring Power Supply

See section 4.4.3

6 WINDSCREEN ENHANCEMENT

6.1 INTRODUCTION

During the Phase I of the project a diagnostic of the Windscreen deficiencies was carried out and a combination of different actuations were proposed in order to optimize the current Windscreen.

The modifications proposed in the document RD4 and listed below were accepted by GRANTECAN:

- Latching system redesign
- Modification of Windscreen guides profiles
- Replacement of the current rollers for cam followers

These solutions have been developed in more detail during the Phase II and there are presented in the following sections.

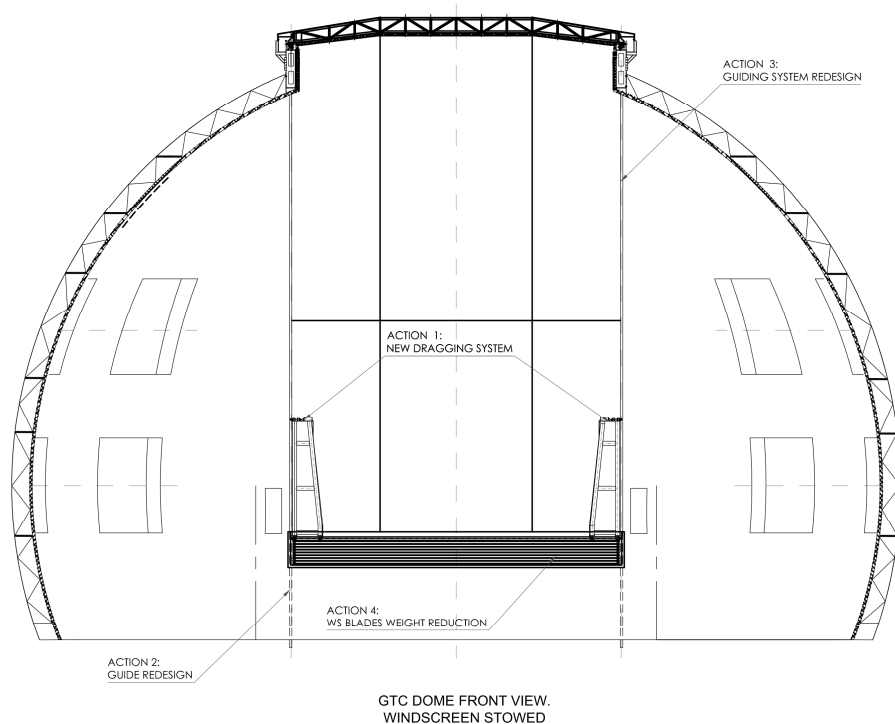


Fig. 6-1 Windscreen enhancement actions

6.2 LATCHING SYSTEM REDESIGN

6.2.1 Aim

The purpose of the new latching system is to reduce the number of deployed blades for the same aperture angle of the Lower Shutter with the Windscreen. This is feasible because in the current configuration there are four blades behind the Lower Shutter.

The solution proposed at the end of the Phase I and accepted by GRANTECAN consisted of maintaining the latching point of the Lower Shutter and raising the latching actuator of the

windscreen accordingly. An intermediate dragging structure would join the attachment point with the first blade. This structure holds the latching actuator at the required height.

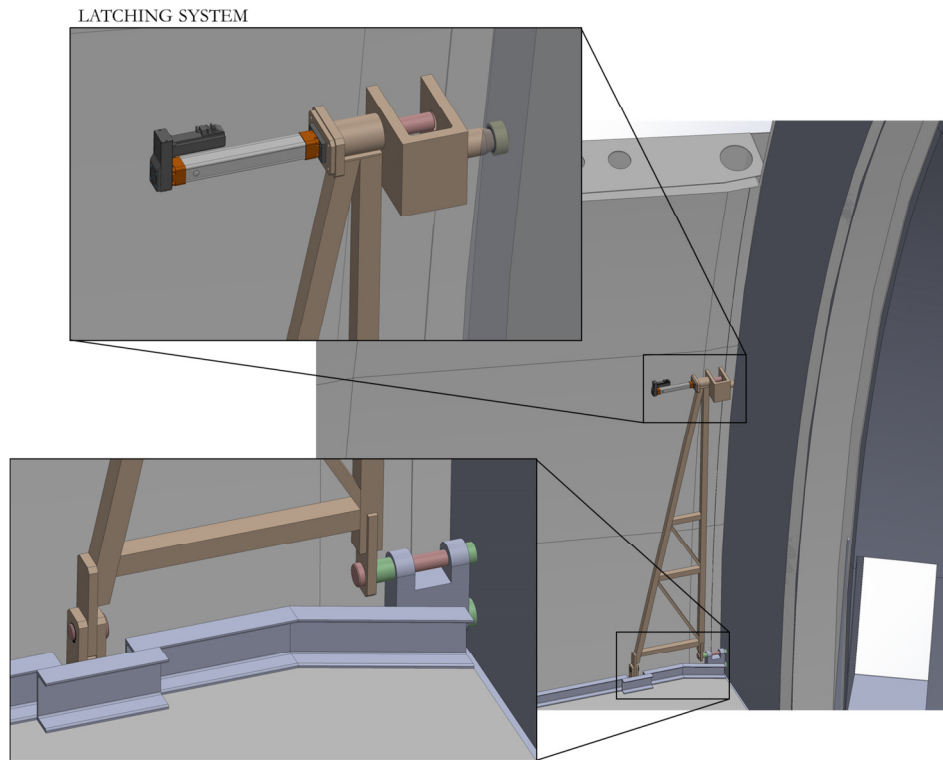


Fig. 6-2. Latching system positioned in the Dome

Initially the height of the dragging system was proposed so the four blades behind the Lower Shutter could have been removed, placing the actuator at 4500mm from the blades stow position. However, with this configuration GRANTECAN considered that the distance of 300mm from the actuator edge to the telescope observation area was too small forcing to control the tracking of the azimuth in a more precise way. Therefore, it was decided to reduce the dragging frame height down to 2600mm.

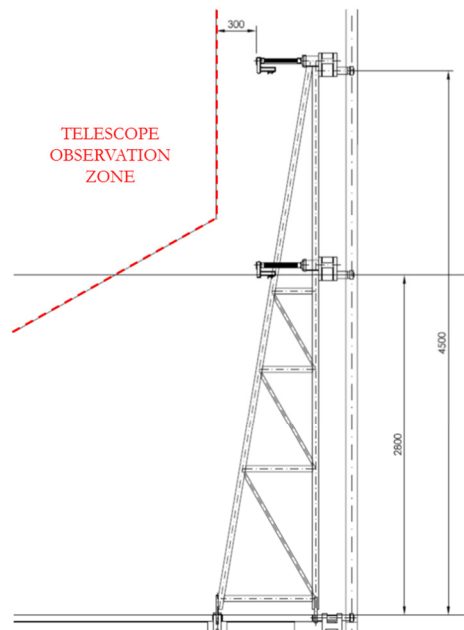


Fig. 6-3. Schematic representation of the change in the height

6.2.2 Dragging Frame

Two dragging structures, one at each side of the aperture, are built up with steel squared tube profiles. The frame is pinned to the first blade by mean of bearings on the lower part and support the latching system with a cam follower on the upper side, which works as an anti-rotation system.

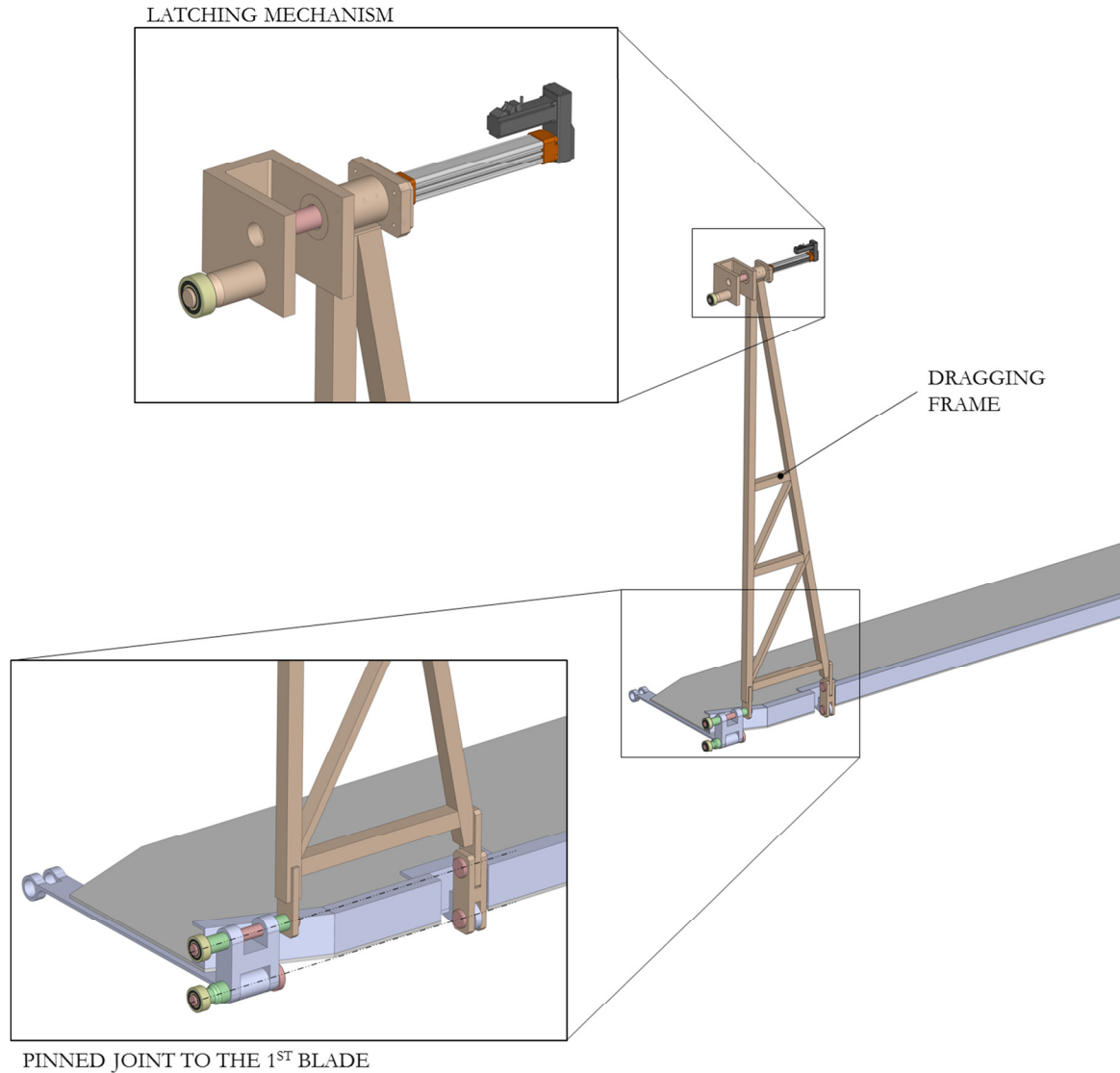


Fig. 6-4. Latching system redesign

6.2.3 Latching Mechanism

The latching system consists of a linear electrical actuator, which pushes the pin in order to attach the dragging frame to the Lower shutter lug. This is shown in the figure below.

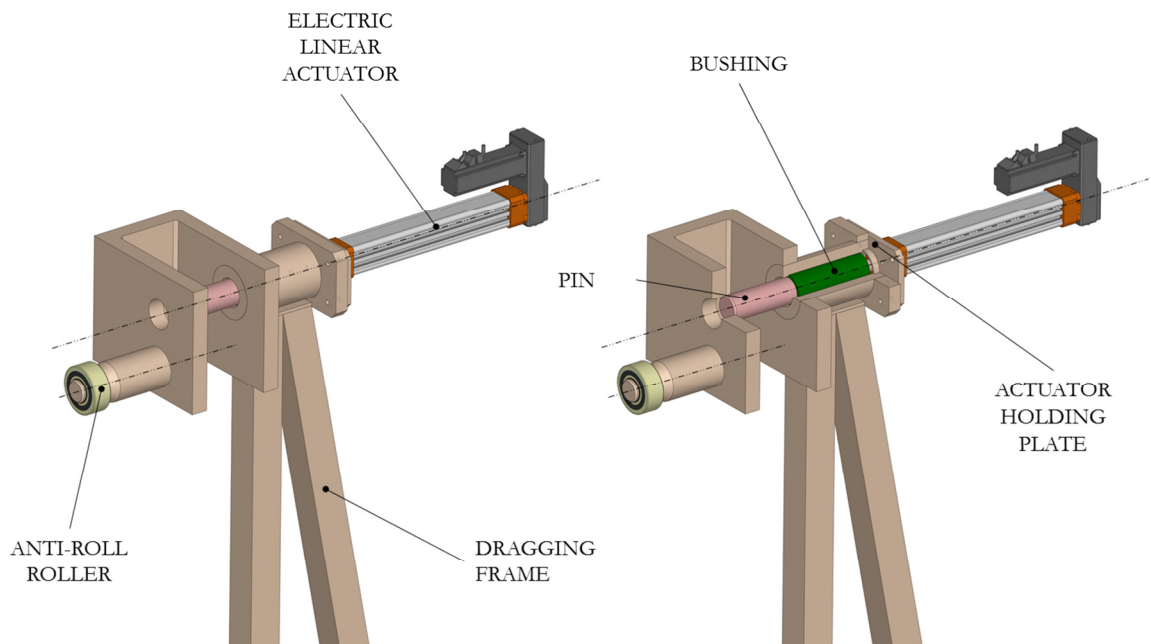


Fig. 6-5. Latching system

6.3 MODIFICATION OF WINDSCREEN GUIDES PROFILE

6.3.1 Aim

In order to reduce the friction at the beginning of the deployment of the blades and the noise during the blade deployment, the following modifications are proposed in the Windscreen guide profiles.

6.3.2 Reduction of friction in the outer roller at the beginning of the deployment of each pair of blades

The high resultant force due to the blades configuration in the stored state creates a friction in the roller of the screen.

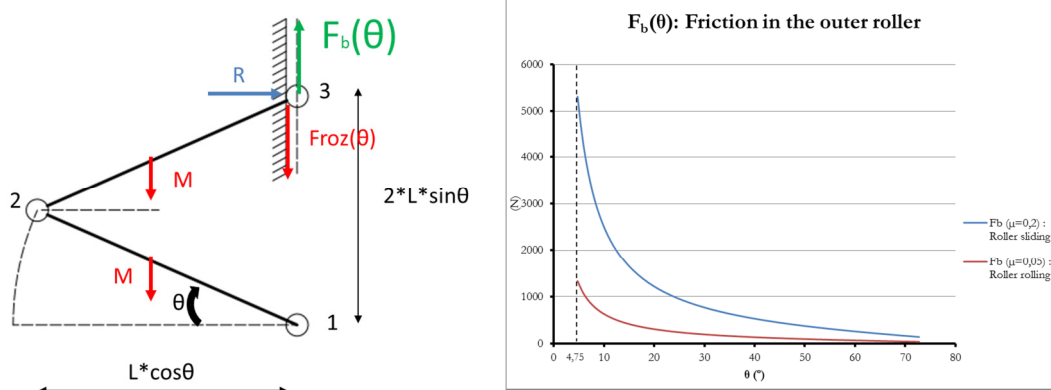


Fig. 6-6. Force diagram (left) and final plot (right) of the friction in the outer roller during windscreen blades deployment

It is proposed to create a small “belly” in the contact guide in order to release the guiding restriction. This would be the schematic representation of the idea:

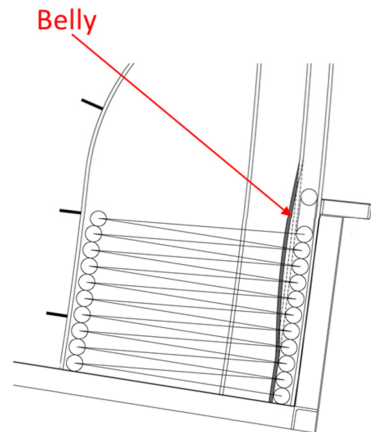


Fig. 6-7. Schematic representation of the “belly” in the outer guide

6.3.3 Reduction of the impact noise during the blades deployment

The impact of the inner roller with the guide produces noise. A possible option is to reduce the noise and the variation of the loads by modifying the inner guide.

It is proposed to add a ramp for the inner roller to guarantee a smooth contact between the roller and the guide. A possible layout is shown in the image below.

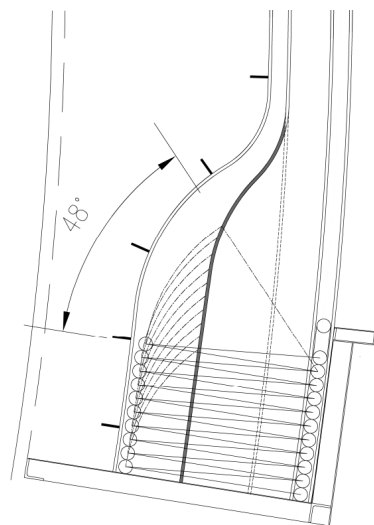


Fig. 6-8. Schematic representation of the modified guide

In the new configuration the speed of the roller at the impact instant is lower and therefore the noise will be reduced. It is proposed to install this new flange to ensure that the impact occurs when $\theta \approx 45^\circ$

6.4 REPLACEMENT OF CURRENT ROLLERS FOR CAM FOLLOWERS

6.4.1 Aim

The main scope of this modification is to reduce the Windscreen roller friction.

As it was explained in the document RD4, in the current situation the Windscreen friction represents 2.5times the weight of the screen in the most demanding position.

Going back to the Figure 5-22 of the document RD 4 which represents the demanded tangential force during the Opening of the Windscreen, the high influence of the friction coefficient in the load value is visible.

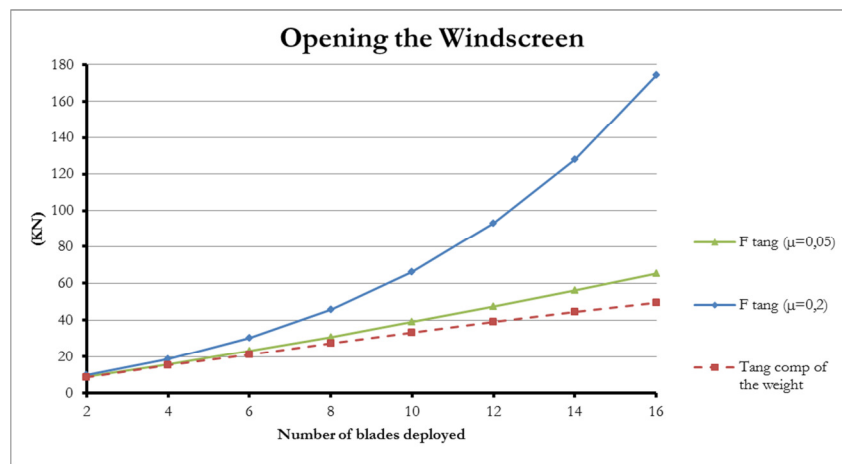


Fig. 6-9. Total tangential force needed to drag windscreen during opening, assuming $\mu=0.05$ and $\mu=0.2$; and component due only to windscreen weight

According to this estimation if a low friction is guaranteed throughout the deployment of the screen, then the required tangential force will be around 150% of the weight component (close to the green curve represented on the previous figure).

This could be considerably improved by replacing the current rollers by cam followers.

6.4.2 Description of New Rollers Design

The following figure shows the proposed solution for the new rollers.

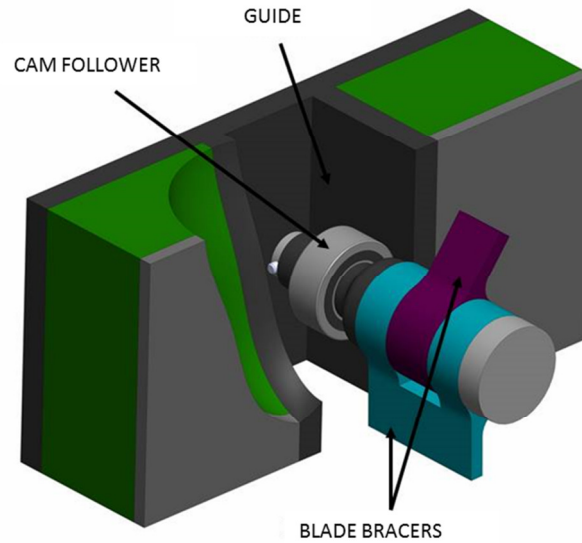


Fig. 6-10 Proposed Bolt-bearing configuration for Windscreen rollers

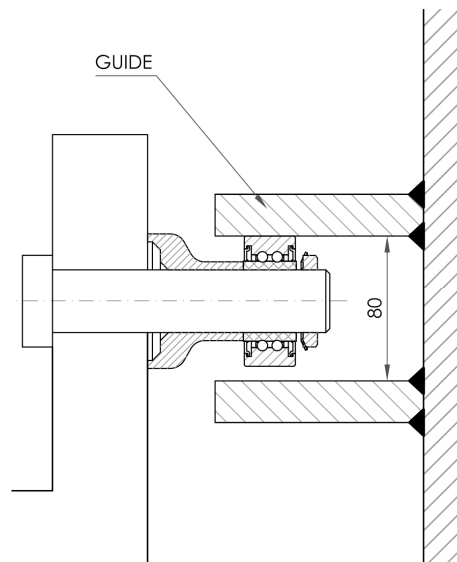
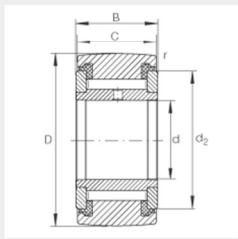


Fig. 6-11. Cross section of the proposed Bolt-bearing configuration for Windscreen rollers

Rodillos de apoyo NATV35-PP

sin jaula, con guía axial, disco de fricción axial en ambos lados



d	35 mm	
D	72 mm	
B	29 mm	
C	28 mm	
d₂	58 mm	
f_{min}	1,1 mm	
m	630 g	Peso
C_{rW}	32500 N	Capacidad de carga dinámica efectiva (radial) como rodillo-guía
C_{0rW}	71000 N	Capacidad de carga estática efectiva (radial) como rodillo-guía
C_{urW}	10100 N	Carga límite de fatiga
n_{DG}	750 1/min	Velocidad de rotación (en funcionamiento continuo y para lubricación con grasa)
		Rodillo de levas con disco de fricción axial, de plástico y perfil optimizado INA. Temperatura de funcionamiento admisible: desde -30°C hasta +100°C (servicio continuo).

Fig. 6-12 Parameters for selected bearing. Source: INA/FAG catalogue

6.5 WEIGHT REDUCTION

6.5.1 Aim

A weight saving study has been performed to assess the available weight saving margin maintaining the current Windscreen.

Two options have been proposed:

- a) Option based on a drilling pattern
- b) Option based on large removal areas covered by fabrics

6.5.2 Option based on a drilling pattern

It considers the option of drilling the Windscreen blades for a maximum of 20% of permeability.

15 holes of 50mm of diameter were fit in each “triangle” of the blade, resulting in an impermeability of 85%. The total weight saving is ≈ 0.6 ton.

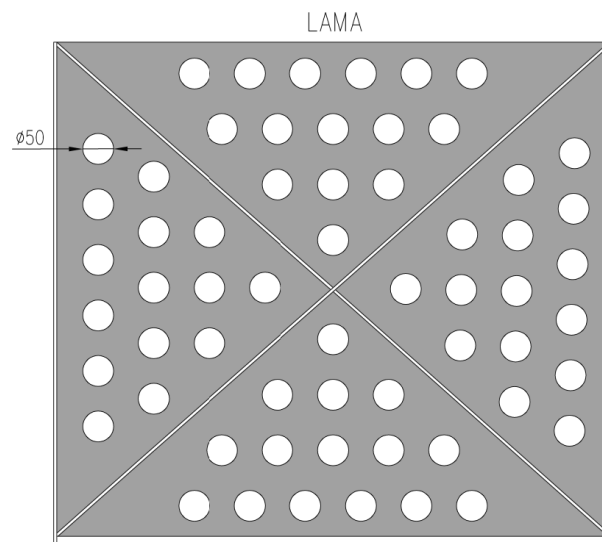


Fig. 6-13. Schematic representation of the drilled blade

6.5.3 Option based on large removal areas covered by fabrics

The second alternative consists of removing a bigger area from the triangular plates, covering the remaining gaps by fabrics (see next figure). The weight saving is around ≈ 1.2 ton.

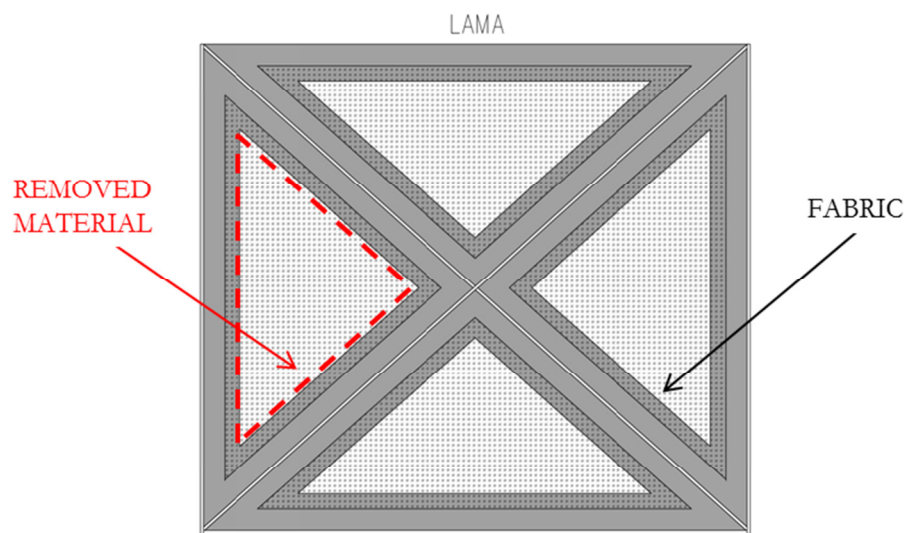


Fig. 6-14. Schematic representation of the blade with the triangular holes and fabric

7 ACTION PLAN FOR THE DAMAGES ON THE DOME ARCH GIRDERS

7.1 INTRODUCTION

The damages on the dome arch beams were observed during the visit to the ORM, with particular interest in the cracks detected. A visual inspection seems to indicate that most fissures are not originated at the weld beads, which are usually starting points for fatigue cracks. In any case, no conclusion can be reached from this initial evaluation without performing a monitoring process along time in order to evaluate whether the crack is stationary and, if not, to estimate the propagation rate.



Fig. 7-1. Detail of a crack in the dome arch girder.

GRANTECAN inspected the dome arch girders and shutters in March, 2013 and registered all the damages detected. An inventory of all the cracks was made. Lately, approximately 1 year later (in January, 2014) the inspection was repeated. The prior and last results from these inspections were documented in AD 4. The conclusion was that no significant propagation of the initial cracks had occurred during the last 10 months. Thus, the cracks seem to remain stationary: they probably appeared during the assembly on site, caused by any other effect different from fatigue stress. For the moment, no action is needed on these cracks. Another inspection in one year is recommended to confirm this hypothesis. In case an increase in length is detected on any of these cracks, different actions are recommended depending on the propagation rate (see Section **¡Error! No se encuentra el origen de la referencia.** for recommendations about crack reparation).

Other types of defects have been detected, as corrosion and lateral deformation (buckling).

It is recommended to repair the corroded areas by removing the oxide layer and applying protective primer and paint to the affected surface, in order to avoid further progress of the corroded zones, with the subsequent strength reduction. Also, the installation of some draining system for rain is recommended in those areas where water accumulation has been detected.

Regarding the lateral deformations, no actions seem to be needed, provided they do not originate collisions or interferences with other parts/systems.

7.2 MONITORING CAMPAIGN

Although the initial cracks detected have not progressed in the last year, inspections along time are still recommended to confirm that they are stationary cracks. The monitoring campaign can start with another inspection just before installing the upgraded system (in May 2015, before Phase II of on-site installation). This will permit two different evaluations:

- Detect if any crack has progressed in the last 1,5 years (since the last inspection in January 2015). Most probably, no crack growth will be detected.
- Register the initial state of defects to check if, after the upgrade of the system, the shutters operation in the entire opening range produces any potential change in the cracks. Again, no crack growth is expected, at least in the arch girders, as the system upgrade actions proposed are intended to reduce the point loads on the rails.

After that, another inspection in 6 months is recommended to detect any early propagation of cracks after the increase of the shutters operation range. For example, the lower shutter has never operated together with the windscreen so particularly the lower front part of the arch girders should be re-inspected. Idem with the lower rear part of the arch girders, where the upper shutter reaches after completing the travel range (90°).

If no propagation of the existing cracks or appearance of new ones is detected, the period until next inspection can be increased. Next table includes a summary of the proposed intervals between inspections.

Inspections	Interval since last inspection	Comments
1 (Initial - March, 2013)	-	Already performed
2 (January, 2014)	10 months	Already performed
3 (before start of Phase II on-site installation)	14 months (approx.)	To register initial state prior to shutters upgrade installation
4	6 months	To detect any modification caused by the increment of upper shutter operation range and lower shutter operation with windscreen.
5 and next ones	12 months	Increment of interval between inspection from 6 months to 1 year only if initial cracks remain stationary and no additional cracks appeared after last inspection

Fig. 7-2. Proposed intervals between inspections

7.3 CRACK REPARATION

For the moment, no crack propagation has been detected so all the cracks are apparently stationary. So no action is required for these cracks so far. However, if at some moment an increase of any crack length is detect as a result of the period monitoring, this section provides a proposal of actions to be taken.

The actions will depend on the velocity of propagation, so once any change in crack size is detected, the inspection should be more frequent to register the change in length along time. Attending to the velocity of propagation registered the following actions should be taken:

- If the cracks remain stationary, as stated before no action will be required. These cracks might have appeared during the assembly on site, caused by any other effect different from fatigue stress. They can be cataloged as stationary cracks.
- If the propagation rate is low, minor actions as cleaning/grinding or drill a crack arrest hole should be considered
- If the propagation rate is high, consideration should be made about the structural risks and the convenience of taking major actions as the introduction of reinforcements.

7.3.1 Minor Actions

7.3.1.1 Cleaning/Re-welding/Grinding

This action includes the following steps:

- Identification of the affected areas.
- Arc-gouging for the cracks located in a welding bead: the area of the weld around the flaw needs to be removed. The extension both in length and in depth to be removed will be, at least, as shown in next figure.

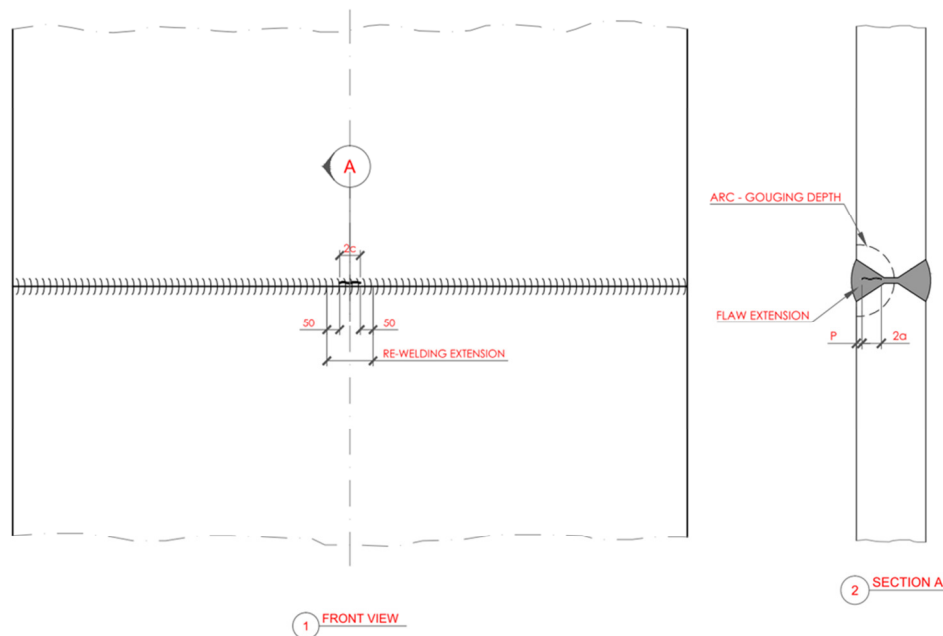


Fig. 7-3. Minimum re-welding extension

- Area preparation: the paint around the affected area will have to be removed by grinding and the metallic surface cleaned of any debris, chip or rest of oil from the grinding and arc-gouging operations.
- Welding: Prior to the welding operation, it shall be verified that the state of the cleaned surfaces is adequate to perform a correct welding operation. Any encountered problem shall be reported, analysed and, if possible, repaired before welding.
- Re-inspection: an inspection of the re-welded area will be done according. Should any of the repaired cracks show any kind of severity, the operation shall be repeated from the beginning, point a)
- Painting: the repaired area shall be re-painted. This includes cleaning and preparation of the surfaces to be painted.

7.3.1.2 Crack Arrest Hole Drilling

Another minor action that can be implemented is an arrest hole. The procedure simply consists of drilling a small hole at the end of the crack that has been advancing. This hole acts as a stopper to the crack propagation and usually works properly for low propagation speeds.

7.3.2 Major Actions

For critical cracks the minor actions previously explained might not achieve the complete stop of the crack progress. In this case, other actions as the introduction of **reinforcing ribs** should be considered. The function of the ribs would be to provide an alternative path to the loads, thus relaxing the acting stresses on the crack location and stopping its propagation.

This repair includes the following steps.

- Identification of the affected areas.
- Ribs installation area preparation for welding: the surfaces where the ribs are to be welded will have to get the paint removed, the metallic surface cleaned of any debris, chip coming from the paint removal.
- Ribs cleaning and positioning: the ribs will be cleaned from any dirt or rust that may have accumulated. Special care will be taken in the edges to be welded. The ribs will be positioned within a tolerance of $\pm 2\text{mm}$ from their theoretical position. It is recommended to use an especially designed positioning tool to facilitate this operation. Also, special care will be taken in the relative positioning of the internal and external ribs. Next figure can be used as reference for the ribs design (a customized solution should be developed for the specific crack typology and location):

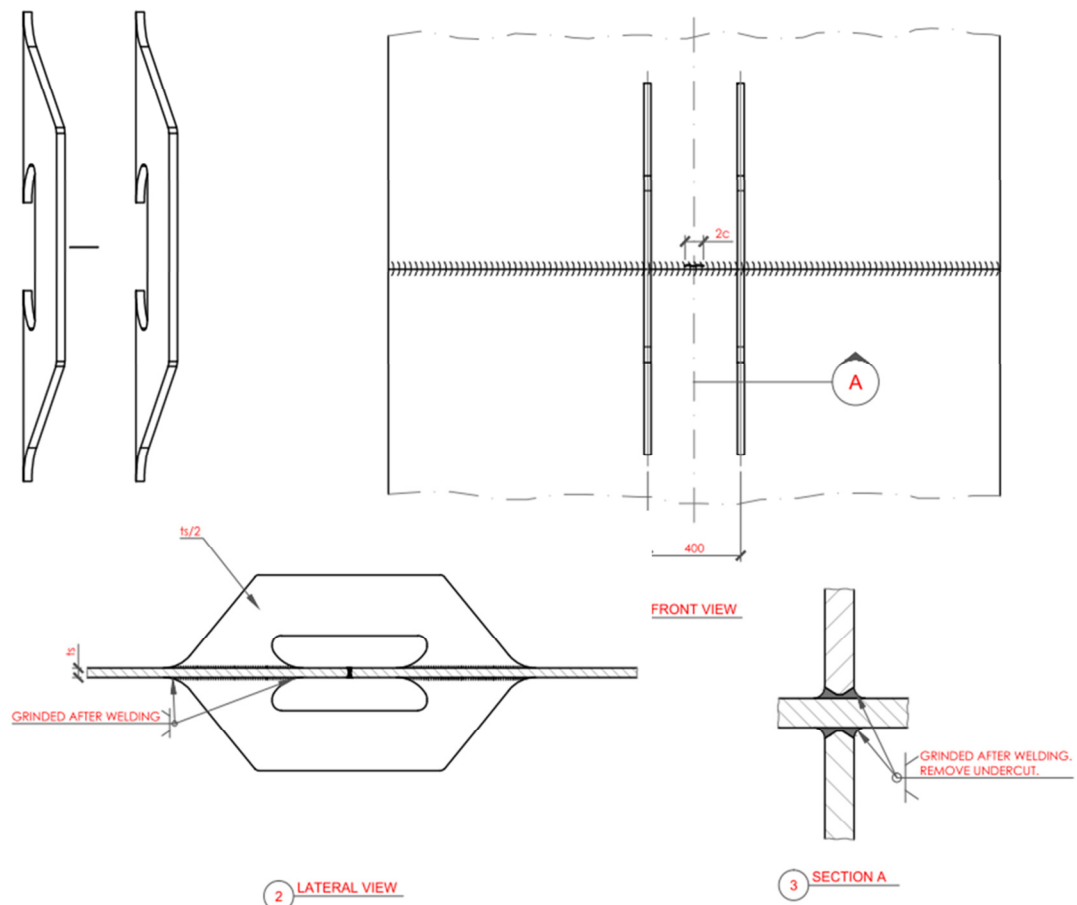


Fig. 7-4. Example of ribs reinforcement for reparation of a weld affected by a crack

- d) Ribs welding. Once the ribs are correctly positioned and secured, the welds will be implemented. The resulting weld shall comply, after grinding, with a detail category 80 acc. to EN 1993-1-9.

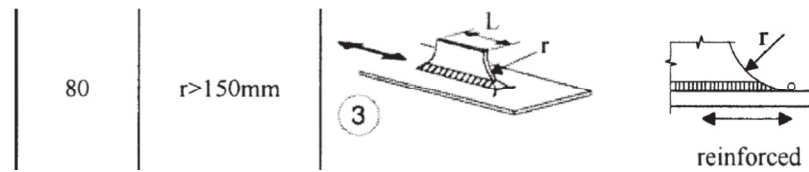
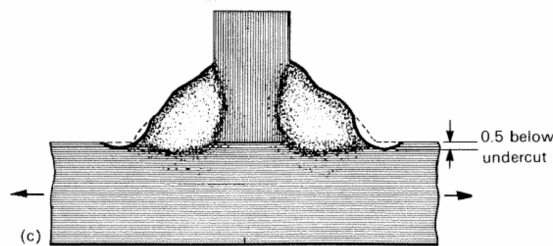


Fig. 7-5 Weld category 80 according to EC-3 Part 1-9 (EN 1993-1-9)

- e) Welds grinding: the welds will be grinded smoothly according to what is stated in the aforementioned detail category. Special care will be taken to remove the undercut.



- f) Welds inspection. Welds will be inspected after grinding, in order to meet the detail category 80.
- g) Painting: the repaired area shall be re-painted. This includes a prior cleaning and preparation of the surfaces.

As an alternative to the rib reinforcement, reparation based on bonded composites (CFRP patches) could be evaluated. This method is used in the repair of ships structures.

8 APPENDIX A: SUPPORT ANALYSES AND CALCULATIONS

8.1 INTRODUCTION

This appendix summarizes the preliminary stress and strength analyses carried out to verify the new support system of the shutters.

In the next sections, the following calculations are presented:

1. Upper Shutter structural analysis in order to verify the support strategy.
2. Structural analysis of the new bogie
 - a. Extreme load analysis
 - b. Fatigue loads

8.2 UPPER SHUTTER SUPPORT STRATEGY

The structural analysis to check the reactions and the displacements of the upper shutter bogies has been done by means of a finite element model (FEM). The load cases include gravity loads at different positions of the upper shutter, i.e. 0, 24, 31.1, 48, 69 and 90 degrees.

To come up with the best strategy, two different configurations have been analysed. The first one, consisting of two hydraulic systems at each side of the upper shutter and the second one with one more group at each side. Moreover, there has been studied both the case in which the tilt of 7° of the chains has been taken into account and the case where one of the chains of one of the sides (B) of the upper shutter is not working, being the forces transferred to the other side (A).

8.2.1 Two hydraulic systems with a chain tilt of 7°

The following figure depicts the FE model of the upper shutter including the four whiffle tree groups which have been modelled with beam and truss elements to simulate the load equalization between the bogies within each hydraulic system.

The upper shutters are fixed in the axial and tangential direction while the whiffle trees are pinned to the fixed frame.

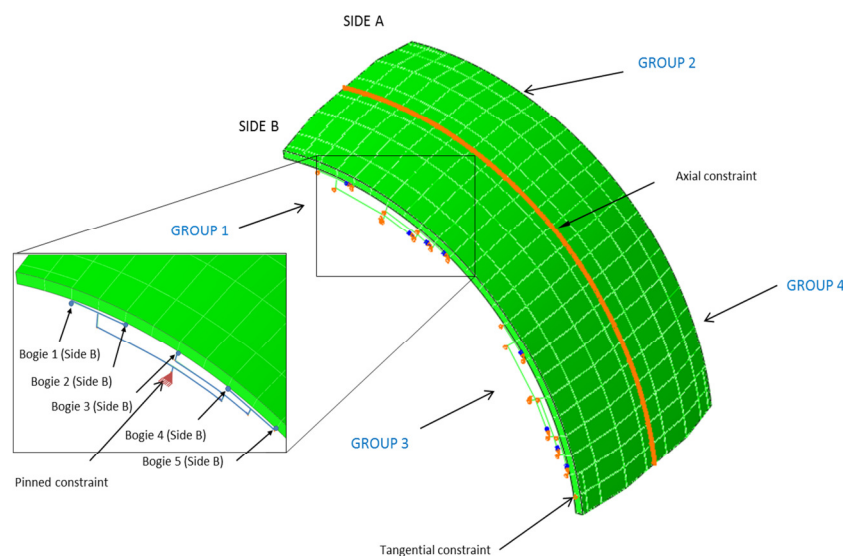


Fig. 8-1 Upper shutter and whiffle tree FE model

For this analysis, the upper shutter FE model provided by GRANTECAN has been used. The model consists of 388224 shells elements and 28388 beam/truss elements.

The shutter structure deformations at 0, 31.1 and 90 degrees are shown in next figures.

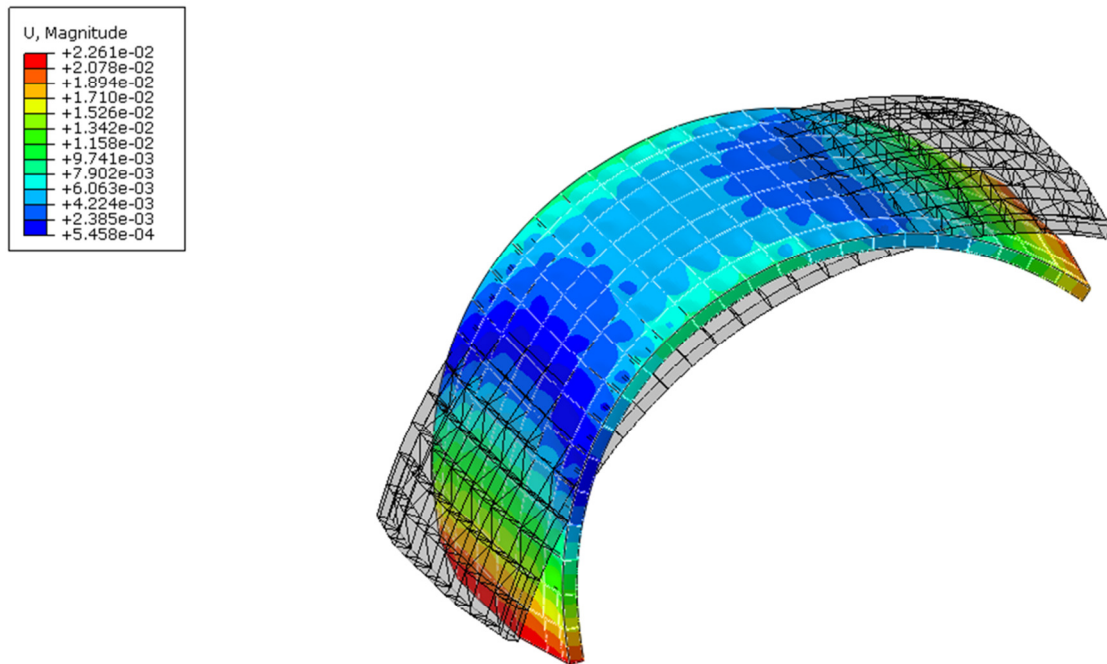


Fig. 8-2 Upper shutter and deformation at 0 degrees

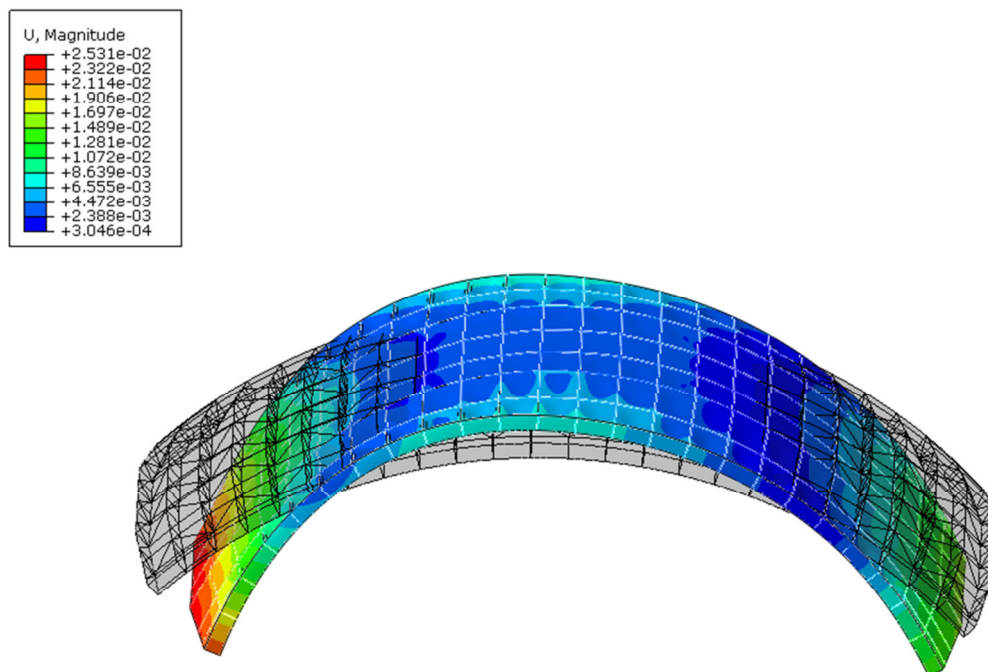


Fig. 8-3 Upper shutter and deformation at 31.1 degrees

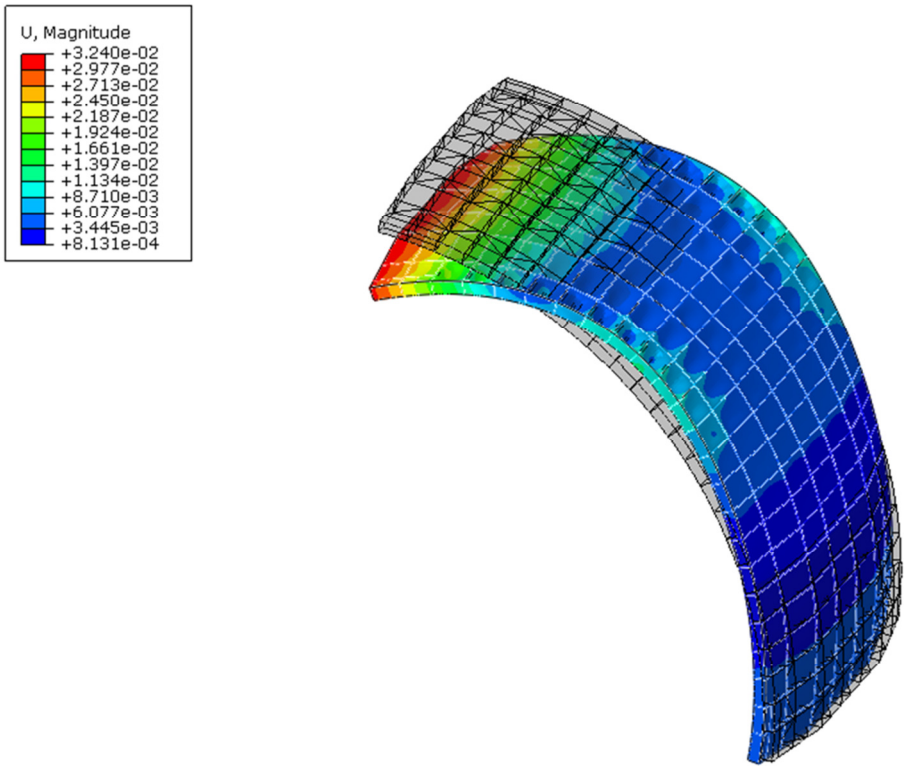


Fig. 8-4 Upper shutter and deformation at 90 degrees

The following figure shows the reactions in the side A and side B bogies.

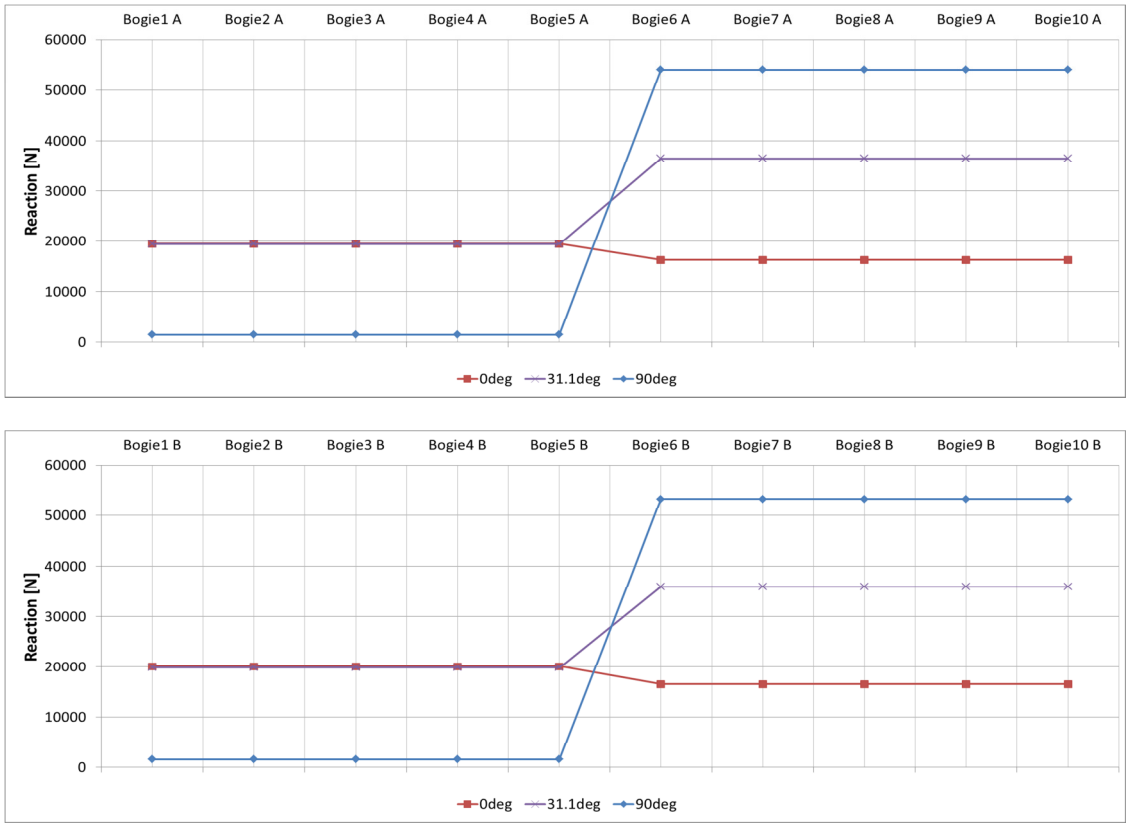


Fig. 8-5 Reaction forces in side A and B bogies

The maximum reaction is 54091 N in the bogies 6, 7, 8, 9 and 10 located in the side A for 90 degrees shutter opening.

Furthermore, the FE allows checking that the cylinders have enough stroke to compensate for the radial displacements of the structure due to the accommodation to the whiffle tree equalization, combined with the radial deviation of the rails. The radial displacements induced by the gravity loads equalized at each bogie of the same group at three different shutter orientations (corresponding to 0, 31.1 and 90 degrees) have been obtained. Then, they have been added to the radial deviations of the rails affecting each bogie.

The results are summarized in the following figures.

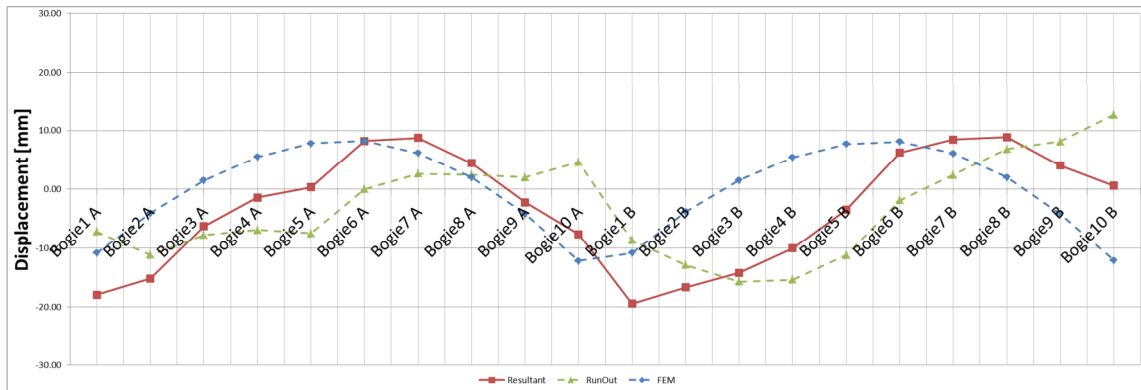


Fig. 8-6 FEM, rail deviation and resultant displacements at 0 degrees

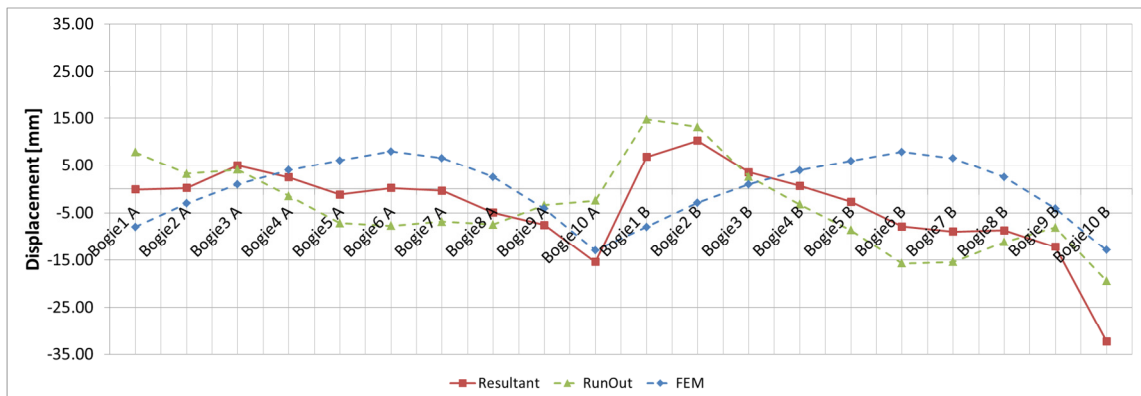


Fig. 8-7 FEM, rail deviation and resultant displacements at 31.1 degrees

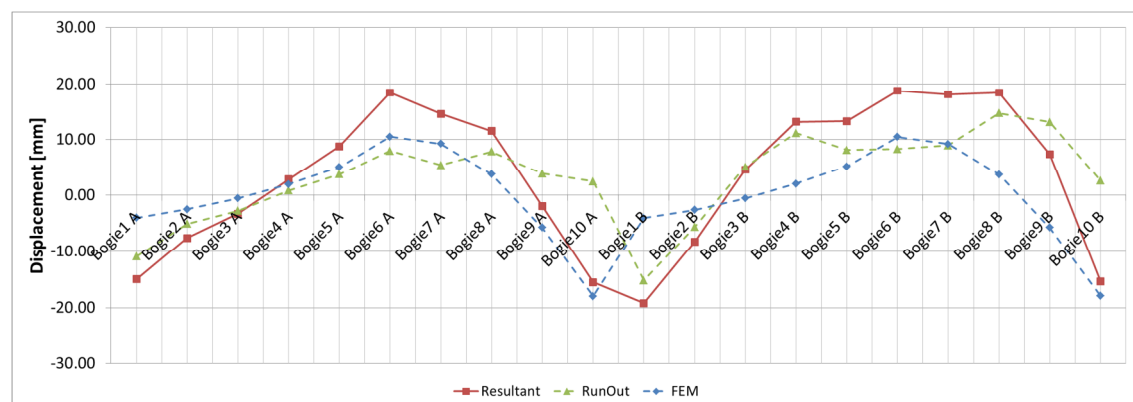


Fig. 8-8 FEM, rail deviation and resultant displacements at 90 degrees

The maximum resultant radial displacement is 32.28 mm, measured in shutter open position (31.1 degrees) at the lowest bogie of Side B.

8.2.1.1 Two hydraulic systems with a chain tilt of 7° and chain B not working

The following figure depicts the FE model of the upper shutter including the four whiffle tree groups which have been modelled with beam and truss elements to simulate the load equalization between the bogies within each hydraulic system.

The upper shutters are fixed in the axial and tangential direction while the whiffle trees are pinned to the fixed frame.

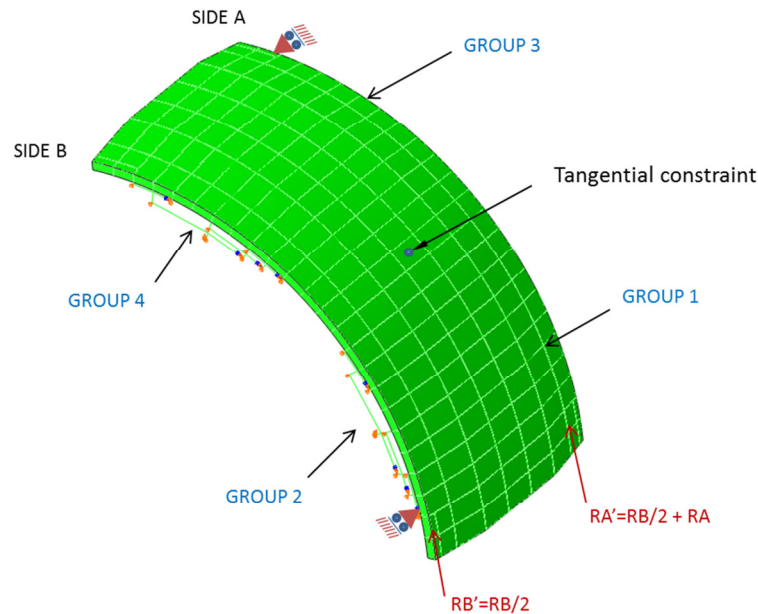


Fig. 8-9 Upper shutter and whiffle tree FE model

For this analysis, the upper shutter FE model provided by GRANTECAN has been used. The model consists of 388000 shells elements and 28400 beam/truss elements.

The shutter structure deformations at 0, 31.1 and 90 degrees are shown in next figures.

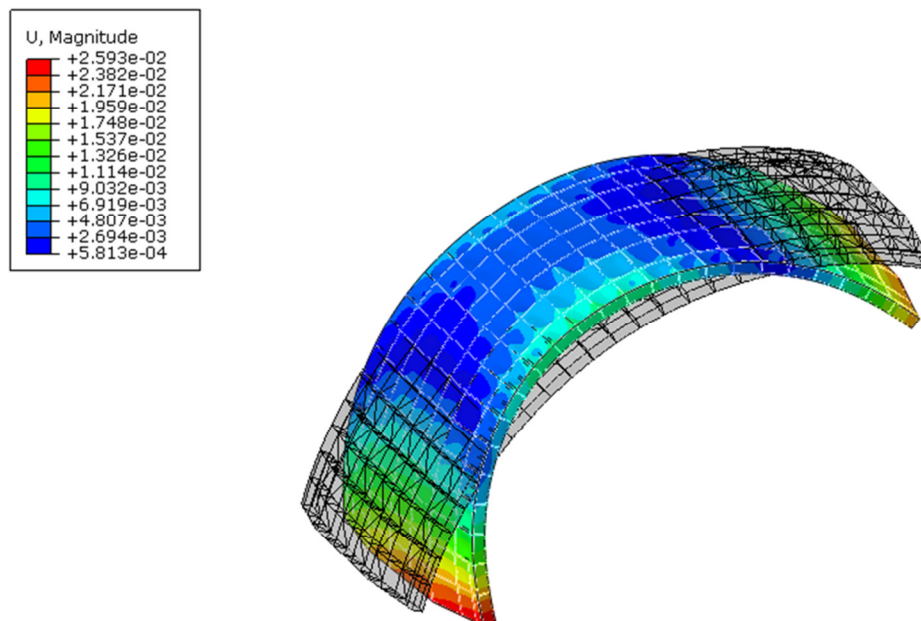


Fig. 8-10 Upper shutter and deformation at 0 degrees

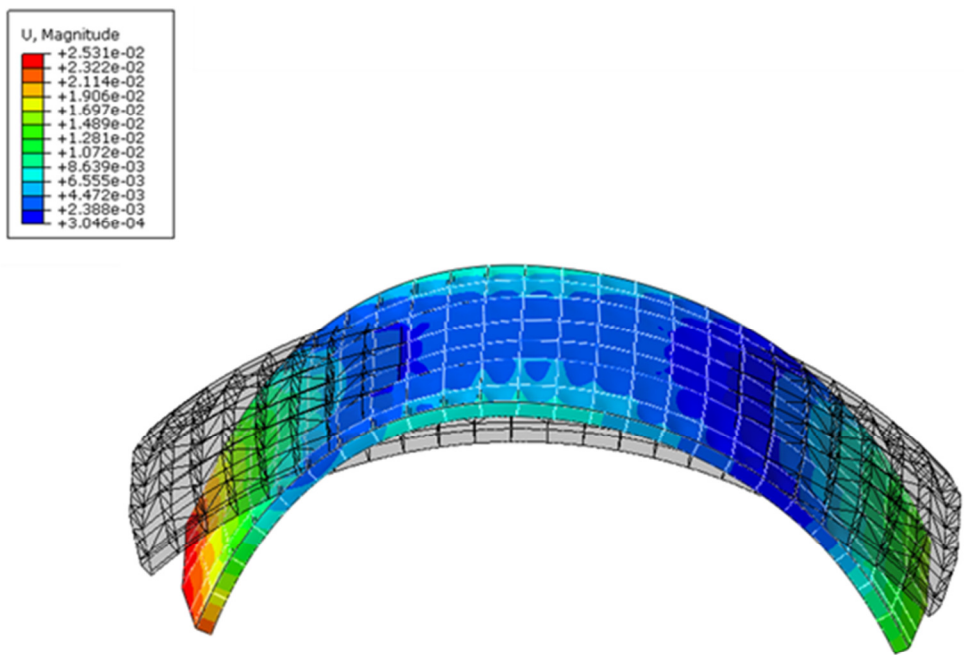


Fig. 8-11 Upper shutter and deformation at 31.1 degrees

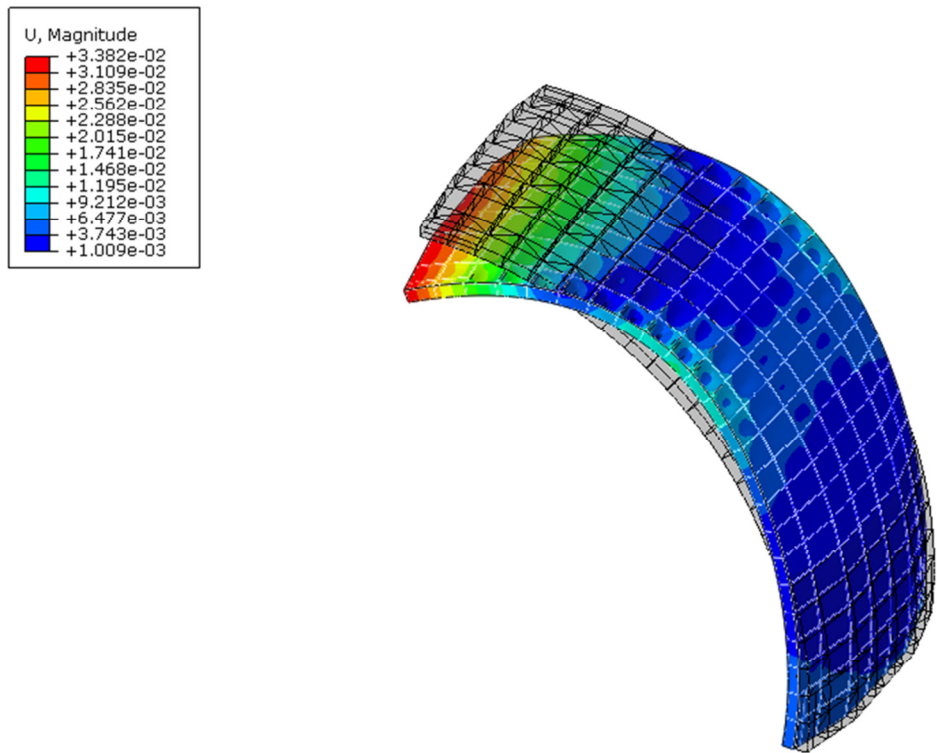


Fig. 8-12 Upper shutter and deformation at 90 degrees

The following figure shows the reactions in the side A and side B bogies

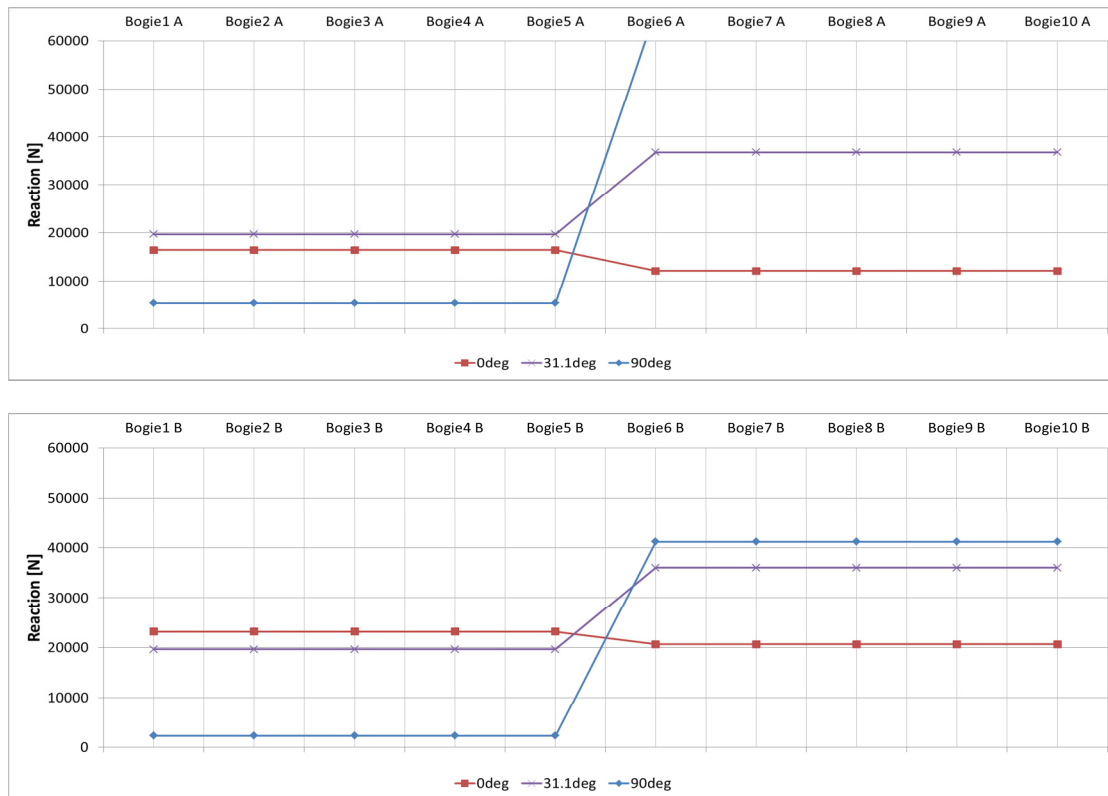


Fig. 8-13 Reaction forces in side A and B bogies

The maximum reaction is 66209.8 N in the bogies 6, 7, 8, 9 and 10 located in the side A for 90 degrees shutter opening.

Furthermore, the FE allows checking that the cylinders have enough stroke to compensate for the radial displacements of the structure due to the accommodation to the whiffle tree equalization, combined with the radial deviation of the rails. The radial displacements induced by the gravity loads equalized at each bogie of the same group at three different shutter orientations (corresponding to 0, 31.1 and 90 degrees) have been obtained. Then, they have been added to the radial deviations of the rails affecting each bogie.

The results are summarized in the following figures.

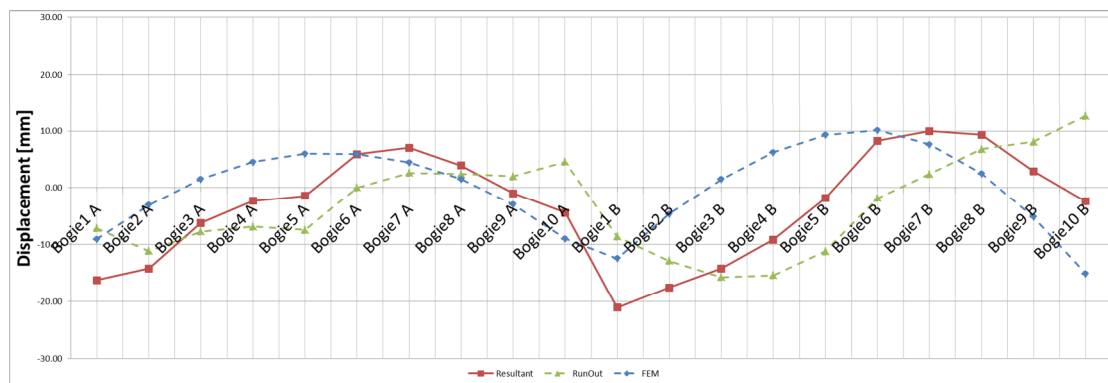


Fig. 8-14 FEM, rail deviation and resultant displacements at 0 degrees

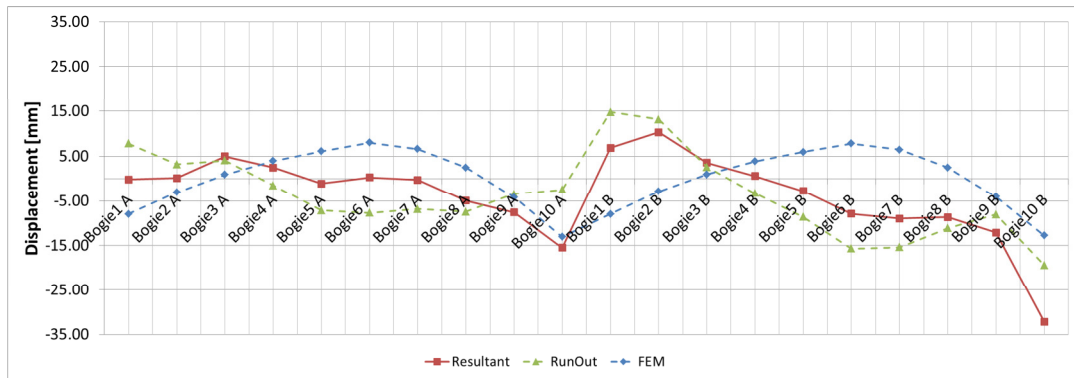


Fig. 8-15 FEM, rail deviation and resultant displacements at 31.1 degrees

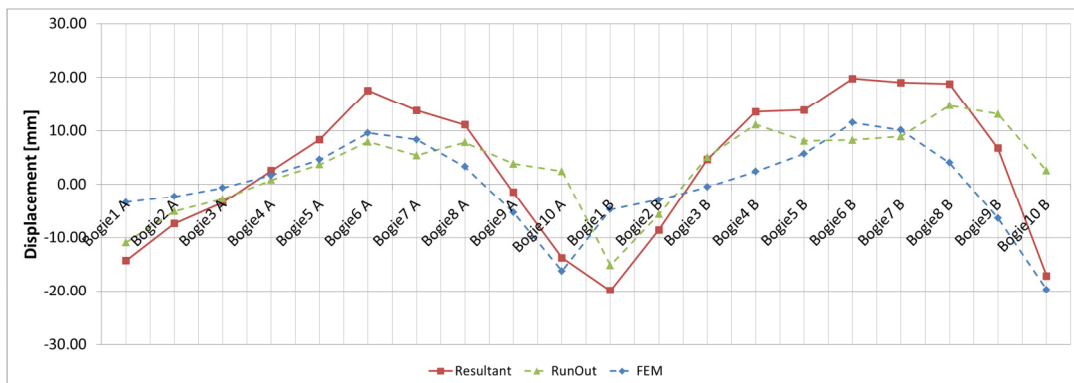


Fig. 8-16 FEM, rail deviation and resultant displacements at 90 degrees

The maximum resultant radial displacement is 32.20 mm, measured in shutter open position (31.1 degrees) at the upper bogie of Side B.

8.2.2 Three hydraulic systems with a chain tilt of 7°

The following figure depicts the FE model of the upper shutter including the six whiffle tree groups which have been modelled with beam and truss elements to simulate the load equalization between the bogies within each hydraulic system.

The upper shutters are fixed in the axial and tangential direction while the whiffle trees are pinned to the fixed frame.

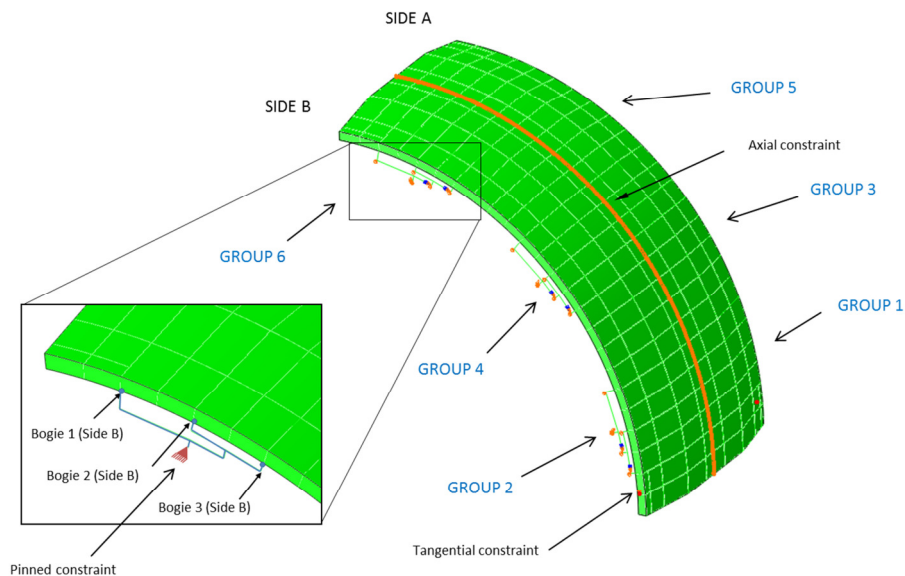


Fig. 8-97 Upper shutter and whiffle tree FE model

For this analysis, the upper shutter FE model provided by GRANTECAN has been used. The model consists of 388224 shells elements and 28374 beam/truss elements.

The shutter structure deformations at 0, 31.1 and 90 degrees are shown in next figures.

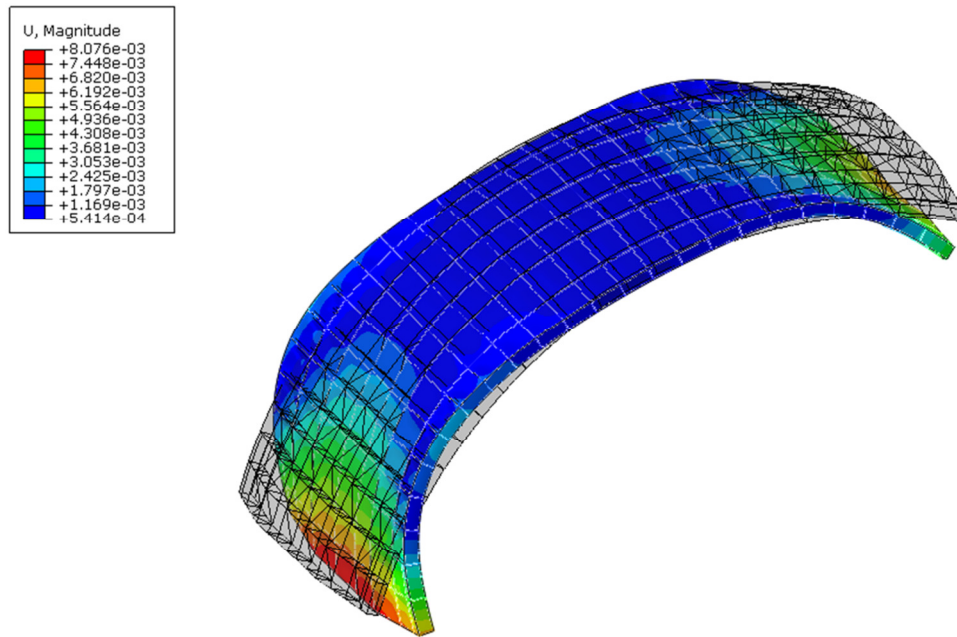


Fig. 8-18 Upper shutter and deformation at 0 degrees

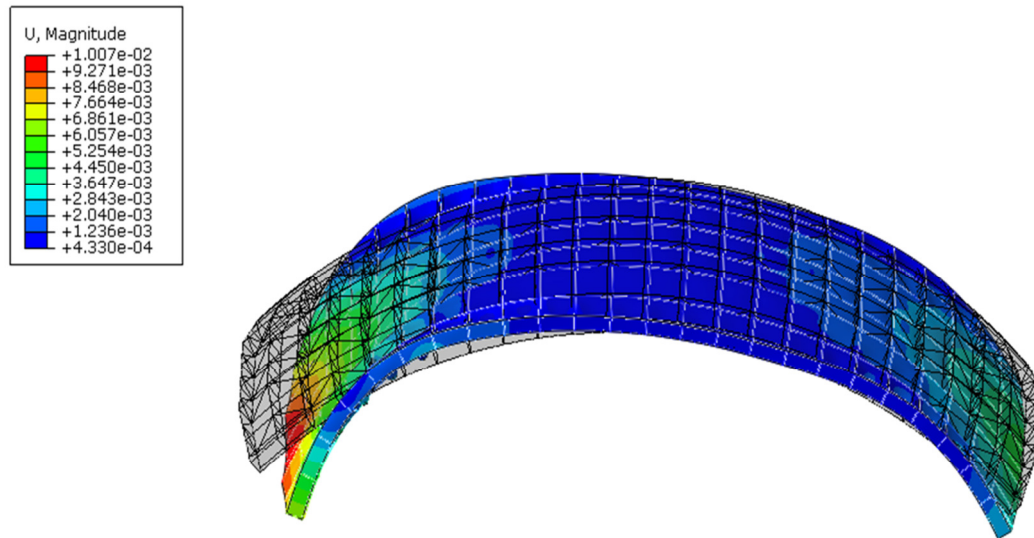


Fig. 8-19 Upper shutter and deformation at 31.1 degrees

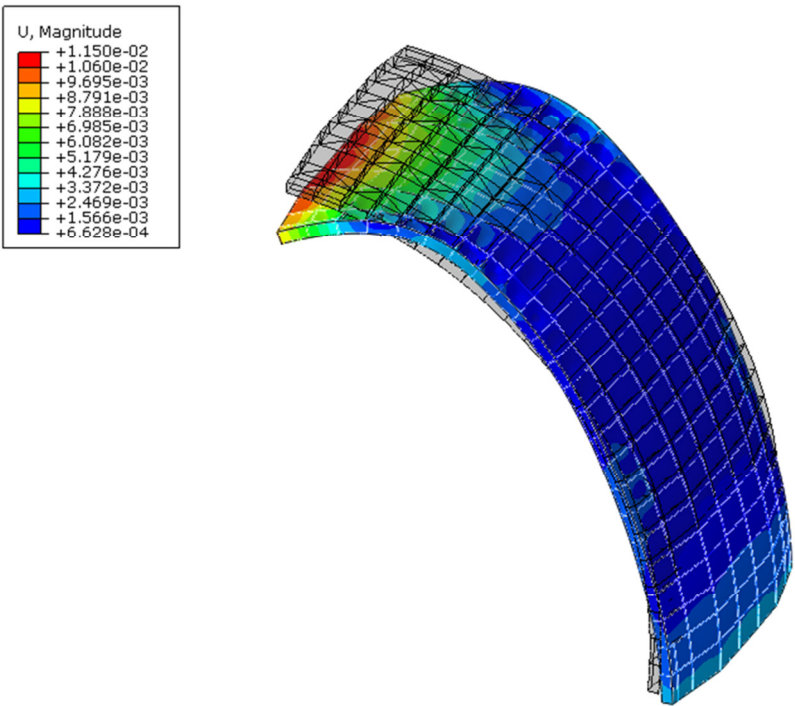


Fig. 8-20 Upper shutter and deformation at 90 degrees

The following figure shows the reactions in the side A and side B bogies.

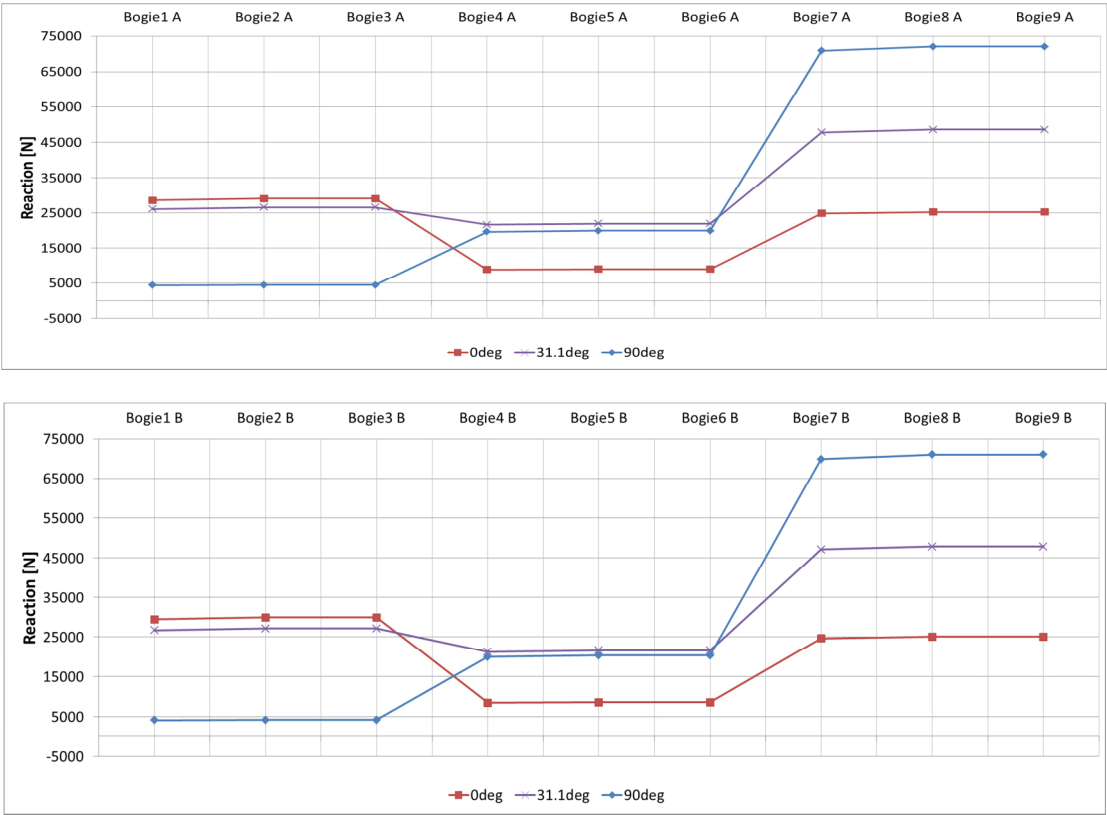


Fig. 8-21 Reaction forces in side A and B bogies

The maximum reaction is 72000 N in the bogies 7, 8 and 9 located in the side A for 90 degrees shutter opening.

Furthermore, the FE allows checking that the cylinders have enough stroke to compensate for the radial displacements of the structure due to the accommodation to the whiffle tree equalization, combined with the radial deviation of the rails. The radial displacements induced by the gravity loads equalized at each bogie of the same group at three different shutter orientations (corresponding to 0, 31.1 and 90 degrees) have been obtained. Then, they have been added to the radial deviations of the rails affecting each bogie.

The results are summarized in the following figures.

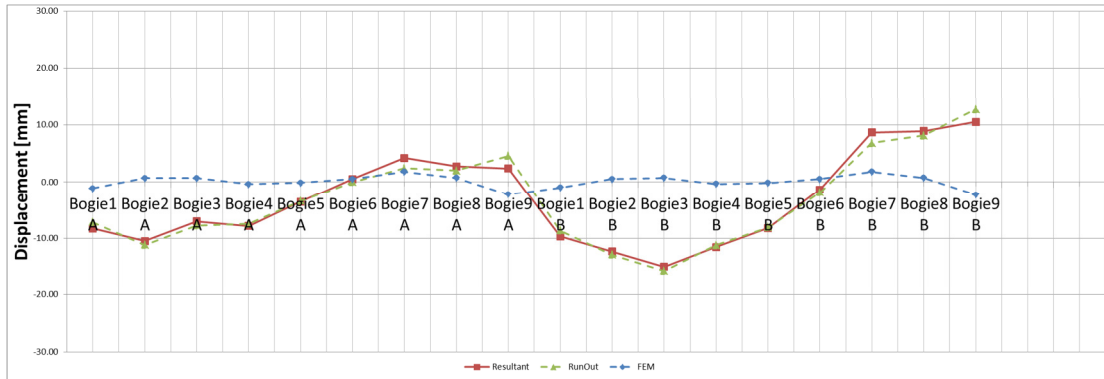


Fig. 8-22 FEM, rail deviation and resultant displacements at 0 degrees

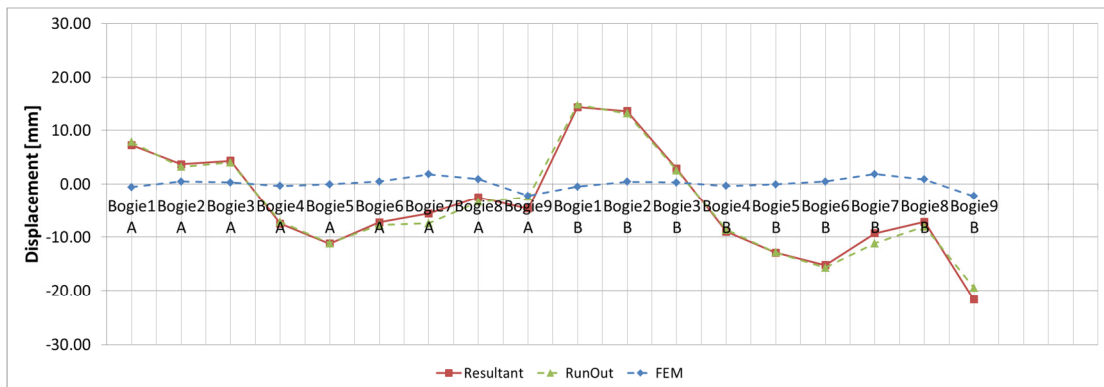


Fig. 8-23 FEM, rail deviation and resultant displacements at 31.1 degrees

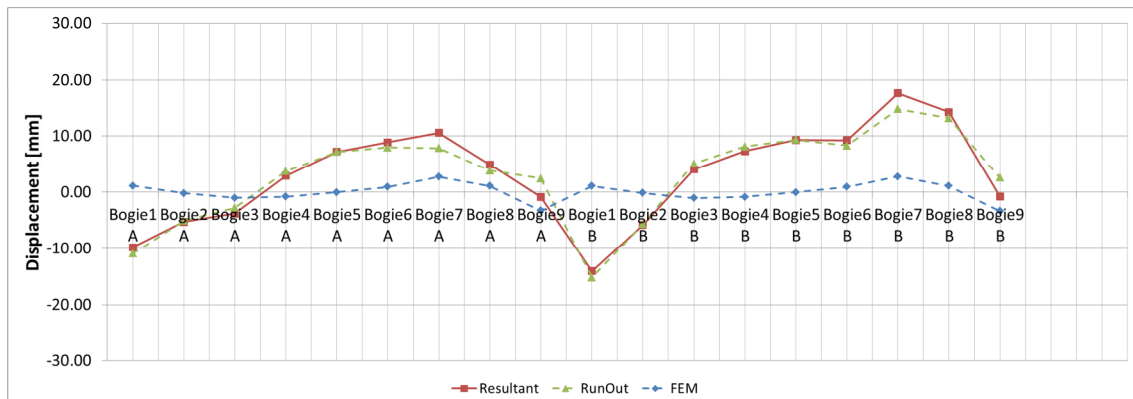


Fig. 8-24 FEM, rail deviation and resultant displacements at 90 degrees

The maximum resultant radial displacement is 21.68 mm, measured in shutter open position (31.1 degrees) at the lowest bogie of Side B.

8.2.2.1 Three hydraulic systems with a chain tilt of 7° and chain B not working

The following figure depicts the FE model of the upper shutter including the six whiffle tree groups which have been modelled with beam and truss elements to simulate the load equalization between the bogies within each hydraulic system.

The upper shutters are fixed in the axial and tangential direction while the whiffle trees are pinned to the fixed frame.

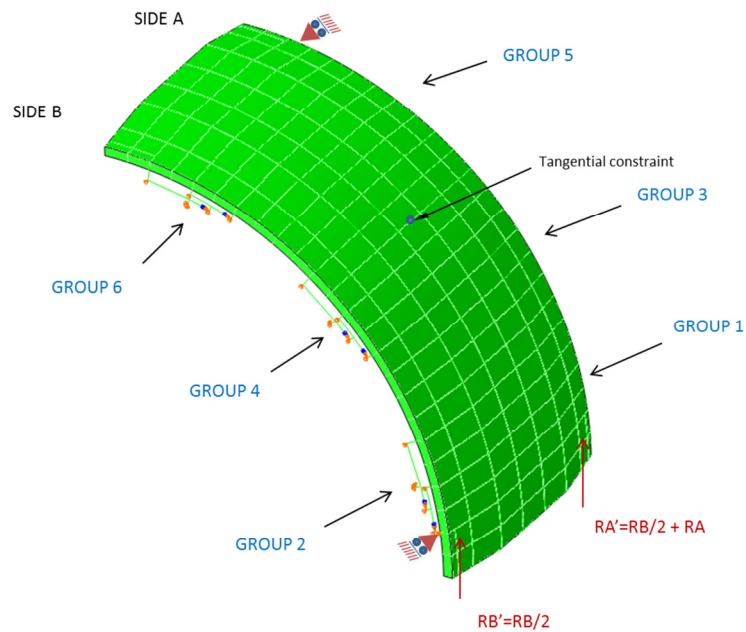


Fig. 8-25 Upper shutter and whiffle tree FE model

For this analysis, the upper shutter FE model provided by GRANTECAN has been used. The model consists of 388224 shells elements and 28374 beam/truss elements.

The shutter structure deformations at 0, 31.1 and 90 degrees are shown in next figures.

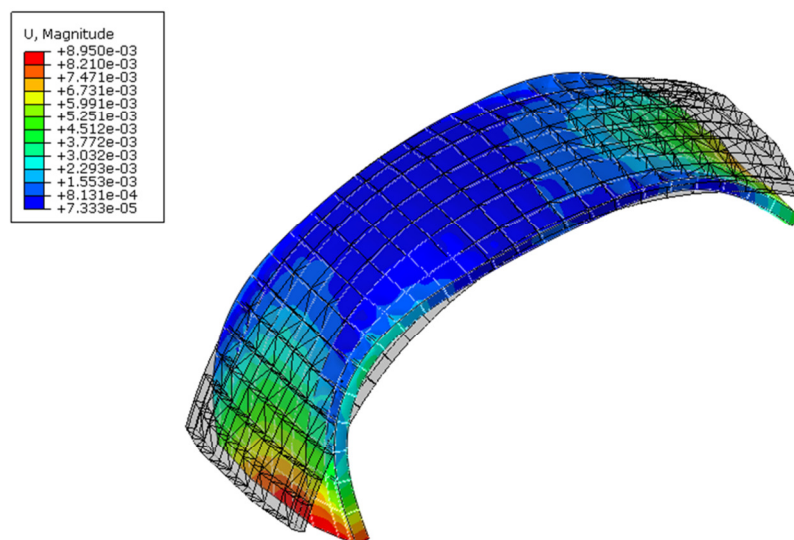


Fig. 8-26 Upper shutter and deformation at 0 degrees

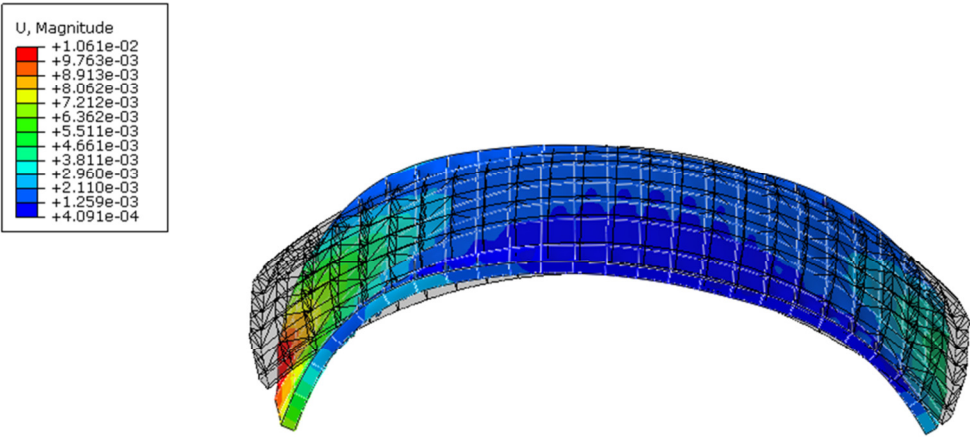


Fig. 8-27 Upper shutter and deformation at 31.1 degrees

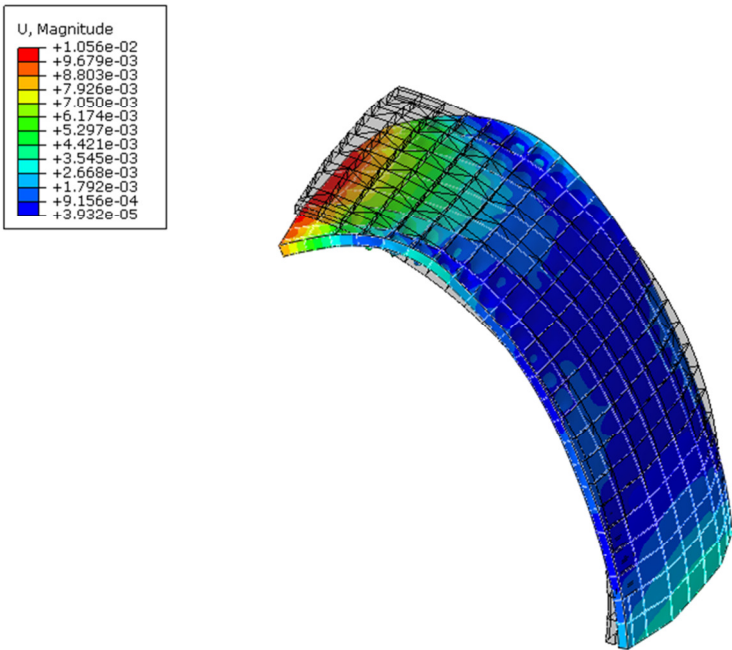
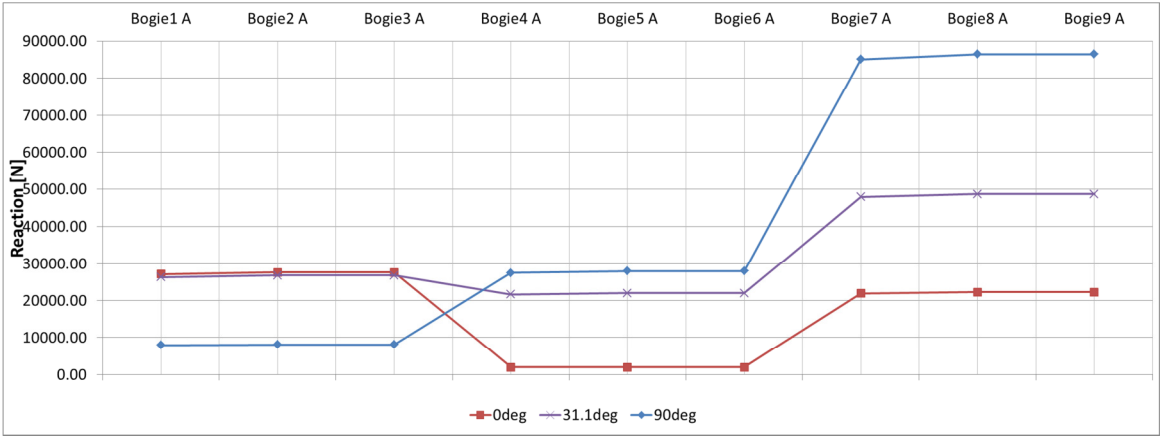


Fig. 8-28 Upper shutter and deformation at 90 degrees

The following figure shows the reactions in the side A and side B bogies.



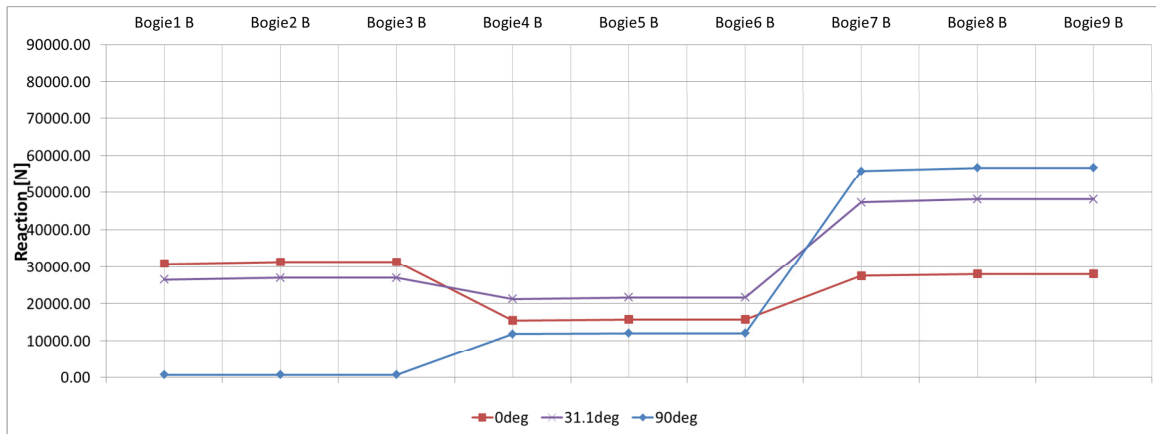


Fig. 8-29 Reaction forces in side A and B bogies

The maximum reaction is 86000 N in the bogies 7, 8 and 9 located in the side A for 90 degrees shutter opening.

Furthermore, the FE allows checking that the cylinders have enough stroke to compensate for the radial displacements of the structure due to the accommodation to the whiffle tree equalization, combined with the radial deviation of the rails. The radial displacements induced by the gravity loads equalized at each bogie of the same group at three different shutter orientations (corresponding to 0, 31.1 and 90 degrees) have been obtained. Then, they have been added to the radial deviations of the rails affecting each bogie.

The results are summarized in the following figures.

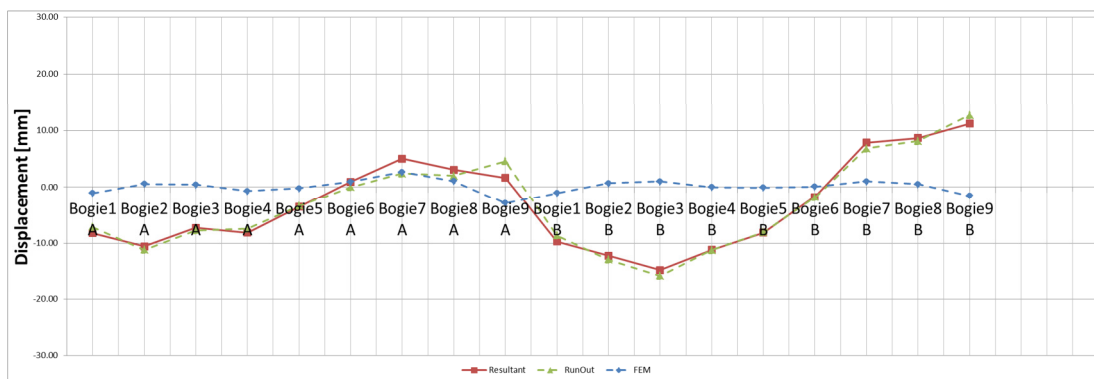


Fig. 8-30 FEM, rail deviation and resultant displacements at 0 degrees

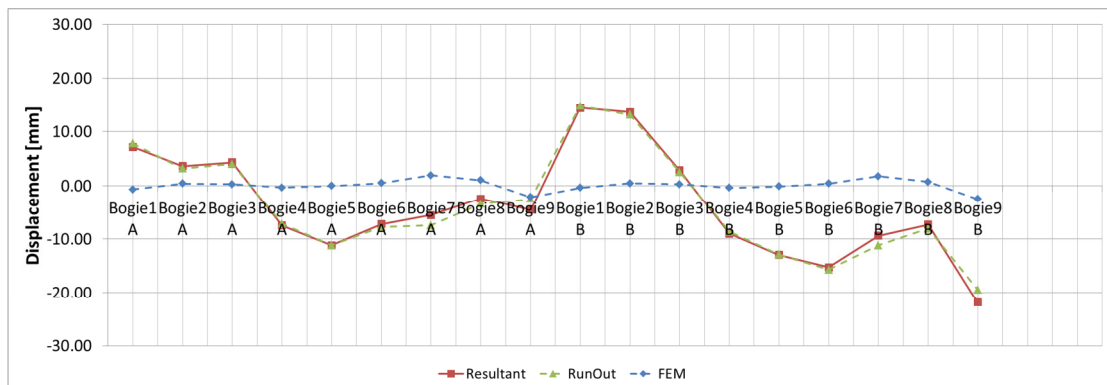


Fig. 8-31 FEM, rail deviation and resultant displacements at 31.1 degrees

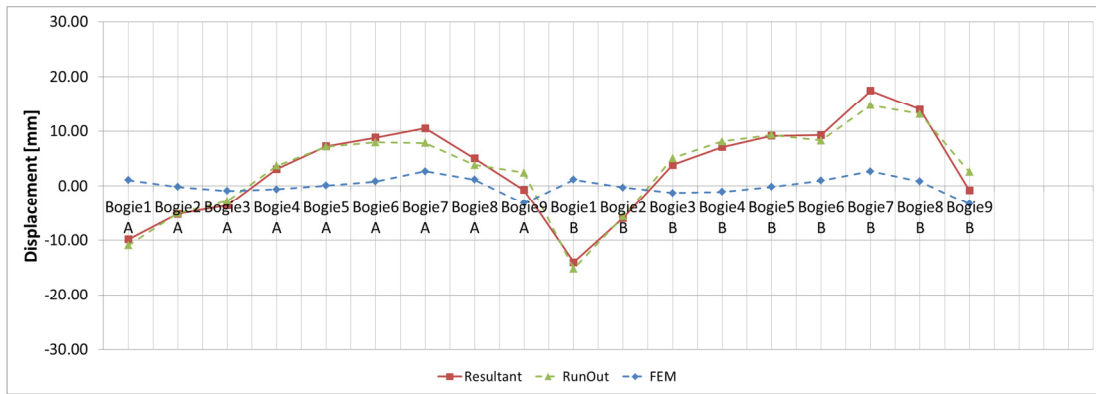


Fig. 8-32 FEM, rail deviation and resultant displacements at 90 degrees

The maximum resultant radial displacement is 21.85mm, measured in shutter open position (31.1 degrees) at the lowest bogie of Side B.

8.2.2.2 Three hydraulic systems support: “1” hydraulic system failure cases study

After the Final Meeting celebrated on 6th March, in La Laguna, GRANTECAN requested to study the scenario of failure of the hydraulic system closest to the Fix Point. The aim of this study is to measure the maximum displacements at the shutter in order to be able to check any unpredicted contact with the surrounding interface of the shutter.

The model used in the previous analysis has been modified by replacing the “whiffletree 1” by GAP elements representing the failure of the hydraulic supporting system. These elements have been set so a reaction force will appear when the shutter displacement at the bogie position is 20mm. Physically, this happens upon making contact the rod of the hydraulic cylinder with the base of the cylinder.

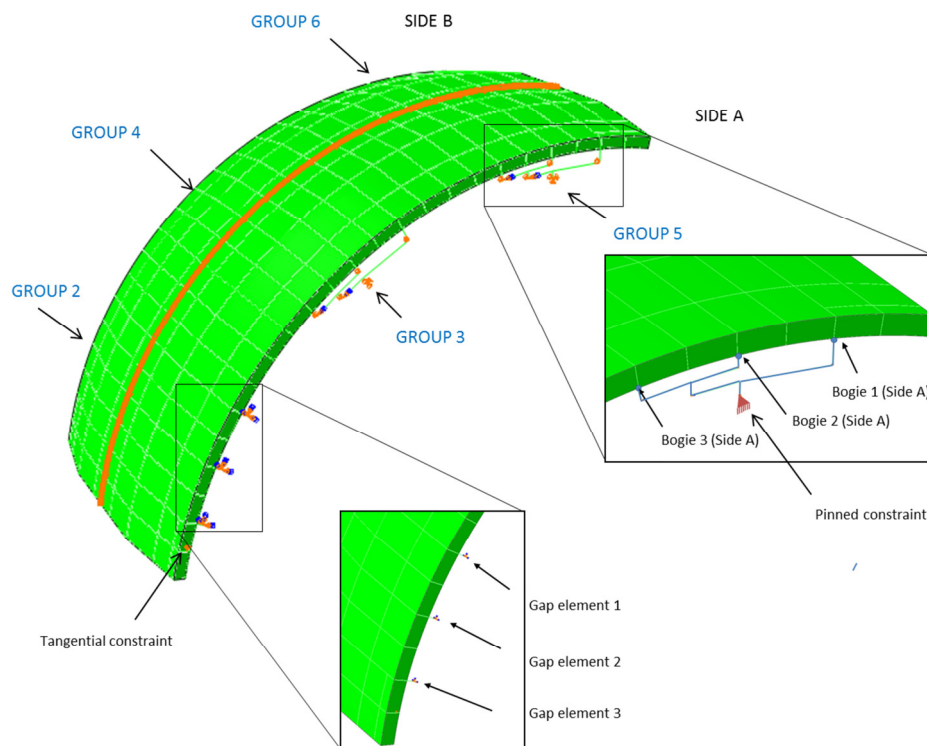


Fig. 8-32 Upper shutter FE model with “hydraulic system 1” failure

The structural analysis to check the displacements of the upper shutter has been done at different positions of the upper shutter, i.e. 0, 24, 31.1, 48, 69 and 90 degrees, being the results shown in the following figures (displacements shown in metres).

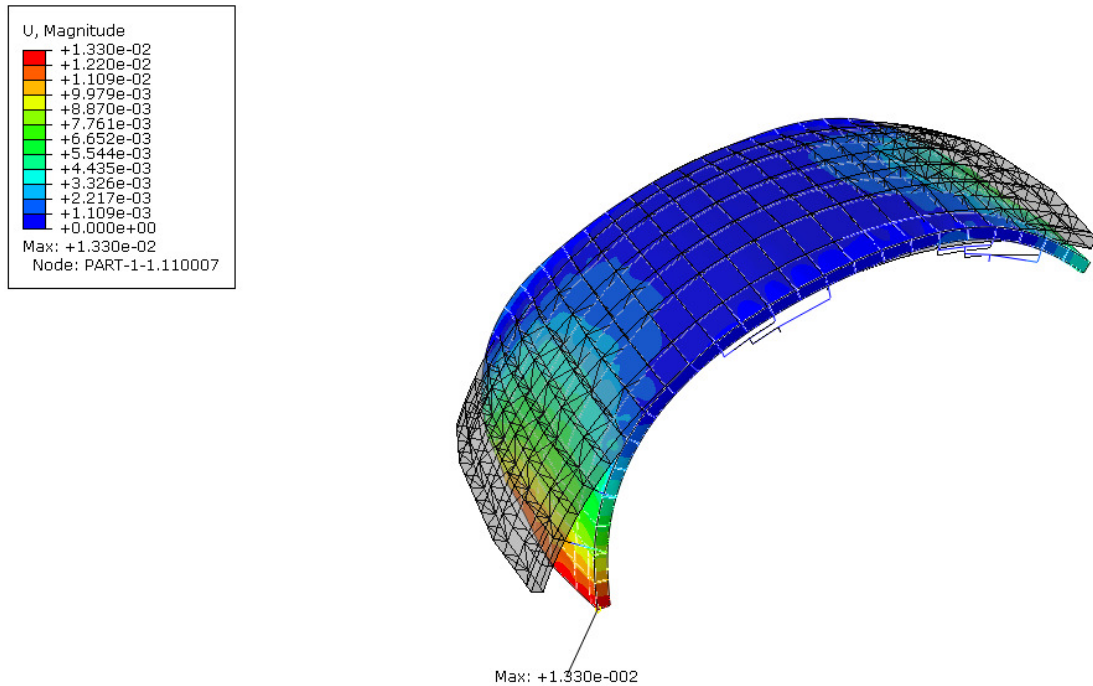


Fig. 8-27 Upper shutter and deformation at 0 degrees

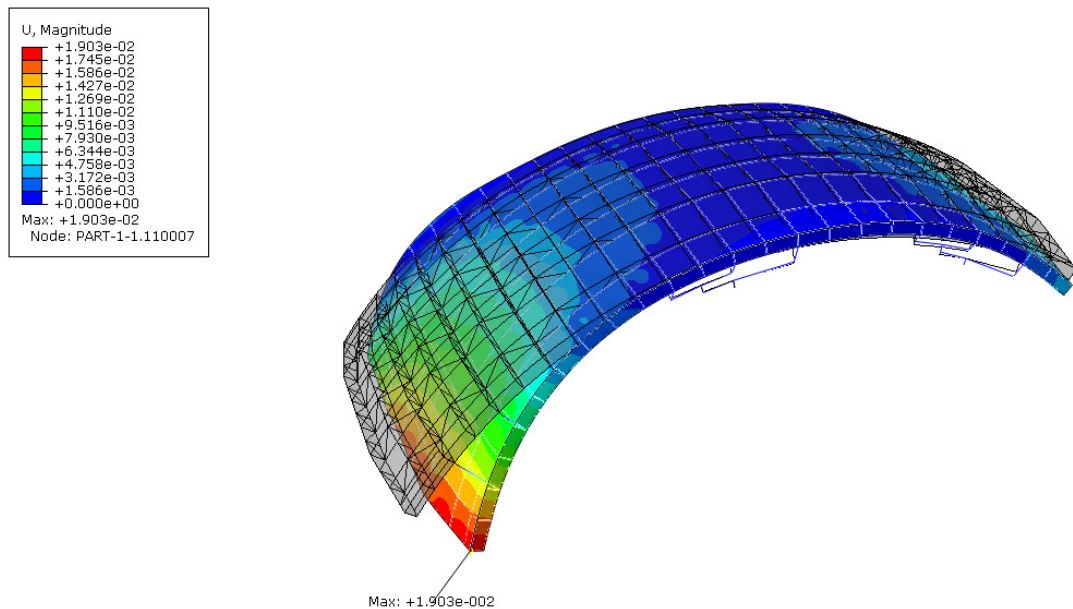


Fig. 8-27 Upper shutter and deformation at 24 degrees

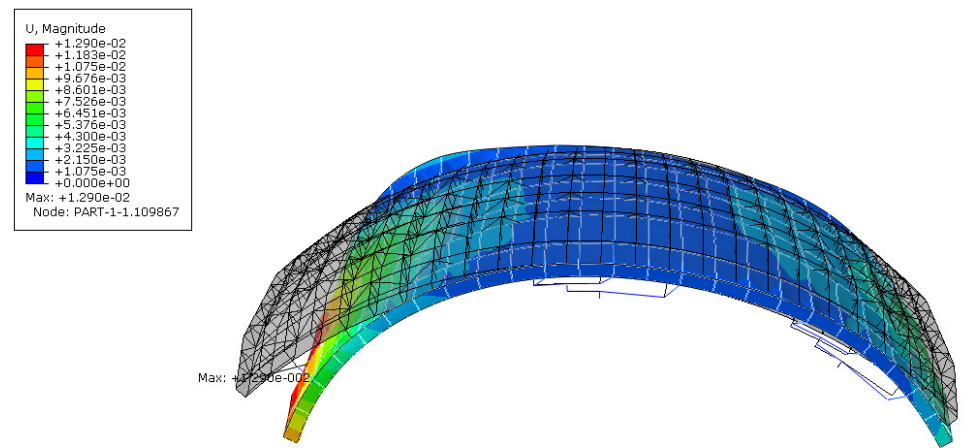


Fig. 8-27 Upper shutter and deformation at 31.1 degrees

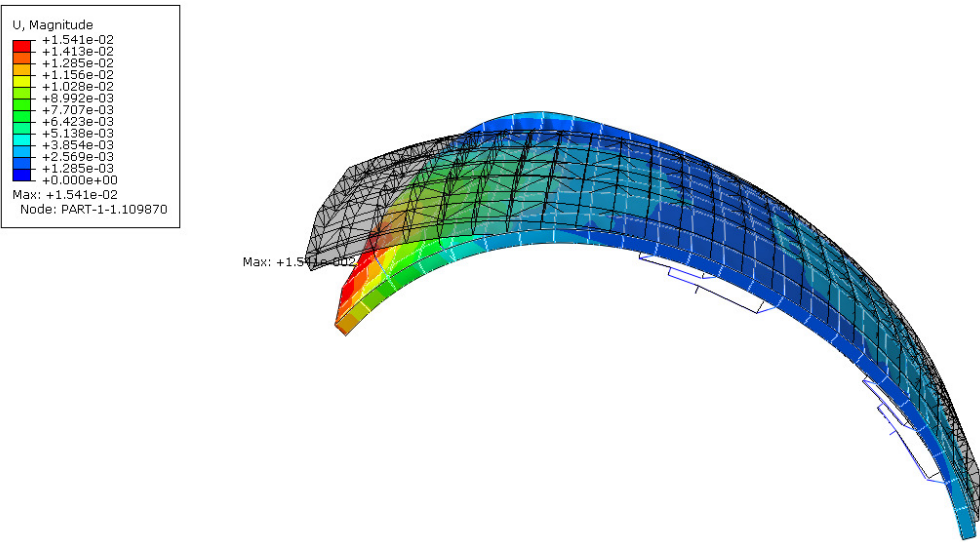


Fig. 8-27 Upper shutter and deformation at 48 degrees

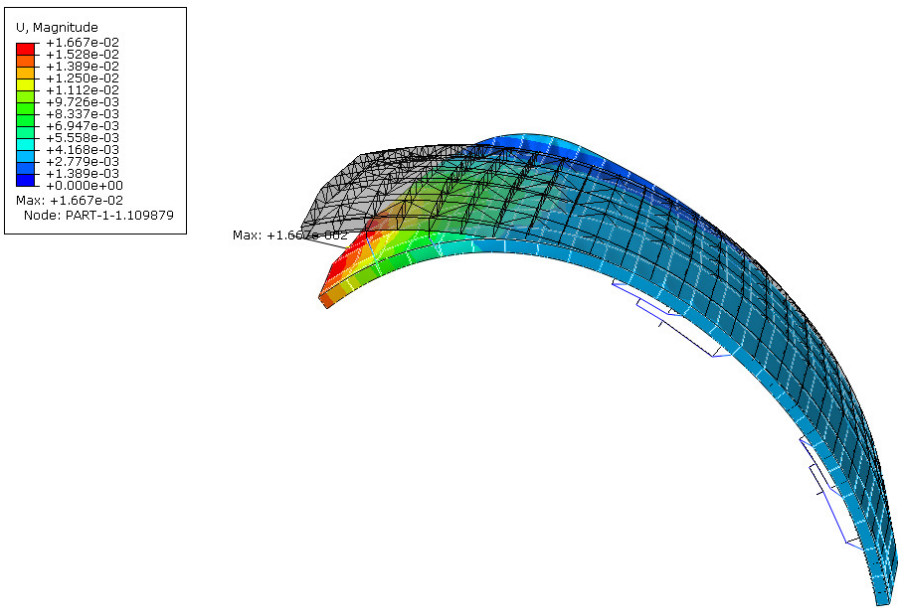


Fig. 8-27 Upper shutter and deformation at 69 degrees

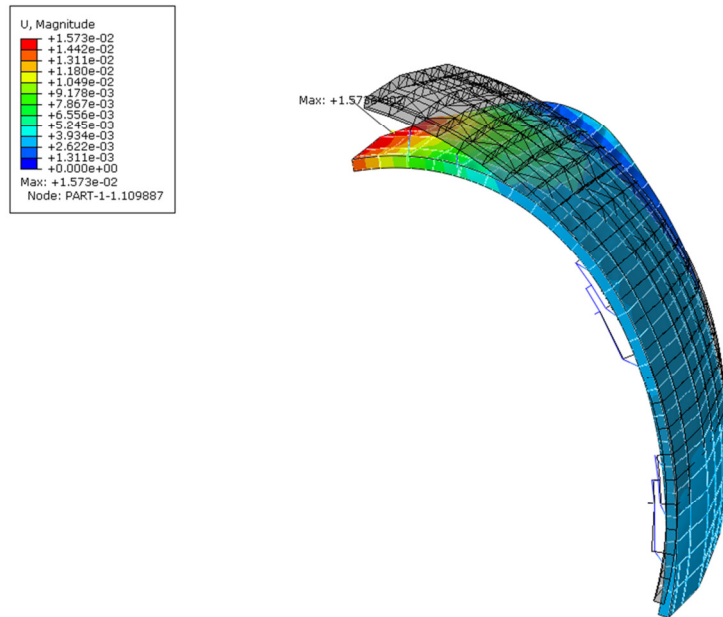


Fig. 8-27 Upper shutter and deformation at 90 degrees

The maximum displacement is 19.03mm, measured at shutter position of 24°.

The following table shows the displacements at the bogie locations measured for the different load cases. Bogie 1 represents the one at the more extreme location.

	0°	24°	31.1°	48°	69°	90°
BOGIE 1	-8.42071	-13.1826	-6.20599	-7.9622	-9.17447	-9.16799
BOGIE 2	-4.27357	-7.91623	-3.22644	-4.21807	-4.9358	-4.99785
BOGIE 3	-1.51012	-3.90153	-1.41401	-1.81529	-2.09276	-2.09223

Fig. 8-27 Table of displacement values at bogie position

8.3 BOGIE (MAIN STRUCTURE AND SECONDARY STRUCTURE)

8.3.1 Input loads

The considered input loads introduced in the FE model are represented in the following figure

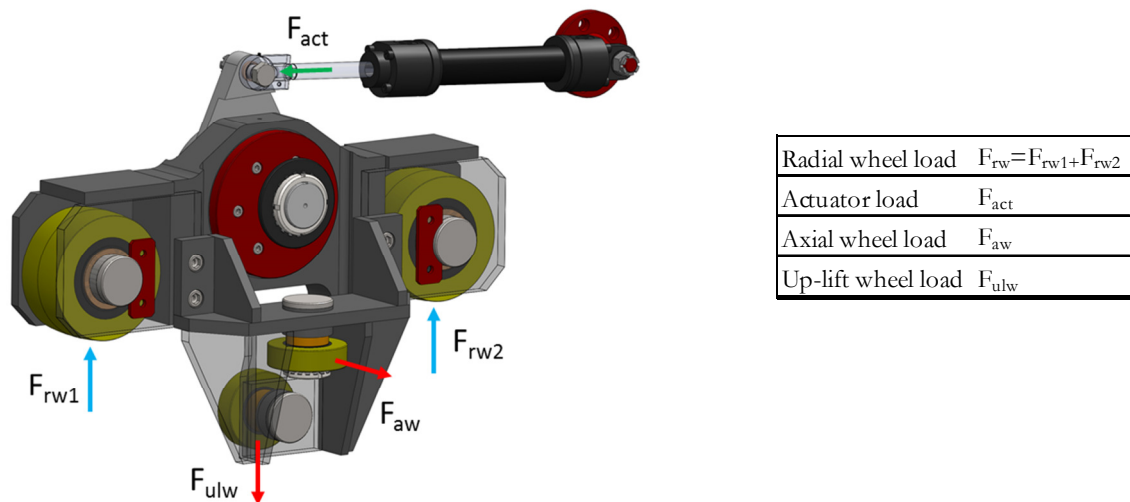


Fig. 8-10 Bogie structure load application points

Ultimate loads:

Factored loads are introduced for this analysis.

- Radial and actuator loads

The maximum radial wheels loads and consequent actuator load are extracted from the Upper shutter support analysis (section **¡Error! No se encuentra el origen de la referencia.**) and a load factor of 1.35 has been considered for the extreme load conditions.

- Axial load

The axial wheel load is computed out of an extreme hypothetical condition; assuming the entire upper shutter is holding from one side (drive systems of the other side not working)

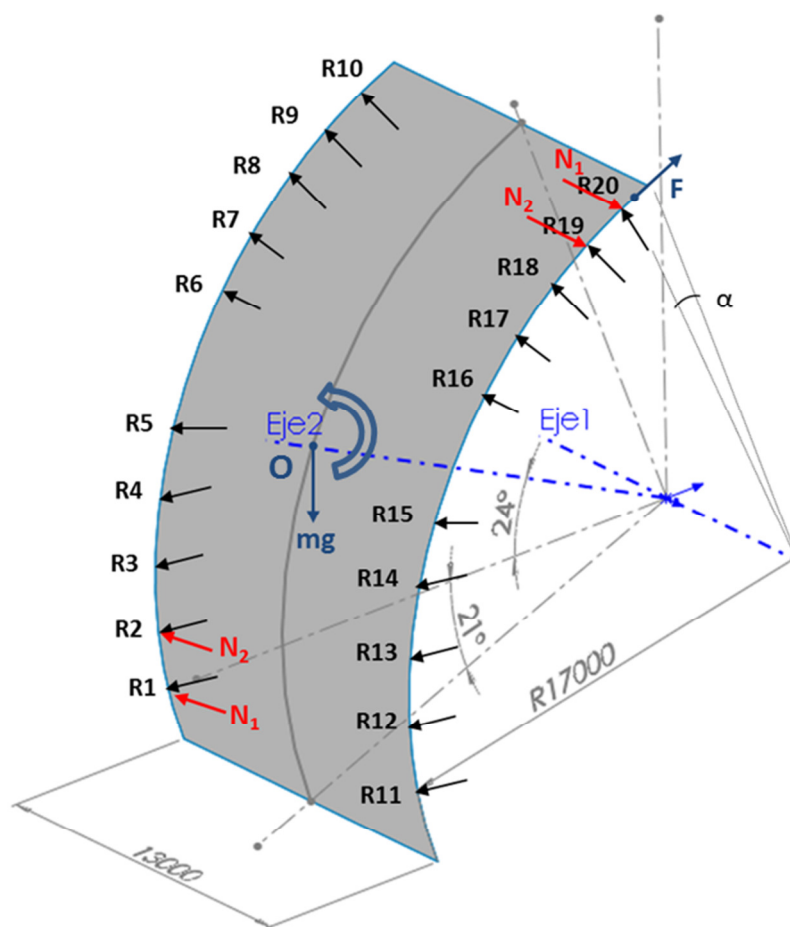


Fig. 8-11 Calculation procedure of maximum axial load

Considering 4 bogies taking the axial load:

$$\sum M_0 = 0 \longrightarrow \begin{cases} N_1 = 45 \text{ kN} \\ N_2 = 37 \text{ kN} \end{cases} \quad \begin{array}{l} \text{MAXIMUM} \\ \text{AXIAL FORCE} \\ \text{ON BOGIE} \end{array}$$

It will be used to verify the bogie structure and the eccentric shaft bolted connection.

In summary, the values of the applied loads for ultimate conditions are:

	Factored extreme load
Radial wheel load $F_{rw}=F_{rw1}+F_{rw2}$ ($F_{rw1}=F_{rw2}$)	67,5KN
Actuator load F_{act}	11,25KN
Axial wheel load F_{aw}	45KN
Up-lift wheel load F_{ulw}	0KN

Fig. 8-12 Applied factored extreme loads

Fatigue loads:

In the analysis of the fatigue strength a partial factor of 1 has been used for the loads. For this analysis, the axial and up-lift wheel loads have been excluded.

The fatigue loads are obtained from the maximum reaction forces in each bogie during the shutter travelling from 0 to 90 degrees.

The reaction forces in bogie number 6 located at the side A, which correspond to the maximum reaction forces, have been used

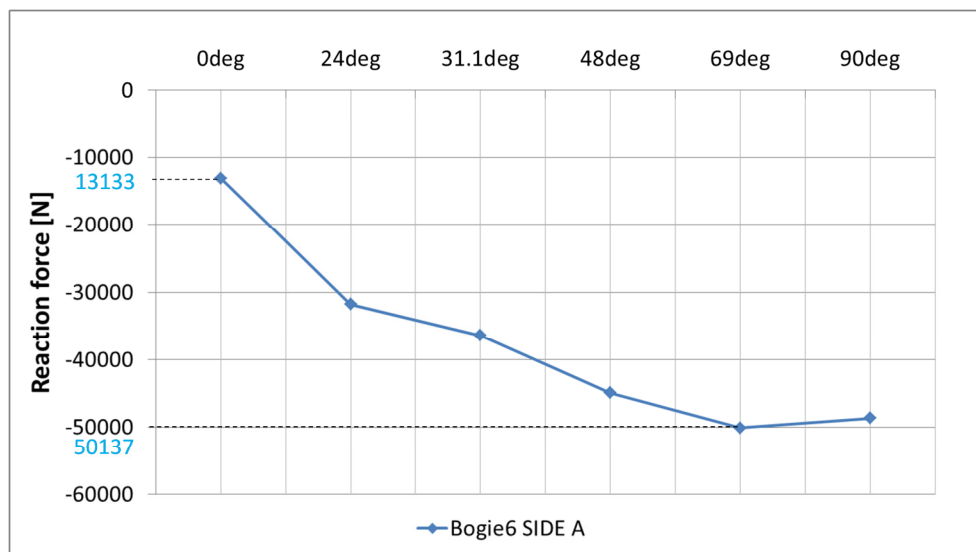


Fig. 8-13 Maximum reaction forces corresponding to bogie number 6 at side A

$$F_{r \max} = 50137 \text{ N}$$

$$F_{r \min} = 13133 \text{ N}$$

$$F_{\text{mean}} = \frac{F_{r \max} + F_{r \min}}{2} = 31635 \text{ N}$$

$$F_{\text{alt}} = \frac{F_{r \max} - F_{r \min}}{2} = 18502 \text{ N}$$

For the fatigue verification the mean load applied is $F_{\text{mean}} = 31.6 \text{ kN}$ and the variable load is $F_{\text{alt}} = 18.5 \text{ kN}$. Thus, for mechanism compensation a $F_{\text{act-mean}} = 5.3 \text{ kN}$ and $F_{\text{act-a}} = 3.1 \text{ kN}$ are applied.

8.3.2 Materials and safety factors

The following materials have been used for steel components

- Rolled elements material: S355 J2G3 according to EN-10025-2

Thickness [mm]	R ₀₂ [MPa]
<16	355
16<t<40	345
40<t<63	335
63<t<80	325
80<t<100	315
100<t<150	295
150<t<200	285
200<t<250	275
250<t<400	265

Fig. 8-14 Yield stress vs. plate thickness according to EN 10025-2

Thickness [mm]	R _m [MPa]
<3	510-680
3<t<100	470-630
100<t<150	450-600

Fig. 8-15 Ultimate strength vs. plate thickness according to EN 10025-2

The following materials partial safety factors have been used

1-Static Analysis:

For the ultimate limit analyses the partial safety factor for material has been taken according to EC3:

- Partial safety factor of $\gamma_s = 1.1$

2-Fatigue Analysis:

For the fatigue limit analyses the partial safety factor for material has been taken according to EC3, Part 1-9 Standard:

Assessment method	Consequence of failure	
	Low consequence	High consequence
Damage tolerant	1,00	1,15
Safe life	1,15	1,35

Fig. 8-16 Partial material safety factors according to EC3, Part 1-9

For the safe life approach and high consequence: 1.35

All structural materials have the following properties

- Density: 7850 kg/m³
- Young Modulus: 2e11 N/m²
- Poisson coefficient: 0.3

8.3.3 Structural FE model

A FE model based on quadratic tetrahedron elements has been created including the following components as shown in the figure.

- Template
- Main shaft
- Eccentric shaft
- Eccentric lid
- Washer

- Main frame
- Main shaft KM
- Radial wheel shaft
- Secondary frame
- Secondary frame shaft

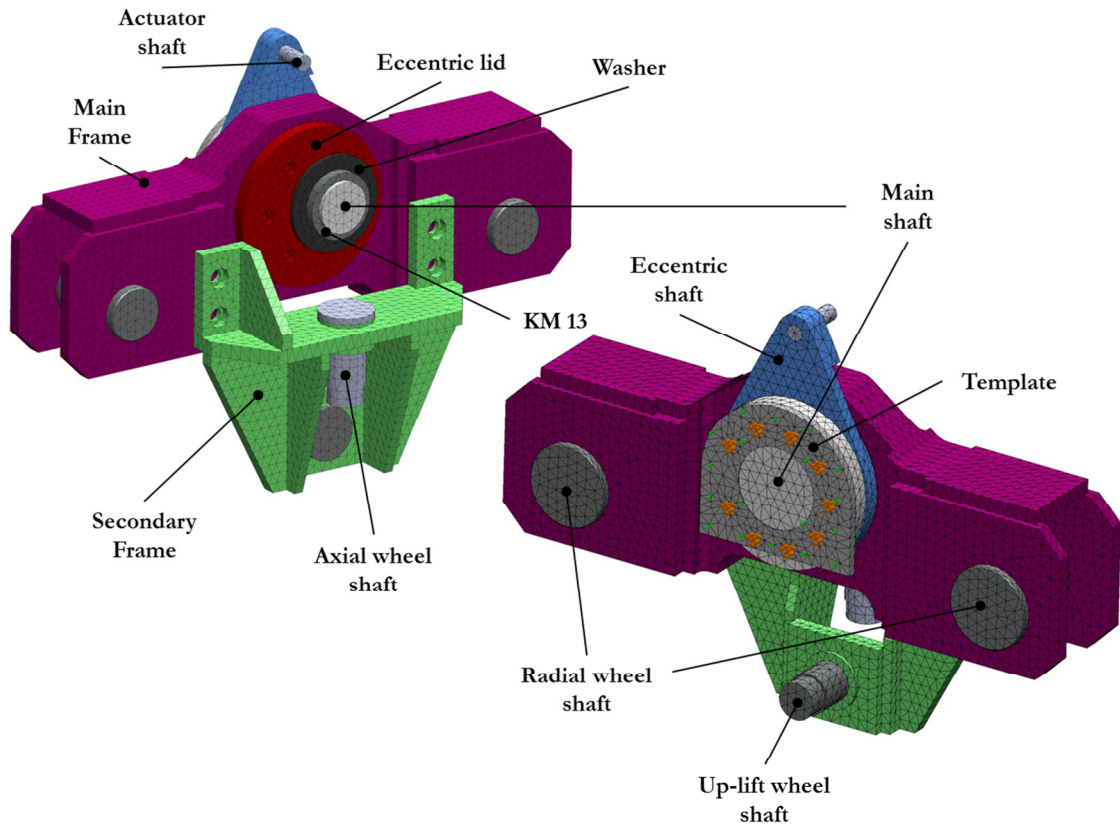


Fig. 8-17 Bogie main and secondary structure FE model

The model has been fixed at the shutter connection surface and the loads have been introduced at the wheels and the actuator (see Fig. 8-10 Bogie structure load application points Fig. 8-10).

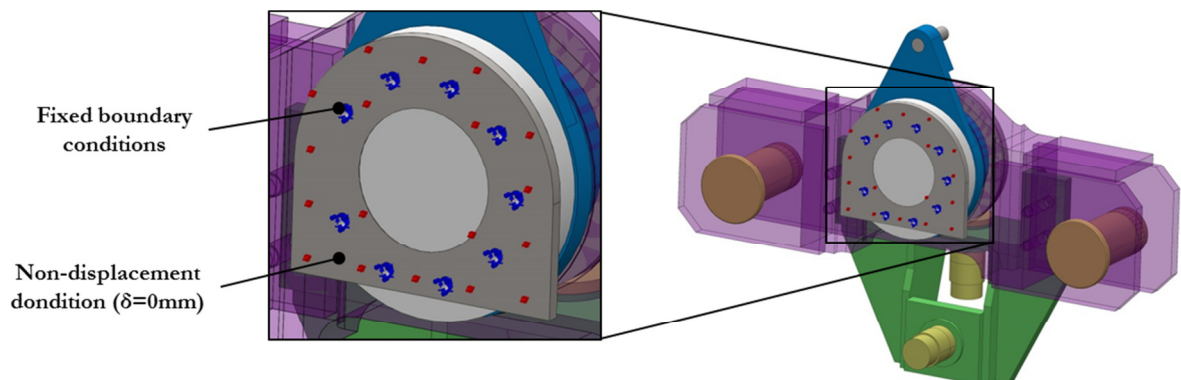


Fig. 8-18 FE model boundary conditions

The following figure shows the non-penetration and fixed rigid conditions used on different interfaces between components.

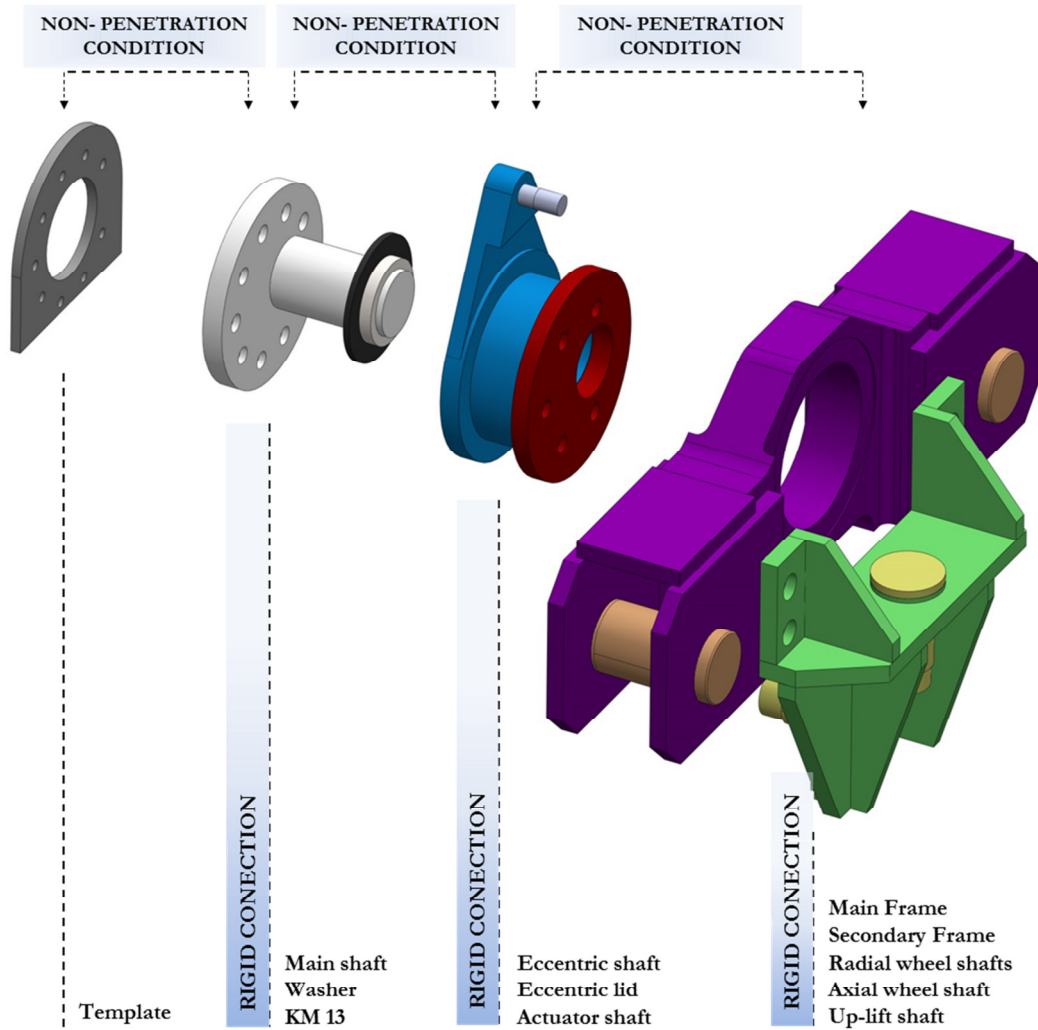


Fig. 8-19 Contact definitions between bogie components

8.3.4 Structural verifications

Extreme loads verification

The steel elements integrity is verified against factored ultimate loads. The validation criterion followed is:

$$|Linear\ stress| < \frac{\sigma_{0.2}}{\gamma_m}$$

where:

- Linear stress is the Von Mises stress from the FE model, calculated using linear material analysis, for factored ultimate loads.
- $\sigma_{0.2}$ is the yield stress.
- γ_m is the reduction coefficient of the material and takes a value of 1.1 for the extreme loads verification..

Fatigue limit state verification

The number of cycles during the dome lifetime, can be estimated as:

- In each Open-Close 1 cycles (see Fig. 8-13)

- 4 open-close cycles per night during 50 years (excluding maintenance work)

Therefore the expected articulations (chain cycles) during GTC lifetime = $1 \times 4 \times 365 \times 50 = 73000$ cycles.

- *Base material fatigue analysis:*

A synthetic S-N curve is calculated for material S355 following the procedure in "Leitfaden für eine Betriebsfestigkeitsrechnung"

R_m = ultimate tensile strength

$R_{m,nom} = 490 \text{ MPa}$

$R_m = 1.06 \cdot R_{m,nom} = 1.06 \cdot 490 \text{ MPa} = 519 \text{ MPa}$

Endurance limit: $\sigma_w = 0.45 \cdot R_m = 233,7 \text{ MPa}$

Stress concentration factor: $\alpha_k = 1$ (since the results are taken directly from the FEM)

Notch sensitivity caused by stress gradient influence and localized plastic deformation at notch base: $n = 1$

$\beta_k = \alpha_k / n = 1$

The surface roughness factor, F_o , is defined as

$$F_o = 1 - 0.22 * (\log R_z)^{0.64} \log R_m + 0.45 (\log R_z)^{0.53}$$

For $R_z = 160 \mu\text{m}$, $F_o = 0.7$

R_z = peak to peak roughness = $6 \cdot R_a$

R_a = average roughness (it has been considered the raw rolled condition)

The combined surface and technology factor, F_{ot} , is defined as

$$F_{ot} = 1 - \sqrt{(1 - F_o)^2 + (1 - F_t)^2} = F_o$$

The technological factor, F_t , is unity for $t \leq 100 \text{ mm}$.

$$F_{otk} = \sqrt{\beta_k^2 - 1 + \frac{1}{F_{ot}^2}} = \frac{1}{F_{ot}} = \frac{1}{F_o} = 1.43$$

As Goodman criteria is to be used to obtain the alternating equivalent stress ($\sigma_m = 0$ and $R = -1$) the factor for influence of mean stress, F_m , is set to unity.

- From 0 to N_D cycles the slope of the curve is $k = \frac{12}{(F_{otk})^2} + 3 = 8.86$ and the number of cycles for endurance limit $N_D = 10^{6.4 - \frac{2.5}{k}} = 1.31e6$
- From N_D cycles to infinity the slope of the curve is relaxed to $k_2 = 2k - 1 = 16.72$

Reduction factors are to be applied to this curve:

- Reduction factor for survival probability. $S_{PS} = 2/3$ for survival probability of 97.7%.
- Reduction factor for stress ratio: This is considered equal to unity, $S_R = 1$, because Goodman criterion is used to transform all cycles to $R = -1$ cycles.

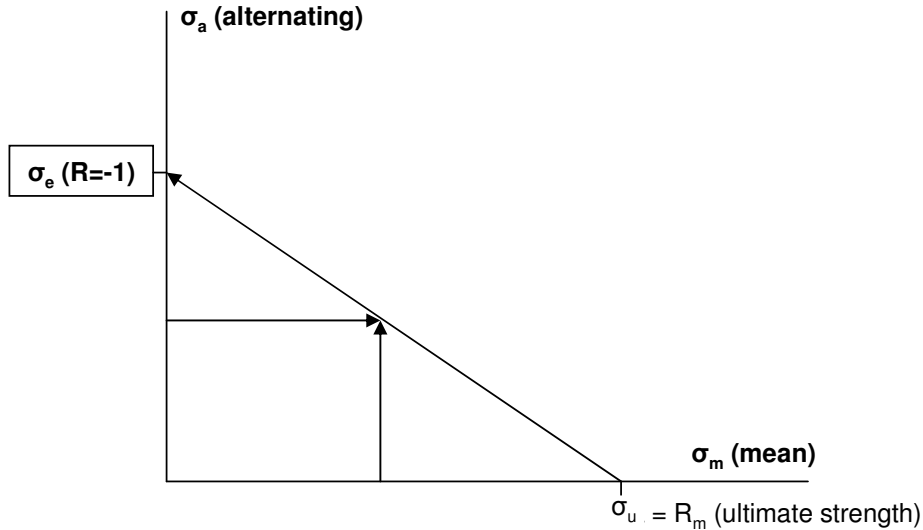


Fig. 8-20 Haigh diagram with Goodman criterion

- Reduction factor for surface roughness $S_{sr} = 1/F_{otk}$, for $R_z = 160 \mu m$

Total reduction factor $S = S_{PS} S_R S_{SR}$

Additionally, a material safety factor is applied: $\gamma_M = 1.15$ (according to safe life approach in Part 1-9 of EC-3).

Following this procedure, an S-N curve is obtained which represents the fatigue limit for the base material, used to compare the results of the FE model. Stress values below this limit involve an infinite life under fatigue loads.

The resulting S-N curve is:

$$N < 1.31e6 \rightarrow \sigma_a = 233,7 \frac{S}{\gamma_m} \left(\frac{N}{1.31e6} \right)^{-1/8.86}$$

$$N > 1.31e6 \rightarrow \sigma_a = 233,7 \frac{S}{\gamma_m} \left(\frac{N}{1.31e6} \right)^{\frac{-1}{2 \cdot 8.86 - 1}}$$

At $1e7$ cycles $\sigma_a = 73 MPa$.

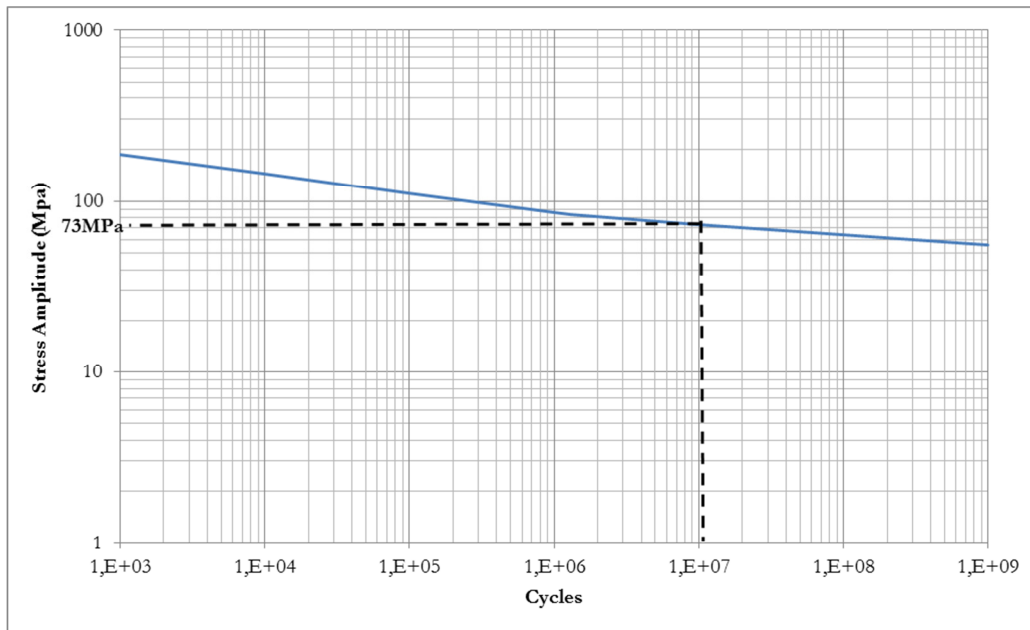


Fig. 8-21 S-N curve for S355 steel

- *Welded zones fatigue analysis:*

The welded zones have lower fatigue resistance than the base material. In order to ensure that the design is correct and it is able to endure the number of cycles estimated without failure, these zones have been analyzed.

The nominal stress analysis criterion has been followed in order to verify the fatigue behavior of the welded material. The maximum nominal stress value in welds is obtained and compared with the S-N curves used to analyze the fatigue behavior of the welded material, extracted from EC-3 Part 1-9. Depending on the joint geometry (construction detail) and on the acting loads, a specific detail category is assigned to each welded joint.

A safety coefficient of 1.35 has been applied according to recommended values for partial factors for fatigue strength EC-3 Part 1-9.

Assessment method	Consequence of failure	
	Low consequence	High consequence
Damage tolerant	1,00	1,15
Safe life	1,15	1,35

Fig. 8-22 Recommended values for partial factors for fatigue strength

The following figures show the fatigue curve for detail category number 80.

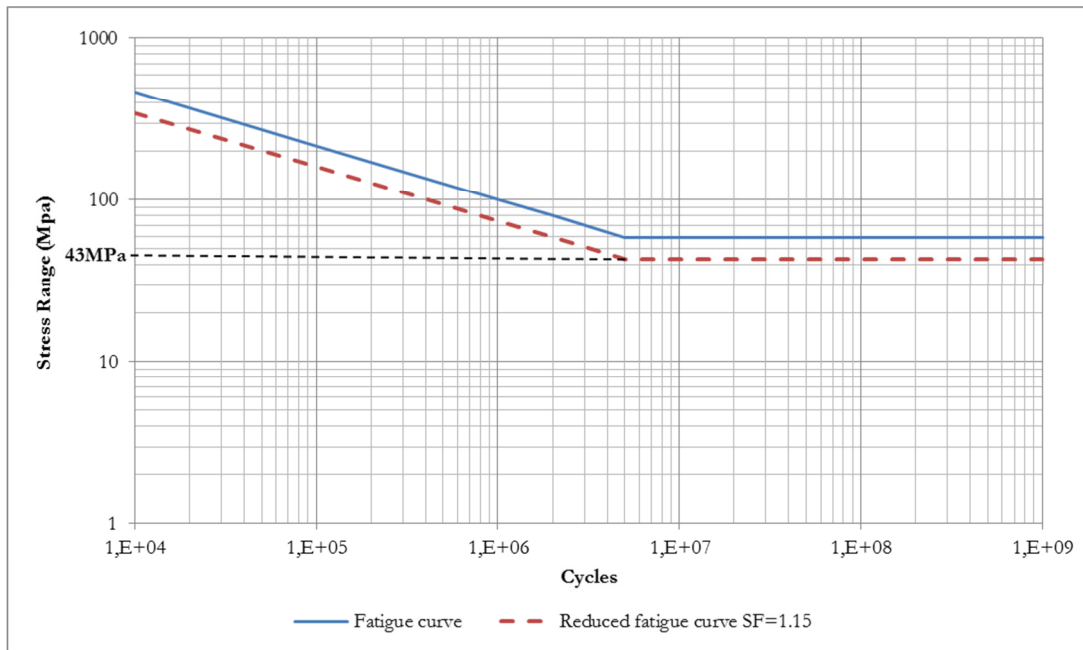


Fig. 8-23 EC3 fatigue curve corresponding to detail 80

The reduced fatigue limit range for the weld detail chosen is $59/1.15=43$ MPa

8.3.5 Structural results

The structural results of the most critical components are shown below.

MAIN SHAFT

Extreme Loads verification:

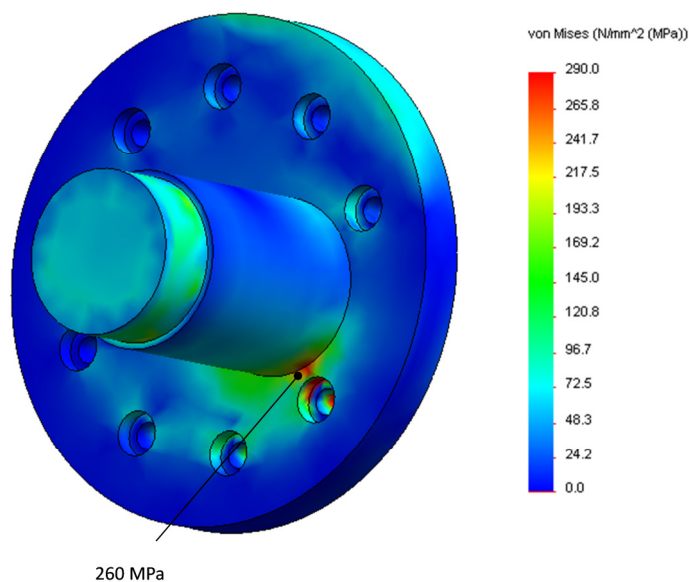


Fig. 8-24 Main axis Von Mises stresses

The maximum Von-Mises stress is 260 MPa. This value is lower than the maximum stress allowed value: $\sigma_{\text{allow}}=315/1.1=286$ MPa

Fatigue Loads verification:

The following figures show the fatigue verification

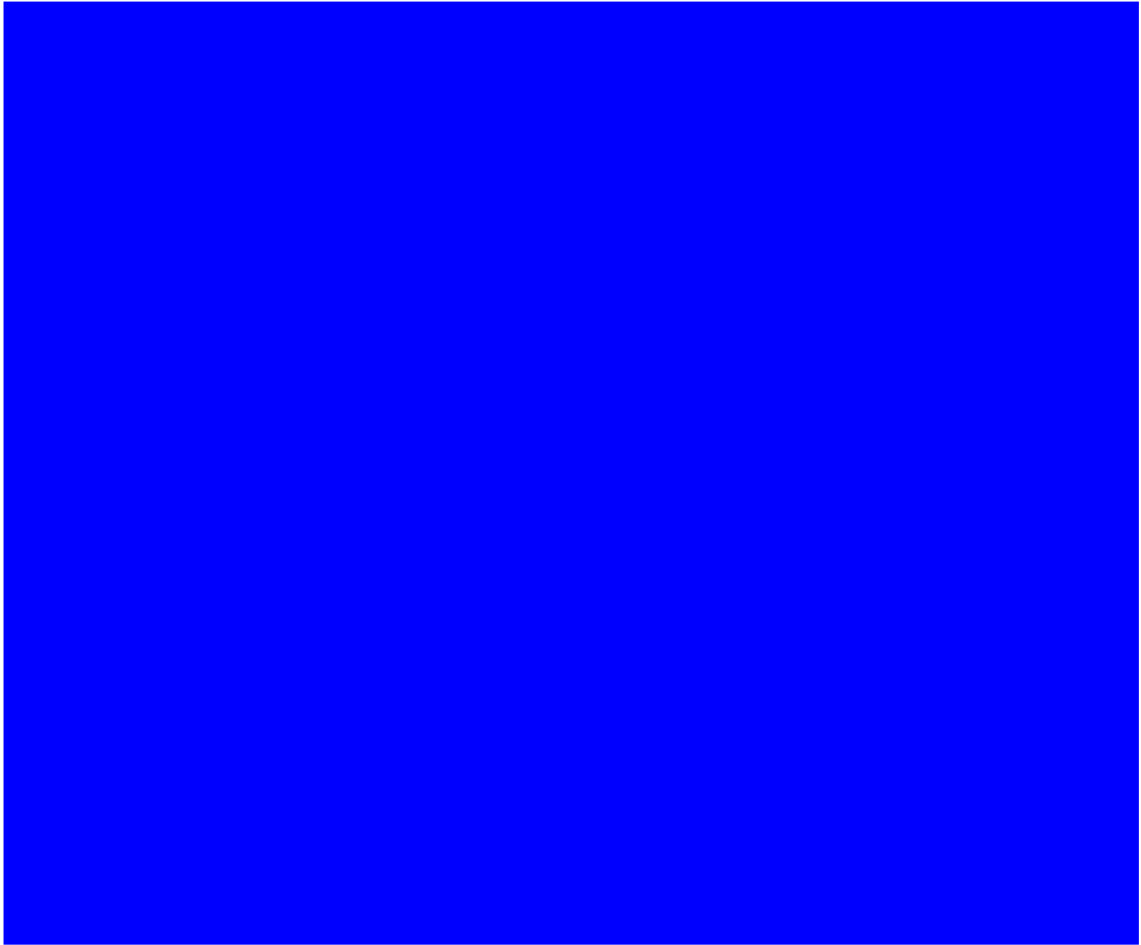


Fig. 8-25 Main axis fatigue stresses

The maximum fatigue equivalent stress (after applying Goodman criterion) is $\sigma_{P1 \text{ goodman}} = 40 \text{ MPa}$, due to the maximum principal stresses, is below the fatigue limit for $1e7$ cycles ($\sigma = 73 \text{ MPa}$)

ECCENTRIC SHAFT*Extreme Loads verification:*

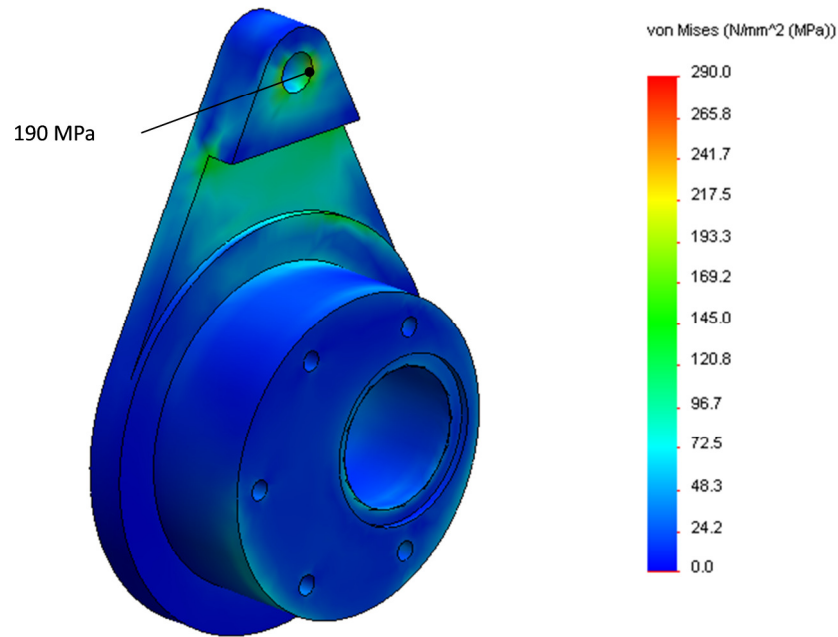


Fig. 8-26 Eccentric axis Von Mises stresses

The maximum Von-Mises stress is 190 MPa. This value is lower than the maximum stress allowed value: $\sigma_{\text{allow}} = 315 / 1.1 = 286$ MPa

Fatigue Loads verification:

The following figures show the fatigue verification

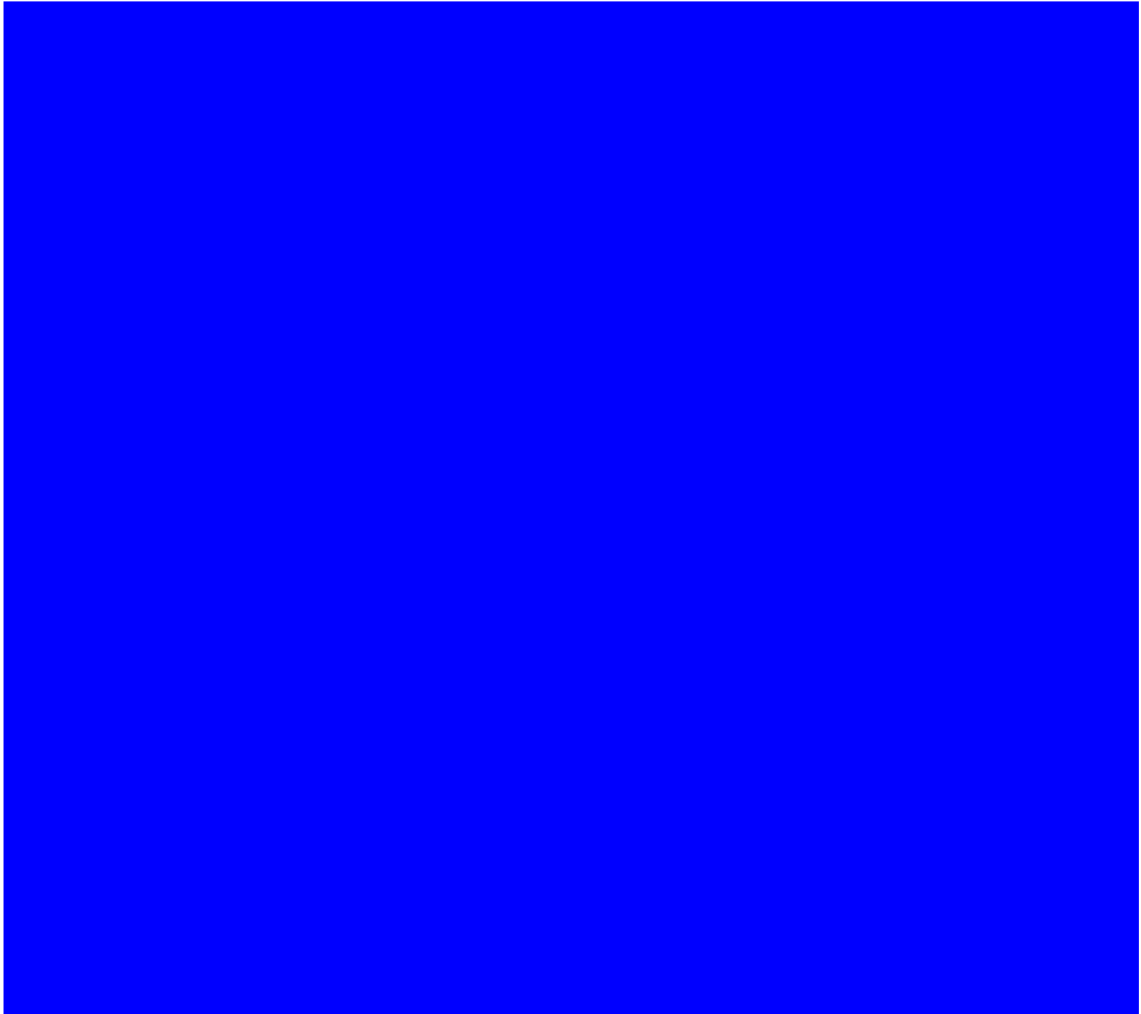


Fig. 8-27 Eccentric axis fatigue stresses

The maximum fatigue equivalent stress (after applying Goodman criterion) $\sigma_{P1 \text{ goodman}} = 46.5 \text{ MPa}$, due to the maximum principal stresses, is below the fatigue limit for $1e7$ cycles ($\sigma = 73 \text{ MPa}$)

MAIN FRAME

Extreme loads verification:

Von Mises stresses under extreme loads are shown in the following figures.

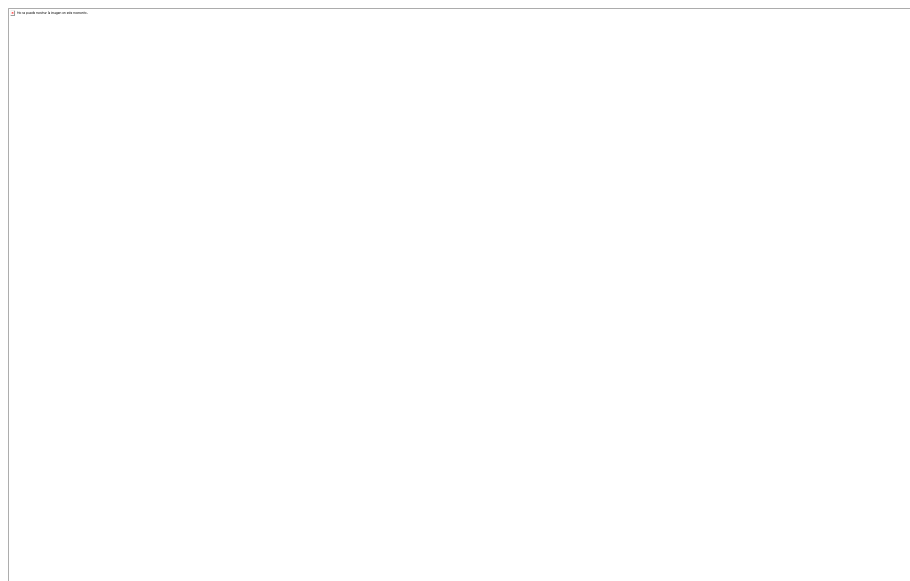


Fig. 8-28 Main frame Von Mises stresses

The maximum Von-Mises stress is 180 MPa. This value is lower than the maximum stress allowed value: $\sigma_{\text{allow}} = 315 / 1.1 = 286$ MPa

Fatigue loads verification

The following figures show the fatigue verification of the main frame base material.



Fig. 8-29 Main frame fatigue stresses

The maximum fatigue equivalent stress (after applying Goodman criterion) $\sigma_{P3 \text{ goodman}} = 16\text{MPa}$, due to the minimum principal stresses, is below the fatigue limit for $1e7$ cycles ($\sigma = 73\text{MPa}$)

The following figures show the fatigue verification of the main frame welds.

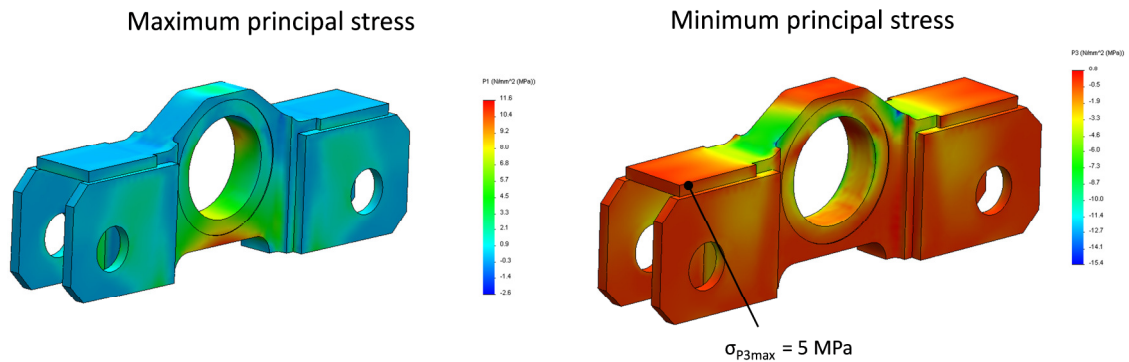


Fig. 8-30 Main frame weld verification

The maximum stress in the welded areas is 5 MPa. This value is lower than the maximum stress allowed value: $\sigma_{\text{allow}} = 43\text{ MPa}$.

SECONDARY FRAME

Extreme loads verification:



Fig. 8-31 Secondary frame Von Mises stresses

The maximum Von-Mises stress is 202 MPa. This value is lower than the maximum stress allowed value: $\sigma_{\text{allow}} = 315/1.1 = 286\text{ MPa}$.

9 APPENDIX B: ENGINEERING DRAWINGS