Preliminary QIKR Motion Design Review
Sample/Detector Table Details

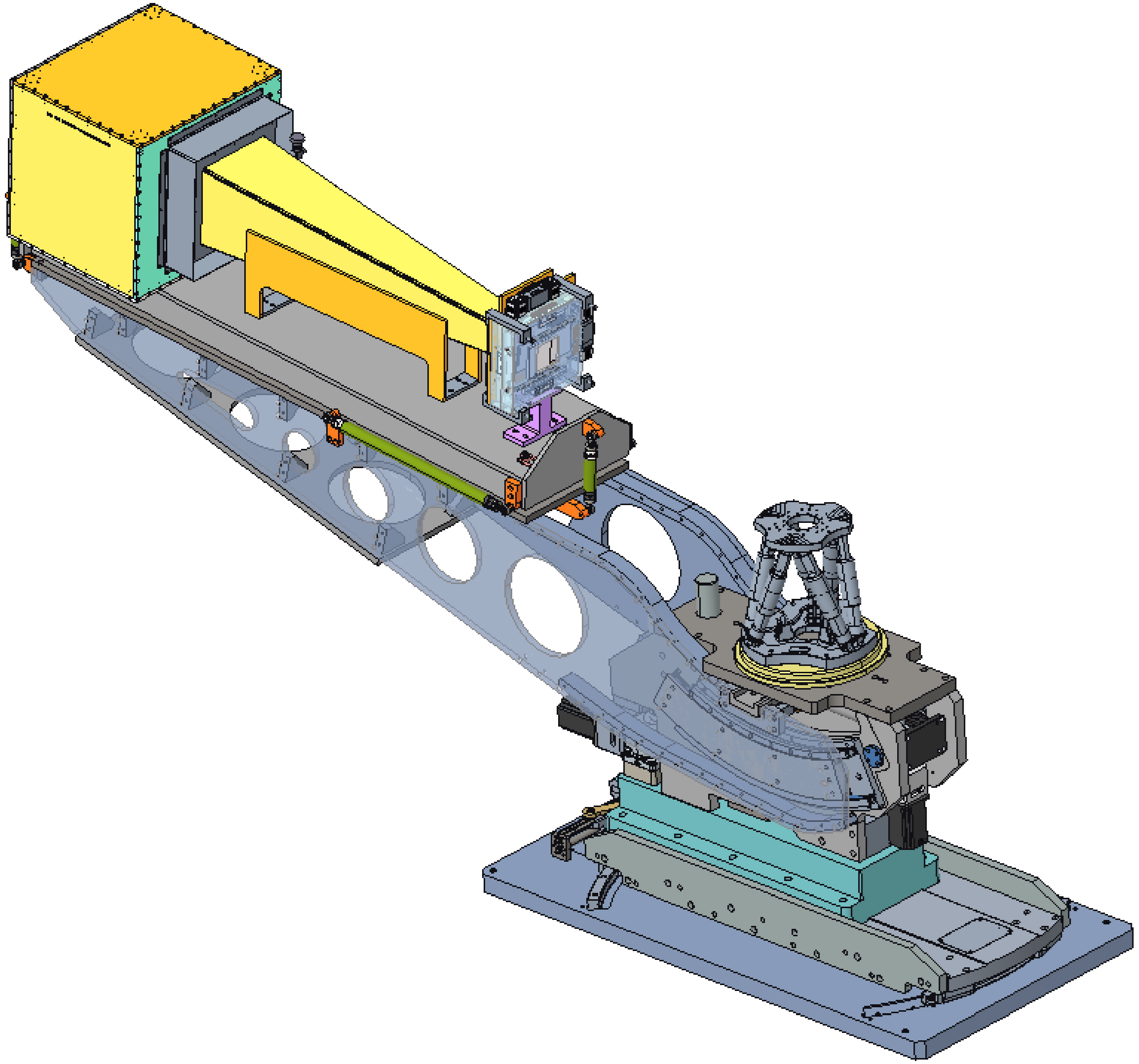
Rudy Thermer, Motion Engineer

November 06, 2024

ORNL is managed by UT-Battelle, LLC for the US Department of Energy

Outline

- Sample/ Detector Table Requirements
- Design Details
- DACs



Detector Table Requirements

- S.04.08.06-R125 – The detector table must place the center of the detector sensing surface at a fixed radial distance 2m $\pm$ 1mm from the nominal sample location.
  - The detector face of the current placeholder model is located a nominal distance of 2m from the sample center location, and can be adjusted within a range of 25mm (1”) to within 50µm. When the final design of the detector is known, the nominal distance will be corrected as needed.
- S.04.08.06-R126 – The detector table must have a footprint in X of $\leq$765 mm.
  - The detector & sample table are 475.8mm wide. The detector arms are 375.1mm wide.
- S.04.08.06-R127 – The detector table must be able to support at least $\geq$227 kg ($\geq$500 lbs) in addition to its own weight.
  - The detector table can support 306kg (674lbs) applied at the downstream edge of the table.
- S.04.08.06-R130 – The detector table must provide $\pm$17.5° of remote rotation about the X-axis with motion resolution of .002°. *Note: Angle measured from horizontal. This angle range provides at least +15° of rotation about the QIKR-A and QIKR-B beam inclinations of 2.5° and -2.5° respectively.*
  - The detector arms rotate through a range of $\pm$17° measured from horizontal with a resolution of $1.1(10)^{-4}$ degrees/ full step.

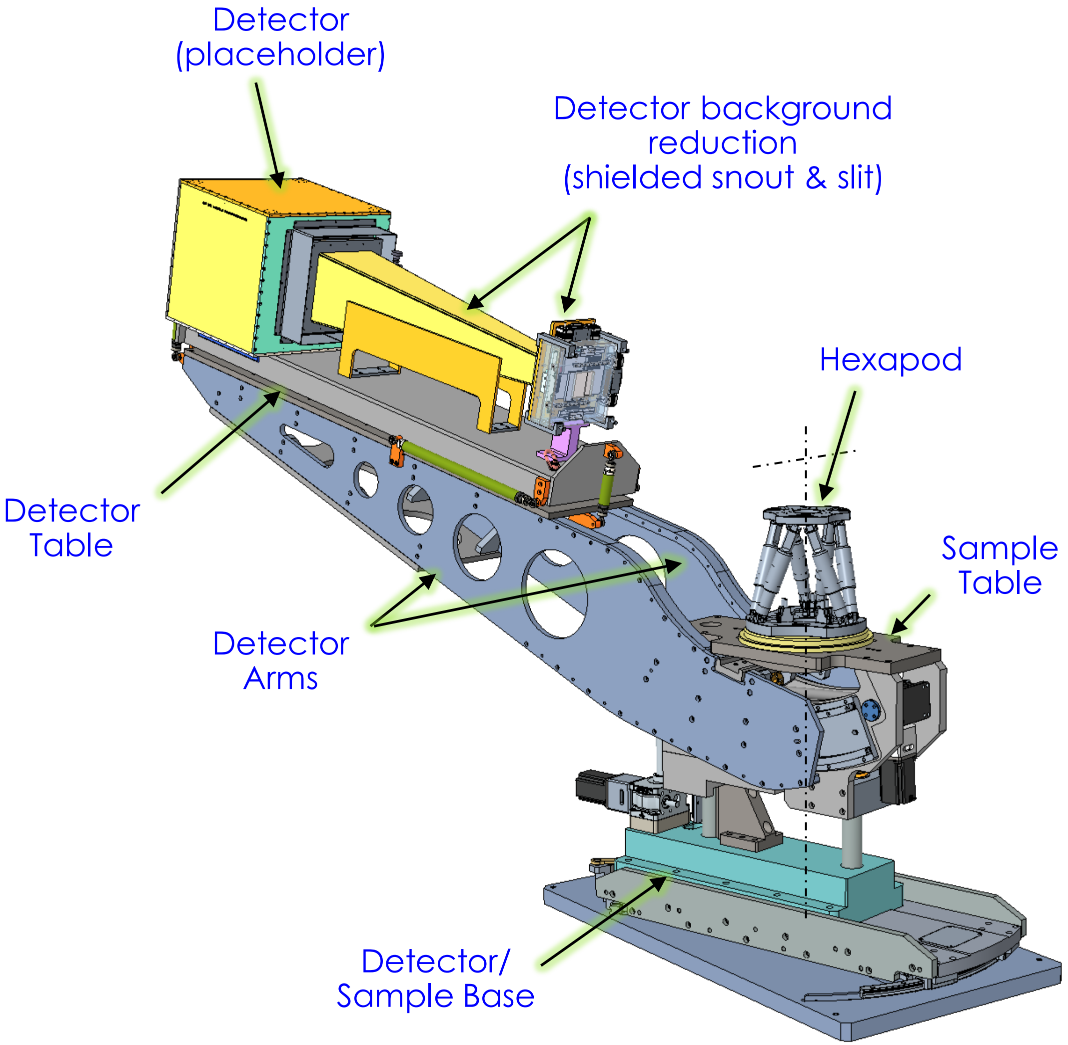
Detector Table Requirements

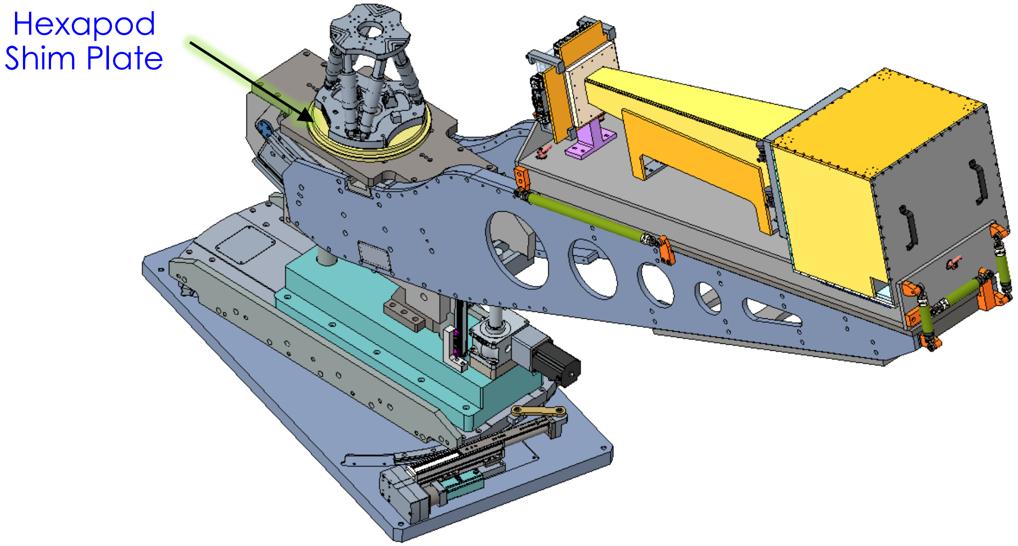
- S.04.08.06-R131 – The detector table must provide $\geq$15° degrees of remote rotation toward the user about the Y-axis with motion resolution of $\pm$.002°.
  - The detector provides 17° of rotation about the Y-axis through the sample center with a resolution of $6.8(10)^{-4}$ degrees/ full step.

Sample Table Requirements

- S.04.08.06-R116 – The sample table must have a footprint in X of $\leq$765 mm.
  - The detector & sample table are 475.8mm wide.
- S.04.08.06-R117 – The sample table must be able to support at least $\geq$455 kg ($\geq$1000 lbs) in addition to its own weight. *Note: Sample environments are expected to weight under 500lbs.*
  - The sample table can support 510kg (1,124lbs) applied at the downstream edge of the table.
- S.04.08.06-R118 – The sample table must provide coarse positioning of the sample to one of three nominal Y locations to within $\pm$.1mm. The Y locations correspond to the height of the three beam components of interest at the nominal sample 'z' distance from the guide end.
  - The sample table has a vertical motion range of 150mm to cover a distance of 105mm between lowest and highest sample positions. Currently, when the sample table sits at its lowest position, its surface sits below the lowest sample position by 560mm. Motion resolution is $1.25(10)^{-3}$mm/ full motor step.

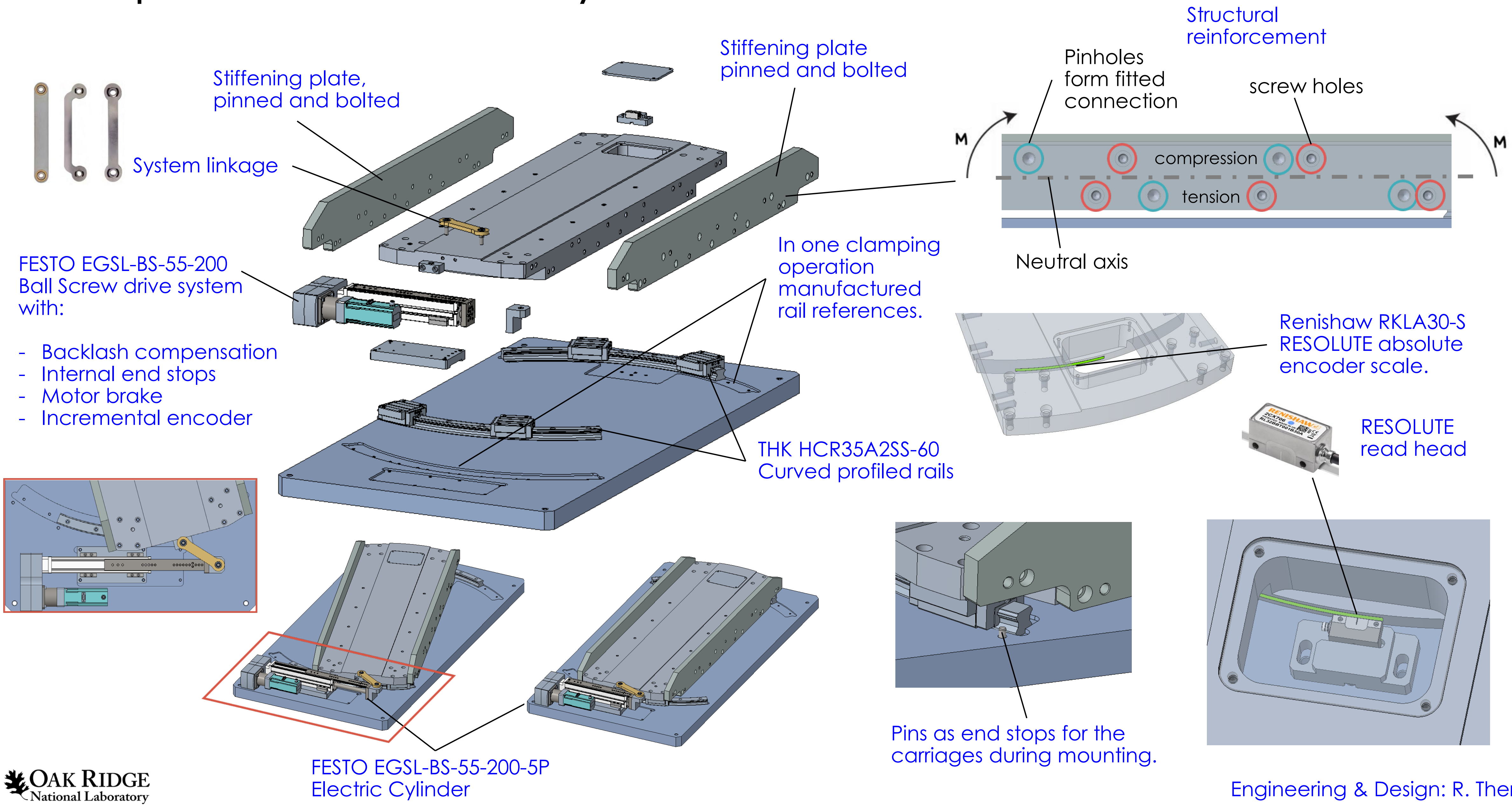
Detector/ Sample Table – Overview





Engineering & Design: R. Thermer

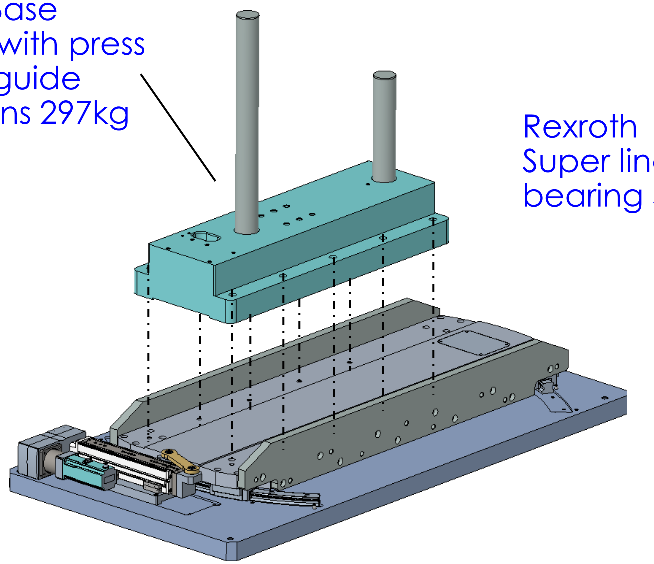
Sample Table – Rotary Table Installation



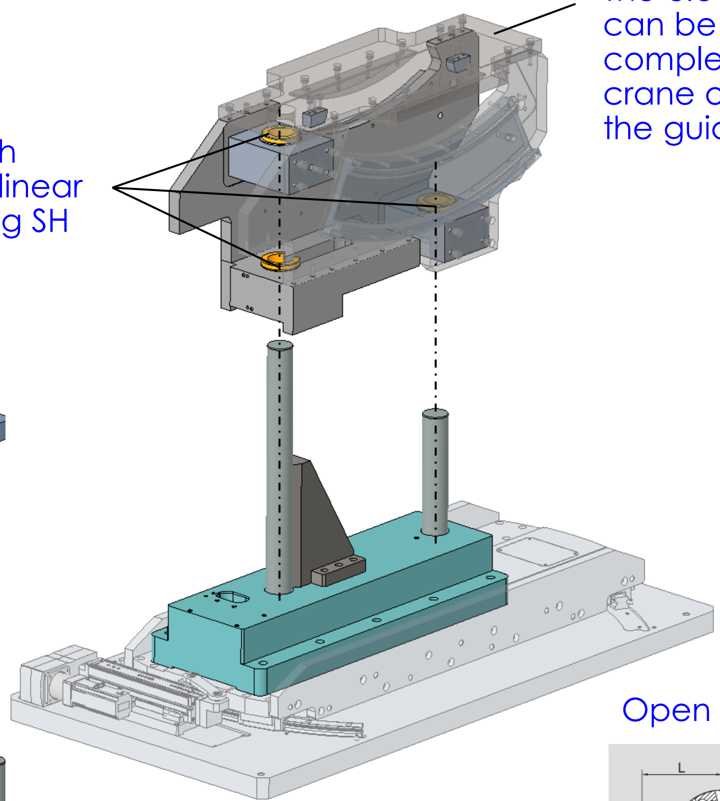
Engineering & Design: R. Thermer

Sample Table – Installation Sequence & Subassemblies

Steel Base block with press fitted guide columns 297kg

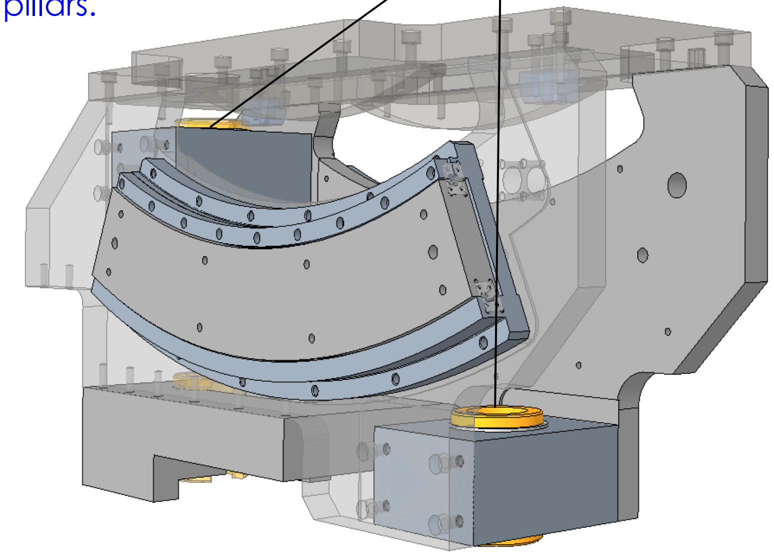


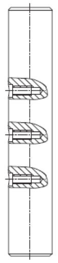
Rexroth Super linear bearing SH

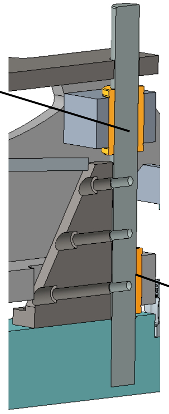


The elevator body (136 kg) can be installed either as a complete unit with the crane or piece by piece on the guide pillars.

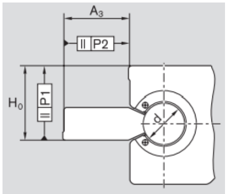
Closed linear bushings secured by retaining rings



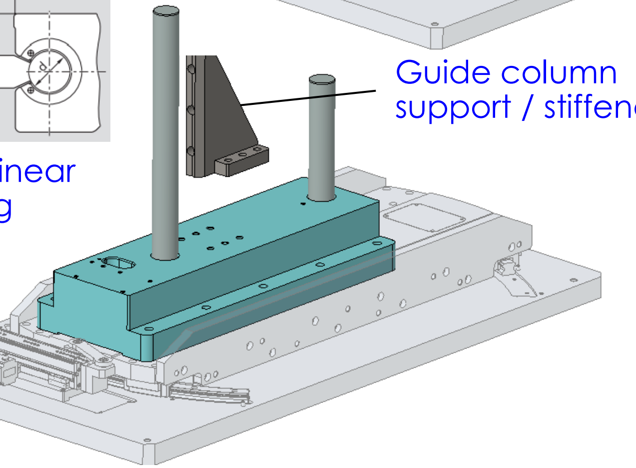




Rexroth offers customized guide columns.

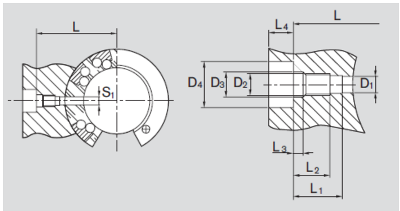


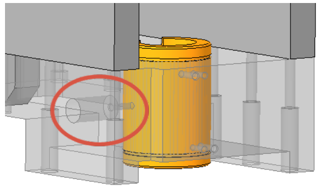
Open linear bearing



Guide column support / stiffener

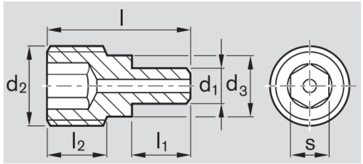
Open ball bushing secured by fixing screws





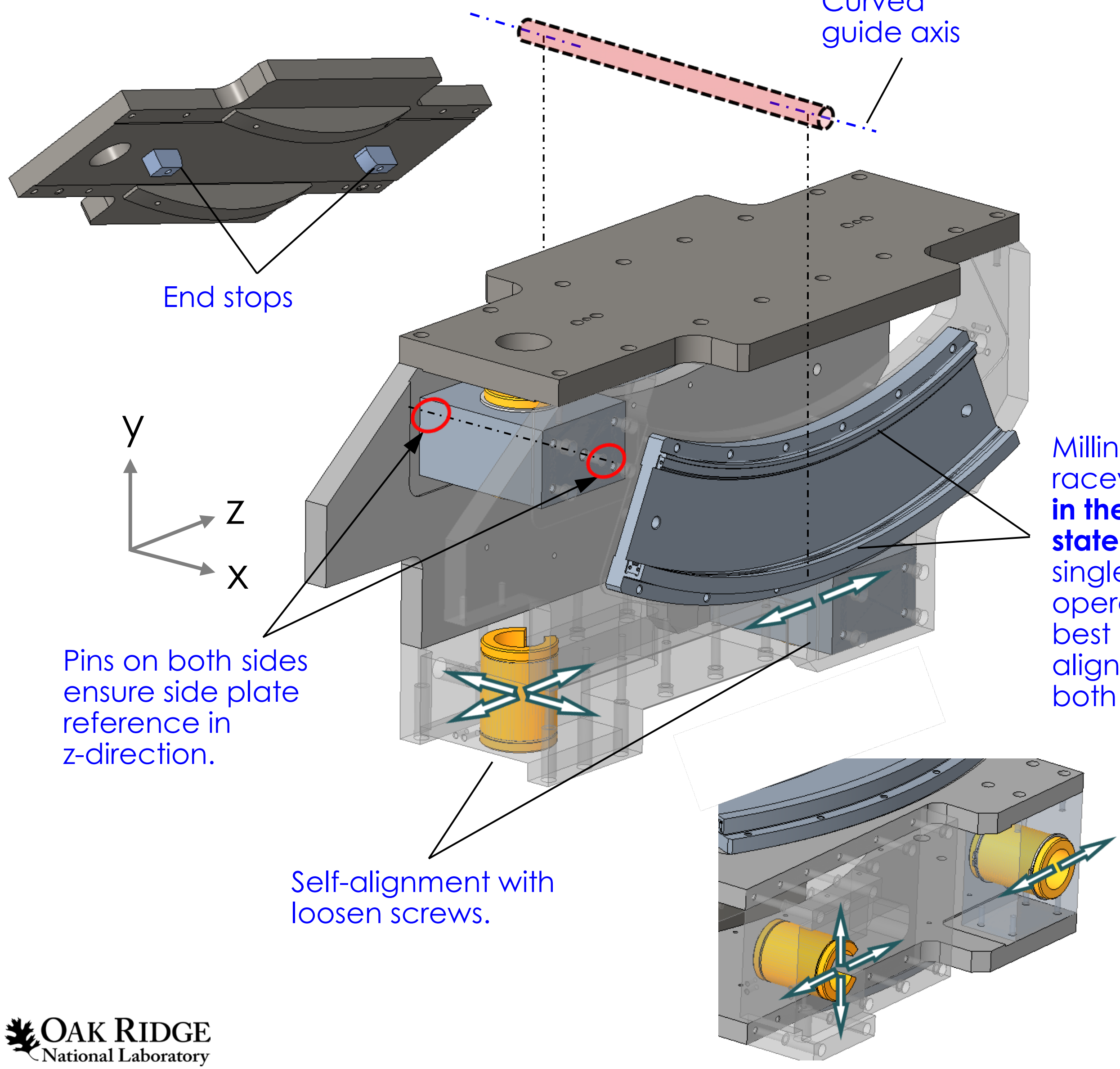
The bores for ball bushings are toleranced so that they can be pressed in by hand

Bore screw for relubricating and fixing the super ball bushing



Engineering & Design: R. Thermer

Sample Table – Post-Manufacturing for Wire Race Ways



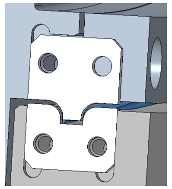
Curved guide axis

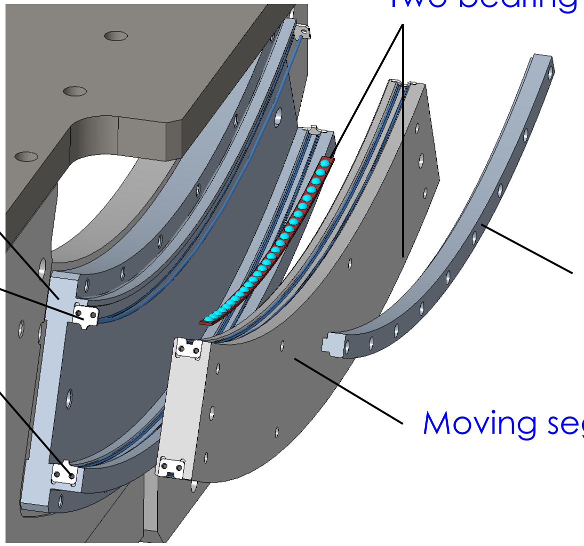
End stops

Pins on both sides ensure side plate reference in z-direction.

Self-alignment with loosen screws.

Milling the wire raceway grooves **in the assembled state** and in a single clamping operation ensures best possible alignment of both axes.





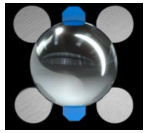
Pinned and bolted guide body

Ball cage stopper

Two bearing cages

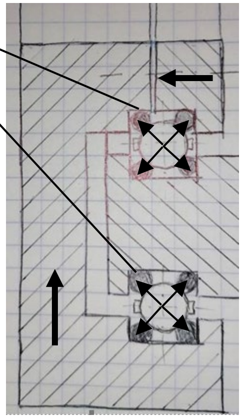
Bracing segment

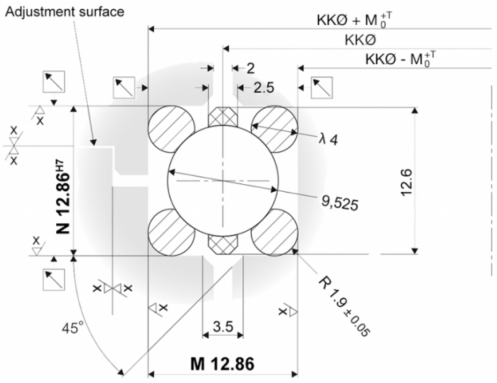
Moving segment



Flexible design using FRANKE 4-Point-wire race bearings.

The four-point bearing is braced by means of suitable tolerancing.

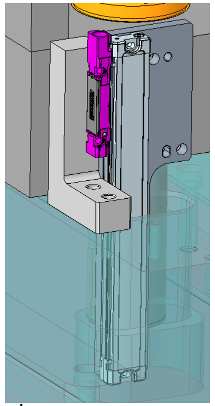




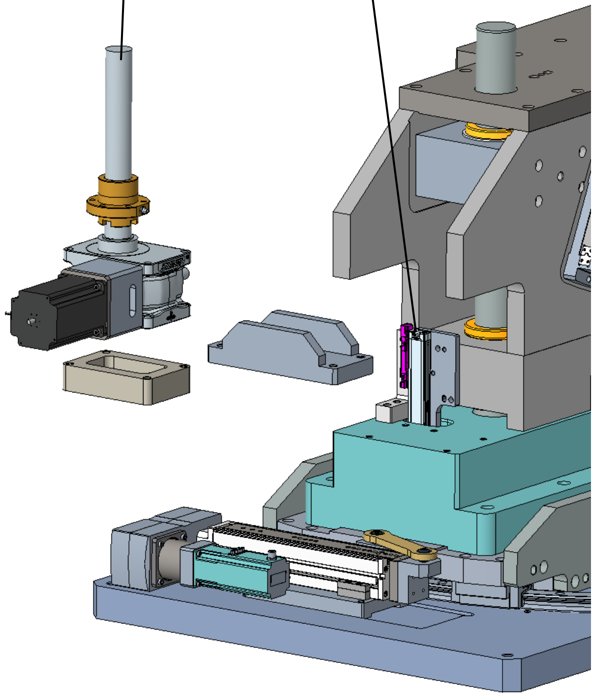
Engineering & Design: R. Thermer

Sample– Installation Sequence & Subassemblies

Renishaw FORTIS enclosed absolute encoder system



Installation of the screw jack system with the crane (95 kg)

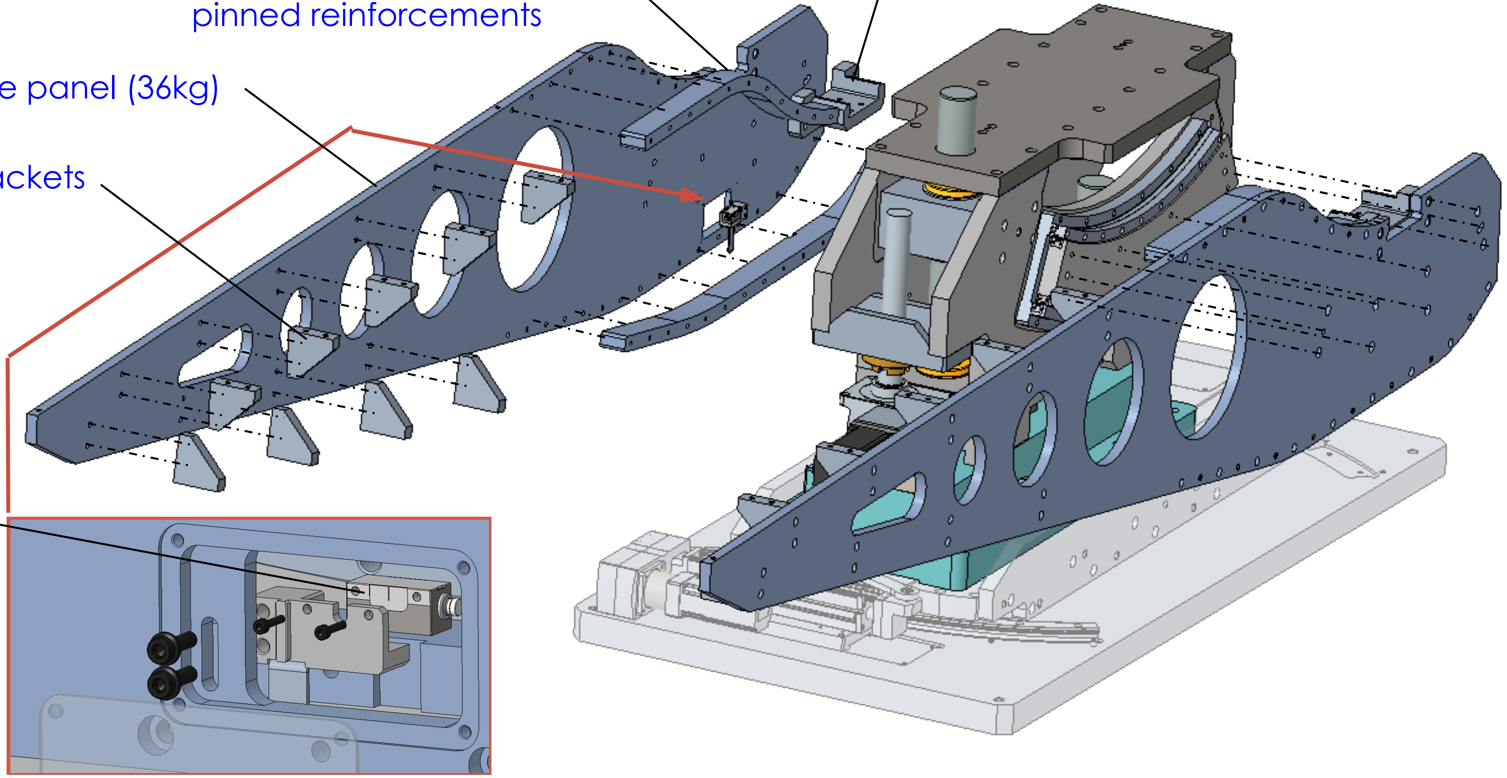


Water jet-cut, bolted and pinned reinforcements

Cross beam mounts

Arm side panel (36kg)

Angular brackets



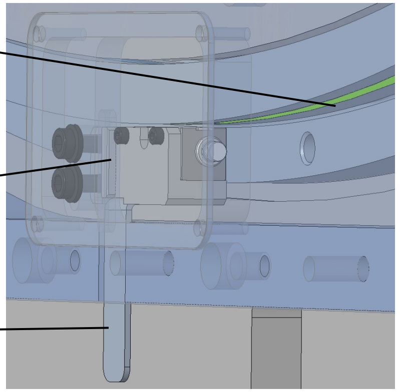
Renishaw RESOLUTE read head

Can be replaced and adjusted on the spot.

Renishaw RKLA30-S RESOLUTE absolute encoder scale.

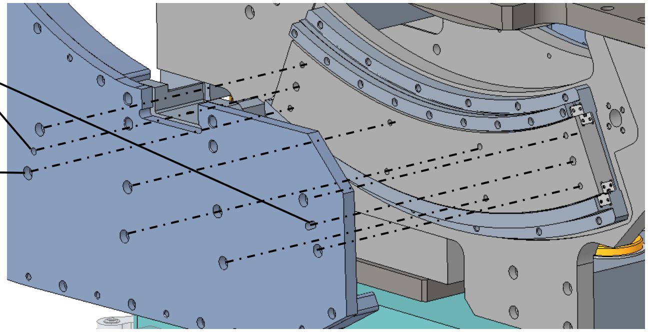
Keyway guide

Bent sheet metal handle for adjustment



2 Locating pins

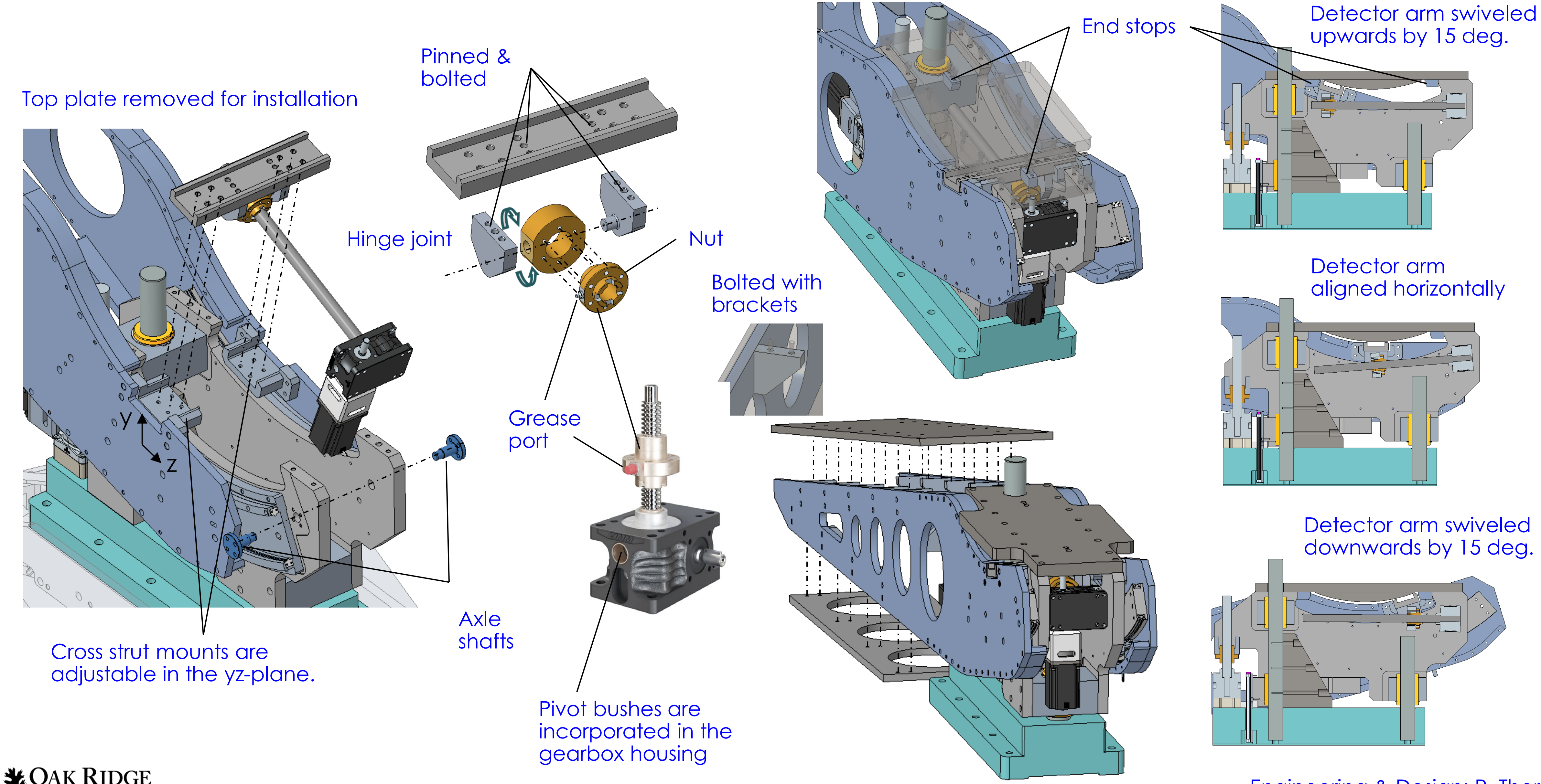
8 Screw holes



Engineering & Design: R. Thermer

OAK RIDGE National Laboratory

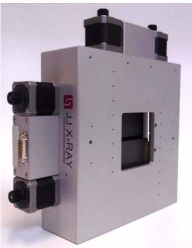
Sample Table – Installation Sequence & Subassemblies



Engineering & Design: R. Thermer

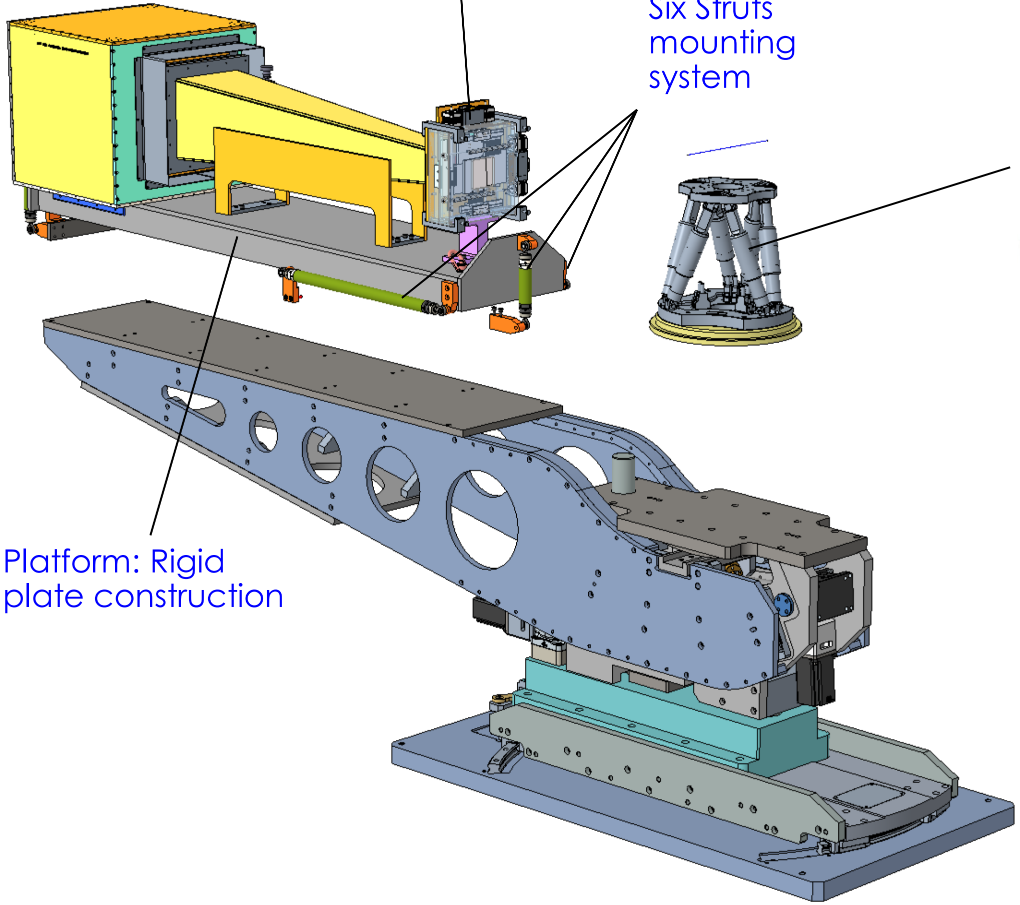
Sample Table – Installation Sequence & Subassemblies

JJ X-Ray
IB-C80-AIR Slits
Aperture size
80 mm x 80 mm



Physical Instrument
H-850.H2A Hexapod 6-Axis
Positioning System

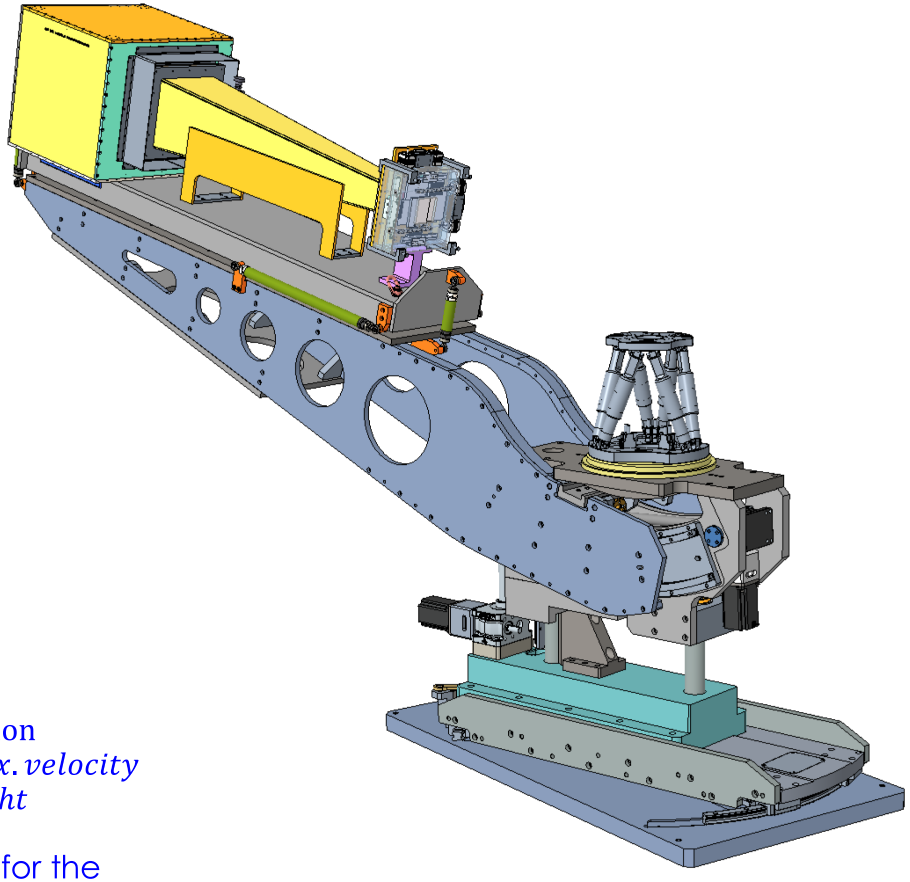
Six Struts
mounting
system



Platform: Rigid
plate construction

$100 \cdot 100 \cdot 50\ mm$ XYZ travel
$30 \cdot 30 \cdot 60°$ rotary travel
$250\ kg$ load
$2nm$ & $50nm$ actuator resolution
$0.5\ mm/sec$ & $8\ mm/$ sec $max.velocity$
$\varnothing\ 348\ mm\ base, 328\ mm\ height$

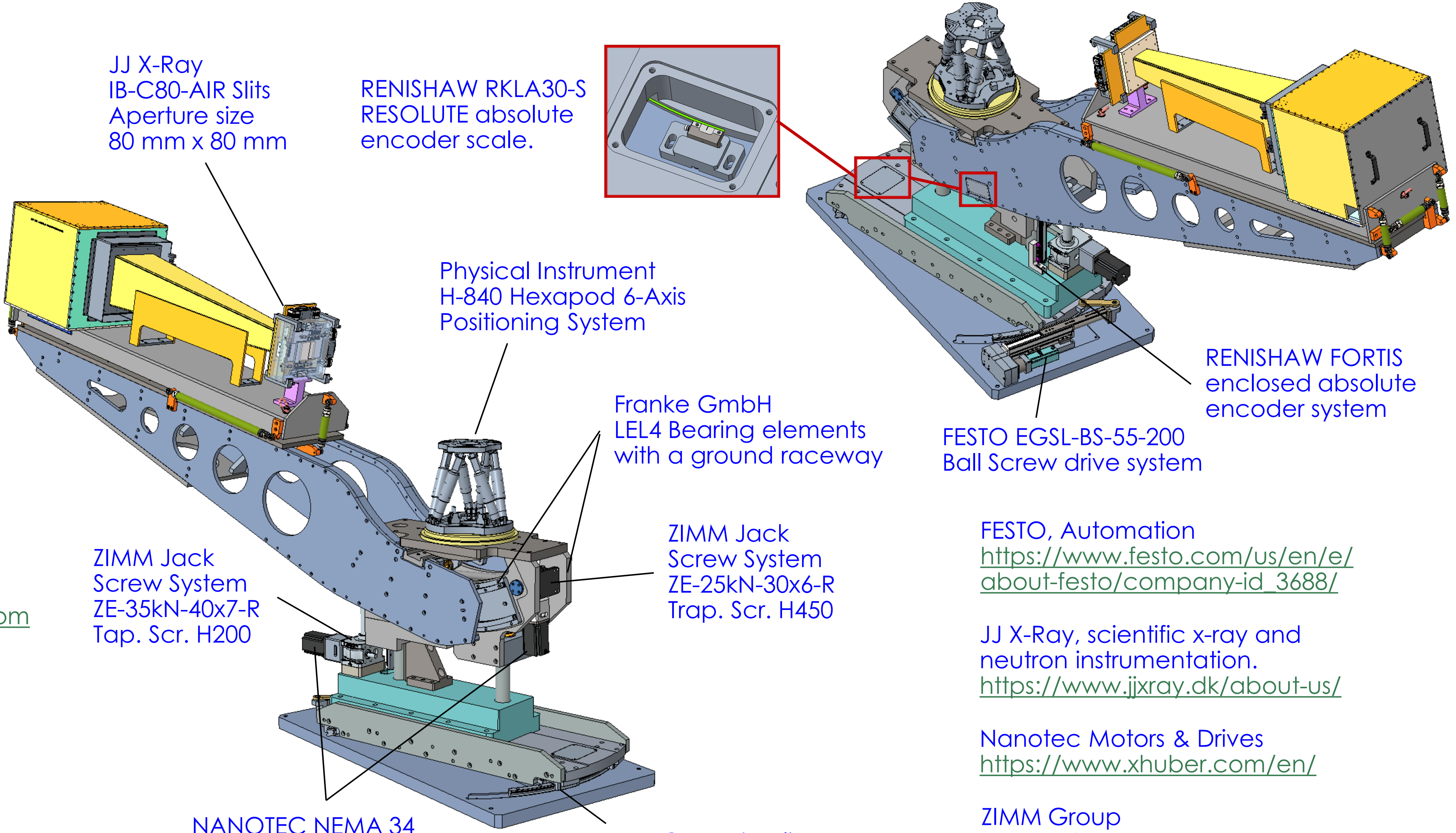
Alternatives can be found for the
Hexapod if needed, but they will
consume more space & will provide
only a fixed pivot point for sample rotation.



Engineering & Design: R. Thermer

Sample Table – Costs and Manufacturing Plan

- All components are from internationally well-established companies which are well represented in the US.
- Multiple vendors exist for Renishaw and THK products.



PI Physical Instruments
https://www.pi-usa.us/en

Franke GmbH, Rolling bearings and linear guides
https://www.franke-gmbh.com

RENISHAW metrology
https://www.renishaw.com/

THK Co, Machine elements
https://www.thk.com/

FESTO, Automation
https://www.festo.com/us/en/e/about-festo/company-id_3688/

JJ X-Ray, scientific x-ray and neutron instrumentation.
https://www.jjxray.dk/about-us/

Nanotec Motors & Drives
https://www.xhuber.com/en/

ZIMM Group
https://zimm.com/en-us/

Sample Table – Column Guides Calculations

Linear bushings

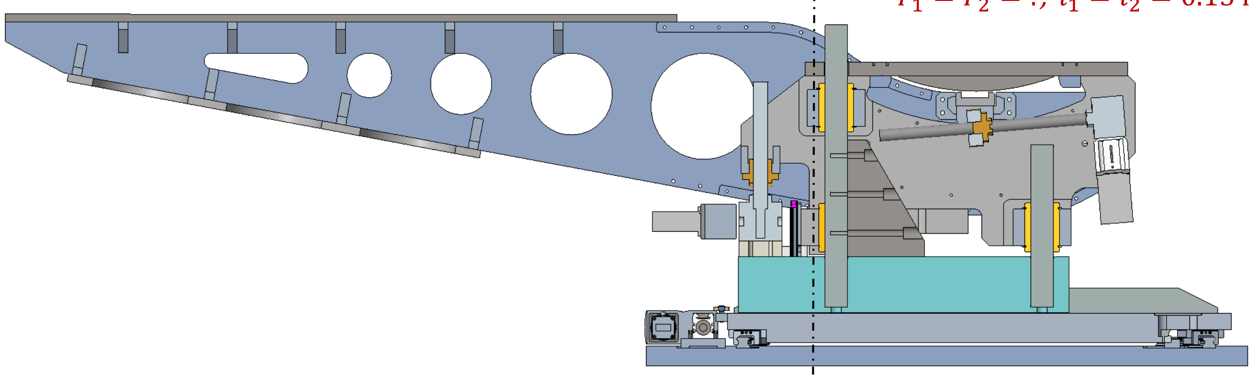
$F_{PL,DA} = 3000\ N,\ l_{PL,DA} = 1.941\ m$ Payload detector arm (R127: ≧227kg)

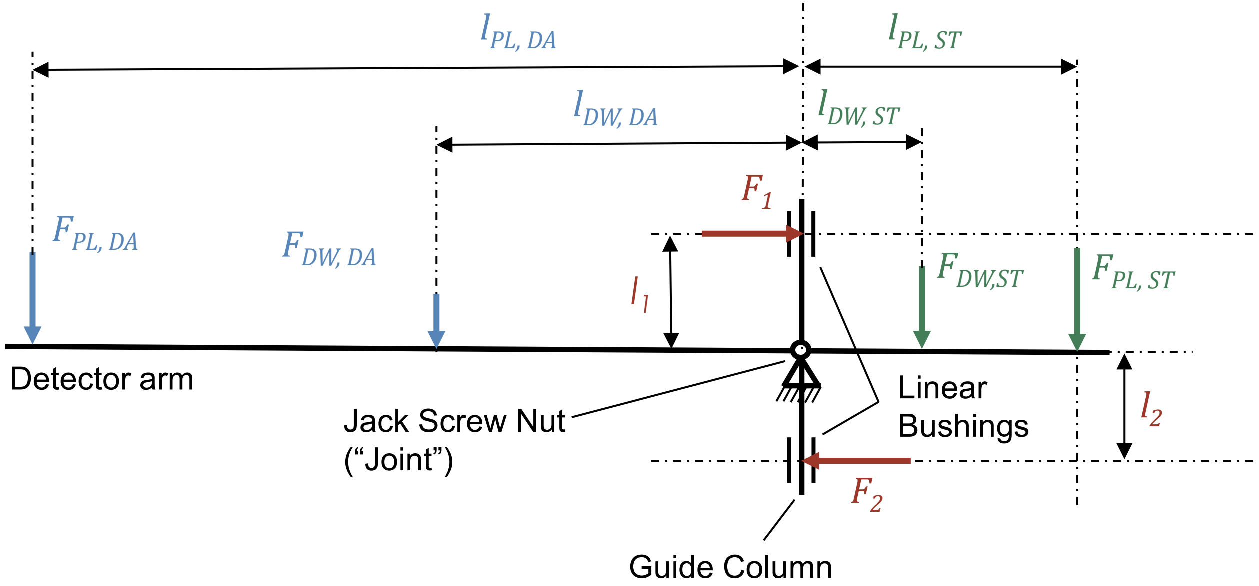
$F_{DW,DA} = 1455\ N,\ l_{DW,DA} = 0.509\ m$ Dead weight force detector arm

$F_{PL,ST} = 5000\ N,\ l_{PL,ST} = 0.555\ m$ Payload sample table (R117: ≧455kg)

$F_{DW,ST} = 1800\ N,\ l_{DW,ST} = 0.486\ m$ Dead weight force sample table (moving part)

$F_1 = F_2 = ?,\ l_1 = l_2 = 0.154\ m$ Reaction forces upper and lower bushing





To find the two forces F1 and F2, we need the following system of equations:

$$F_{PL,DA} \cdot l_{PL,DA} + F_{DW,DA} \cdot l_{DW,DA} - F_{PL,ST} \cdot l_{PL,ST} - F_{DW,ST} \cdot l_{DW,ST}$$

$$-F_1 \cdot l_1 - F_2 \cdot l_2 = 0 \qquad (1)$$

$$\frac{F_1}{F_2} = \frac{l_2}{l_1} \qquad (2)$$

We have an even force distribution on the ball bushings if $l_1 = l_2$, thus $F_1 = F_2$. In this case, we simply obtain

$$F_1 = F_2 = \frac{F_{PL,DA} \cdot l_{PL,DA} + F_{DW,DA} \cdot l_{DW,DA} - F_{PL,ST} \cdot l_{PL,ST} - F_{DW,ST} \cdot l_{DW,ST}}{2 \cdot l_1}$$

$\underline{F_1 = 9.5\ kN}$

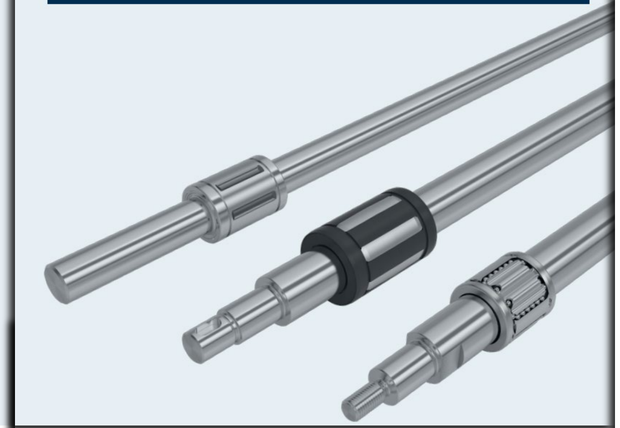
The manufacturer BOSCH REXROTH specifies a dynamic and static load rating of **23.5 kN** and **18.700 kN** respectively.

The dynamic load rating of 23.5 kN is based on a travel distance of **100 km** (see next slide).

Sample Table – Column Guides Calculations

rexroth
A Bosch Company

Linear bushings



The benefits

- High-precision linear bushing for moving heavy loads
- Steel bearing plates with machined ball guide grooves and machined back for high precision
- Super H with more tracks than super A
- Super SH with even more tracks for maximum load rating and rigidity
- High travel speed (up to 5 m/s)
- Compensates for shaft deflection and misalignment
- Optional integrated wiper seals
- Optional linear seal for open linear bushing
- Linear sets with aluminum housing

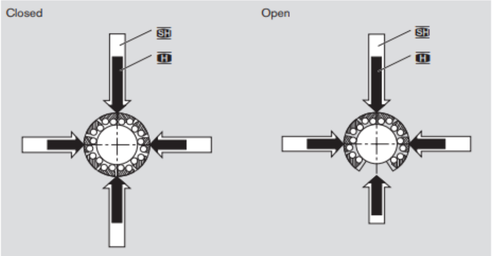
R0732 page 84
Super H (closed)

R0733 page 84
Super H (open)

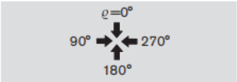
R0730 page 86
Super SH (closed)

Super linear bushings SH

Effect of load direction on load rating



Main load directions



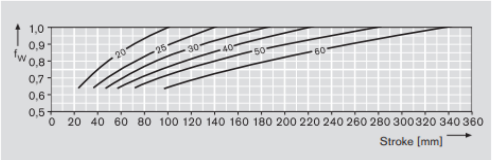
Load direction factors

The load ratings C and C_0 apply for the load direction $\varrho = 0°$. For all other load directions, the load ratings must be multiplied by the factors f_ϱ (dynamic load rating) or $f_{\varrho 0}$ (static load rating C_0).

Shaft Ø d (mm)	Load direction factor f_ϱ											
	Super linear bushing H						Super linear bushing SH					
	↓	→ ←	↑	↓	→ ←	↑	↓	→ ←	↑	↓	→ ←	↑
20–25	1	0.80	0.98	1	0.80	0.67	1	0.79	1	1	0.79	0.52
30–60	1	0.70	0.91	1	0.70	0.62	1	0.86	1	1	0.86	0.59
	Load direction factor $f_{\varrho 0}$											
20–25	1	0.70	0.87	1	0.70	0.67	1	0.68	1	1	0.68	0.50
30–60	1	0.62	0.80	1	0.62	0.61	1	0.83	1	1	0.83	0.55

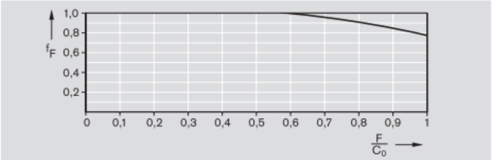
Reduced load rating with short stroke

When short stroke is present, the service life of the shaft is less than that of the super linear bushing. The load ratings C in the tables must therefore be multiplied by the factor f_W.



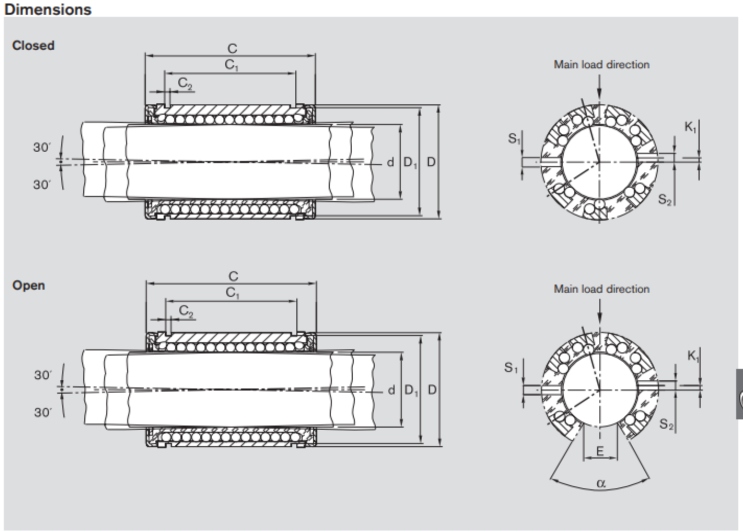
Reduced load rating with heavy load

The load rating is reduced under heavy load F. The dynamic load rating must be multiplied by the load factor f_F.



Super linear bushings SH

Dimensions



Dimensions (mm) Ø d	D	C h13	C_1 H13	C_2	D_1	S_1[1] +0.1	S_2[1] +0.1	K_1	E[2]	Rows of balls ◯	Rows of balls (open)	Angle α (°)	Radial clearance (µm) Shaft/bore h6/H7	h6/K7	h6/M7	Load ratings[2] (N) dyn. C	stat. C_0
20	32	45	31.2	1.6	30.5	3.0	–	–	9.5	7	6	60	+43 +11	+25 −7	+18 −14	2,520	1,880
25	40	58	43.7	1.85	38.5	3.5	3	−1.5	12.0	7	6	60	+43 +11	+25 −7	+18 −14	4,430	3,360
30	47	68	51.7	1.85	44.5	3.5	3	2.0	12.8	7	6	60	+43 +11	+25 −7	+18 −14	6,300	5,230
40	62	80	60.3	2.15	59.0	3.5	3	1.5	16.8	7	6	60	+50 +12	+29 −7	+20 −18	9,680	7,600
50	75	100	77.3	2.65	72.0	4.5	5	2.5	22.1	7	6	60	+50 +12	+29 −7	+20 −18	16,000	12,200
60	90	125	101.3	3.15	86.5	6.0	–	–	27.0	7	6	60	+56 +14	+31 −11	+21 −21	23,500	18,700

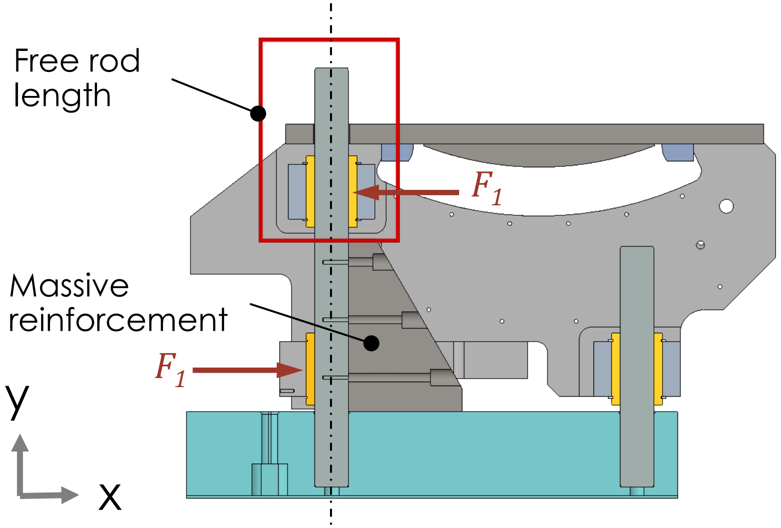
1) Holes at center of dimension C
2) Minimum size in relation to Ø d
3) The load ratings apply for the main load direction

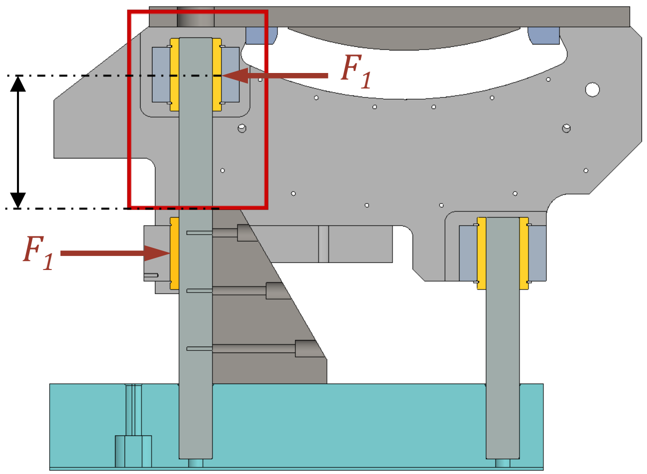
The dynamic load ratings are based on a total travel of 100,000 m.
When based on 50,000 m, the C values in the table are multiplied by 1.26.

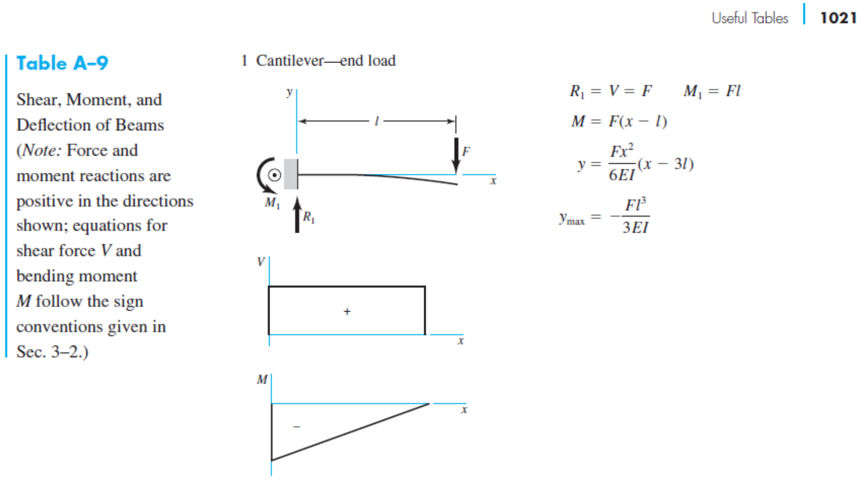
⚠ Refer to the diagrams on page 78 for load in the direction of opening.

Sample Table – Column Guides Calculations

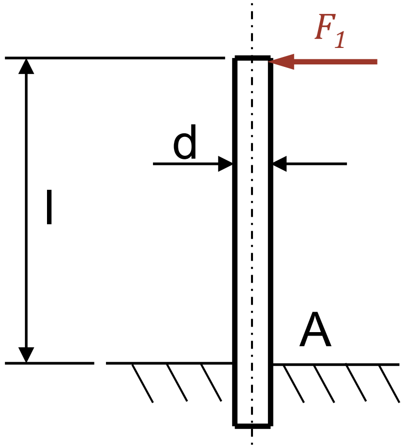
Column Guide Deflection







J. K. Nisbett, R.G. Budynas (2015). Mechanical Engineering Design (10th ed.). McGraw-Hill, p.1021



From the previous calculation we have the force $F_1 = 9500\ N$

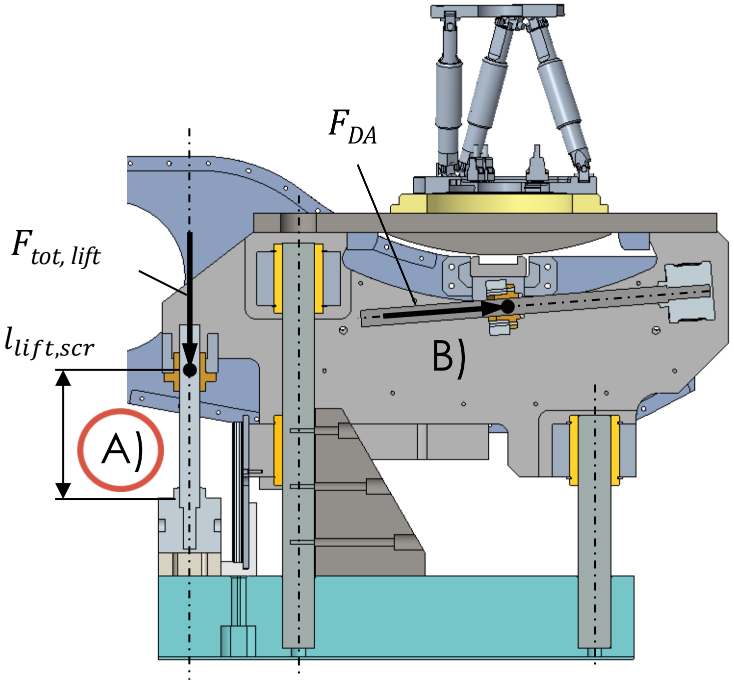
$l = 172\ mm$	Free rod length
$d = 60\ mm$	Rod diameter
$E = 210000\ N/mm^2$	E – Modulus
$F_1 = 9500\ N$	Force on bushing

Using the second moment of inertia $I = \pi \frac{d^4}{64}$, we obtain

$$x_{max} = \frac{F_1 \cdot l^3}{3 \cdot E \cdot I} = \frac{9500\ N \cdot (172\ mm)^3}{3 \cdot 210000 \frac{N}{mm^2} \cdot \pi \frac{(60\ mm)^4}{64}} = \underline{0.12\ mm}$$

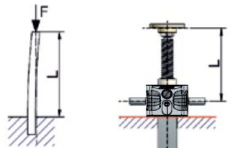
Sample Table – Lifting Stage Actuator Calculations

Critical buckling force of the screws



ZIMM Product Catalogue
2021 p.162

Conservative assumption: unguided case



$I = \frac{F \times v \times (L \times 2)^2}{\pi^2 \times E}$ then $d = \sqrt[4]{\frac{I \times 64}{\pi}}$

The ZIMM Group's various lifting screw sizes have been designed for different load ranges. For a given spindle length and load, the appropriate spindle diameter can be determined using the ZIMM calculation sheet.

A) Lift Screw

$F_{PL,DA} = 3000\ N$	Payload detector arm	(R127: ≧227kg)
$F_{DW,DA} = 1455\ N$	Dead weight detector arm	
$F_{PL,ST} = 5000\ N$	Payload sample table	(R117: ≧ 455 kg)
$F_{DW,ST} = 1800\ N$	Dead weight sample table (moving part)	
$F_{tot,\,lift} = F_{PL,DA} + F_{DW,DA} + F_{PL,ST} + F_{DW,ST} = 11255\ N$	Total weight force acting on vertical spindle	
$l_{lift,\,scr} = 225\ mm$	Free screw length	
$E = 210000\ N/mm^2$	E – modulus	
$v = 3$	Safety factor	

Thus, we obtain for the 2nd moment of inertia

$$I = \frac{F_{tot,\,lift} \cdot v \cdot 4 \cdot l_{lift,\,scr}{}^2}{\pi^2 \cdot E} = \frac{11255\ N \cdot 3 \cdot 4 \cdot (225\ mm)^2}{\pi^2 \cdot 210000\ N/mm^2} = 3299\ mm^4$$

and the minimum core diameter of the screw

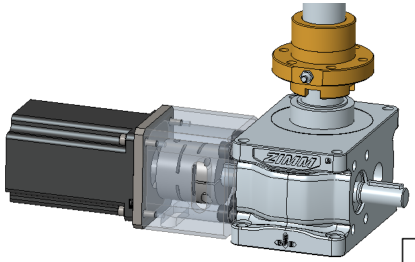
$$d = \sqrt[4]{\frac{I \cdot 64}{\pi}} = \sqrt[4]{\frac{3299\ mm^4 \cdot 64}{\pi}} = \underline{16.1\ mm}$$

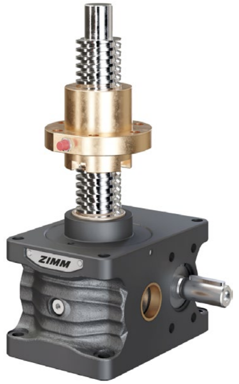
Screw Jack Series	GSZ-2	ZE-5	ZE-10	ZE-25	ZE-35/50
Trapezoidal screw Tr	16x4	18x4	20x4	30x6	40x7
Core-Ø in mm (minimum)	10,9	12,9	14,9	22,1	31
Ball screw KGT Ømm	16	16	25	32	40
Core-Ø in mm (minimum*)	12,9	12,9	21,5	27,3	34,1

Sample Table – Lifting Stage Actuator Calculations

ZIMM jack screw systems

35kN-40×7-R-Trapezoidal Screw





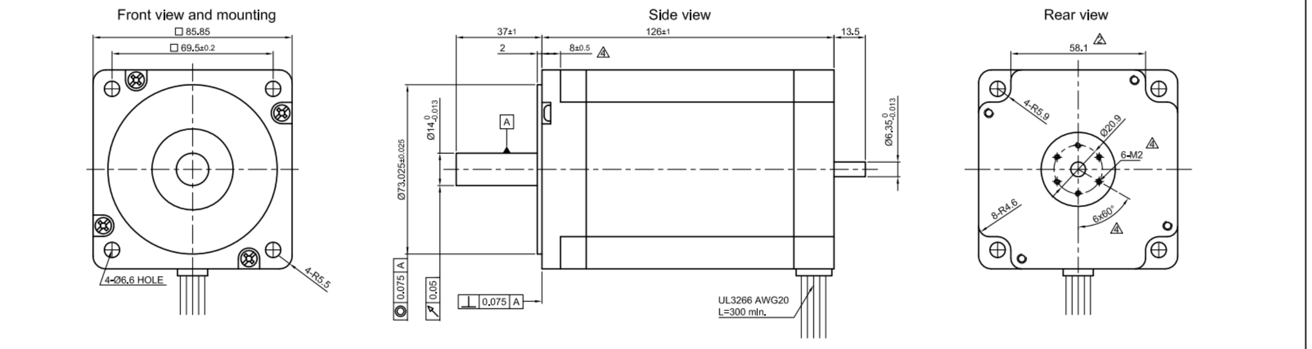
Size:	35 kN
Nominal speed:	1500 rpm
Max. drive shaft speed:	1800 rpm
Screw size standard:	40x7
Housing material:	GGG-50, corrosion-resistant
Worm shaft:	Steel, case-hardened, ground
Weight of screw jack body:	95 kg
Weight of screw/m:	8 kg
Gearbox lubrication:	Synthetic fluid grease
Screw lubrication:	Grease lubrication
Gearbox operating temperature:	max. 60°, higher on request
Moment of inertia:	N: 0.97 kg cm² / L: 0.67 kg cm²
Input torque (at 1500 rpm):	max. 19.8 Nm (N) / max. 9 Nm (L)
Drive-through torque:	max. 130 Nm
Screw:	Rotating (R)

Standard configuration

Code	Gearbox (series)	Size	Version (variant)	Ratio	Screw	Stroke per drive shaft rotation
ZE-35-RN	ZE	35	R (rotating screw)	N (normal) 7:1	Tr 40x7	1,00 mm
ZE-35-RL				L (low) 28:1		0,25 mm

Nanotec hybrid stepper motor

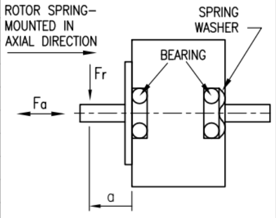
ST8918L6708-B – STEPPER MOTOR – NEMA 34



CONNECTION / SPECIFICATION	UNIPOLAR OR BIPOLAR–1 WINDING	BIPOLAR SERIAL	BIPOLAR PARALLEL
VOLTAGE (VDC)	3.6		
AMPS/PHASE	6.7	4.7	9.5
RESISTANCE/PHASE (Ohms)@25°C ⚠3	0.54±15%	1.08±15%	0.27±15%
INDUCTANCE/PHASE (mH) @1KHz	2.7±20%	10.8±20%	2.7±20%
HOLDING TORQUE (Nm) [lb–in]	6.6 [58.41]	9.33 [82.57]	9.33 [82.57]
DETENT TORQUE (Nm) [lb–in]	0.20 [1.77]		
STEP ANGLE (°)	1.8 ⚠5		
STEP ACCURACY (NON–ACCUM)	± 5% ⚠5 ⚠1		
ROTOR INERTIA (Kg–m²) [lb–in²]	$3.0x10^{-4}$ [1.025]		
WEIGHT (Kg) [lb]	3.95 [8.71]		

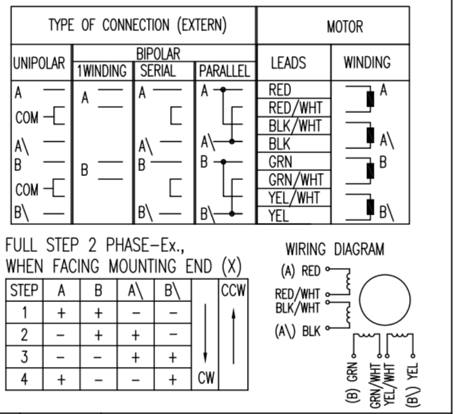
TEMPERATURE RISE: MAX.80°C (MOTOR STANDSTILL; FOR 2 PHASE ENERGIZED)
AMBIENT TEMPERATURE –20°~ 50°C [–4°F ~ 122°F]
INSULATION RESISTANCE 100 MOhm (UNDER NORMAL TEMPERATURE AND HUMIDITY)
INSULATION CLASS B 130° [266°F] ⚠5
DIELECTRIC STRENGTH 500VAC FOR 1 MIN. (BETWEEN THE MOTOR COILS AND THE MOTOR CASE)
AMBIENT HUMIDITY MAX. 85% (NO CONDENSATION)

PERMISSIBLE RADIAL+AXIAL FORCE



AXIAL–FORCE Fa (N)	Fa=65			
DISTANCE a (mm)	5	10	15	20
RADIAL–FORCE Fr (N)	535	355	256	200

	AXIAL	RADIAL
SHAFT PLAY (mm)	0.075	0.025
AT LOAD MAX: (N)	10	5.0



FULL STEP 2 PHASE–Ex., WHEN FACING MOUNTING END (X)

STEP	A	B	A\	B\	CW	CCW
1	+	+	–	–	↓	↑
2	–	+	+	–		
3	–	–	+	+		
4	+	–	–	+		

REV	DESCRIPTION	DATE	APVD
5	REMOVE TEXT	03.03.15	A.S.
4	NEW THREAD/THICKNESS OF FLANGE	25.07.13	A.S.
3	VALUE OF RESISTANCE	11.11.10	J.W.

Nanotec® PLUG & DRIVE
ST8918L6708–B

SCALE FREE
X ±0.5
1PL ±0.2
2PL ±0.1
ANGLE ±30'

	SIGNATURE	DATE
APVD	S.Ha.	09.01.07
CHKD		
DRN	J.W.	13.06.06

STEPPING MOTOR
DWG.NO ST8918L6708–B

Sample Table – Lifting Stage Actuator Calculations

Necessary drive torque

ZIMM 35kN-40×7-R-Trapezoidal Screw

$F_{tot,lift} = F_{PL,DA} + F_{DW,DA} + F_{PL,ST} + F_{DW,ST} = 11255\ N$	Lifting Load (max. payloads + dead weights)
$p = 7\ mm$	Screw pitch
$\eta_{gear} = 0.52$	Gearbox efficiency (worm gear screw jack)
$\eta_{screw} = 0.35$	Screw efficiency
$i = 28$	Gearbox ratio
$M_{G,35} = ?$	Necessary drive torque

$$M_{G,35} = \frac{F_{tot} \cdot p}{2 \cdot \pi \cdot \eta_{gear} \cdot \eta_{screw} \cdot i} = \frac{11255\ N \cdot 0.007\ m}{2 \cdot \pi \cdot 0.52 \cdot 0.35 \cdot 28} = \underline{2.5\ Nm}$$

Jack screw system efficiencies

Efficiencies of the screw jack $\eta_{Gearbox}$ (without screw)

i	rpm	GSZ-2	ZE-5	ZE-10	ZE-25	ZE-35
N	3000	0,87	0,81	0,83	0,87	-
N	1500	0,87	0,82	0,84	0,87	0,87
N	1000	0,86	0,82	0,82	0,86	0,87
N	750	0,86	0,82	0,84	0,85	0,86
N	500	0,85	0,82	0,84	0,83	0,85
N	100	0,74	0,77	0,79	0,78	0,78
L	3000	0,78	0,74	0,78	0,76	-
L	1500	0,77	0,70	0,74	0,72	0,64
L	1000	0,75	0,67	0,72	0,7	0,64
L	750	0,74	0,65	0,7	0,68	0,64
L	500	0,71	0,62	0,67	0,65	0,63
L	100	0,54	0,53	0,59	0,54	0,52

Efficiencies of the screws η_{Screw}

Tr-screw, single-pitch	16x4	18x4	20x4	30x6	40x7	50x8
Efficiency	0,45	0,42	0,39	0,39	0,35	0,33
Tr-screw, double-pitch	16x8P4	18x8P4	20x8P4	30x12P6	40x14P7	50x16P8
Efficiency	0,62	0,59	0,56	0,56	0,53	0,50

Torque, resolution and speed

Motor: Nanotec ST8918L6708-B (Stepper motor)

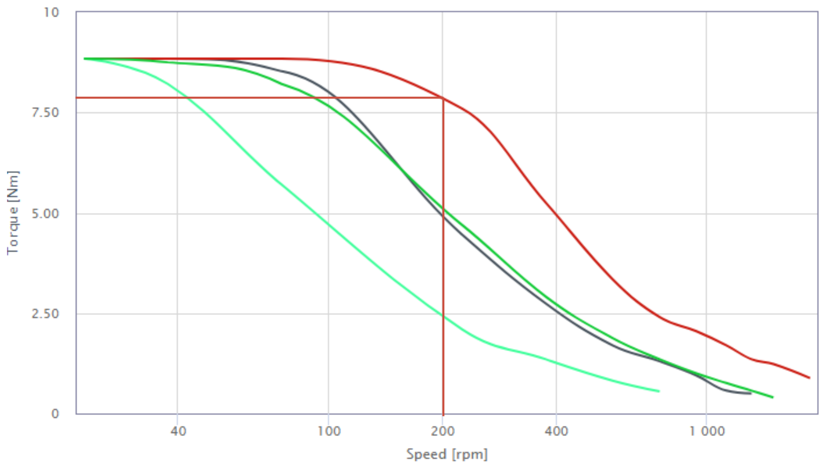
Selection: *Stepper motor with 8 Nm torque at 200 rpm* $(3.2 \cdot M_{G,35})$

Resolution: *Stroke per drive shaft rotation* $= 0.25\ mm$

$$Stroke\ per\ step\ (1.8°) = \frac{0.25\ mm}{200} = \underline{0.00125\ mm}$$ (R118: ± 0.1mm)

Speed: $$Duration\ for\ the\ whole\ stroke = \frac{150\ mm}{200\ rpm \cdot 0.25\ mm} = \underline{3\ min}$$

Torque curve ST8918L6708-B

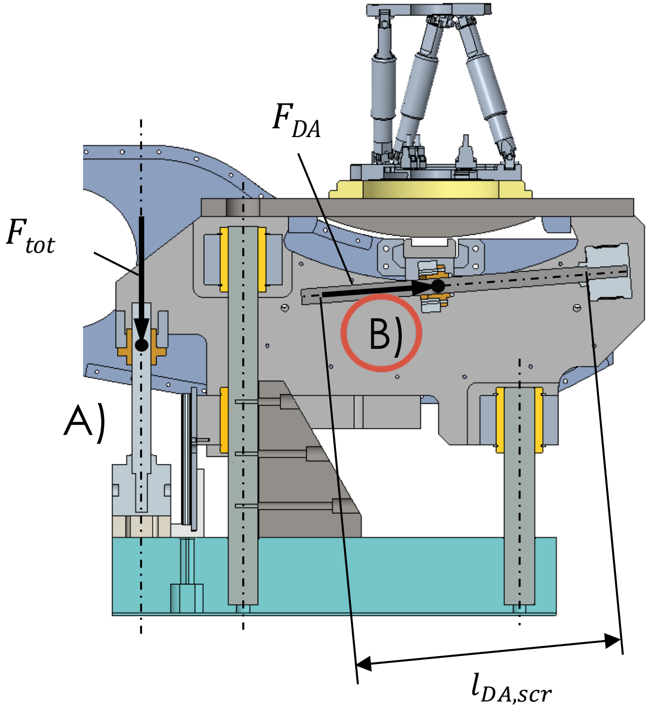


Encoder: Renishaw Fortis-NML170 (Enclosed linear optical encoder)

Resolution: $\pm 5\ \mu m$

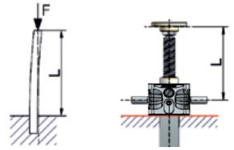
Detector Arm – Actuator Calculations

Critical buckling force of the screws



ZIMM Product Catalogue 2021 p.162

Conservative assumption: unguided case



$I = \frac{F \times v \times (L \times 2)^2}{\pi^2 \times E}$ then $d = \sqrt[4]{\frac{I \times 64}{\pi}}$

B) Detector Arm Screw

$l_{DA,scr} = 471\ mm$	Free screw length	
$E = 210000\ N/mm^4$	E – modulus	
$v = 2$	Safety factor	
$F_{PL,DA} = 3000\ N,\ l_{PL,DA} = 2500\ mm$	Payload detector arm	(R127: ≧227kg)
$F_{DW,DA} = 1455\ N,\ l_{DW,DA} = 1064\ mm$	Dead weight detector arm	

Force F_{DA} , the screw nut reaction force is:

$$F_{PL,DA} \cdot l_{PL,DA} + F_{DW,DA} \cdot l_{DW,DA} = F_{DA} \cdot l_{3,DA}$$

$$F_{DA} = \frac{F_{PL,DA} \cdot l_{2,DA} + F_{DW,DA} \cdot l_{1,DA}}{l_{3,DA}}$$

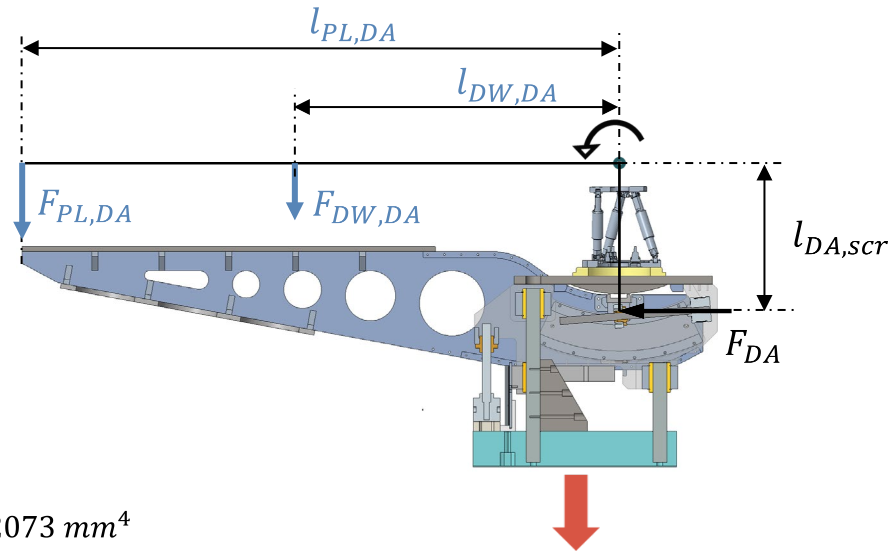
$$= \frac{3000\ N \cdot 2.5\ m + 1455\ N\ \cdot 1.064\ m}{0.643\ m} \cong 14.1\ kN$$

Thus, we obtain for the 2nd moment of inertia

$$I = \frac{F_{DA} \cdot v \cdot 4 \cdot l_{DA,scr}{}^2}{\pi^2 \cdot E} = \frac{14100\ N \cdot 2 \cdot 4\ (471\ mm)^2}{\pi^2 \cdot E} = 12073\ mm^4$$

and the minimum core diameter of the screw

$$d = \sqrt[4]{\frac{I \cdot 64}{\pi}} = \sqrt[4]{\frac{12073\ mm^4 \cdot 64}{\pi}} = \underline{22.3\ mm}$$

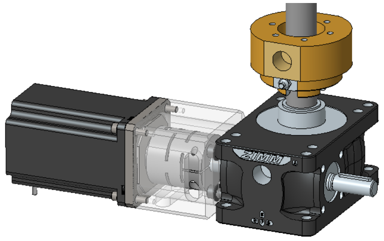


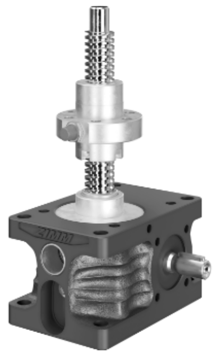
Screw Jack Series	GSZ-2	ZE-5	ZE-10	ZE-25	ZE-35/50
Trapezoidal screw Tr	16x4	18x4	20x4	30x6	40x7
Core-Ø in mm (minimum)	10,9	12,9	14,9	22,1	31
Ball screw KGT Ømm	16	16	25	32	40
Core-Ø in mm (minimum*)	12,9	12,9	21,5	27,3	34,1

Detector Arm – Actuator Calculations

ZIMM jack screw systems

25kN-30×6-R-Trapezoidal Screw





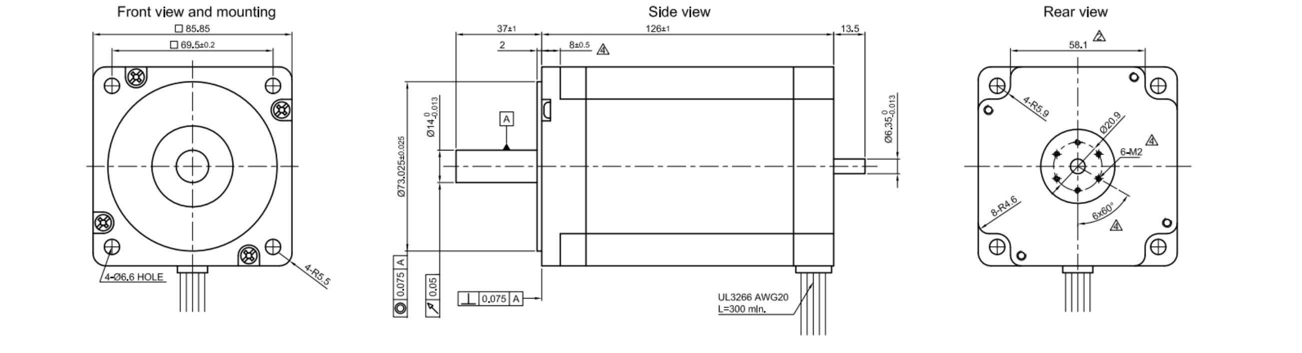
Size:	25 kN
Nominal speed:	1500 rpm
Max. drive shaft speed:	3000 rpm
Screw size standard:	30x6
Housing material:	Aluminium, corrosion-resistant
Worm shaft:	Steel, case-hardened, ground
Weight of screw jack body:	38 kg
Weight of screw/m:	45 kg
Gearbox lubrication:	Synthetic fluid grease
Screw lubrication:	Grease lubrication
Gearbox operating temperature:	max. 60°, higher on request
Moment of inertia:	N: 0.667 kg cm² / L: 0.443 kg cm²
Input torque (at 1500 rpm):	max. 18 Nm (N) / max. 10 Nm (L)
Drive-through torque:	max. 108 Nm
Screw:	Rotating (R)

Standard configuration

Code	Gearbox (series)	Size	Version (variant)	Ratio	Screw	Stroke per drive shaft rotation
ZE-25-RN	ZE	25	R (rotating screw)	N (normal) 6:1	Tr 30x6	1,00 mm
ZE-25-RL				L (low) 24:1		0,25 mm

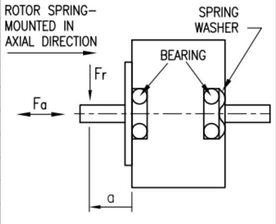
Nanotec hybrid stepper motor

ST8918L6708-B – STEPPER MOTOR – NEMA 34



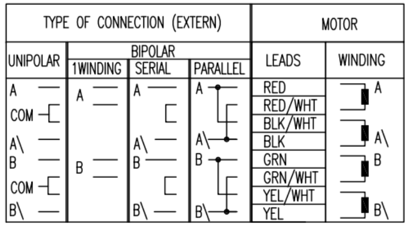
CONNECTION / SPECIFICATION	UNIPOLAR OR BIPOLAR–1 WINDING	BIPOLAR SERIAL	BIPOLAR PARALLEL
VOLTAGE (VDC)	3.6		
AMPS/PHASE	6.7	4.7	9.5
RESISTANCE/PHASE (Ohms)@25°C ⚠3	0.54±15%	1.08±15%	0.27±15%
INDUCTANCE/PHASE (mH) @1KHz	2.7±20%	10.8±20%	2.7±20%
HOLDING TORQUE (Nm) [lb–in]	6.6 [58.41]	9.33 [82.57]	9.33 [82.57]
DETENT TORQUE (Nm) [lb–in]	0.20 [1.77]		
STEP ANGLE (°)	1.8 ⚠5		
STEP ACCURACY (NON–ACCUM)	± 5% ⚠5 ⚠1		
ROTOR INERTIA (Kg–m²) [lb–in²]	3.0x10⁻⁴ [1.025]		
WEIGHT (Kg) [lb]	3.95 [8.71]		
TEMPERATURE RISE: MAX.80°C (MOTOR STANDSTILL; FOR 2 PHASE ENERGIZED)			
AMBIENT TEMPERATURE –20°~ 50°C [–4°F ~ 122°F]			
INSULATION RESISTANCE 100 MOhm (UNDER NORMAL TEMPERATURE AND HUMIDITY)			
INSULATION CLASS B 130° [266°F] ⚠5			
DIELECTRIC STRENGTH 500VAC FOR 1 MIN. (BETWEEN THE MOTOR COILS AND THE MOTOR CASE)			
AMBIENT HUMIDITY MAX. 85% (NO CONDENSATION)			

PERMISSIBLE RADIAL+AXIAL FORCE



AXIAL–FORCE Fa (N)	Fa=65			
DISTANCE a (mm)	5	10	15	20
RADIAL–FORCE Fr (N)	535	355	256	200

	AXIAL	RADIAL
SHAFT PLAY (mm)	0.075	0.025
AT LOAD MAX: (N)	10	5.0

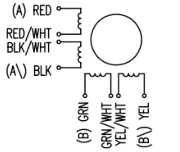


FULL STEP 2 PHASE–Ex., WHEN FACING MOUNTING END (X)

STEP	A	B	A\	B\
1	+	+	–	–
2	–	+	+	–
3	–	–	+	+
4	+	–	–	+

CW ↓ / CCW ↑

WIRING DIAGRAM



REV	DESCRIPTION	DATE	APVD
5	REMOVE TEXT	03.03.15	A.S.
4	NEW THREAD/THICKNESS OF FLANGE	25.07.13	A.S.
3	VALUE OF RESISTANCE	11.11.10	J.W.

Nanotec® PLUG & DRIVE
ST8918L6708–B

SCALE FREE		APVD	S.Ha.	09.01.07
X	±0.5	CHKD		
1PL	±0.2	DRN	J.W.	13.06.06
2PL	±0.1		SIGNATURE	DATE
ANGLE	±30'			

STEPPING MOTOR
DWG.NO ST8918L6708–B

OAK RIDGE National Laboratory

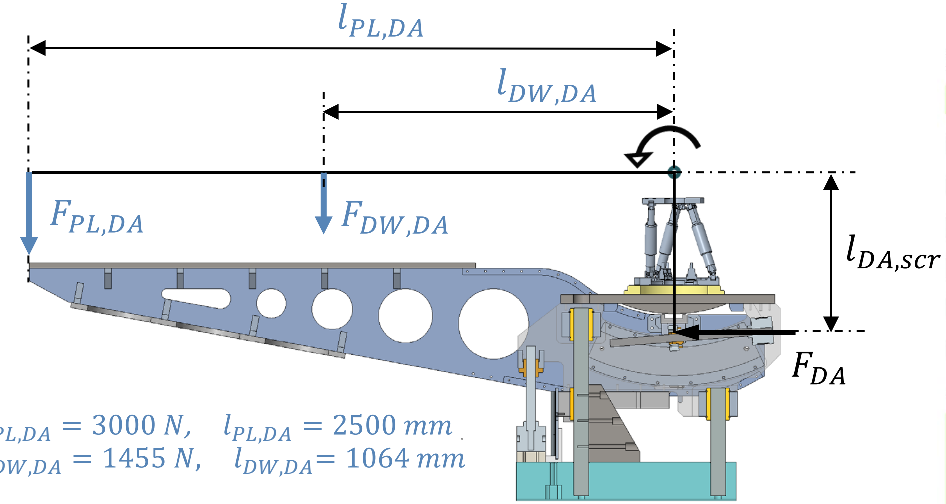
Detector Arm – Actuator Calculations

Necessary drive torque

ZIMM 25kN-30×6-R-Trapezoidal Screw

$F_{DA} = 14.1\ kN$	Lifting Load (max. payloads + dead weights)
$l_{DA,scr} = 643\ mm$	Arm length curved guides
$p = 6\ mm$	Screw pitch
$\eta_{gear} = 0.54$	Gearbox efficiency (worm gear screw jack)
$\eta_{screw} = 0.39$	Screw efficiency
$i = 24$	Gearbox ratio
$M_{G,35} = ?$	Necessary drive torque

$$M_{G,35} = \frac{F_{DA} \cdot p}{2 \cdot \pi \cdot \eta_{gear} \cdot \eta_{screw} \cdot i} = \frac{14\,100\ N \cdot 0.006\ m}{2 \cdot \pi \cdot 0.54 \cdot 0.39 \cdot 24} = \underline{2.7\ Nm}$$



$F_{PL,DA} = 3000\ N,\quad l_{PL,DA} = 2500\ mm$
$F_{DW,DA} = 1455\ N,\quad l_{DW,DA} = 1064\ mm$

Jack screw system efficiencies

Efficiencies of the screw jack $\eta_{Gearbox}$ (without screw)

i	rpm	GSZ-2	ZE-5	ZE-10	ZE-25	ZE-35
N	3000	0,87	0,81	0,83	0,87	-
N	1500	0,87	0,82	0,84	0,87	0,87
N	1000	0,86	0,82	0,82	0,86	0,87
N	750	0,86	0,82	0,84	0,85	0,86
N	500	0,85	0,82	0,84	0,83	0,85
N	100	0,74	0,77	0,79	0,78	0,78
L	3000	0,78	0,74	0,78	0,76	-
L	1500	0,77	0,70	0,74	0,72	0,64
L	1000	0,75	0,67	0,72	0,7	0,64
L	750	0,74	0,65	0,7	0,68	0,64
L	500	0,71	0,62	0,67	0,65	0,63
L	100	0,54	0,53	0,59	0,54	0,52

Efficiencies of the screws η_{Screw}

Tr-screw, single-pitch	16x4	18x4	20x4	30x6	40x7	50x8
Efficiency	0,45	0,42	0,39	0,39	0,35	0,33
Tr-screw, double-pitch	16x8P4	18x8P4	20x8P4	30x12P6	40x14P7	50x16P8
Efficiency	0,62	0,59	0,56	0,56	0,53	0,50

Torque, resolution and speed

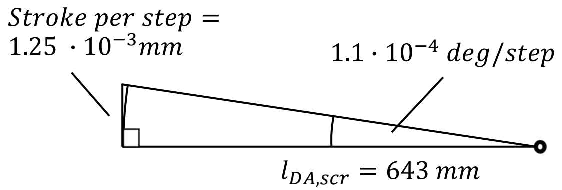
Stepper motor: Nanotec ST8918L6708-B

Selection: *Stepper motor with 8 Nm torque at 200 rpm* $(3 \cdot M_{G,35})$

Resolution: *Stroke per drive shaft rotation* $= 0.25\ mm$

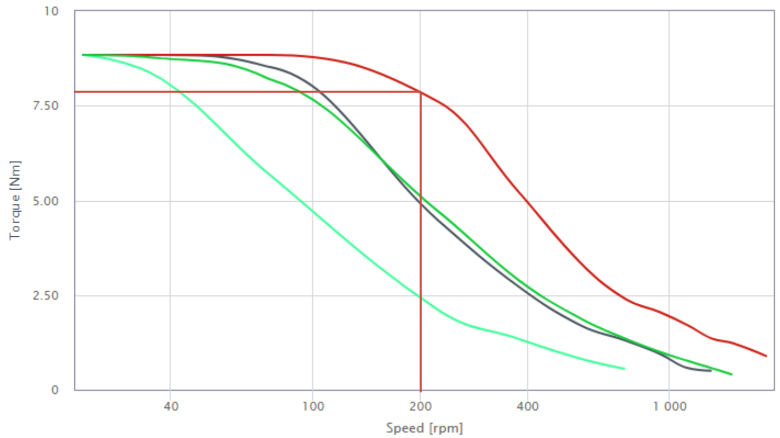
Stroke per step $(1.8°) = \frac{0.25\ mm}{200} = 0.00125\ mm$

Angular Resolution $\approx \text{asin}\frac{1.25 \cdot 10^{-3} mm}{643 mm} = \underline{1.1 \cdot 10^{-4}\ deg}$ (R130: $2 \cdot 10^{-3}\ deg$)



Speed: *Duration for the whole stroke* $= \frac{450\ mm}{200\ rpm \cdot 0.25\ mm} = \underline{9\ min}$

Torque curve ST8918L6708-B

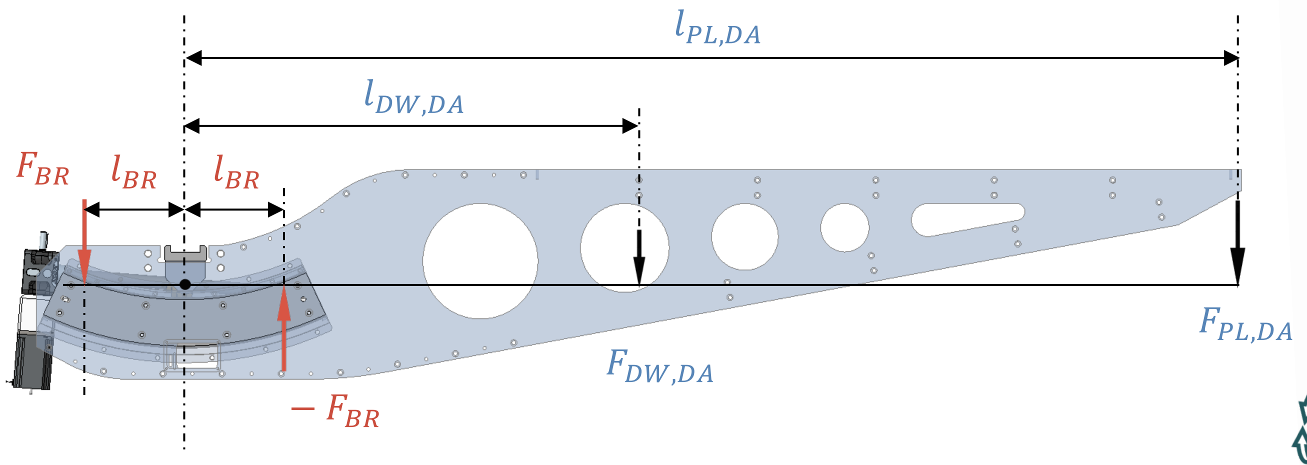


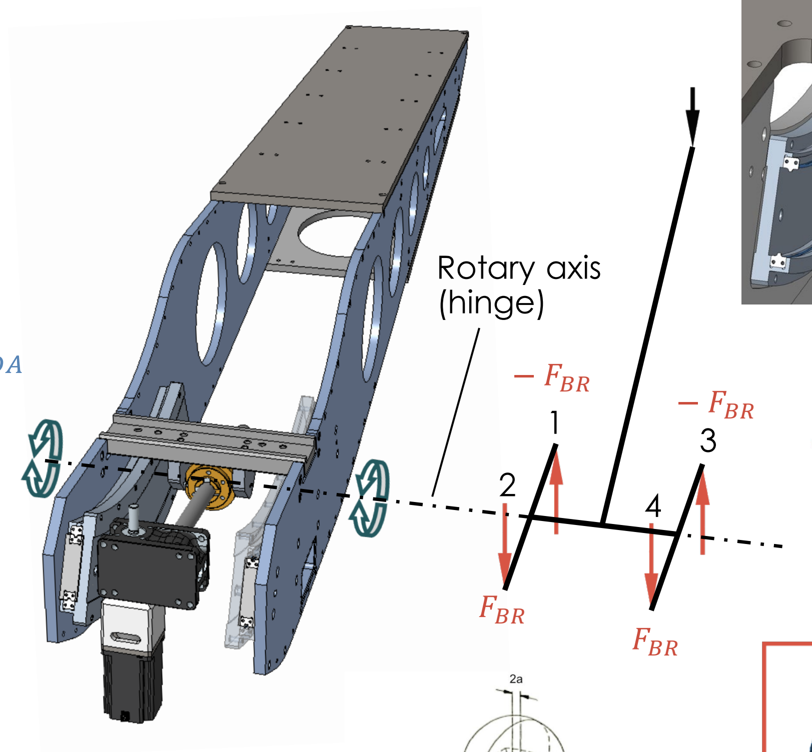
Encoder: Renishaw RKLA30-S RESOLUTE absolute encoder scale.

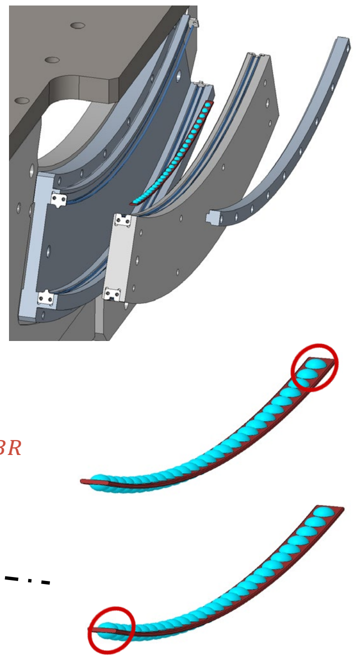
Resolution: *BiSS C*: $50\ nm$

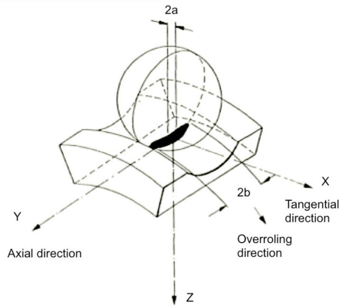
Engineering & Design: R. Thermer

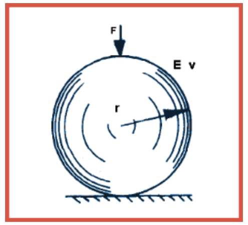
Sample Table – Pivot Bearing Calculation

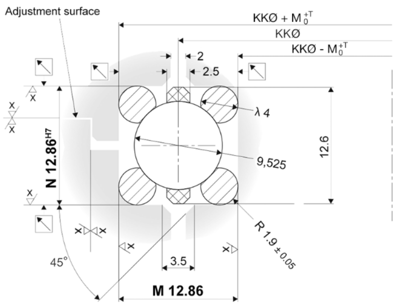












$F_{PL,DA} = 3000\ N,\ l_{PL,DA} = 2500\ mm$ — Payload detector arm

$F_{DW,DA} = 1455\ N,\ l_{DW,DA} = 1064\ mm$ — Dead weight detector arm

$l_{BR} = 150\ mm$ — Bearing Reaction

We find the Reaction force F_{BR}:

$$F_{PL,DA} \cdot l_{PL,DA} + F_{DW,DA} \cdot l_{DW,DA} = 4 \cdot F_{BR} \cdot l_{BR}$$

$$F_{BR} = \frac{F_{PL,DA} \cdot l_{PL,DA} + F_{DW,DA} \cdot l_{DW,DA}}{4 \cdot l_{BR}} =$$

$$= \frac{3000\ N \cdot 2.5m + 1455\ N \cdot 1.064m}{4 \cdot l_{BR}} = 15080.2\ N$$

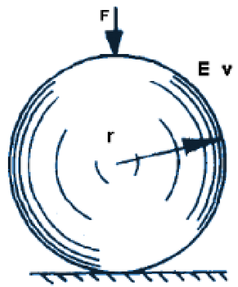
$$= 15.1\ kN$$

Conservative assumptions:

- The load will be fully absorbed by only 8 balls, 2 at each end of the curved rail. Each ball cage normally holds 27 balls.
- Bearing Ball is in contact wit a flat surface

Sample Table – Pivot Bearing Calculation

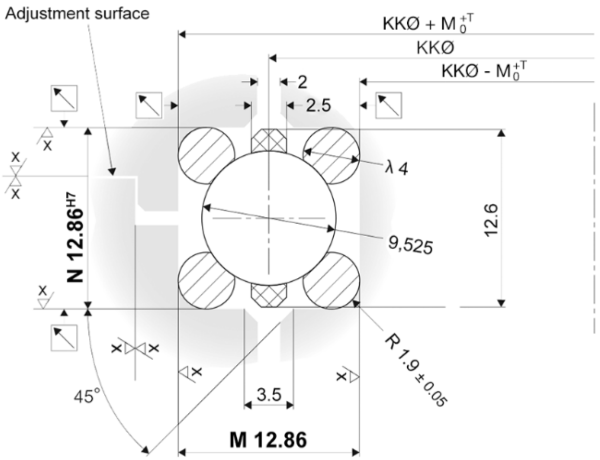
Contact Stress estimate
Sphere in Contact Flat Surface



Rolling bearing material **100Cr6**

$E = 208000\ N/mm^2$	E-Modulus
$F = \frac{1}{4} \cdot F_{BR} = 3775\ N$	Load (*)
$r \cong 4.8\ mm$	radius
$v = 0.3$	Poisson's ratio

FRANKE GmbH 4-point contact bearing element LEL 4



(*) We assume 2 points of contact per ball and with two balls these are 4 points of contact. The Hertzian contact stress equations deliver:

Contact area radius:

$$a = \sqrt[3]{\frac{6F(1-v^2)r}{E}} = \sqrt[3]{\frac{6 \cdot 3775N \cdot (1-0.3^2) \cdot 4.8mm}{208000N/mm^2}} = 0.78\ mm$$

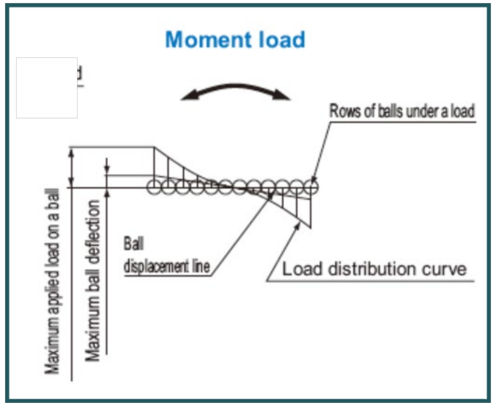
J. K. Nisbett, R.G. Budynas (2015). Mechanical Engineering Design (10th ed.). McGraw-Hill,

Contact Stress:

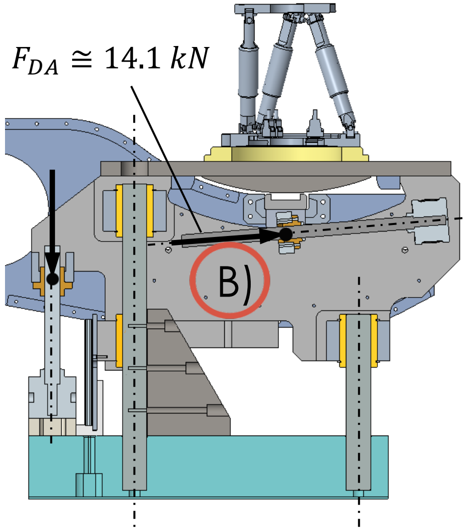
$$\sigma_c = \frac{3F}{2\pi a^2} = \frac{3 \cdot 3775N}{2 \cdot 3.1415 \cdot 0.78\ ^2 mm^2} = 2958 \frac{N}{mm^2}\ (MPa)\ \ to\ 4\ \ s.f.$$

Which is about **1.6** - **2.3 times more** than the yield strength of **100Cr6** (1300 - 1800 MPa).

Detailed clarification with Franke GmbH needed !



Detector Arm – Drive Link Calculations



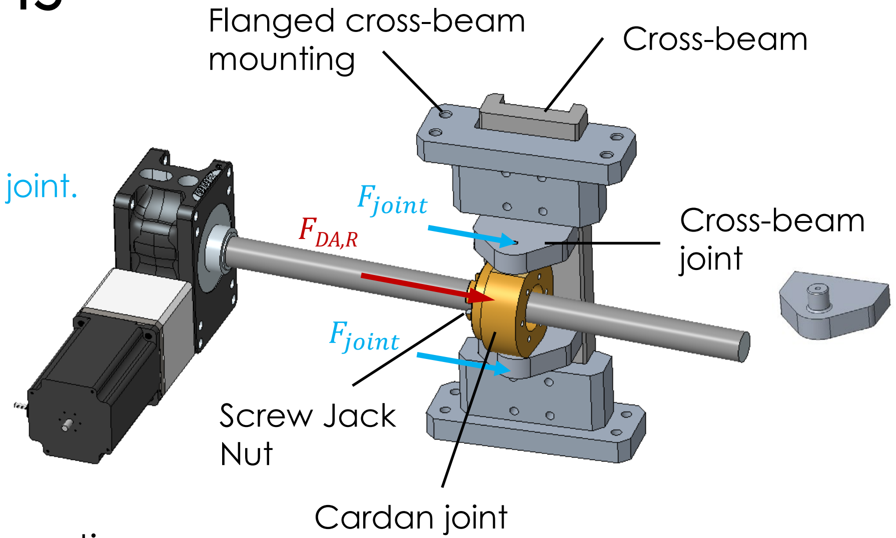
$F_{joint} = F_{DA}/2$ Force absorbed by one half of the joint.

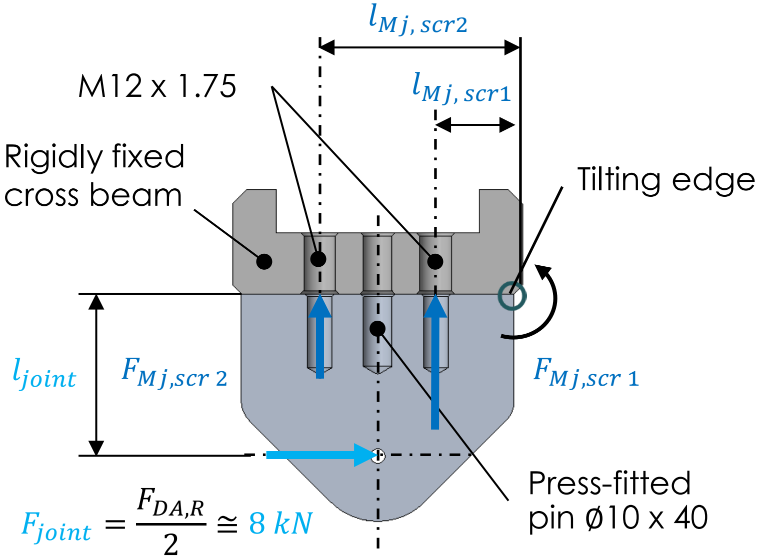
$l_{joint} := 56\ mm$ Lever length

$F_{JM,\ scr\ 1,2} :=\ ?$ Torque compensating bolt axial forces

$l_{JM,\ scr1} := 27\ mm$ Lever length

$l_{JM,\ scr2} := 67\ mm$ Lever length





$F_{joint} = \frac{F_{DA,R}}{2} \cong 8\ kN$

Axial bolt forces required for torque absorption:

$$F_{joint} \cdot l_{joint} - F_{JM,\ scr\ 1} \cdot l_{JM,\ scr1} - F_{JM,\ scr\ 2} \cdot l_{JM,\ scr2} = 0 \qquad (1)$$

$$\frac{F_{JM,scr\ 1}}{F_{JM,scr\ 2}} = \frac{l_{JM,\ scr2}}{l_{JM,\ scr1}} \qquad (2)$$

The torque is absorbed via the axial bolt forces, assuming a linear distribution of the bolt forces.

From eq. (2) follows

$$F_{JM,scr\ 1} = \frac{67}{27} \cdot F_{JM,scr\ 2} \qquad (3)$$

substituting this in eq. (1) gives

$$F_{joint} \cdot 56\ mm - \frac{67}{27} \cdot F_{JM\ scr\ 2} \cdot 27\ mm - F_{JM,\ scr\ 2} \cdot 67\ mm = 0$$

$$F_{joint} \cdot 56\ mm = 134\ mm \cdot F_{JM,\ scr\ 2}$$

$$F_{JM,\ scr\ 2} = \frac{28}{67} \cdot F_{joint} = \underline{3.3\ kN\ to\ 2\ s.f.} \qquad (4)$$

Detector Arm – Drive Link Calculations

Crossbeam Joint Calculation

Substituting the result of eq. (4) in eq. (3) gives

$$F_{JM,\,scr\,1} = \frac{67}{27} \cdot \frac{28}{67} \cdot F_{joint} = \frac{28}{27} \cdot F_{joint} = \underline{8.3\ kN\ \ to\ 2\ s.f.} \qquad (5)$$

Preload force for the shear force absorption:

With a static friction coefficient of $\mu_{st} = 0.15$ for steel on steel, we obtain the minimum clamping force $F_{Jsh,\,scr}$ per screw

$$2 \cdot F_{Jsh,\,scr} \cdot \mu_{St} \geq F_{DA,R}/2$$

$$F_{Jsh,\,scr} \geq \frac{F_{DA,R}}{2 \cdot \mu_{St}} = \frac{8\ kN}{2 \cdot 0.15} = \underline{27\ kN\ to\ 2\ s.f.} \qquad (6)$$

Total required clamping force per screw with safety factor 1.5:

$$F_{scr,joint} = \left(F_{JM,\,scr\,1} + F_{Jsh,\,scr}\right) \cdot 1.5 = \qquad (*)$$

$$(8.3\ kN + 27\ kN) \cdot 1.5 = \underline{52.4\ kN\ to\ 3\ s.f.} \qquad (7)$$

(*) We want the same preload on all the screws and prefer $F_{JM,\,scr\,1}$ for the torque absorption since $F_{JM,\,scr\,1} > F_{Mj,\,scr\,2}$

Selection of the bolt property class and dimension:

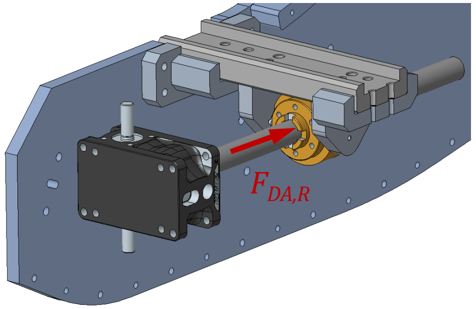
Maximum preload and tightening torque at 90 % utilization of the yield point.

Threads	Friction coeff. $\mu_K = \mu_G$	Maximum preload $F_{M\,max}$ [kN]							Maximum tightening torque $M_{A\,max}$ [Nm]						
		Property class based on ISO 898/1							Property class based on ISO 898/1						
		3.6	4.6	5.6/4.8	6.8	8.8	10.9	12.9	3.6	4.6	5.6/4.8	6.8	8.8	10.9	12.9
M10	0,08	8,7	11,6	14,5	23,2	31,0	45,6	53,3	10,2	13,6	17,0	27,2	36	53	62
	0,10	8,4	11,3	14,1	22,5	30,3	44,5	52,1	12	16,1	20,1	32,3	43	63	73
	0,12	8,2	11,0	13,7	21,9	29,6	43,4	50,8	13,7	18,3	22,9	36,5	48	71	83
	0,14	8,0	10,7	13,3	21,3	28,8	42,2	49,4	15,2	20,3	25,3	40,6	54	79	93
M12	0,08	12,7	16,9	21,1	33,8	45,2	66,3	77,6	17	23	29	47	63	92	108
	0,10	12,3	16,4	20,5	32,8	44,1	64,8	75,9	20	27	34	55	73	108	126
	0,12	12,0	16,0	20,0	32,0	43,0	63,2	74,0	23	31	39	62	84	123	144
	0,14	11,6	15,5	19,4	31,1	41,9	61,5	72,0	26	34	43	69	93	137	160

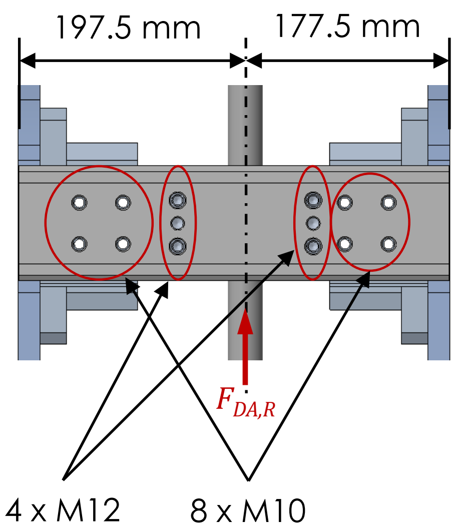
Bossard technical resources:
https://media.bossard.com/global-en/-/media/bossard-group/website/documents/technical-resources/en/f-047-en.pdf

Detector Arm – Drive Link Calculations

Crossbeam Calculation



The slightly asymmetrical force introduction can be neglected in the following calculations.



Required clamping force per screw:

If we compare the lever lengths in the calculation for the beam joint with the lever lengths in the calculation for the cross beam there is practically no difference.
We can therefore use half the value of the result in eq. (7) as we have twice as many bolts.

Thus, the required force per screw is

$$F_{scr,beam} = \frac{F_{scr,joint}}{2} = \frac{52.4\ kN}{2} = 26.2\ kN\ to\ 3\ s.f. \tag{8}$$

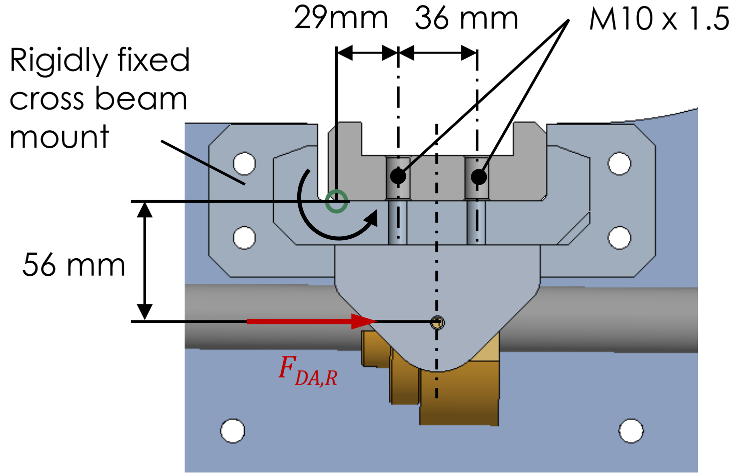
Selection of the bolt property class and dimension:

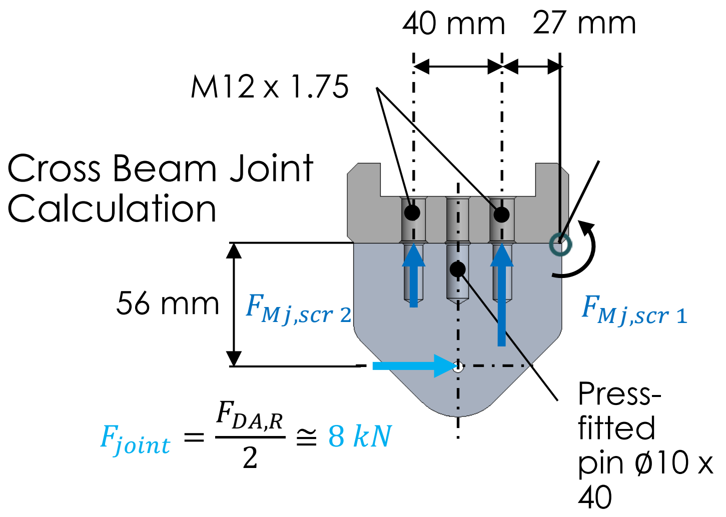
Maximum preload and tightening torque at 90 % utilization of the yield point.

Threads	Friction coeff. $\mu_K=\mu_G$	Maximum preload $F_{M\,max}$ [kN]							Maximum tightening torque $M_{A\,max}$ [Nm]						
		Property class based on ISO 898/1							Property class based on ISO 898/1						
		3.6	4.6	5.6/4.8	6.8	8.8	10.9	12.9	3.6	4.6	5.6/4.8	6.8	8.8	10.9	12.9
M10	0,08	8,7	11,6	14,5	23,2	31,0	45,6	53,3	10,2	13,6	17,0	27,2	36	53	62
	0,10	8,4	11,3	14,1	22,5	30,3	44,5	52,1	12	16,1	20,1	32,3	43	63	73
	0,12	8,2	11,0	13,7	21,9	29,6	43,4	50,8	13,7	18,3	22,9	36,5	48	71	83
	0,14	8,0	10,7	13,3	21,3	28,8	42,2	49,4	15,2	20,3	25,3	40,6	54	79	93
M12	0,08	12,7	16,9	21,1	33,8	45,2	66,3	77,6	17	23	29	47	63	92	108
	0,10	12,3	16,4	20,5	32,8	44,1	64,8	75,9	20	27	34	55	73	108	126
	0,12	12,0	16,0	20,0	32,0	43,0	63,2	74,0	23	31	39	62	84	123	144
	0,14	11,6	15,5	19,4	31,1	41,9	61,5	72,0	26	34	43	69	93	137	160

Bossard technical resources:
https://media.bossard.com/global-en/-/media/bossard-group/website/documents/technical-resources/en/f-047-en.pdf

Cross Beam Calculation

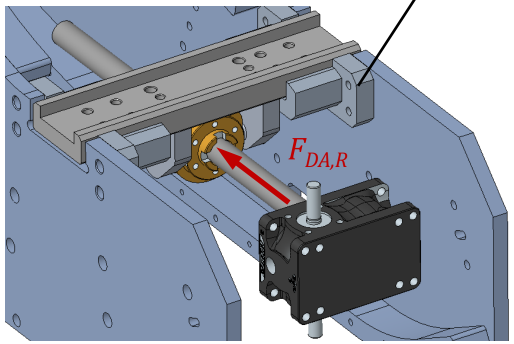




Detector Arm – Drive Link Calculations

Crossbeam Mount Calculation

$F_{DA,R} := 14.1\ kN$ Screw nut reaction force
$l_{DA,R} := 56\ mm$ Lever length
$F_{sht,scr} := ?$ Shear force per screw due to torsion
$l_{sht,scr} := 87.5\ mm$ Lever length
$F_{sh} := ?$ Shear force



Axial bolt forces required to absorb torsional load:

$$F_{DA,R} \cdot l_{DA,R} - 8 \cdot F_{sht,scr} \cdot l_{sht,scr} = 0 \quad (9)$$

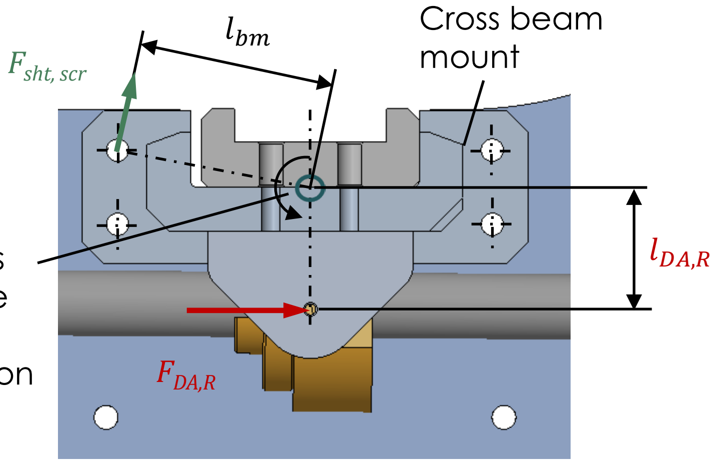
The torsional load is absorbed by 8 screws, 4 per flange.

$$F_{sht,scr} = \frac{F_{DA,R} \cdot l_{DA,R}}{8 \cdot l_{sht,scr}} = \frac{14.1\ kN \cdot 56\ mm}{8 \cdot 87.5\ mm} = 1128\ N \quad (10)$$

With a static friction coefficient of $\mu_{st} = 0.15$ the required force for the absorption of the torsional load becomes

$$F_{bm,tors} \cdot \mu_{St} \geq F_{sht,scr}$$

$$F_{bm,tors} \geq \frac{F_{sht,scr}}{\mu_{St}} = \frac{1128\ N}{0.15} = \underline{7.5\ kN} \quad (11)$$



Clamping force for the shear force absorption:

The minimum clamping force $F_{bm,sh}$ per screw

$$8 \cdot F_{bm,sh} \cdot \mu_{St} \geq F_{DA,R}$$

$$F_{bm,sh} \geq \frac{F_{DA,R}}{8 \cdot \mu_{St}} = \frac{14.1\ kN}{8 \cdot 0.15} = \underline{11.75\ kN} \quad (12)$$

Detector Arm – Drive Link Calculations

Crossbeam Mount Calculation

Total required clamping force per screw with safety factor 1.5:

$$F_{scr,joint} = \left(F_{bm,\,tors} + F_{bm,\,sh}\right) \cdot 1.5$$

$$(7.5\ kN + 11.75\ kN) \cdot 1.5 = 28.9\ kN\ to\ 3\ s.f. \qquad (13)$$

Selection of the bolt property class and dimension:

Maximum preload and tightening torque at 90 % utilization of the yield point.

Threads	Friction coeff. $\mu_K = \mu_G$	Maximum preload $F_{M\,max}$ [kN]							Maximum tightening torque $M_{A\,max}$ [Nm]						
		Property class based on ISO 898/1							Property class based on ISO 898/1						
		3.6	4.6	5.6/4.8	6.8	8.8	10.9	12.9	3.6	4.6	5.6/4.8	6.8	8.8	10.9	12.9
M10	0,08	8,7	11,6	14,5	23,2	31,0	45,6	53,3	10,2	13,6	17,0	27,2	36	53	62
	0,10	8,4	11,3	14,1	22,5	30,3	44,5	52,1	12	16,1	20,1	32,3	43	63	73
	0,12	8,2	11,0	13,7	21,9	29,6	43,4	50,8	13,7	18,3	22,9	36,5	48	71	83
	0,14	8,0	10,7	13,3	21,3	28,8	42,2	49,4	15,2	20,3	25,3	40,6	54	79	93
M12	0,08	12,7	16,9	21,1	33,8	45,2	66,3	77,6	17	23	29	47	63	92	108
	0,10	12,3	16,4	20,5	32,8	44,1	64,8	75,9	20	27	34	55	73	108	126
	0,12	12,0	16,0	20,0	32,0	43,0	63,2	74,0	23	31	39	62	84	123	144
	0,14	11,6	15,5	19,4	31,1	41,9	61,5	72,0	26	34	43	69	93	137	160

Bossard technical resources: https://media.bossard.com/global-en/-/media/bossard-group/website/documents/technical-resources/en/f-047-en.pdf

Detector Arm – Drive Link Calculations

Crossbeam Bending Stress Calculation

We assume a rectangular cross beam with cross sectional area b x c, (b > c) that is fixed on both sides. The Bending stress that occurs in the given load case is defined as

$$\sigma_b = \frac{M_{A,B}}{W} \qquad (14)$$

In our case, with $l_1 = l_2 = l/2$, the bending moment at position A is

$$M_A = \frac{F_{DA,R} \cdot l_1 \cdot {l_2}^2}{l^2} := \frac{F_{DA,R} \cdot {l_2}^3}{l^2} = \frac{F_{DA,R}}{l^2} \cdot \left(\frac{l}{2}\right)^3 = \frac{F_{DA,R} \cdot l}{8} \qquad (15)$$

and at C

$$M_C = \frac{2 \cdot F_{DA,R} \cdot l \cdot {l_1}^2 \cdot {l_2}^2}{l^4} = \frac{2 \cdot F_{DA,R} \cdot l}{l^4} \cdot \frac{l^2}{4} \cdot \frac{l^2}{4} = M_A$$

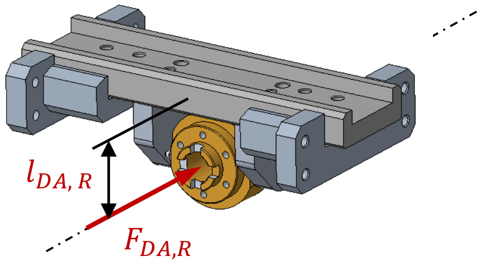
and the axial moment of resistance

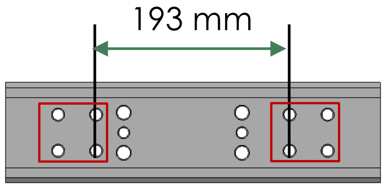
$$W := \frac{b^2 \cdot c}{6} \qquad (16)$$

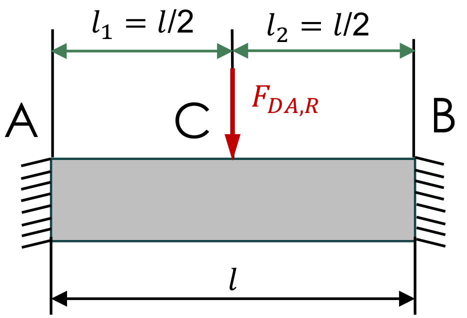
Substituting eq. (15) and (16) into eq. (14) gives

$$\sigma_b = \frac{M_B}{W} = \frac{F_{DA,R} \cdot l}{8} \cdot \frac{6}{b^2 \cdot c} = \frac{3 \cdot F_{DA,R} \cdot l}{4 \cdot b^2 \cdot c} = \frac{3 \cdot 14.1\ kN \cdot 374\ mm}{4 \cdot (100\ mm)^2 \cdot 21mm} = \underline{18.8\ \frac{N}{mm^2}\ (MPa)\ \ to\ 3\ s.f.}$$

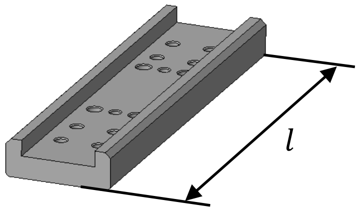
Using **stainless steel 316** (yield strength 241 MPa) the safety factor is ≅13 !





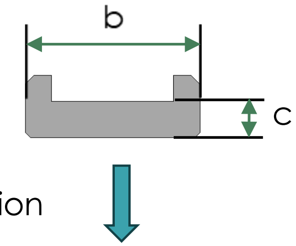


$F_{DA,R} = 14\ kN$	Force
$l = 374\ mm$	Crossbar length
$b = 100\ mm$	Bar side
$c = 21\ mm$	Bar side

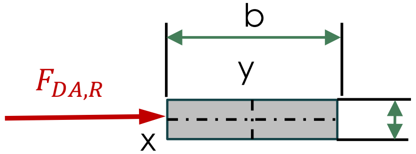


GIECK
Engineering Formulas
7th Edition P10, P12

Cross Section



Assumption



$$W_y := \frac{b^2 \cdot c}{6} \qquad W_x := \frac{b \cdot c^2}{6}$$

Detector Arm – Drive Link Calculations

Crossbeam Torsion

We consider again a cross bar with cross sectional area b x c, (b > c).

From Shigley's Handbook follows that the max. shear stress for the given torsional loading occurs in the middle of the longest side b and is of magnitude

$$\tau_{max} \approx \frac{F_{DA,R} \cdot l_{DA,R}}{b \cdot c^2} \cdot \left(3 + \frac{1.8 \cdot c}{b}\right) \quad (17)$$

$$= \frac{14.1\ kN \cdot 56mm}{100\ mm \cdot (21mm)^2} \cdot \left(3 + \frac{1.8 \cdot 21mm}{100mm}\right) = 60.5\ \frac{N}{mm^2}\ (MPa)\ \text{to 3 s.f.} \quad (18)$$

Using stainless steel 316 (yield strength 241 MPa) the safety factor is ≅ 4 !

With the modulus of rigidity $G = \frac{E}{2(1+v)} = \frac{193000\ MPa}{2(1+0.3)} = 74231\ MPa$

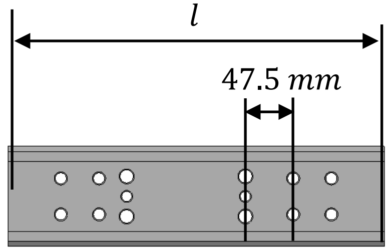
and the factor $\beta = 0.281$ (see table) we obtain the angle of twist

$$\theta = \frac{F \cdot l_{DA,R} \cdot l/2}{\beta \cdot b \cdot c^3 \cdot G} = \frac{14000N \cdot 66.5\ mm \cdot 187\ mm}{0.281 \cdot 100mm \cdot (21mm)^3 \cdot 74231 \frac{N}{mm^2}} =$$

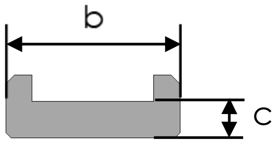
$$4 \cdot 10^{-5}\ rad\ (= 0.01\ deg)\ \text{to 1 s.f.} \quad (19)$$

$F_{DA,R} = 14\ kN$	Force
$l_\tau := l_{DA,R} + \frac{c}{2} = 56\ mm + 10.5\ mm = 66.5\ mm$	Lever length
$l = 374\ mm$	Crossbar length
$b = 100\ mm$	Bar side
$c = 21\ mm$	Bar side
$v = 0.3$	Poisson's ratio
$E = 193\ GPa$	Modulus of elasticity

Shigley Mechanical Engineering Design 10th Edition p. 116, eq. (3 – 40) and (3 – 41)

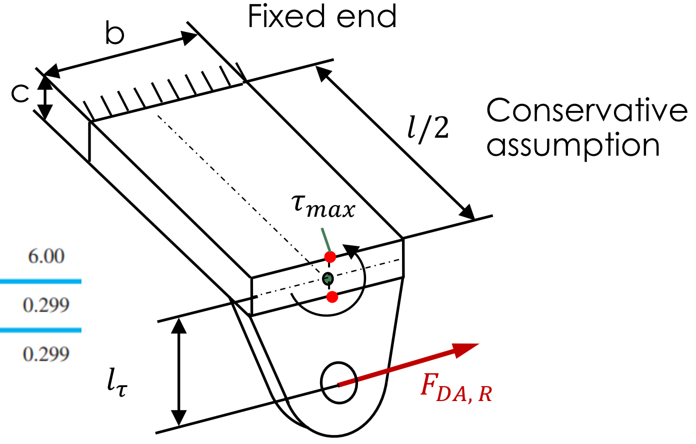


Cross Section



$$\frac{b}{c} = \frac{100\ mm}{21\ mm} = 4.76$$

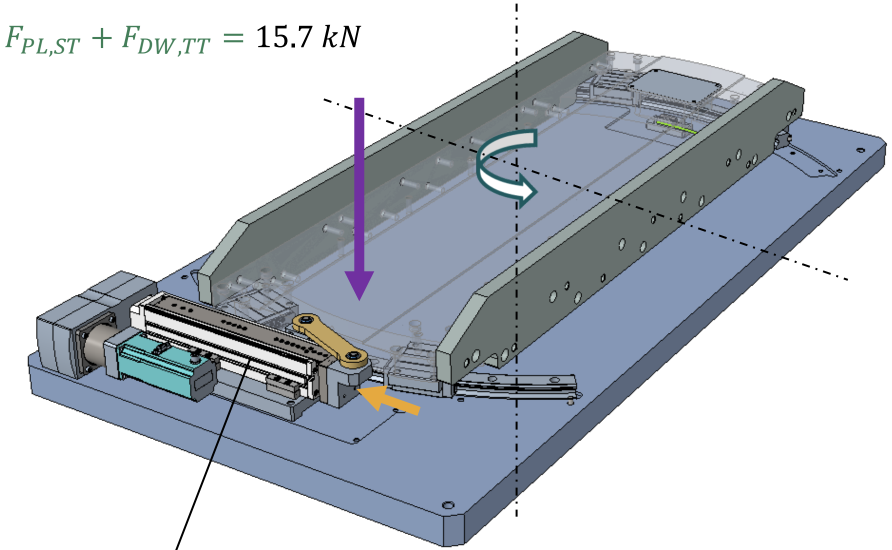
b/c	1.00	1.50	1.75	2.00	2.50	3.00	4.00	6.00
α	0.208	0.231	0.239	0.246	0.258	0.267	0.282	0.299
β	0.141	0.196	0.214	0.228	0.249	0.263	0.281	0.299



Sample Table – Rotary Table Calculation

Actuator calculation & selection

$F_{PL,DA} + F_{DW,STDA} + F_{PL,ST} + F_{DW,TT} = 15.7\ kN$



Given are the following loads

$F_{PL,DA} = 3000\ N,$	Payload detector arm	(R127: ≧ 227kg)
$F_{PL,ST} = 5000\ N,$	Payload sample table	(R117: ≧ 455kg)
$F_{DW,STDA} = 6482\ N,$	Dead weight sample table & detector arm	
$F_{DW,TT} = 1180\ N$	Dead weight turntable	
$\mu_r = 0.003$	Friction coefficient of the curved guides	

Thus, the friction force is therefore

$$F_\mu = \mu_r \cdot \left(F_{PL,DA} + F_{DW,STDA} + F_{PL,ST} + F_{DW,TT}\right) = 0.003 \cdot 15662N \underline{\cong 47\ N}$$

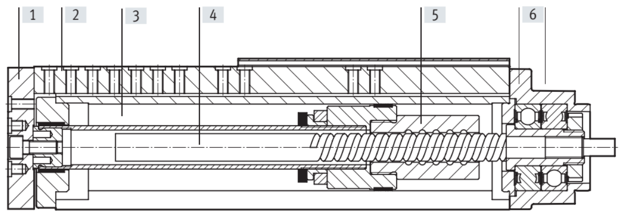
We select the ball screw driven electric cylinder FESTO EGSL-BS-55-200-5P

General technical data								
Size		35	45		55		75	
Spindle pitch	[mm/rev]	8	3	10	5	12.7	10	20
Design		Electric mini slide						
		With ball screw						
		With guide						
Guide		Ball bearing cage guide						
Type of mounting		Via female thread						
		With centring sleeve						
		Via accessories						
Mounting position		Any						
Working stroke	[mm]	50	100, 200		100, 200, 250		100, 200, 300	
Guide value for payload, horizontal	[kg]	2	6		10		14	
Guide value for payload, vertical	[kg]	2	6		10		14	
Continuous feed force F_x	[N]	50	100		200		300	
Max. feed force F_x	[N]	75	150		300		450	
Max. no-load driving torque	[Nm]	0.015	0.090	0.080	0.150	0.135	0.265	0.165
Max. driving torque[1]	[Nm]	0.127	0.205	0.415	0.415	1.017	1.654	2.231
Max. radial force[2]	[N]	20	120		260		300	
Max. speed	[m/s]	0.5	0.3	1.0	0.4	1.0	0.65	1.3
Nominal acceleration	[m/s²]	15						
Max. acceleration[3]	[m/s²]	25						
Repetition accuracy	[mm]	±0.015						
Max. reversing backlash[4]	[µm]	≤50						

1) Friction and acceleration torque of the rotating mass taken into consideration
2) At the drive shaft
3) The max. acceleration is dependent on the moving mass, the driving torque and the max. feed force
4) In new condition

Materials

Sectional view



Axis		
[1]	Yoke plate	Anodised wrought aluminium alloy
[2]	Guide rail	Rolled steel
[3]	Housing	Anodised wrought aluminium alloy
[4]	Spindle	Rolled steel
[5]	Spindle nut	Rolled steel
[6]	Cover	Painted aluminium
	Note on materials	RoHS-compliant
		Contains paint-wetting impairment substances

Sample Table – Rotary Table Calculation

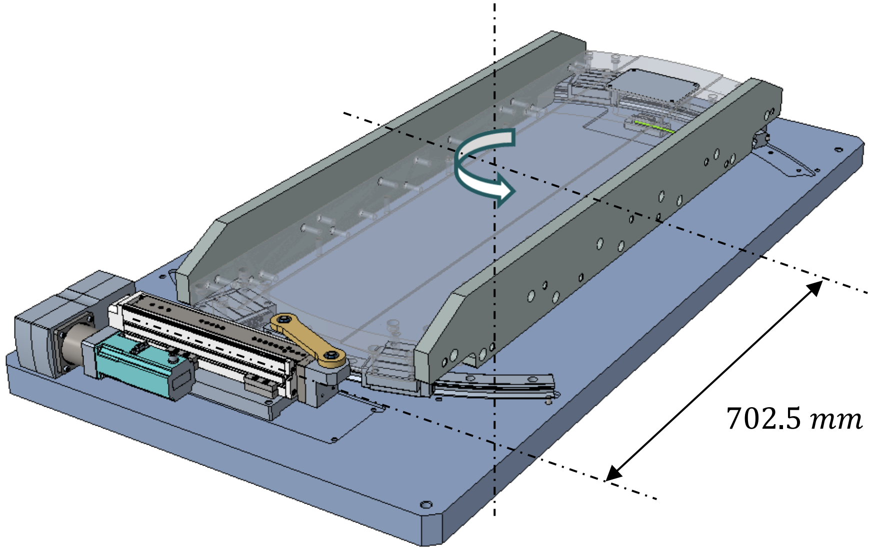
Torque, resolution and speed

FESTO Electric Cylinder

Motor : FESTO Stepper Motor EMMS-ST-57-S-SEB-G2
Torque: 0.8 Nm at 100 rpm; integrated break and rot. Encoder

Gearbox: FESTO Planetary Gear EMGA-60-P-G3-SST-57
Gear Ratio: 3:1

El. Cylinder: FESTO EGSL-BS-55-200 Ball Screw Drive
Ball Screw Pitch: 5 mm; Stroke: 200 mm

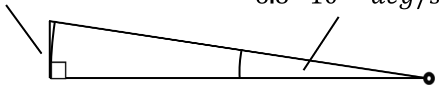


Resolution: $Stroke\ per\ drive\ shaft\ rotation = \frac{5\ mm}{3} \cong 1.7\ mm$

$Stroke\ per\ step\ (1.8°) = \frac{1.7\ mm}{200} = 0.0083\ mm$

$Angular\ Resolution \approx \text{asin}\frac{8.3\cdot10^{-3} mm}{702.5\ mm} = 6.8\cdot10^{-4}\ deg$ (R130: $2\cdot10^{-3}\ deg$)

Speed: $Duration\ for\ the\ whole\ stroke \cong \frac{200\ mm}{100\ rpm \cdot 1.7\ mm} = 1.16\ \text{min} \cong 70\ s$

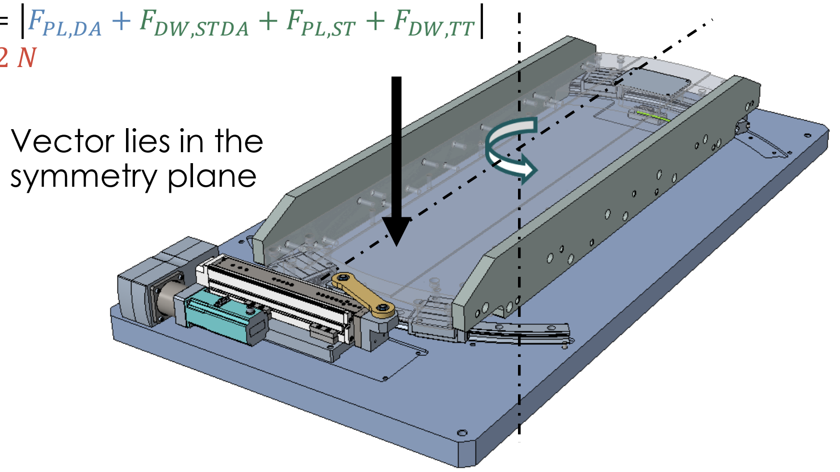


Encoder: Renishaw RKLA30-S RESOLUTE absolute encoder scale.

Resolution: $BiSS\ C$: 50 nm

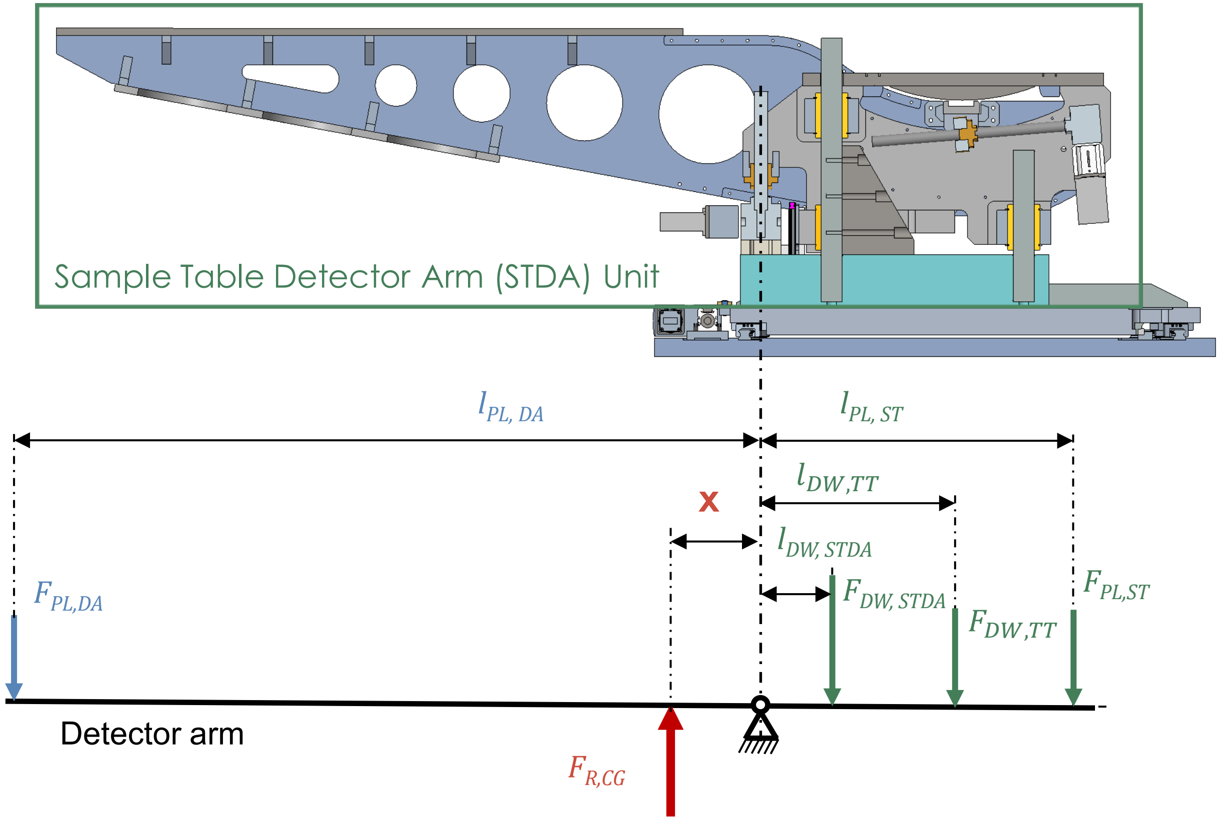
Sample Table – CG Calculation

$$|F_{R,CG}| = |F_{PL,DA} + F_{DW,STDA} + F_{PL,ST} + F_{DW,TT}|$$
$$= 15662\ N$$



Calculation of the position of the load acting on the turntable.

Given are the following loads:



$F_{PL,DA} = 3000\ N$, $l_{PL,DA} = 1.941\ m$	Payload detector arm	(R127: ≧ 227kg)
$F_{PL,ST} = 5000\ N$, $l_{PL,ST} = 0.555\ m$	Payload sample table	(R117: ≧ 455kg)
$F_{DW,STDA} = 6482\ N$, $l_{DW,STDA} = 0.190\ m$	Dead weight sample table & detector arm unit	
$F_{DW,TT} = 1180\ N$, $l_{DW,TT} = 0.555\ m$	Dead weight turntable	
$x :=$	Correction with respect to lifting nut	

We refer to the positions of the given forces relative to the lifting nut and calculate the position of a supporting force in case of a free-cut body.

$$F_{PL,DA} \cdot (l_{PL,DA} + x) - F_{CG,STDA} \cdot (l_{CG,STDA} - x) - F_{PL,ST} \cdot (l_{PL,ST} - x) - F_{CG,TT} \cdot (l_{CG,TT} - x) = 0$$

$$x = \frac{-F_{PL,DA} \cdot l_{PL,DA} + F_{PL,ST} \cdot l_{PL,ST} + F_{CG,STDA} \cdot l_{CG,STDA} + F_{CG,TT} \cdot l_{CG,TT}}{F_{PL,DA} + F_{CG,STDA} + F_{PL,ST} + F_{CG,TT}}$$
$$= -0.074\ m$$

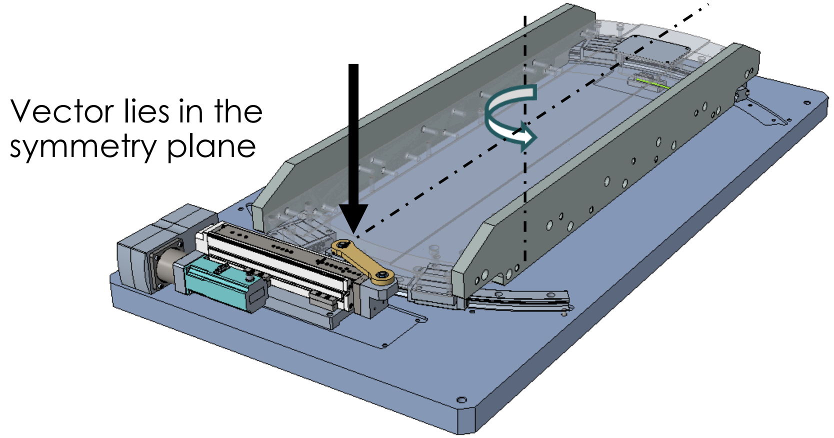
As the distance between the lifting spindle and the turntable axis is 555 mm, the total load acts on the turntable at 629 mm from the axis of rotation.

Engineering & Design: R. Thermer

Oak Ridge National Laboratory

Sample Table– CG Calculation

$$|F_{R,CG}| = |F_{PL,DA} + F_{DW,STDA} + F_{PL,ST} + F_{DW,TT}| = 15662\ N$$



Load distribution on turntable guides

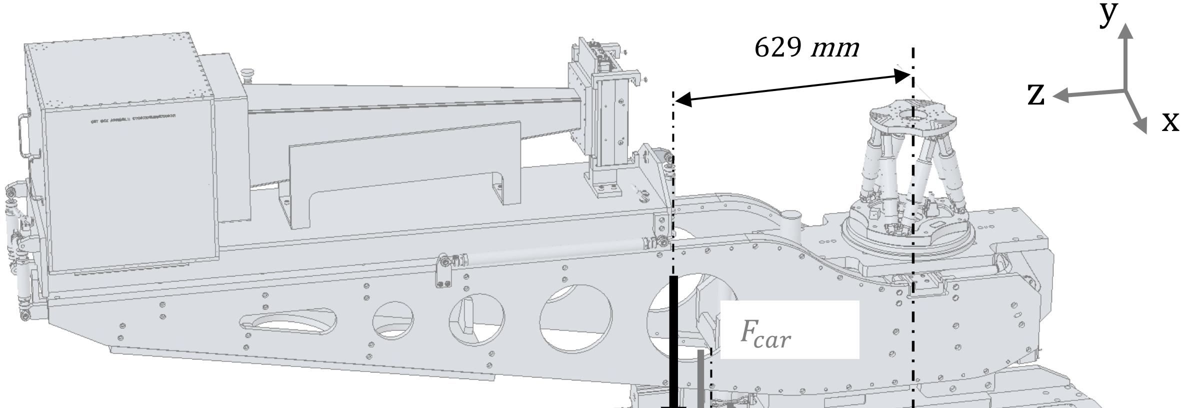
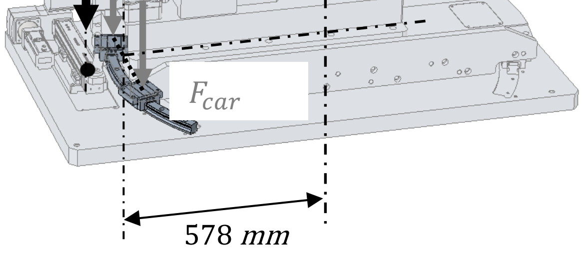
According to the calculation in the previous sheet, the total load is mainly distributed over the rear two carriages.

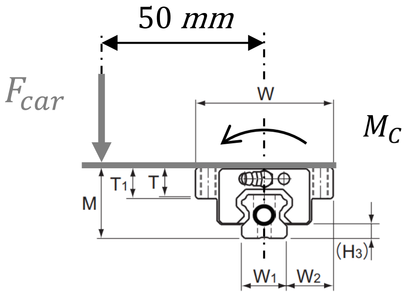
Thus, the force acting of each of the two carriages is

$$F_{car} = \frac{|F_{R,CG}|}{2} = 7.85\ kN\ (\cong \frac{1}{4.75} \cdot C)$$

and the corresponding moment is (conservative assumption with only two supporting carriages)

$$M_{car} = F_{car} \cdot 0.050\ m = 7850\ N \cdot 0.050\ m \cong 0.39\ kNm\ (\cong \frac{1}{2.3} M_C)$$



THK

	Basic load rating		Static permissible moment kN·m*				
	C	C0	MA		MB		MC
	kN	kN	1 block	Double blocks	1 block	Double blocks	1 block
HCR 35A+60/600R HCR 35A+60/800R HCR 35A+60/1000R HCR 35A+60/1300R	37.3	61.1	0.782	3.93	0.782	3.93	0.905

OAK RIDGE National Laboratory

Engineering & Design: R. Thermer

Questions?