

Passage to the next academic year 26/27

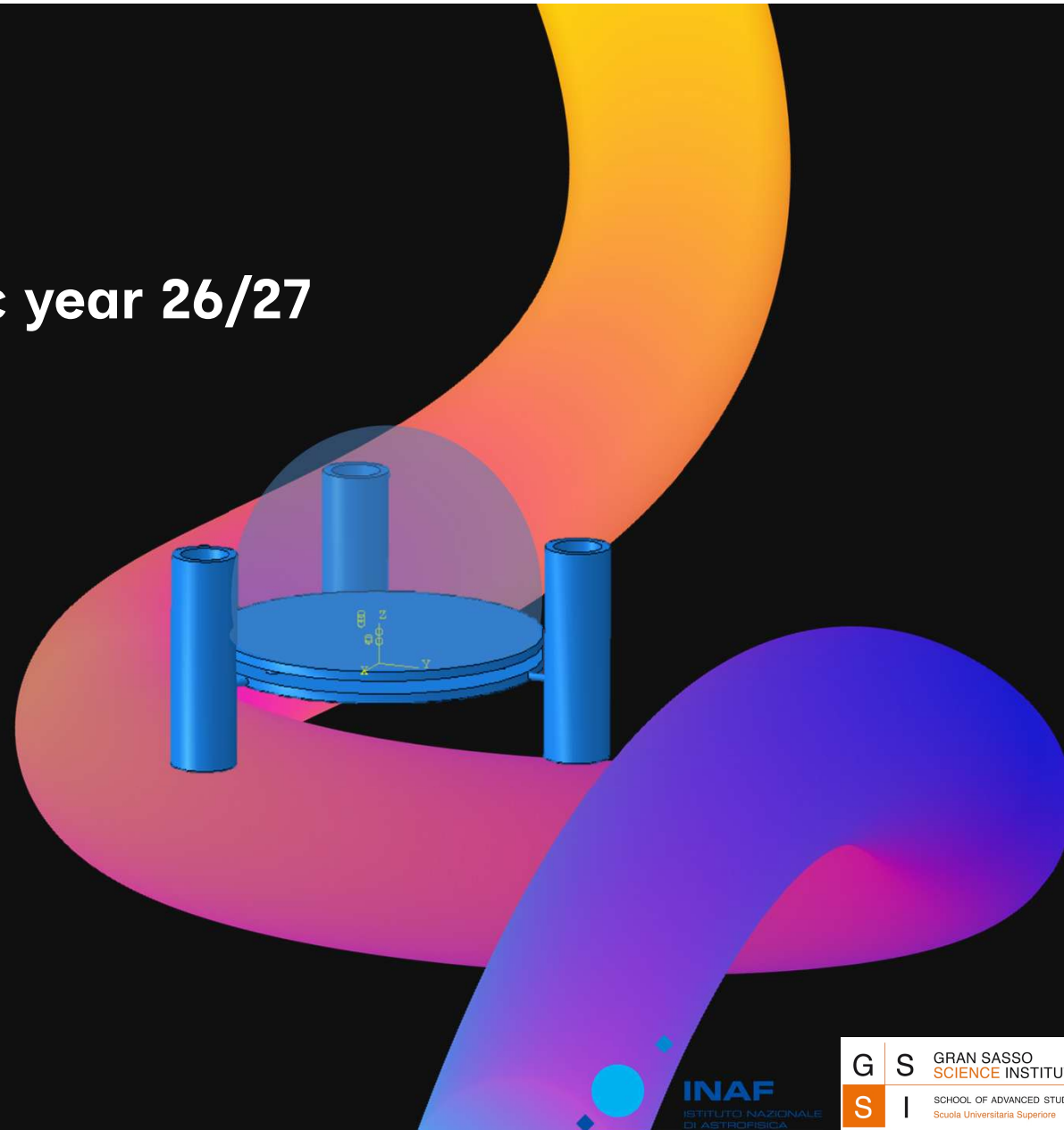
Development of the LGWA platform and leveling system

PhD Student:
Glazkova Veronika (GSSI)

Supervisor & Coordinator:
Professor Jan Harms

Co-supervisor:
Dr. Gianluca Di Rico (INAF)

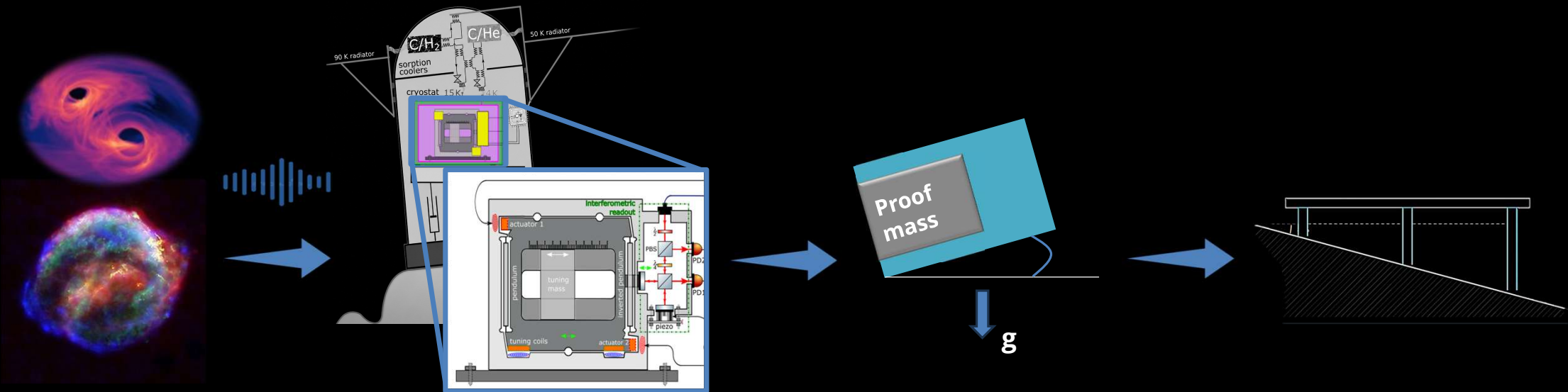
Gran Sasso Science Institute



INAF
ISTITUTO NAZIONALE
DI ASTRONOMIA

G	S	GRAN SASSO SCIENCE INSTITUTE
S	I	SCHOOL OF ADVANCED STUDIES Scuola Universitaria Superiore

Basic idea



Equipment with low resonance frequency provides the mechanical **sensitivity** needed **to detect** the extremely faint surface displacements induced by **GW**

However, the extremely **soft suspension** inside this mechanism makes the sensor highly susceptible to **tilt-to-horizontal coupling**.

To solve all these problems, a **levelling platform** is needed. It should compensate for the slope of the **ground** and align to within a **few μrad** .

Deployment Strategy

Lander

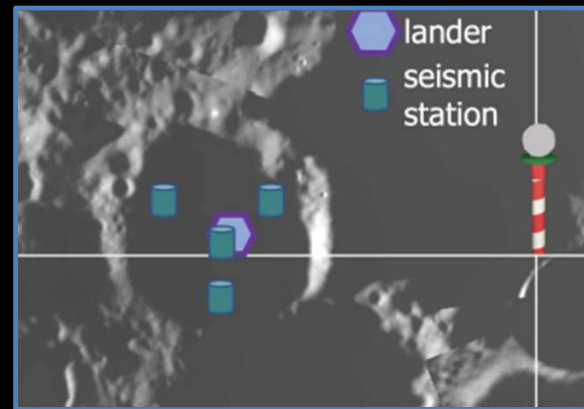
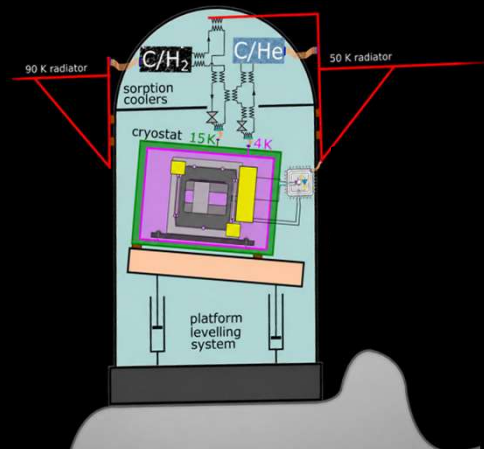
→ The vehicle is expected to land at the designated point in PSR

Rover

→ The rover transports the stations (x4) to a location some distance away from the landing point

Station

→ Seismic station is to be deployed



N

→ ETH Zürich study

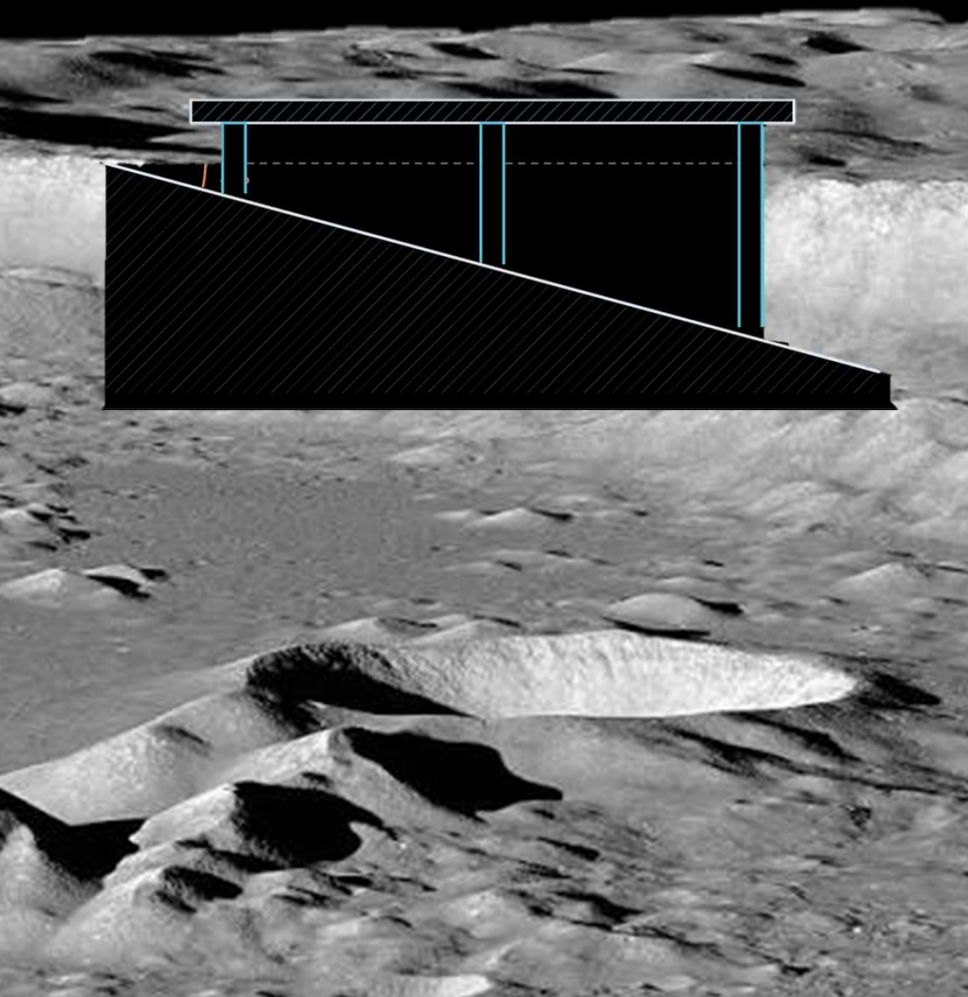
→ LGWA payload paper suggestion[5]

S

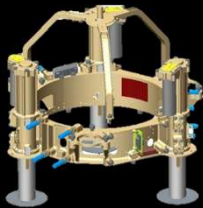
The objective of this work is therefore to develop a leveling mechanism that first **aligns the platform with the local lunar surface, compensating for ground slope**, and then maintains this alignment with an accuracy of about $10 \mu\text{rad}$, so that gravitational-wave signals can be reliably recorded

Previous work and heritage

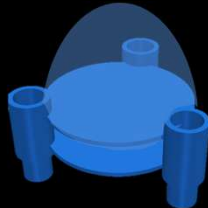
Conditions and Challenges



MarsInsight



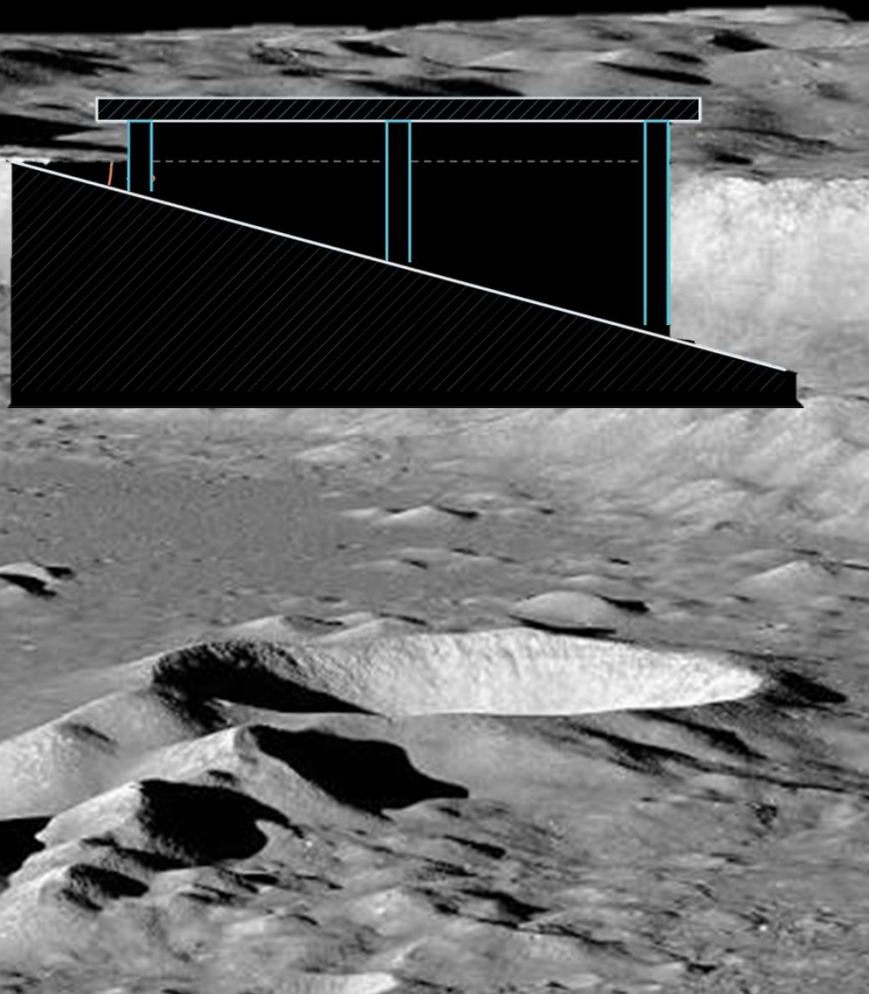
LGWA



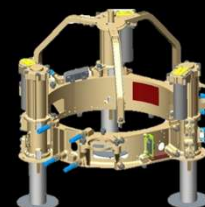
Source	What it gives	How to use it
The payload of the Lunar Gravitational-wave Antenna (arXiv:2301.13685), Sec. IV	Leveling accuracy of a few μrad , 30 μm proof-mass positioning, comparison with InSight	Requirements for the fine stage
LGWA: Mission Studies and Science Case (arXiv:2404.09181), Sec. 2.3.4	Ground tilt drivers (topography, regolith, thermal, tides), tens of μrad	Need for continuous releveled
Doravari (IUCAA), LGWA meeting 2024, Tilt-Platform Design and Performance Constraints	Design concept, constraints, feasibility	Baseline concept, general mission requirements
InSight/SEIS leveling system (Bierwirth et al. 2019; Fayon et al.)	$\pm 15^\circ$ legs with 6 cm stroke, MEMS and precision tiltmeters	Heritage for the coarse stage, telescopic legs and technical structure

Previous work and heritage

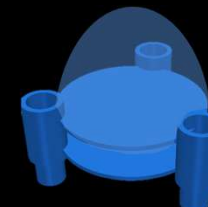
Conditions and Challenges



MarsInsight



LGWA



Temperature
Environment
Weight
Size
Duration
Tilt Accuracy

80-300°K
Low-pressure / Medium vacuum
8.5 kg + 3 kg payload
25 cm
4 yrs
0.044 rad

40-100°K
Ultra-high vacuum
15 kg +11kg payload
40 cm
10 yrs
10 μ rad

Thermal and Environmental Challenges:

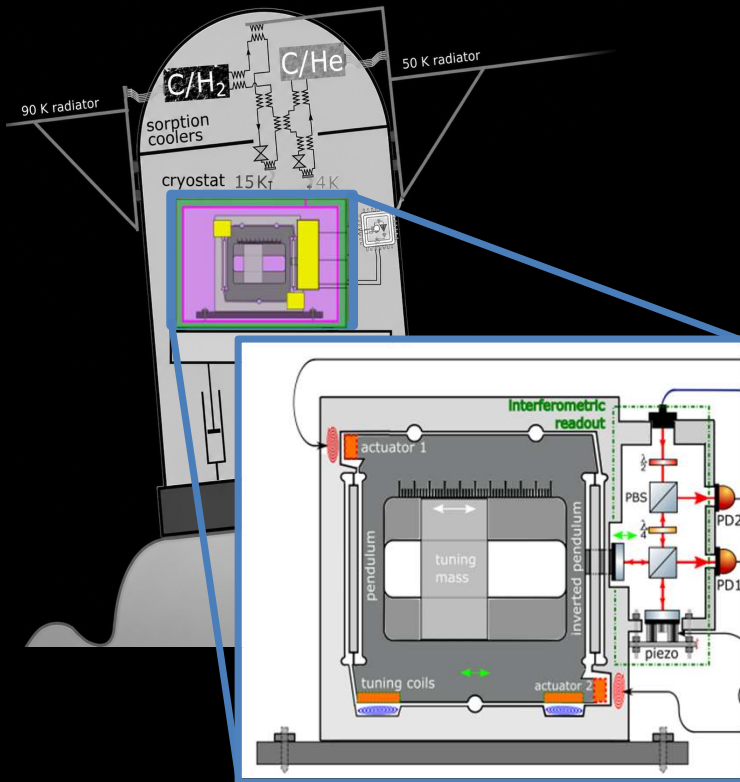
- **CTE** mismatch between dissimilar materials (Invar vs Ti)
- For **moving parts** dry/solid lubricants or self-lubricating ceramics (wet lubricants freeze at 15-40°K)
- Special actuators/**stepper motors** and other parts
- Structural materials with extremely **low outgassing rate**

Key Technical Challenges:

- Contact mechanics with regolith (penetration depth, bearing capacity)
- Bearing capacity (weight+ 11kg payload) held by mechanism statically for 10 yrs
- Potentially self-locking screw , need holding force >100N without power

Requirements:

Tilt angle (precision alignment system)

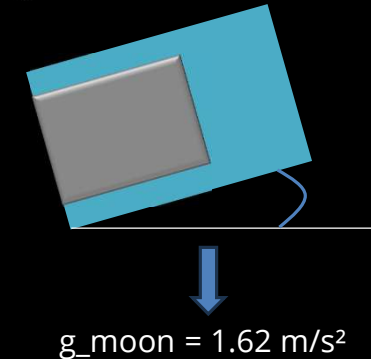


Residual tilt from the coarse stage (for example $\sim 1^\circ$)

Compensation from fine stage
 $\Delta z_{\text{needed}} = R \cdot \tan(\theta_{\text{residual_tilt}})$

Resolution/step requirement
 $\theta_{\text{max}} = 45.7 \mu\text{rad}$

angular frequency
 $\theta_{\text{max}} = d_{\text{pm}} \cdot \omega_0^2 / g_{\text{moon}}$
 $>30 \mu\text{m}$
 Requirement from the paper [5]



$\omega_0 = 2\pi \cdot \text{frequency (from [5])} = 1.57 \text{ rad/s}$

$\theta_{\text{max}} = 45.7$ microradians of tilt angle to prevent the mass from running onto the coil => otherwise there is a saturation of readout

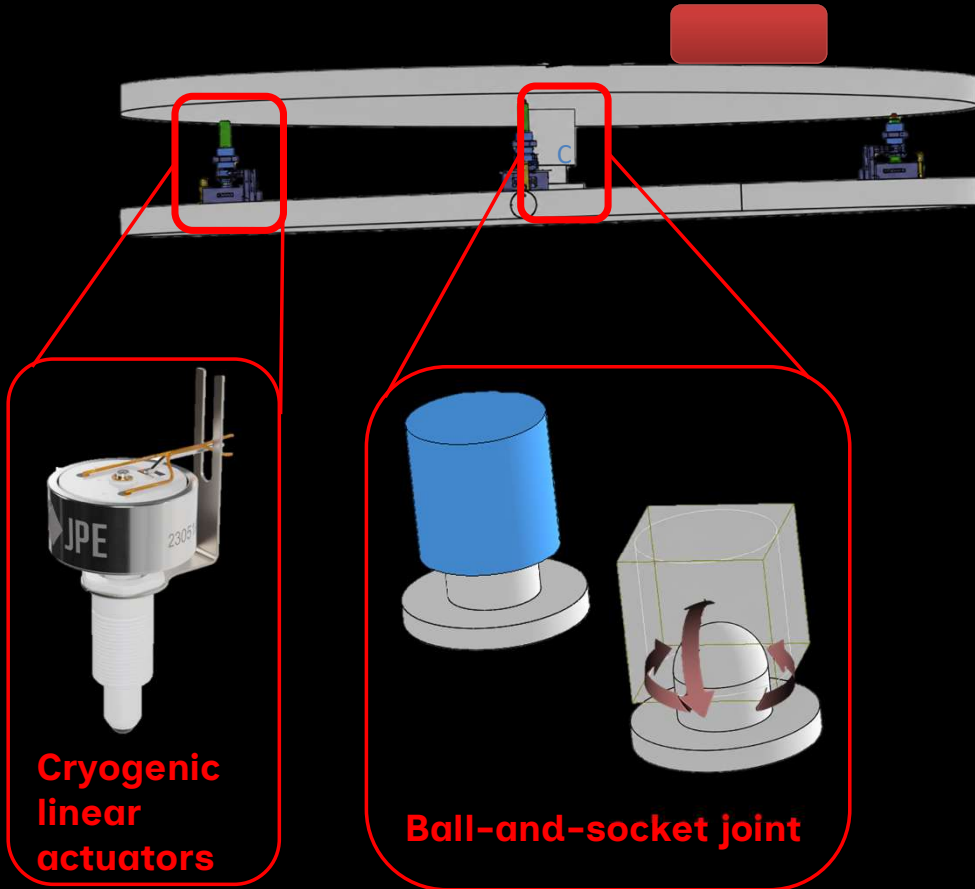


CLA2603 - CRYO LINEAR ACTUATOR

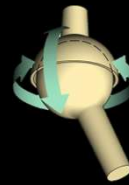
$\delta\theta_{\text{step}} = \delta z_{\text{step}} / R = 5\text{nm (min step size of actuator)} / 200\text{mm} = 25 \text{ nrad}$

the required accuracy window ($45.7 \mu\text{rad}$) is 1828 times wider than minimum adjustment (25 nrad) working in cryogenic conditions

Fine (high-precision) stage



The ball-and-socket joint



Purpose: central pivot of the fine stage decouples the 3 CLA actuators from radial/shear load, so they act in pure push-pull (axial) mode for tip-tilt + piston control.

Material: Ti-6Al-4V (ball + socket), matching platform structure.

Challenges: Ti-on-Ti cold-welding at cryo T (dry-film coating needed); preload shift from thermal contraction; no organic lubricants.

Cryogenic linear actuators



Purpose: 3× JPE CLA2603 @120° on the base disk drive fine piston/tip-tilt of the top plate against the central ball-socket pivot.

Material: body is from stainless steel 316L; screw & nut from ceramic, 250μm pitch.

Challenges: piezo step size drifts with T; dry ceramic-on-ceramic wear/backlash; Harness routing inside ø100mm leg.

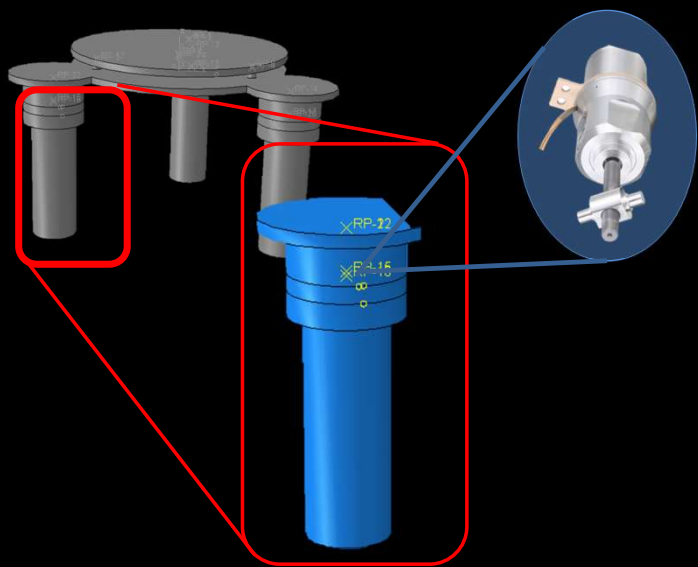
Cryogenic tilt sensor



Purpose: close the leveling loop, resolve tilt to few-μrad while surviving 40–100K in a PSR. Fine stage needs a tilt sensor here, but the *specific implementation is an open question*.

* there is a possibility of using accelerometers as an alternative

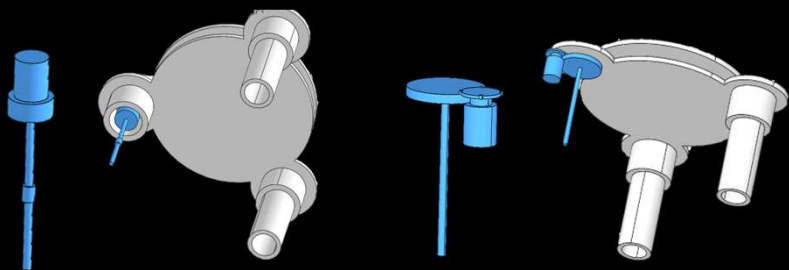
Coarse stage idea



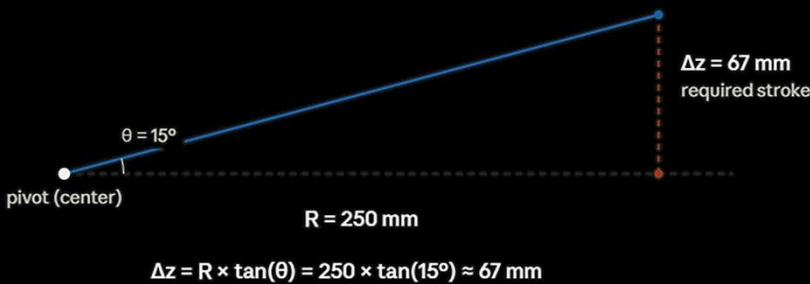
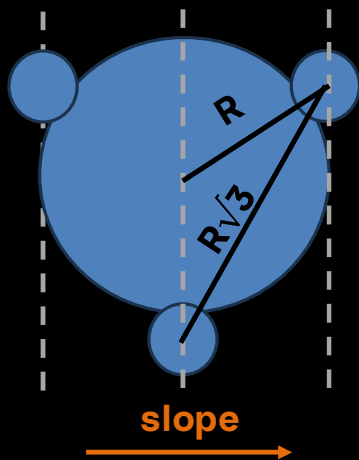
The Heavy-Duty Spindle Systems (Phyton GmbH)

Compact modular

Sidemount



Worst-case scenario: The three legs are positioned at different heights on an inclined plane due to the slope of the ground. Based on the geometry (radius), we can determine the required stroke.



LEG RADIUS R (mm)	SIDE S = $R\sqrt{3}$ (mm)	TILT ALONG LEG AXIS (mm)	TILT BETWEEN LEGS (mm)	REQUIRED STROKE (mm)
160	277	[+42.9, -21.4, -21.4]	[+37.1, 0, -37.1]	74.3
250	433	[+67.0, -33.5, -33.5]	[+58.0, 0, -58.0]	116.0
290	502	[+77.7, -38.9, -38.9]	[+67.3, 0, -67.3]	134.6

Mass Budget Estimate

#	Component	Used in	Qty	Outer D (mm)	Inner D (mm)	L/t (mm)	Material	Density	Mass/unit (kg)	Total (kg)
1	Top disc + Bottom disc (solid, 500mm)	Case 1	2	500	0	15	Ti-6Al-4V	4.43	13.047	26.09
2	Disc (solid, 400mm)	Case 2 (top) + Case 3	1	400	0	15	Ti-6Al-4V	4.43	8.350	8.35
3	Bottom disc, Y-shape, $A \approx 0.0518 \text{ m}^2$ (51,754 mm ²)	Case 2 only	1	—	—	15	Ti-6Al-4V	4.43	3.439	3.44
4a	Mounting bosses, 200mm	Case 1	3	200	0	15	Ti-6Al-4V	4.43	2.088	6.26
4b	Mounting bosses, 150mm	Case 2 + Case 3	3	150	0	15	Ti-6Al-4V	4.43	1.174	3.52
5	Telescopic leg tube	All cases	3	100	75	330	Ti-6Al-4V (assumed)	4.43	5.023	15.07
6	Leg cap / sleeve	All cases	3	135	100	100	Ti-6Al-4V (assumed)	4.43	2.862	8.59
7	Actuator, JPE CLA2603 (cryo linear actuator)	Case 1 + Case 2	3	—	—	—	Stainless steel 316L	—	0.039*	0.12
8	Ball-and-socket joint (custom)	Case 1 + Case 2	3	—	—	—	Ti-6Al-4V	—	0.020*	0.06
9	Stepper motor + 48:1 gearhead (Phytron VSS 32.200)	All cases	3	—	—	—	—	—	0.238*	0.71

Case 1 — disk 50cm (baseline)

$1 + 4a + 5-9$

56.90 kg



Case 2 — disk 40cm, Y-shape bottom

$2 + 3 + 4b + 5-9$

39.86 kg

$\Delta -17.04 \text{ kg } (-29.9\%)$



Case 3 — single disc, no fine stage

$2 + 4b + 5 + 6 + 9$

36.24 kg

$\Delta -20.66 \text{ kg } (-36.3\%)$



* Component mass taken from vendor datasheet / rough custom-part estimate (not computed from geometry)
Material: Ti-6Al-4V, $\rho = 4.43 \text{ g/cm}^3$

Requirements Summary

Vibration tolerance during launch

Vacuum compatibility

Tilt range (coarse stage)

Tilt angle (precision alignment system)

Hold position without power consumption

Holding strength against linear force

Frequency range

Mass budget

100 g

10^{-9} Torr

$\sim 15^\circ$

$\sim 10 - 45.7 \mu\text{rad}$

consistent with
the self-locking
property of
CLA2603

consistent with
self-locking,
holding capacity
of Phytron GmbH

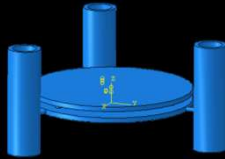
100 N

~ 100 Hz

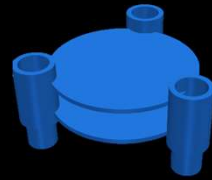
~ 15 kg

Comparison of Leveling platform configurations

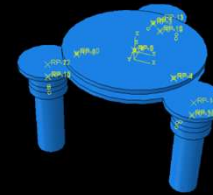
A: Wide Tripod (Flexure Joint)



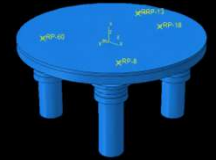
B: Compact Tripod



C: Dual-Radius Hybrid



D: Table Mount (Compact)



Stability
(tip-over margin)

Good support polygon → big margin

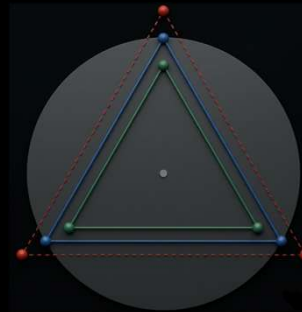
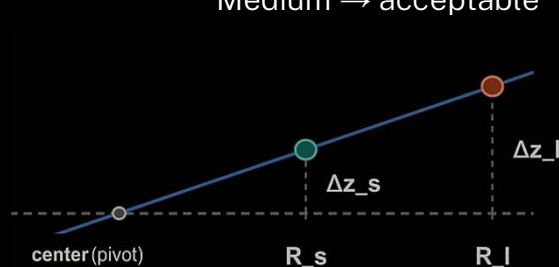
General support polygon → acceptable

Small support polygon → issue

Absolute leg travel
(Δ stroke) for $\pm 15^\circ$

Medium → acceptable

Correlated with
angular resolution
per mm of travel
($d\theta/dL = 1/R$)



The largest → the telescoping mechanism must provide a greater linear stroke, entails greater mechanism volume and mass, and involves more friction surfaces in the cryo-vacuum environment

The smallest one is more compact, has fewer telescopic sections and fewer wear surfaces

Complexity /
manufacturability

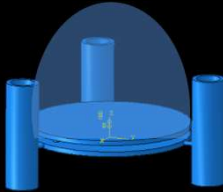
High. Flexures for legs operating in cryogenic conditions (15–100 K) and a 10^{-9} Torr vacuum represent an open problem; there is no COTS solution,

Medium. COTS-adjacent joint possible

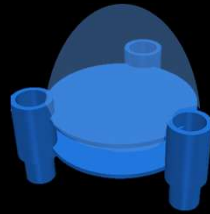
Acceptable. Direct connection legs → disk without intermediate flexures, fewer moving/wear interfaces in cryo

Same as C, but the tighter packaging might conflict with actuator/ball joint clearance in the center

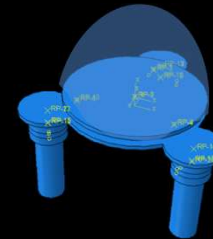
A: Wide Tripod (Flexure Joint)



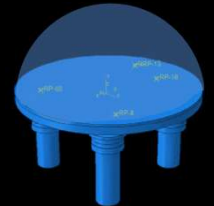
B: Compact Tripod



C: Dual-Radius Hybrid



D: Table Mount (Compact)



Thermal / CTE risk

The legs on the outside of the disk allow for **freer thermal expansion** and result in lower internal stress

Accumulated thermal stress without flexure buffer

The legs are integrated into the disk edge; a CTE mismatch between the legs and the disk can give rise to internal stresses

Higher risk: closer to the center means closer to the rest of the mechanism (actuators, ball joint) → greater likelihood of a statically indeterminate load path during cryogenic contraction

Load path clarity

A **clean separation:** legs → coarse stage, independent of actuators → fine stage

Partially clean. Legs still geometrically separate from actuators (unlike D), but rigid connection risks a statically indeterminate load path at the leg-ring interface under cryogenic contraction.

It is also partitioned, but geometrically the legs are "inside" the disk footprint

The **least clean:** the legs, actuators, and ball joint compete for the same central volume

Mass / volume envelope

Mass estimate 1 level: 32.90 kg
Mass estimate 2 level: 36.34 kg
Volume estimate with the dome: 0.0921 m³

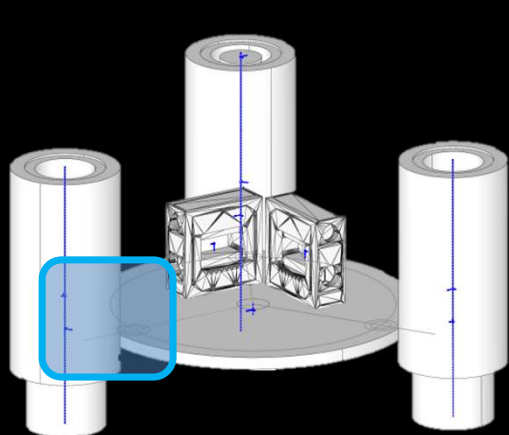
Mass estimate 1 level: 32.90 kg
Mass estimate 2 level: 36.34 kg
Volume estimate with the dome: 0.0921 m³

Mass estimate 1 level: **36.42kg**
Mass estimate 2 level: **39.86kg**
Volume estimate with the dome: **0.1147 m³**

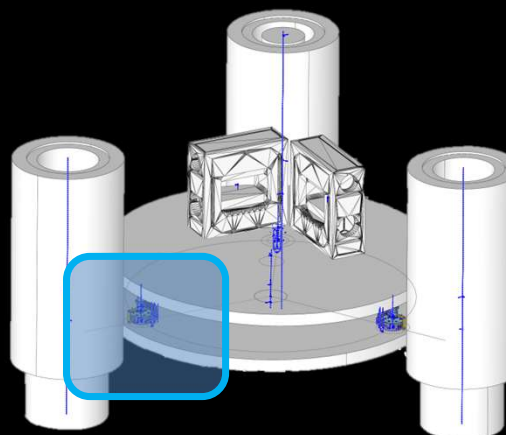
Mass estimate 1 level: 32.90 kg
Mass estimate 2 level: 36.34 kg
Volume estimate with the dome: **0.0834 m³**

SolidWorks preliminary configurations:

Different joint [leg-disk] structure



One level platform



Two level platform

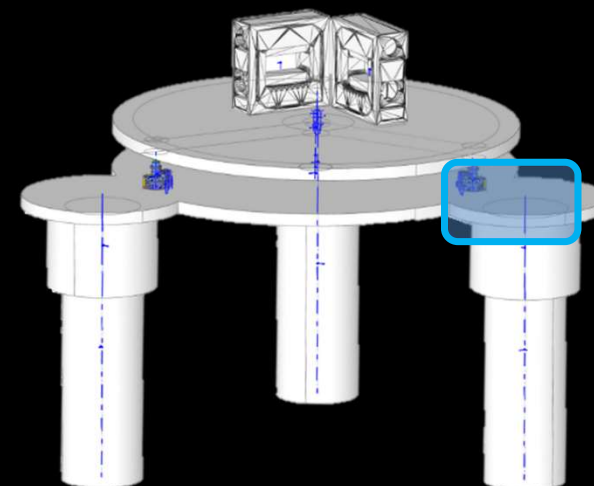
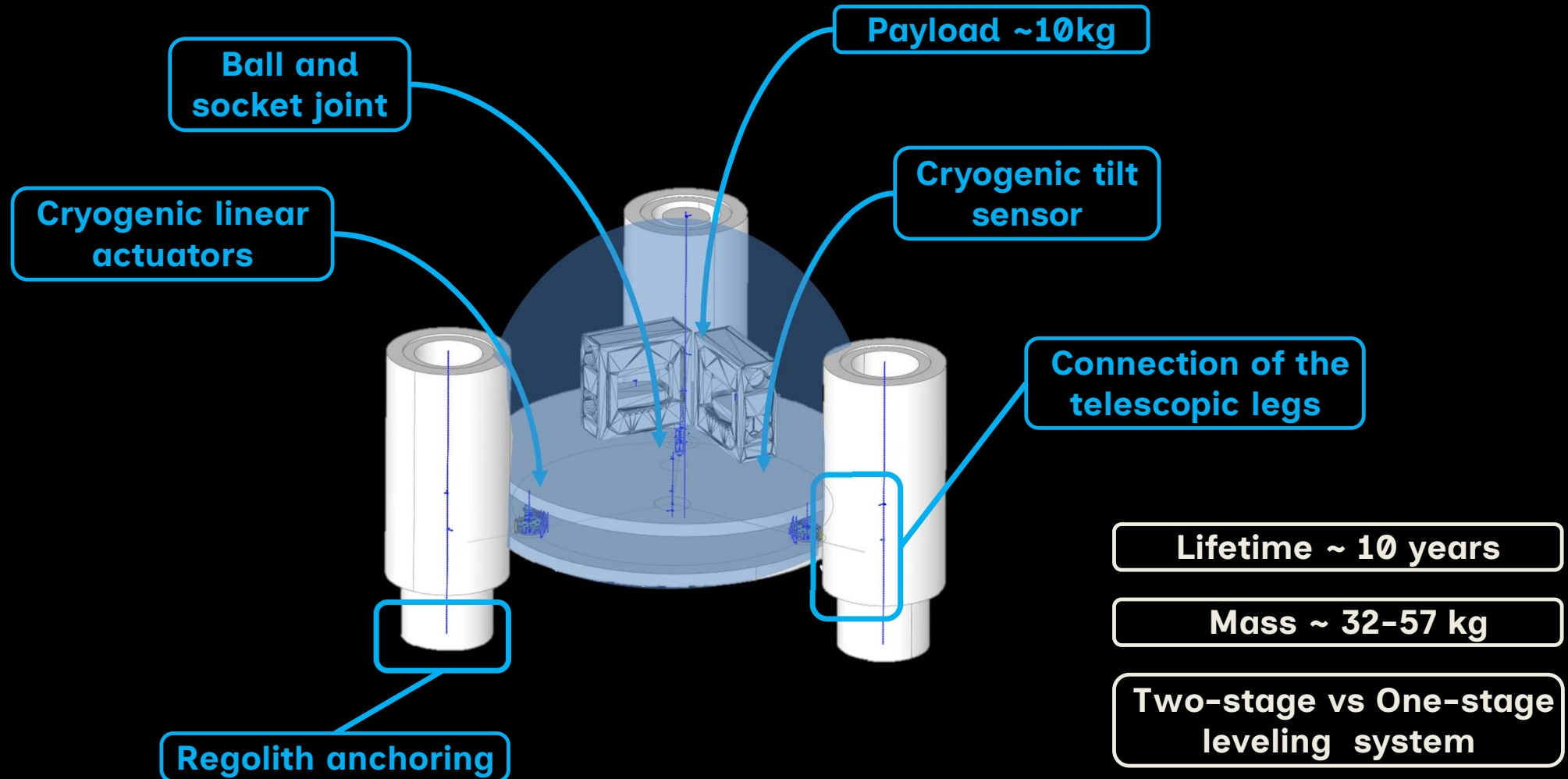


Table
two level platform

** Joint development in progress with National Institute for Nuclear Physics
Gran Sasso National Laboratory (LNGS) Mechanics Service and Istituto
Nazionale di Astrofisica (INAF)*

General idea of the Soundcheck

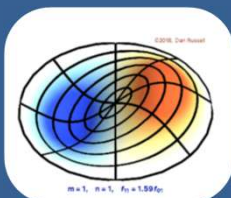
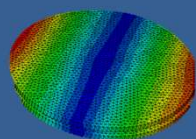


* This is not a final design of the platform; it has been presented for general consideration only

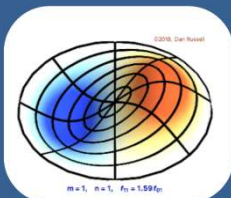
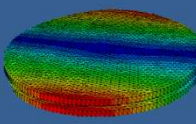
Numerical simulation results:

Staged verification: two disks platform

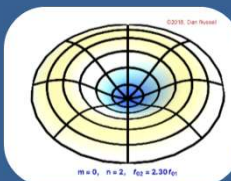
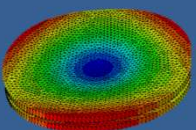
Mode1: 163 Hz



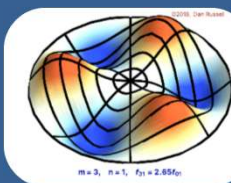
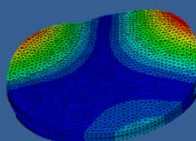
Mode2: 163 Hz



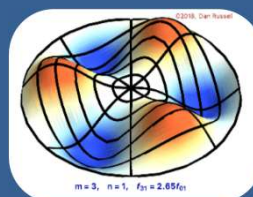
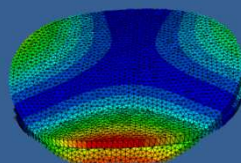
Mode3: 250 Hz



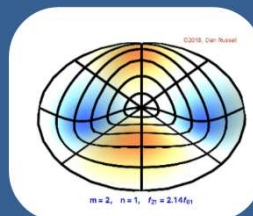
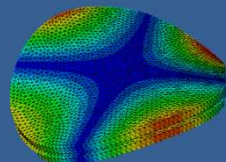
Mode4: 330 Hz



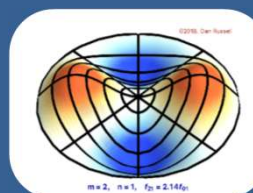
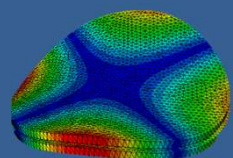
Mode5: 330 Hz



Mode6: 415 Hz



Mode7: 415 Hz



*Acoustics and Vibration
Animations
Daniel A. Russell

Scope & Assumptions of the Numerical Model

Boundary condition: free-free (unconstrained), matching Russell's free circular membrane reference

Actuator representation: idealized as a linear spring ($k \approx 8 \times 10^7$ N/m, JPE datasheet), real stick-slip screw behavior not captured

Ball-and-socket joint representation: it locks all 3 translational degrees of freedom (DOFs 1, 2, and 3) while leaving all 3 rotational degrees of freedom (DOFs 4, 5, and 6) completely free

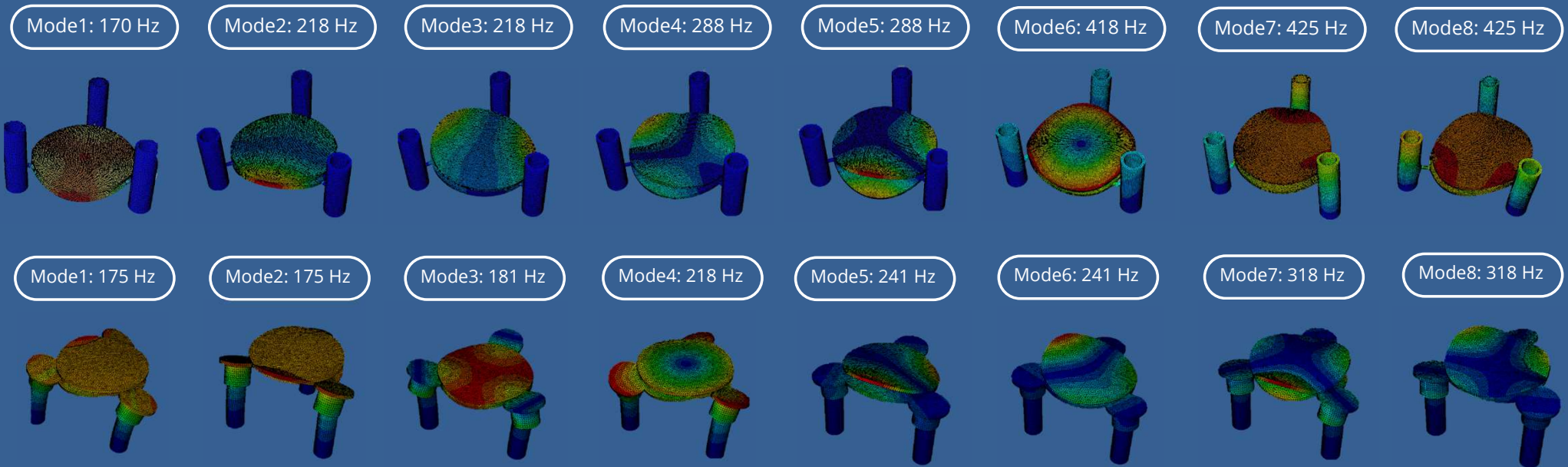
Load cases: static/thermal loads (100g launch, thermal sweep) not yet included in this modal analysis

Geometry: simplified (tie constraints, no fasteners/tolerances/manufacturing imperfections)

Qualitative mode-shape verification (not frequency validation!) Nodal diameter/circle topology against Russell's circular membrane analytical solutions

Numerical simulation results:

Frequency modes for configurations A/B and C



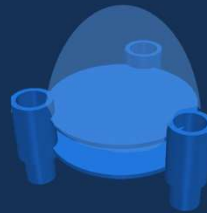
These are first-order / preliminary estimates from a simplified FE model intended to establish relative trends and order-of-magnitude margins, not final qualification values. Results are verified for internal consistency (mass recovery, sensitivity checks) but not yet validated against physical test data.

Absolute frequency values may shift once these are refined relative comparison between configurations is expected to be more robust than absolute numbers

Conclusion

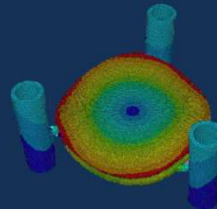
Trade study (Configs A/B/C/D) points to Config B as the most balanced option

The configuration B (compact tripod) benefits from a good mass envelope and a compact design, while still being a manufacturable configuration using COTS elements



Numerical simulation results

First-order / preliminary estimates from a simplified FE models showed frequency range > 100Hz



Two-level stage system → one-level stage

Phytron actuators with a microstepping mechanism can offer microrad accuracy. This leads to a single mechanical stage, which is helpful in most cases, such as reducing mass, volume and complexity.



Future study

Open Technology Risks

Cryogenic Sensing

market survey
technology substitution
analysis

Dry Lubrication Qualification

(i) creep and stress relaxation
(ii) wear life

Regolith BC

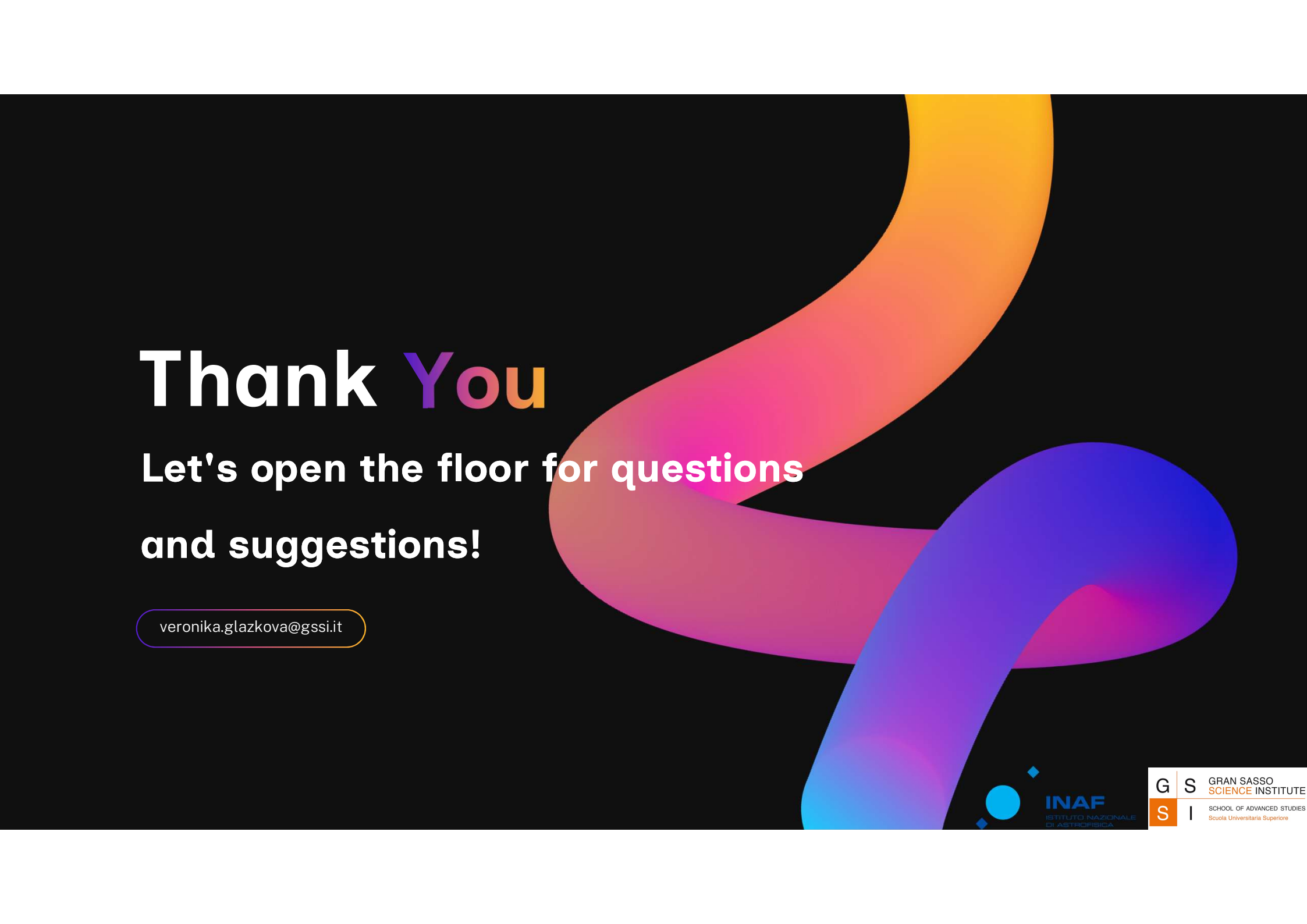
Sweep Winkler foundation springs ($K_z = 4Gr/(1-\nu)$, Apollo LSPE shear velocity data) instead of the current boundary conditions (ENCASTRE/fixed)

Geometry / stiffness coefficients

Compare different configurations (in-line, sidemount, compact modular) and diameter of leg, cap

Load + temperature

Static load cases
(payload, vibration analysis, 100g launch)
Thermal sweep (+20°C to -150°C)



Thank You

Let's open the floor for questions
and suggestions!

veronika.glazkova@gssi.it

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Backup slides



Progress Tracker

Live status across the platform's workstreams — update as you go, notes stay attached to each part.

36%

OVERALL COMPLETE

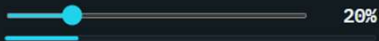
■ Not started · 1 ■ In progress · 7 ■ Blocked · 0 ■ Done · 0

Numerical analysis

DELETE

IN PROGRESS

PROGRESS



NOTES / NEXT STEPS

general algorithm of implementing model ,
verification of disk natural mode

Manufacture / COTS elements

DELETE

IN PROGRESS

PROGRESS



NOTES / NEXT STEPS

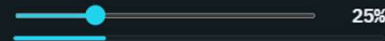
most parts can be done in LNGS, cryo

List of requirements

DELETE

IN PROGRESS

PROGRESS



NOTES / NEXT STEPS

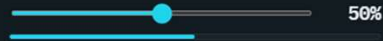
Some requirements depend on chosen
configuration/launcher and constraints,

Configuration of the platform

DELETE

IN PROGRESS

PROGRESS



NOTES / NEXT STEPS

approximate legs structure, waiting to decide
about 1/2 level system

Stepper motor system

DELETE

IN PROGRESS

PROGRESS



NOTES / NEXT STEPS

Phytron system investigation with different
configurations

Actuators

DELETE

IN PROGRESS

PROGRESS



NOTES / NEXT STEPS

to finilize waiting the decision about
phytron usage (accuracy of micriradians,
max Block with micriradians)

Regolith study data

DELETE

NOT STARTED

PROGRESS



NOTES / NEXT STEPS

waiting for data from Soundcheck/archives

Loading and temperature

DELETE

IN PROGRESS

PROGRESS



NOTES / NEXT STEPS

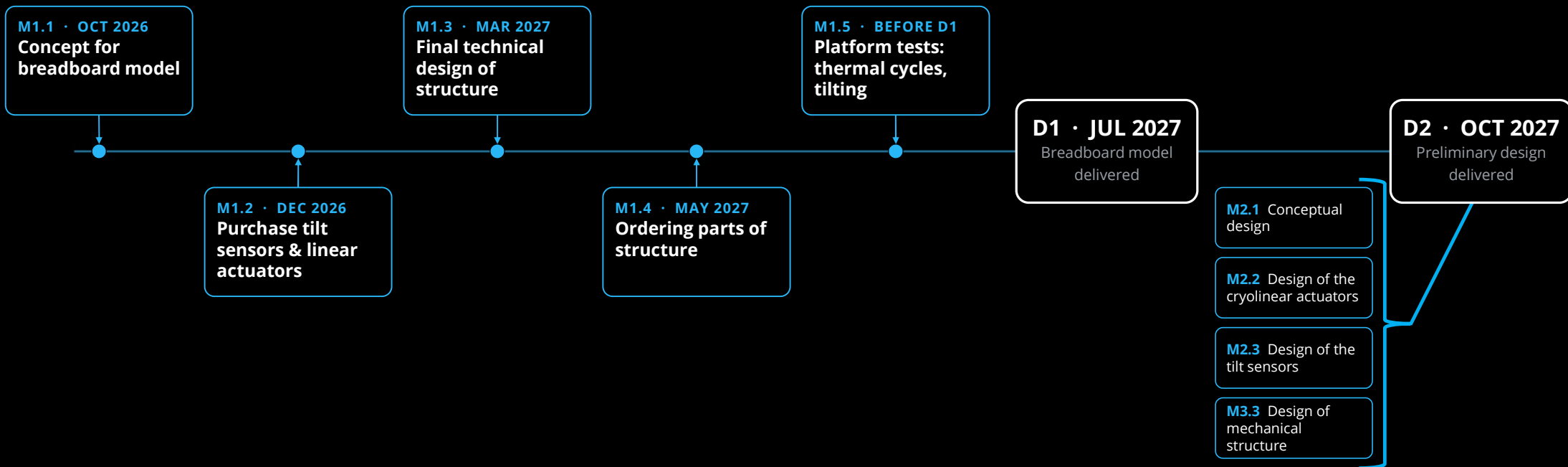
Inertia load from payload - implemented,
gravity, other loads in progress.

Roadmap

LGWA leveling platform → breadboard model → preliminary technical design

9 milestones

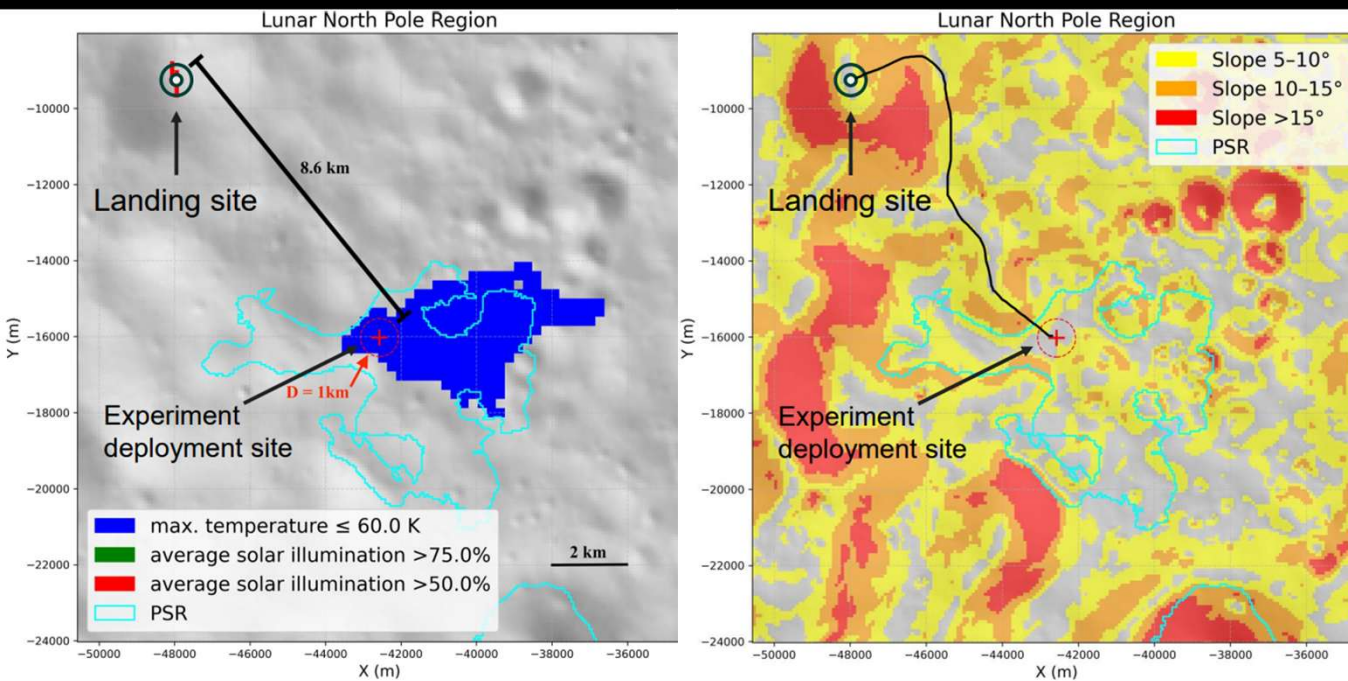
~12-month plan



PSR selection

Trade-off between *landing site requirements* and *experimental deployment site*

- High average solar visibility (>50%)
- Low temperature amplitude over a lunar day
- Terrain slope 5° - 10° favorable
- $T_{max} < 60^{\circ}K$
- Terrain slope < 10° favorable



*data from ETH Zürich PSR study [22]



Requirements:

Vibration tolerance during launch

	Ariane 64	LVM3	UVIT/AstroSat, PSLV
Data source	Ariane 6 UM Table 3.2.7.a	Not publicly available (using generic estimate)	UVIT proto-flight test spec (published, arXiv:1208.4672)
Peak shock (SRS)	~1000g	Not publicly available (assume comparable order)	ISRO test-bench capability (URSC) confirms ~1500g order of magnitude*
Sine vibration	0.6–1.0g 2–100Hz	Not publicly available	3.5–10g, 16–100Hz (X-axis) 3–5g, 11–100Hz (Y/Z)
Critical band (structural)	100–300 Hz (universal risk zone)	100–300 Hz (same) + possible 10–50Hz (solid booster-specific, unconfirmed)	100–300 Hz (universal risk zone)
Design target	100g quasi-static, GEVS-class conservative envelope (applies to both, searching official LVM3 numbers)		

*ISRO shock qualification order-of-magnitude: ~1500g (inferred from URSC Pyroshock Test Bench design capacity, not a direct payload-level published spec proxy only)



Analytical calculation [verification1]

1. Calculation of free disk based on the Kirchhoff plate theory:

$$f = \frac{a^2}{2\pi R^2} \sqrt{\frac{D}{\rho h}} \quad D = \frac{Eh^3}{12(1-\nu^2)} = \frac{110 \cdot 10^9 \cdot 0,015^3}{12(1-0,336^2)} = 34874,71$$

2. Finding the **stable/main part** (which doesn't change from mode to mode)

$$f_{main} = \frac{1}{2\pi R^2} \sqrt{\frac{D}{\rho h}} = \frac{1}{2\pi \cdot 0,125^2} \sqrt{\frac{34874,71}{4430 \cdot 0,015}} = 233,35$$

3. Changing the coefficient a^2

**(usually the coeff is λ^2 Leissa (NASA book) but it is connected to the diameter, as we have radius we need to divide by 2^2)*

got the mode values:

modes	λ^2 Leissa	$a^2 = \lambda^2$ Leissa /4	Frequency ABAQUS [Hz]	Frequency by hand [Hz]
1-6	0	0	0	0
7	5,25	1,3125	303	306
8	5,25	1,3125	303	306
9	9,07	2,2675	526	529
10	12,23	3,0575	702	713

A. W. Leissa, "Vibration of Plates," NASA SP-160 [verification2] ✓

CIRCULAR PLATES

11

TABLE 2.5.—Values of $\lambda^2 = \omega a^2 \sqrt{\rho/D}$ for a Completely Free Circular Plate; $\nu=0.33$

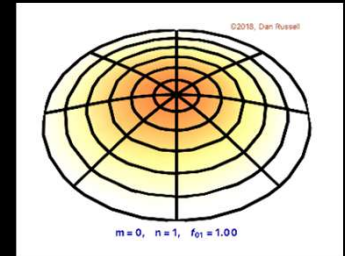
s	λ^2 for values of n of—						
	0	1	2	3	4	5	6
0							
1	9.084		5.253	12.23			
2	38.55	20.52	35.25	52.91	* 21.6	* 33.1	* 46.2
3	87.80	59.86	83.9	111.3	* 73.1	* 95.8	* 121.0
4	157.0	119.0	154.0	192.1	142.8	175.0	210.3
5	245.9	198.2	242.7	290.7	232.3	274.6	319.7
6	354.6	296.9	350.8	408.4	340.4	392.4	447.3
7	483.1	415.3	479.2	546.2	467.9	529.5	593.9
8	631.0	651.8	627.0	703.3	615.0	686.4	760.1
9	798.6	711.3	794.7	880.3	781.8	864.4	952.3
10	986.0	888.6	981.6	1076	968.5	1061	1158.7
		1086	1188	1292	1401	1513	1631

*the nomenclature for labelling the modes is **(d,c)** where **d** is the number of nodal diameters and **c** is the number of nodal circles.

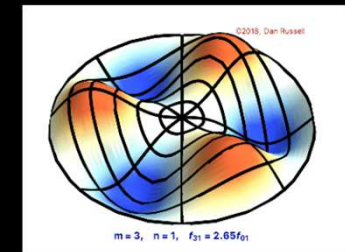
A **node** is a point (or line) on a structure that does not move while the rest of the structure is vibrating.

On the pictures, the nodal diameters and circles show up as white regions that don't oscillate, while the red and blue regions indicate positive and negative displacements.

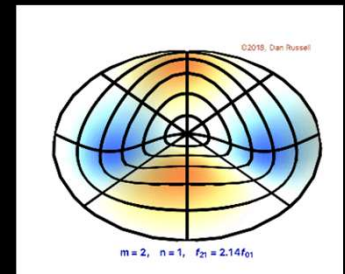
Mode (0,1) = (1,0)



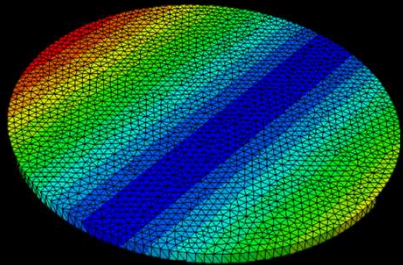
Mode (3,1) = (0,3)



Mode (2,1) = (0,2)

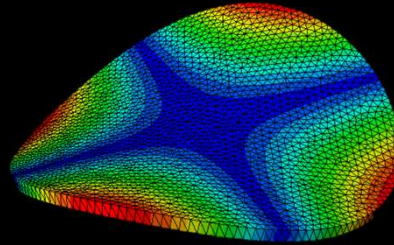


ABAQUS calculation [verification2] ✓



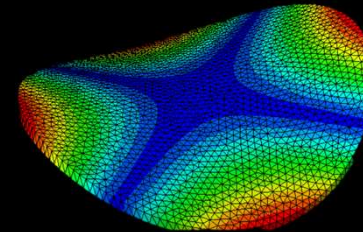
Mode 2: 0 Hz

Shape : lines



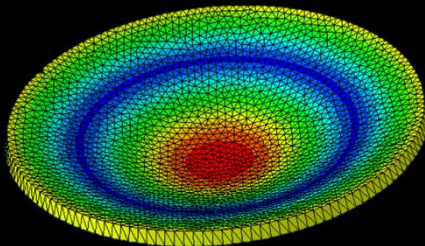
Mode 7: 303 Hz

Shape : saddle



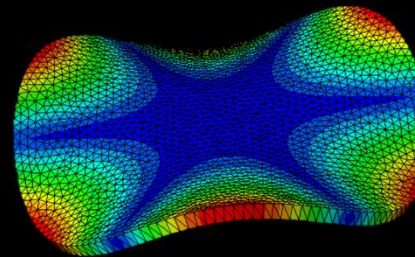
Mode 8: 303 Hz

Shape : saddle



Mode 9: 526 Hz

Shape : umbrella



Mode 10: 702 Hz

Shape : flower

**RIGID BODY MODE/
FREE FLOATING STRUCTURE**

2 disks platform

VS

1 disk

$$\sqrt{\frac{k}{M}}$$

Index	Description
0	Increment 0: Base State
1	Mode 1: Value = 1.05245E+06 Freq = 163.28 (cycles/time)
2	Mode 2: Value = 1.05900E+06 Freq = 163.78 (cycles/time)
3	Mode 3: Value = 2.45358E+06 Freq = 249.30 (cycles/time)
4	Mode 4: Value = 4.26795E+06 Freq = 328.80 (cycles/time)
5	Mode 5: Value = 4.27594E+06 Freq = 329.11 (cycles/time)
6	Mode 6: Value = 6.80623E+06 Freq = 415.22 (cycles/time)
7	Mode 7: Value = 6.80701E+06 Freq = 415.24 (cycles/time)
8	Mode 8: Value = 8.39462E+06 Freq = 461.13 (cycles/time)
9	Mode 9: Value = 2.36438E+07 Freq = 773.89 (cycles/time)
10	Mode 10: Value = 2.41441E+07 Freq = 782.03 (cycles/time)

Index	Description
0	Increment 0: Base State
1	Mode 1: Value = -3.60875E-05 Freq = 0.0000 (cycles/time)
2	Mode 2: Value = -2.33262E-05 Freq = 0.0000 (cycles/time)
3	Mode 3: Value = 3.45505E-07 Freq = 9.35507E-05 (cycles/time)
4	Mode 4: Value = 9.56283E-06 Freq = 4.92168E-04 (cycles/time)
5	Mode 5: Value = 1.85019E-05 Freq = 6.84586E-04 (cycles/time)
6	Mode 6: Value = 2.95021E-05 Freq = 8.64463E-04 (cycles/time)
7	Mode 7: Value = 3.64318E+06 Freq = 303.78 (cycles/time)
8	Mode 8: Value = 3.64319E+06 Freq = 303.78 (cycles/time)
9	Mode 9: Value = 1.09615E+07 Freq = 526.93 (cycles/time)
10	Mode 10: Value = 1.94916E+07 Freq = 702.66 (cycles/time)

free modes

elastic modes

Mode (0,2)

circular node
no nodal diameters

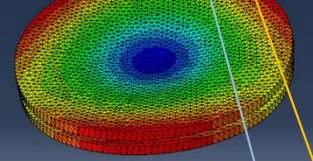
Mode (2,1)

possible explanation
joints behave as stiff
details -> $k \gg$ two
disks has more
frequency value

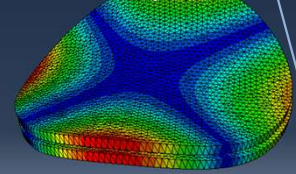
Mode (3,1)

high order modes
characterized by more
stiffness itself -> no
changes between

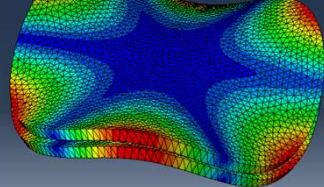
Mode 3: 249 Hz



Mode 6,7: 415 Hz



Mode 10: 782 Hz

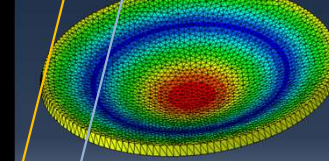


Shape : umbrella

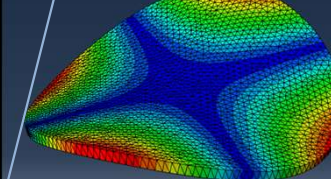
Shape : saddle

Shape : flower

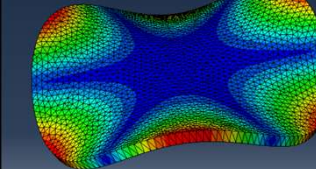
Mode 9: 526 Hz



Mode 7,8: 303 Hz



Mode 10: 702 Hz



K – not increasing
M- increase x2

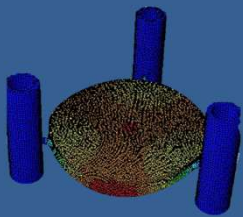
K – increasing by
supports
M- „don't affect“

K >>>

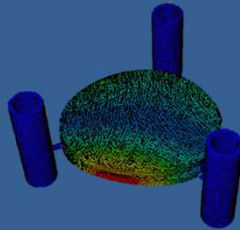
Numerical simulation results:

Staged verification: Configuration A

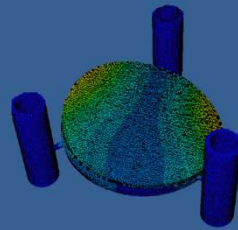
Mode1: 170 Hz



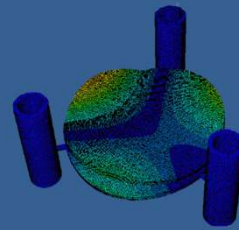
Mode2: 218 Hz



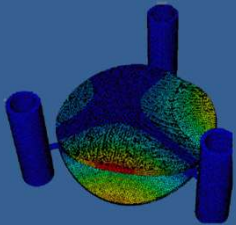
Mode3: 218 Hz



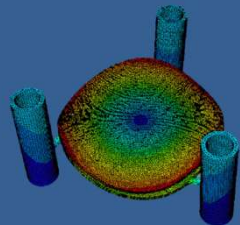
Mode4: 288 Hz



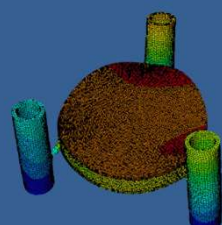
Mode5: 288 Hz



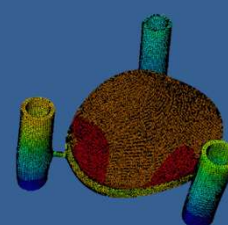
Mode6: 418 Hz



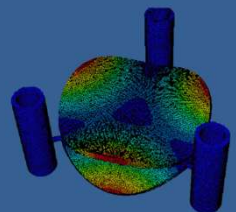
Mode7: 425 Hz



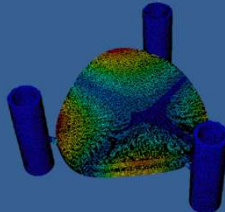
Mode8: 425 Hz



Mode9: 439 Hz



Mode10: 528 Hz



MODE NO	E F F E C T I V E M A S S		
	X-COMPONENT	Y-COMPONENT	Z-COMPONENT
1	40.871	0.10349	5.43837E-05
2	0.10350	40.872	3.42126E-06
3	7.98893E-05	2.57015E-06	29.651
4	1.29738E-06	6.85895E-08	1.69780E-06
5	4.12830E-02	0.20389	9.03917E-06
6	0.20614	4.09656E-02	2.11472E-07
7	0.15439	2.3624	6.87626E-07
8	2.3597	0.15467	3.16654E-07
9	0.24235	1.1472	3.81017E-05
10	1.1491	0.24135	1.90124E-05
TOTAL	45.128	45.127	29.651

MODES 1-5

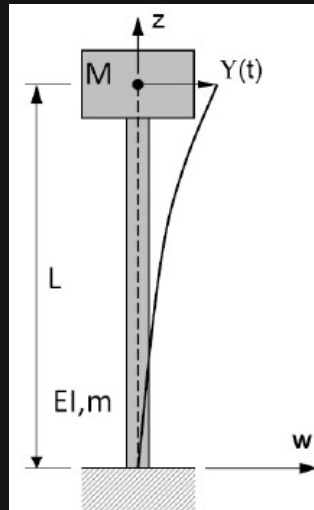
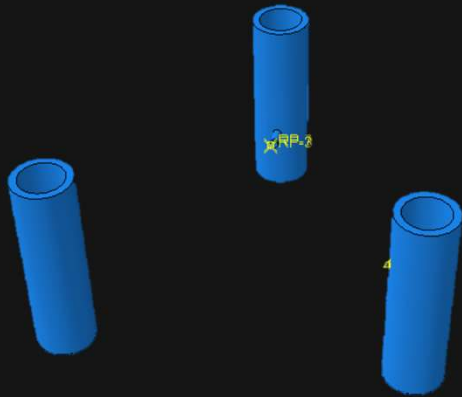
At this point, the legs act like **rigid springs**. They experience a tremendous shear force at their very base, but they don't bend or vibrate independently of the body. They simply swing along with the entire structure, like a pendulum.

Modes 1-5, 9-10: disk-bending dominated, legs remain quasi-stationary — consistent with disk flexural stiffness (not leg stiffness) governing these modes; frequencies close to isolated two-disk case.

Modes 6-8: [to confirm via strain-energy split] — candidate leg-participation modes

Numerical simulation results:

Calculation of the leg frequency



Cantilever beam with a tip mass at the end

$$f = \frac{1}{2\pi} \sqrt{\frac{3EI}{ML^3}}$$

f — first natural frequency , Hz

E — Young's modulus , Pa

110Gpa = 110*10⁹ Pa

I — second moment of area , m⁴

3,356*10⁻⁶ m⁴

M — tip/end mass , kg

26,17 kg/3 = 8,723 kg

L — beam length , m

0,33 m

The answer is 299 Hz

Numerical simulation results:

Staged verification: Configuration A, results with the flexure from Delft University

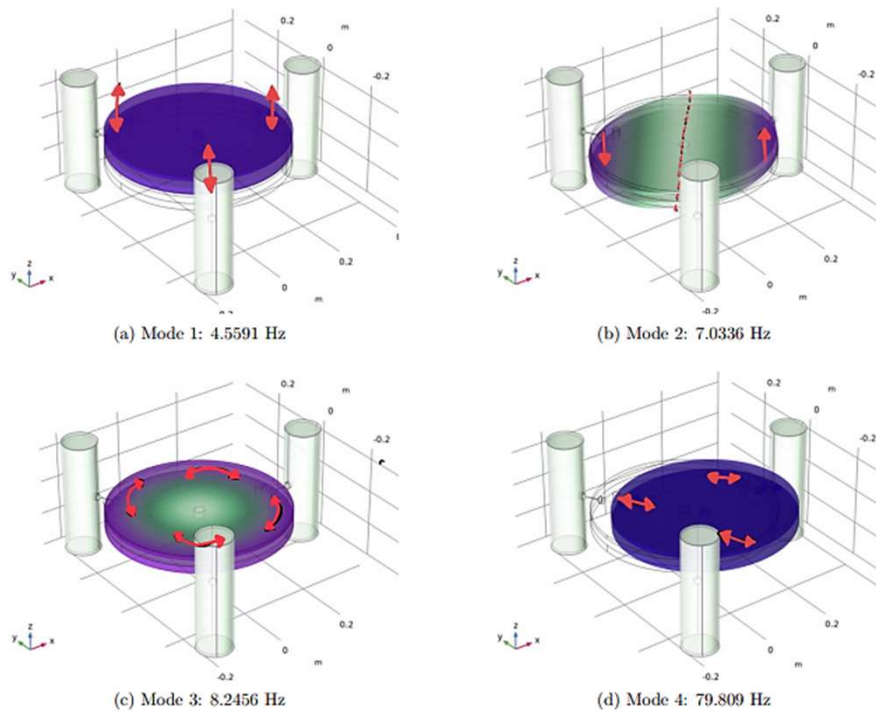
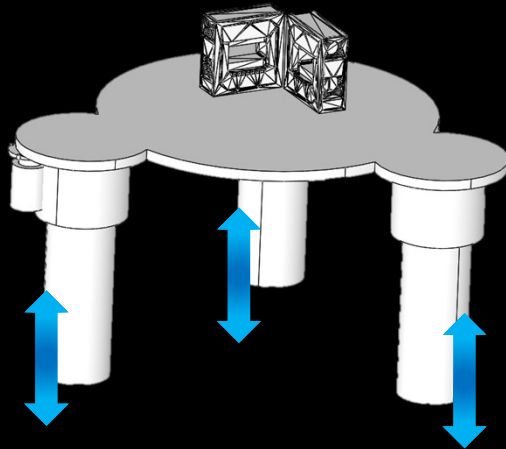


Figure 17: The four modes that were found from the eigenmode analysis using COMSOL.

For the joint system we used a simplified flexure joint, a solid cylinder. (Disclaimer: we know this is not the best suitable flexure joint, however it is an easy implementation to show how and what the resulting eigenmodes are). The required stiffness for a flexure joint should be 1661.5 N/m, calculated in section 3.5.1. When we know the material of the cylindrical flexure joint (specifically its Young's Modulus E), its length (L_{flex}) and radius (r_{flex}) we can calculate the stiffness with Equation 9 (Blom, 2015).

Solution with one level stage



Data: Arm (R=250mm)
 Steinmeyer 1214 screw (5mm lead)
 Phytron VSS/VSH motor (200 full steps/rev)

Nominal full-step resolution:
 5mm = 5000 μ m / 200 steps = 25 μ m/step
 → 25 μ m/250mm (w/o microstepping) = 100 μ rad/step
 → x2 over requirement

Requirement: θ_{max} = 45.7 microradians of tilt angle to prevent the mass from running onto the coil => otherwise there is a saturation of readout

Instead of using **two-leveling system** (CLA2603 + phytron stepper motor scheme) the idea is simplify it to the **one level system** with Phytron stepper motor+screw

Microstep	μ m / step (linear)	μ rad / step (at R=250mm)	Basis
1× (full step)	25	100	1.8°/step, 200 steps/rev (Phytron datasheet)
8×	3.1	12.5	nominal microstep (illustrative divisor)
32×	0.78	3.1	nominal microstep (illustrative divisor)
64×	0.39	1.6	nominal microstep (illustrative divisor)

The 100 μ rad/step is geometrically ideal case but real accuracy/repeatability $\approx$ **200 μ rad** vs 45.7 μ rad

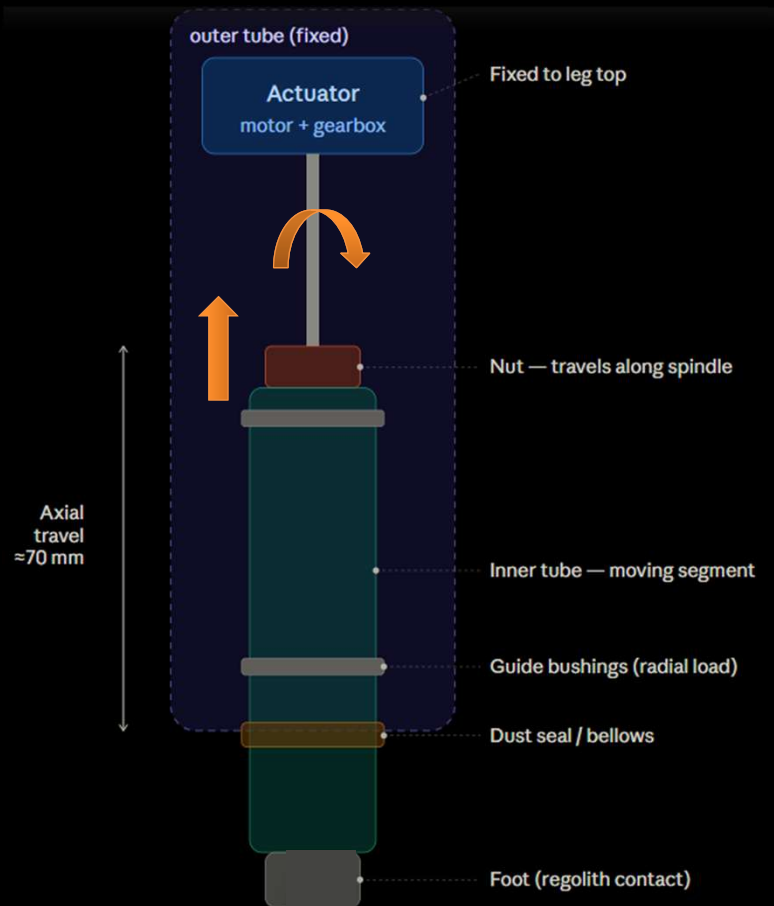
$$\text{Accuracy} = \Delta\theta_{step} + \Delta\theta_{screw} + \theta_{backlash}$$

$$\text{Repeatability}_{bl} \approx \theta_{backlash} + \Delta\theta_{friction}$$

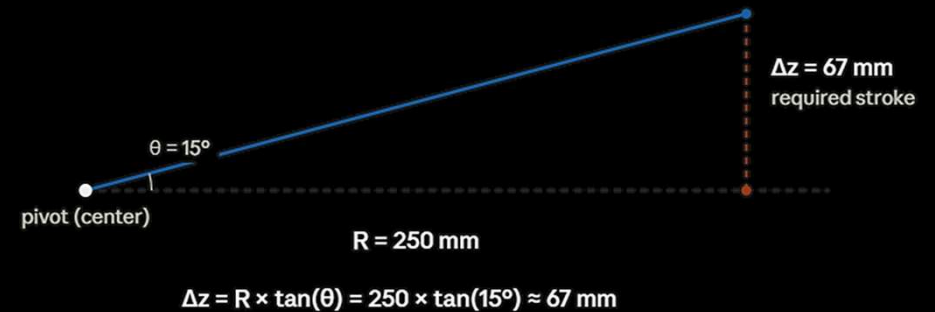
Single Phytron stage alone cannot meet the tilt requirement without either preload+closed-loop feedback (encoder) or the CLA fine stage reinforcing the two-stage architecture

Requirements:

Tilt range (coarse stage)



The Heavy-Duty Spindle Systems (Phytron GmbH)



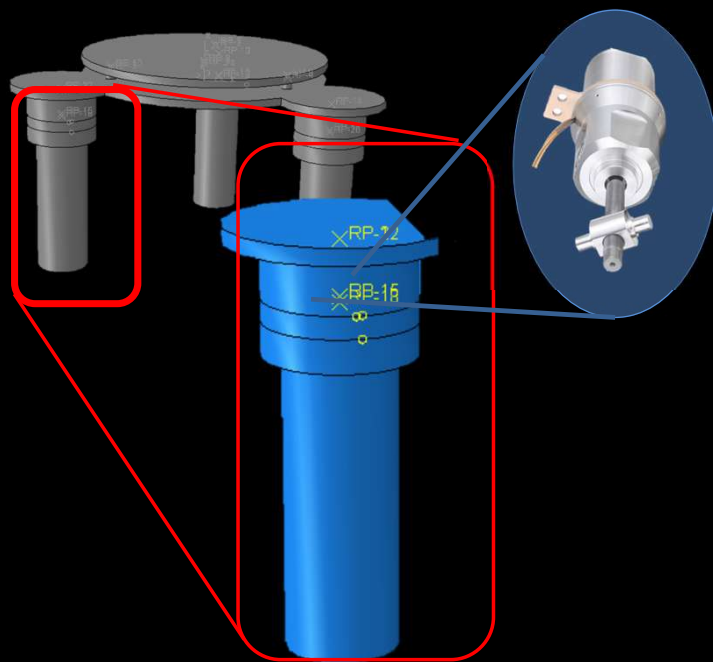
- 1300 N axial force is a capability spec (max load the actuator can push/hold against)
- Phytron LAV is dry-lubricated by design, rated to -270°C out of the box
- Domain heritage: developed for SRF cavity tuning (LCLS-II, Fermilab), extensively cryo/vacuum field-tested
- self-locking property depends on thread lead angle vs friction angle (self-locking, holding capacity of same order as 1300N rated axial force)

$$\lambda \text{ (lead angle)} = \arctan(1 \text{ mm} / (\pi \times \sim 11 \text{ mm})) \approx 1.66^\circ$$

$$\text{Friction angle (dry-lubricated, DL-5 coating)} \approx \arctan(0.1-0.15) \approx 5.7^\circ-8.5^\circ$$

$$\lambda (1.66^\circ) \ll \text{friction angle } (5.7^\circ-8.5^\circ) \rightarrow \text{screw self-locking}$$

Coarse stage idea



The Heavy-Duty Spindle Systems (Phytron GmbH)

Compact modular version

fully enclosed within the leg tube, the tube itself serves as the protective housing, minimizing external seals and impact-exposed surfaces.

Component

VSS52 motor
VGPL gearhead
Steinmeyer 1214 ball
screw
Travelling nut

Cryo/Vacuum

✓ Confirmed (-270°C , 10^{-11} hPa)
⚠ not documented separately (Likely OK)
⚠ Vacuum: yes (custom order)
Cryo: unconfirmed
⚠ Vacuum: yes (custom order)
Cryo: unconfirmed

Sidemount version

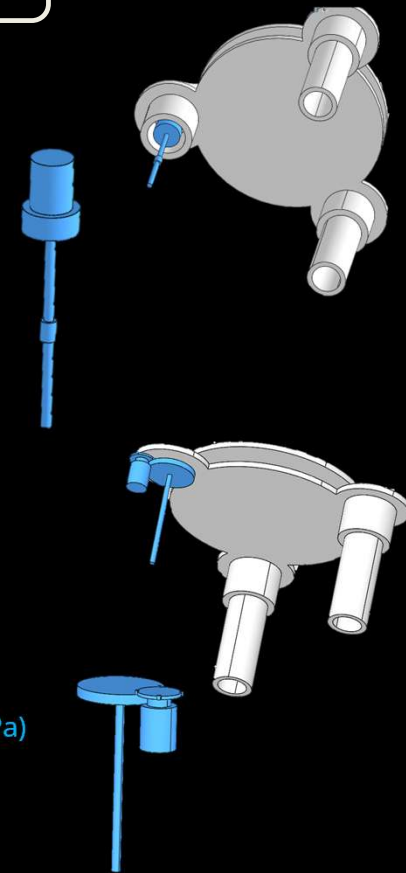
motor and gearbox sit external to the leg, requiring a separate housing and a sealed shaft penetration

Component

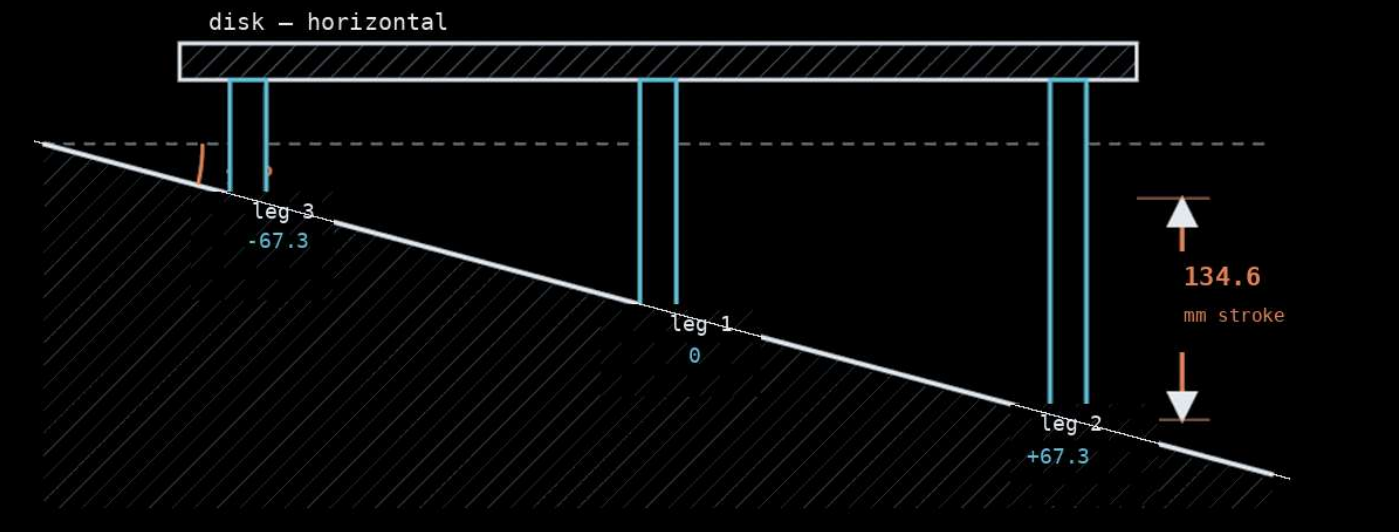
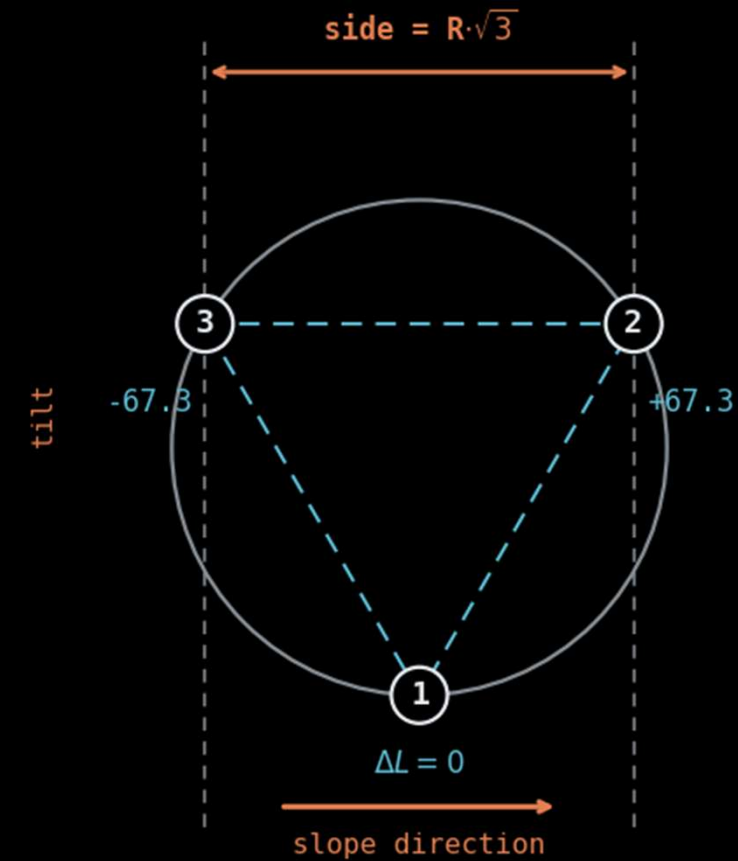
VSH motor ($\varnothing 52 \times 135 \text{ mm}$)
Right-angle gearbox
Coupling shaft
M5 bolts

Cryo/Vacuum

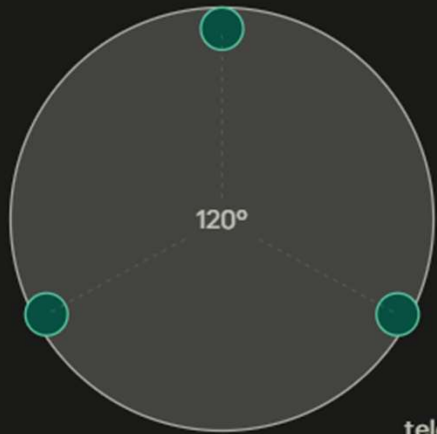
✓ Confirmed VSS/VSH family, -270°C , 10^{-11} hPa)
⚠ no cryo-rated part found yet
⚠ Custom-machined, not a catalog item
⚠ Standard - use vented type for vacuum



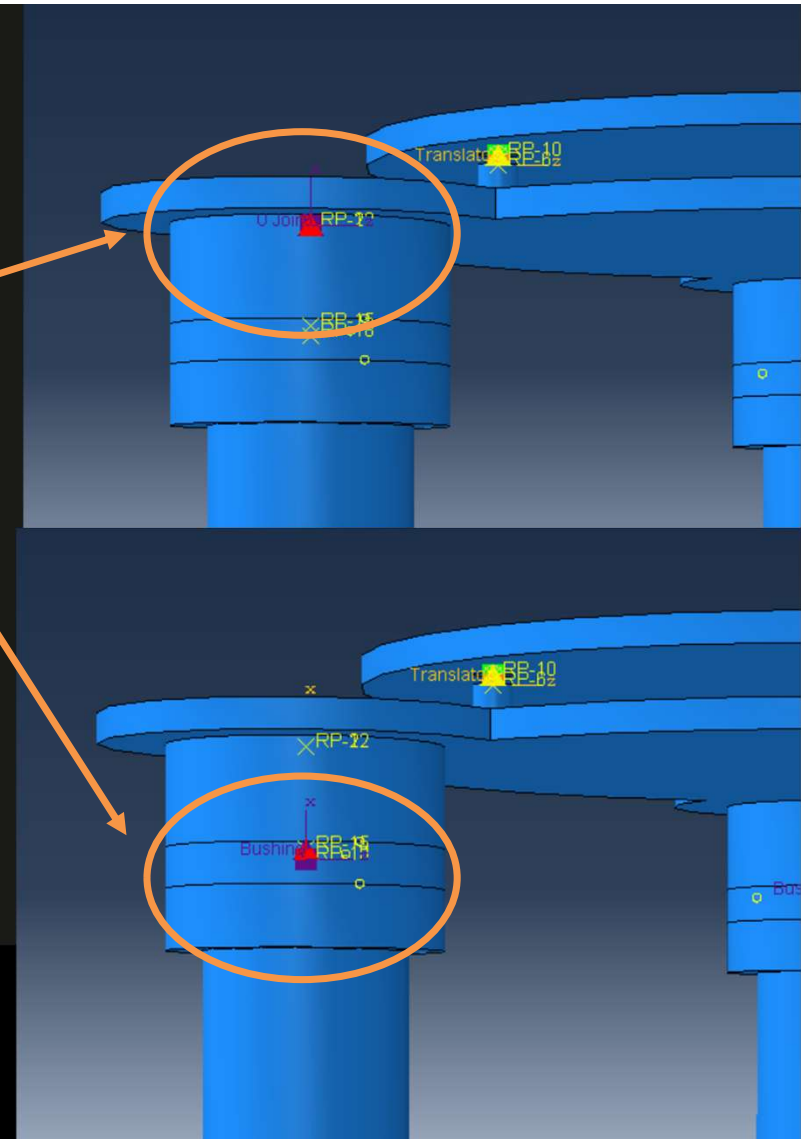
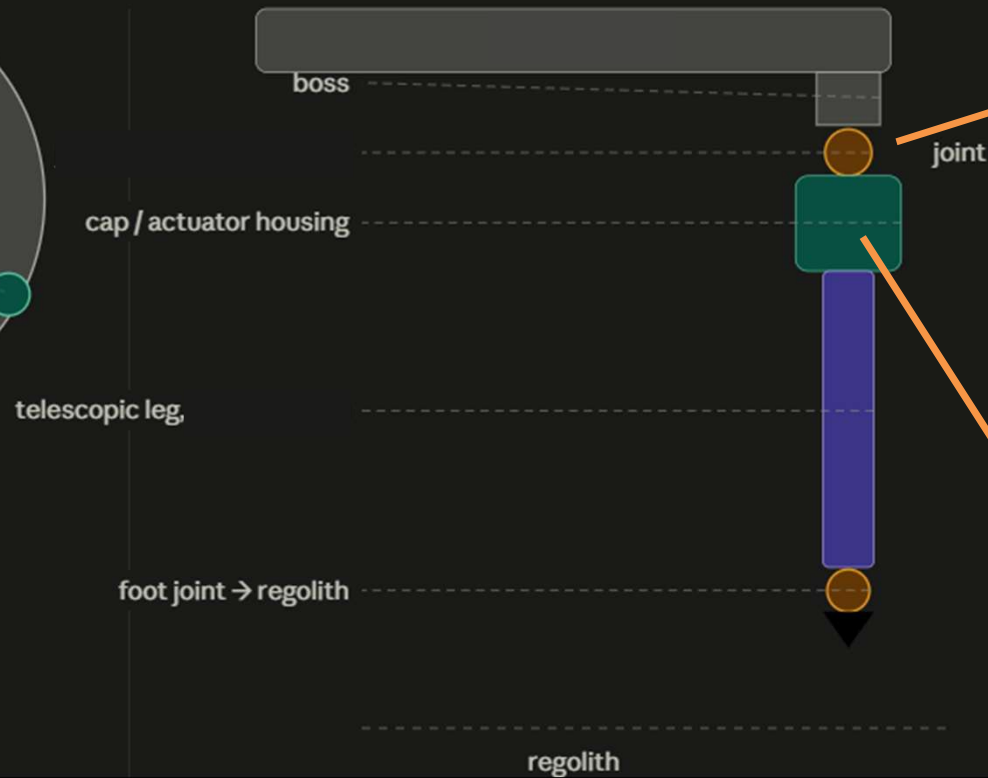
Worst case scenario (slope)



Top view



Side cross-section



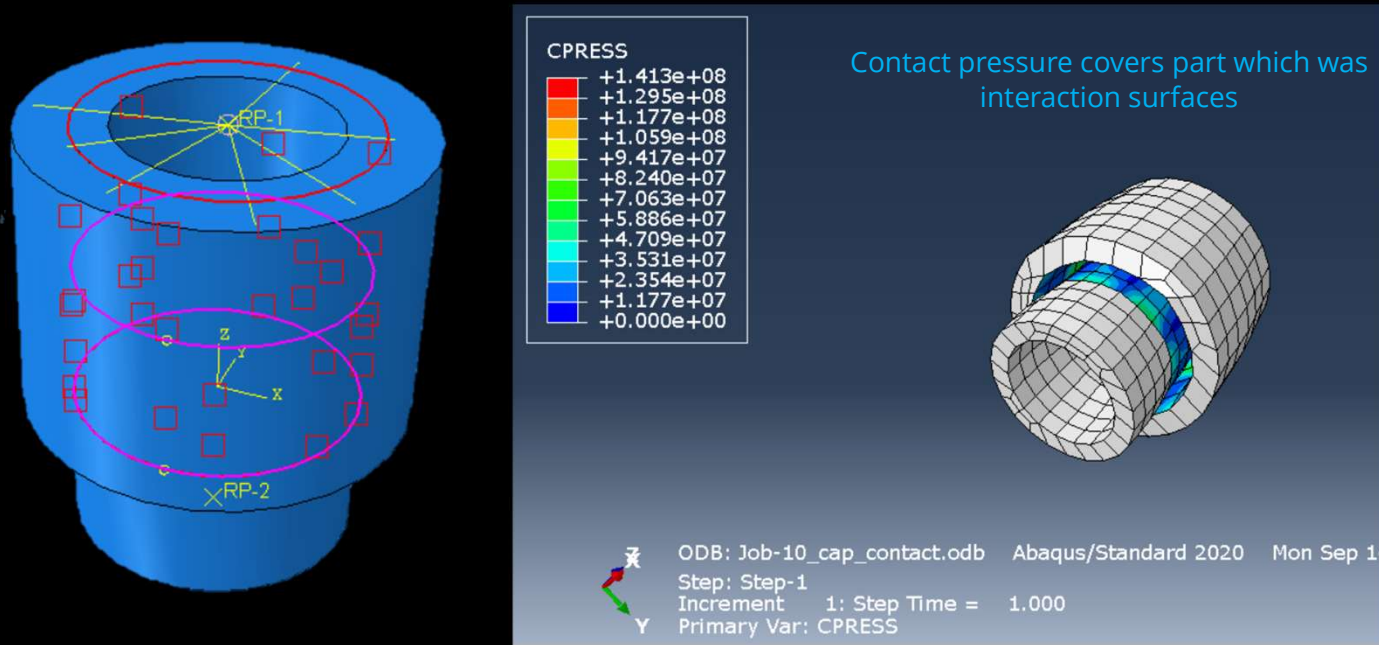
$$k_{\text{radial_per_plane}} \approx k_{\text{radial_total}} / 2 \approx 1.5 \times 10^8 / 2 \approx 7.5 \times 10^7 \text{ N/m} \quad (\text{one plane})$$

$$L = 0.045 \text{ m} \rightarrow L/2 = 0.0225 \text{ m}$$

$$k_{\text{bending}} \approx k_{\text{radial_per_plane}} * L^2 = 7.5 \times 10^7 \times (0.0225)^2 \approx 7.5 \times 10^7 \times 5.06 \times 10^{-4} \approx 3.8 \times 10^4 \text{ Nm/rad}$$

$$k_{\text{bending}} (\text{DOF5/6}) \approx 3.8 \times 10^4 \text{ Nm/rad}$$

Stiffness modelling



To model properly the joints in the telescopic leg it is needed to know coefficients.

In Abaqus it is for D11, D22/D33, D44, D55, D66

D11 calculated from the stiffness of the actuator.

For CLA2603 it is datasheet value = 8e7 N/m (axial)

For configuration with phytron motor it is calculated through 'screw as a rod' and total compliance by the formulae:

$$(\text{compliance})\lambda = \left(\frac{1}{k_{\text{tot}}} = \frac{1}{k_{\text{nut}}} + \frac{1}{k_{\text{screw}}} \right)$$

So, overall **D11** = 5e7N/m

For D22/D33 submodel of a CAP is presented

For D55/D66 they can be estimated from D22/D33

Between bottom disk and the CAP is U Joint connection => Available CORM: UR1, UR3

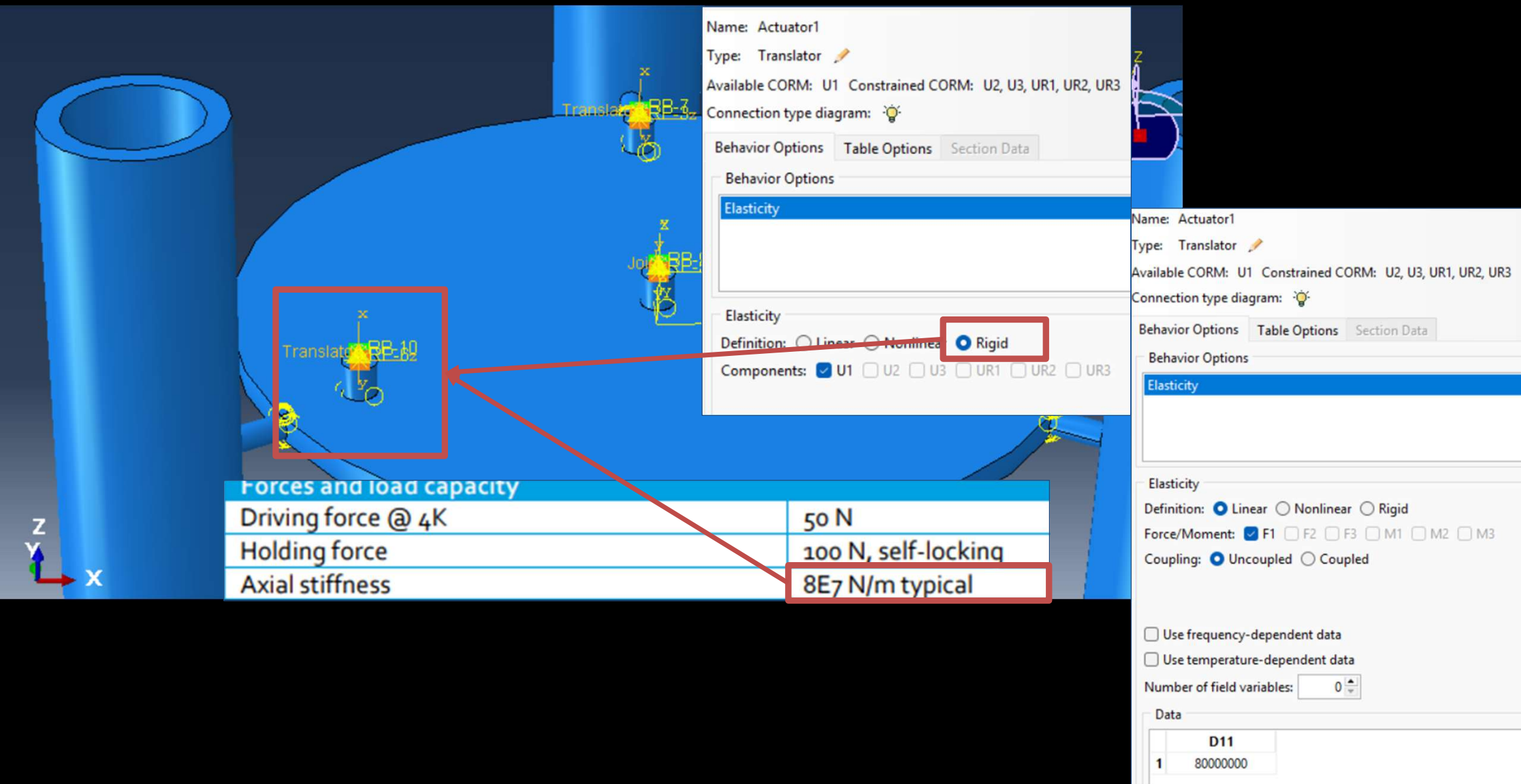
Constrained CORM: U1, U2, U3, UR2

Between the CAP and the leg Bushing connection => Available CORM: U1 (CLA/spindle), U2 (rad), U3(rad), UR1(torsion=0), UR2(bend), UR3(bend)

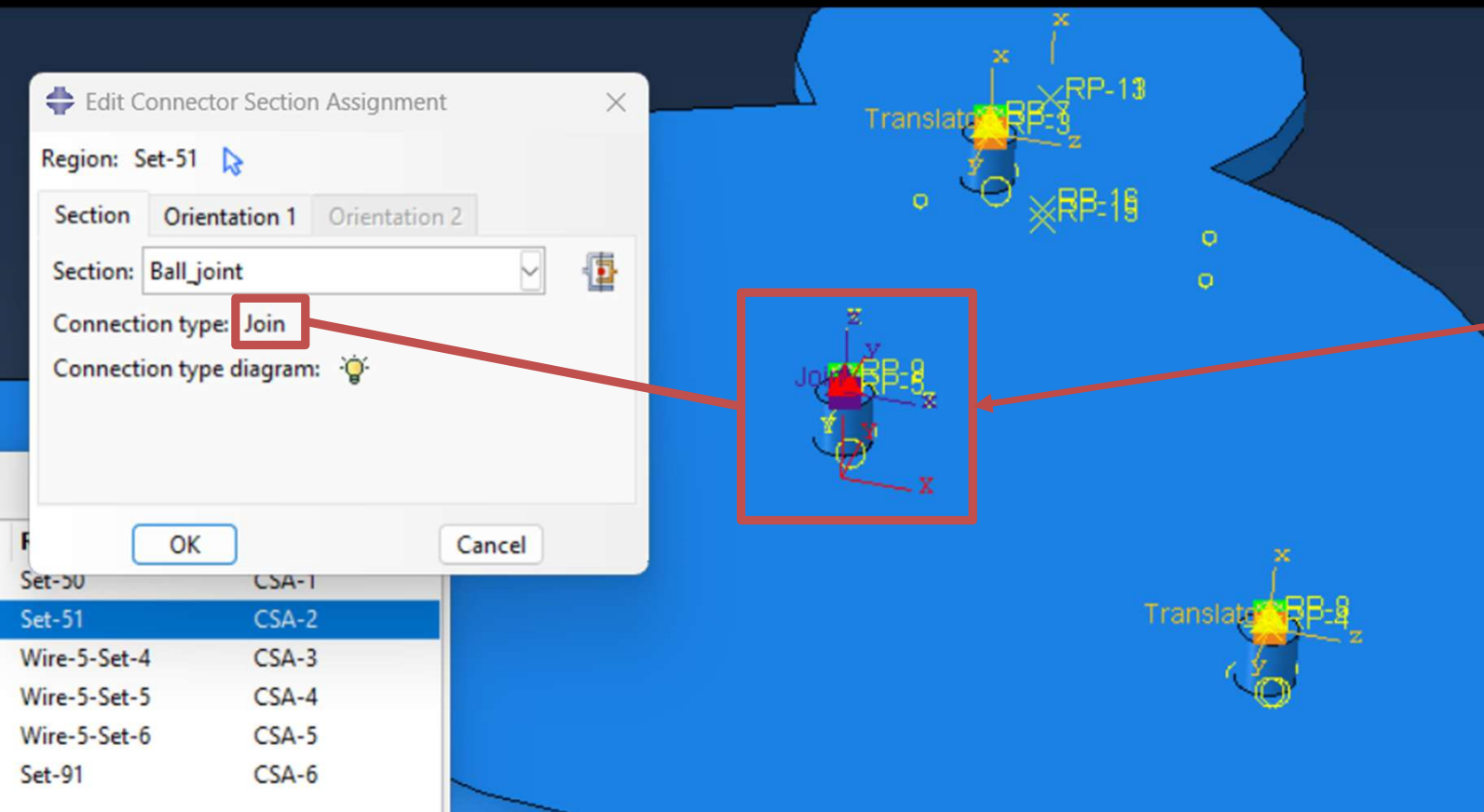
Constrained CORM: None

5 × 10⁷ 1,5 × 10⁸ 1,5 × 10⁸ (w/o D44) 3,8 × 10⁴ 3,8 × 10⁴

Simplified *Job-9_tie_CLA* model results:



Simplified *Job-9_tie_CLA* model results:



Modelling the **Ball-and-Socket joint**

In Abaqus, the **JOIN** connection is a **translational basic connector type** that forces the position of two nodes to remain identical throughout the analysis.

Specifically, it **locks all 3 translational degrees of freedom (DOFs 1, 2, and 3)** while leaving all 3 rotational degrees of freedom (DOFs 4, 5, and 6) completely **free**

Requirements

Platform Levelling System Specifications for LGWA (by Suresh)

Operating Temperature:	15 K - 100 K	(operation during testing at room T?)
Thermal cycling		
15 K to Room T	100 cycles	
15 k to 100 K	1000 cycles	
Life cycle		
roundtrips of 60 mm	50	(Phytron/coarse)
roundtrips of 5 mm	100	(CLA fine range / micro-step)
roundtips of 0.1 mm	10000	(CLA fine range / micro-step)
Payload mass on the platform:	30 kg	
Levelling platform width:	50 cm	
Duration of the mission:	10 yrs	
Load per actuator on earth:	4 kg → 40 N	(only rotational inertia load)
Load per actuator on Lunar Surface:	4 kg → 7 N	(only rotational inertia load)
Holding strength	10 kg → 100 N	(matched to CLA2603 from JPE)
Safety factor:	x2	
Linear actuator range:	10 mm	(gross tilt corrected by external steppers)
Linear actuator precision:	$\theta = d/R$	1 microns → 2 micro-rad (R=500mm) 1 microns → 4 micro-rad (R=250mm)
Linear actuator accuracy (repeatability)	$\theta = d/R$	5 microns → 10 micro-rad (R=500mm) 5 microns → 20 micro-rad (R=250mm)

Continuous motion range (stepless extension)

Room Temp	10 microns
Cryogenic	3 microns
Linear velocity:	
Min	4 nm / sec
Max	1.5 mm / sec
Settling time to final hold position	10 sec
Hold position without power consumption	required (consistent with the self-locking property of CLA2603)
Holding strength against linear force	100 N
Vibration tolerance during launch	100 g acceleration
Vacuum compatibility	10 ⁻⁹ Torr

Payload LGWA.Table 1 [1]

TABLE I. Mechanical, readout, and electronic parameters for the analyzed inertial sensor configuration. The second set of parameters pertain to the used proof-mass readouts: the homodyne interferometer and inductive SQUID readout.

Parameter	Value	Unit
Proof mass	10	kg
Natural frequency	0.25	Hz
Temperature	5	K
Nb quality factor	$1 \cdot 10^4$	...
Si quality factor	$1 \cdot 10^6$	...
Coil-superconductor gap	0.1	mm
Actuator strength	5	$\mu\text{N/A}$
Frequency noise ^a	$500 \times f^{-1/2}$	Hz/ $\sqrt{\text{Hz}}$
Static differential arm length	0.5	mm
Injected laser power	10	mW
Wavelength	1550	nm
TIA feedback resistor	20	k Ω
SQUID energy resolution, E_A	$50 \hbar$	J/Hz
Signal to SQUID coupling efficiency, $\eta\beta$	0.25^{42}	...
$1/\sqrt{f}$ corner frequency, f_c	10^{43}	Hz

^aTypical value for high-end lasers, e.g., The RockTM from NP Photonics.⁴¹

Cryo Linear Actuator 2603

CLA2603 – CRYO LINEAR ACTUATOR 2603



Features

- CLA model with highest force output
- Nanometer step size
- Very large driving force
- 20 mK to 375 K, vacuum compatible
- Extreme stability due to self-locking nature
- No power dissipation when "off"
- Robust design for easy handling
- Full ceramic screw and nut
- Position feedback option -COE

Description / Applications

The CLA2603 is the model with the highest force output in the Cryo Linear Actuator (CLA) series. A CLA can be considered as a piezo driven setscrew with high force output and very good stability, ideal for set and forget applications in a cryo-vacuum environment. Piezo ceramics in the disc-shaped head generate torque pulses which cause the connected screw to rotate in the stationary nut, yielding linear motion at the ball tip. Both screw and nut are ceramic and run without coatings or lubrication.

Specifications

General info	
Type of motion	Linear, with rotating ball tip
Dimensions	See drawings below
Operational environmental conditions	20 mK to 375 K, ambient to UHV
Weight	39 g, -COE adds 9 g
Preload	The ball tip must be axially loaded >1 N for proper operation
Stepping motion	
Travel range	12 mm, reduced to 6 mm for -COE
Velocity @ 300 K	25 $\mu\text{m/s}$
Velocity @ 4 K	15 $\mu\text{m/s}$
Minimal step size @ 300 K	15 nm
Minimal step size @ 4 K	5 nm
Drive voltage	-30 V to 120 V
Forces and load capacity	
Driving force @ 4K	50 N
Holding force	100 N, self-locking
Axial stiffness	8E7 N/m typical
Materials	
Main body	Stainless steel 316L
Piezo actuator	Low voltage multilayer, ceramic insulated
Screw and nut	Ceramic, thread pitch 250 μm
Model specific information	
-COE	Cryo Optical Encoder, 850 pulses per revolution, equivalent to 294 nm axial displacement
-V, -HV or -UHV	Level of vacuum compatibility, see the glossary on the homepage for details
Electronics CPSC	
Controller Base Cabinet	CAB
Driver for stepping and scanning	CADM
Position readout	OEM

MarsInsight VS LGWA

Specifications	InSight	LGWA
<i>Tilt accuracy</i>	0.044 rad	10 μ rad
<i>Tilt range</i>	15°	15°
<i>Platform Diameter</i>	25 cm	50 cm
<i>Temperature Range</i>	80-300 K	40-100 K

Gravity modelling

Edit Load [X]

Name: Gravity

Type: Gravity

Step: Step-1 (Static, General)

Region: (Whole Model) [Mouse Icon]

Distribution: Uniform [Dropdown] $f(x)$

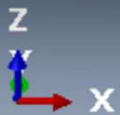
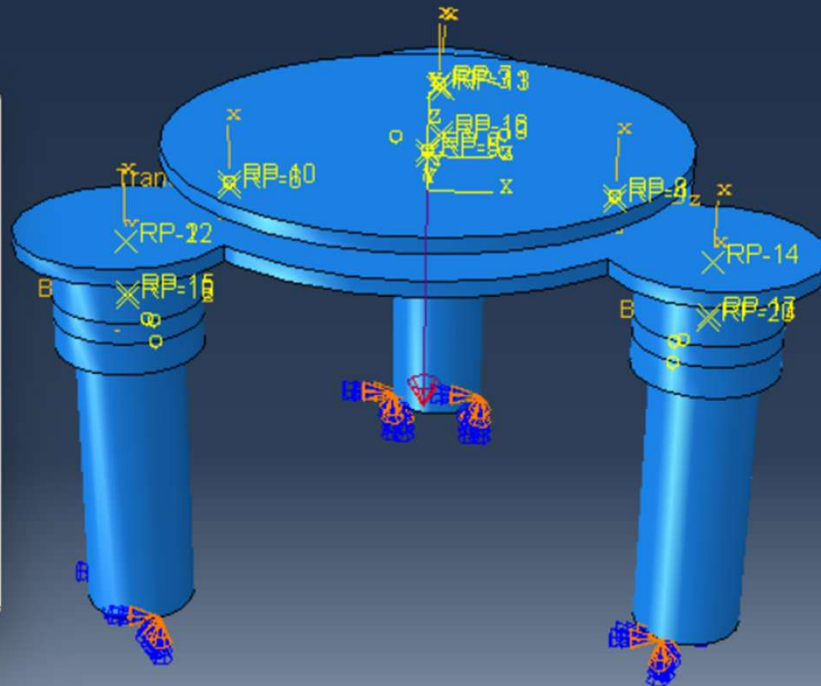
Component 1: 0

Component 2: 0

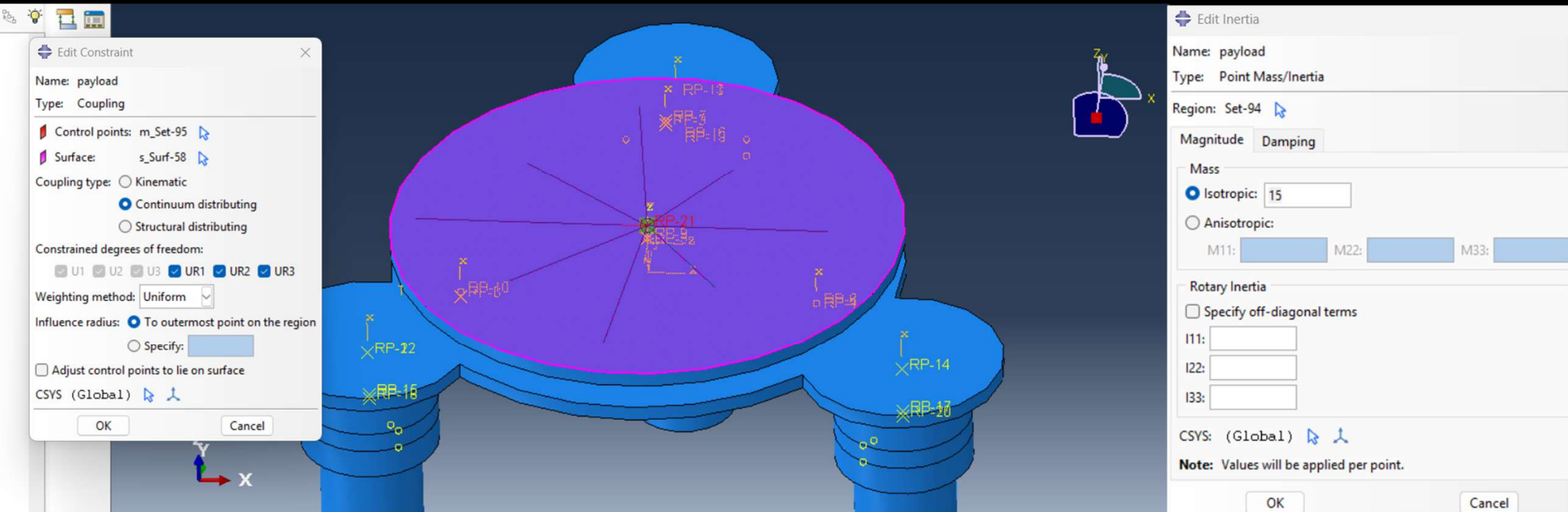
Component 3: -1.62

Amplitude: (Ramp) [Dropdown] [Waveform Icon]

OK Cancel



Payload modelling



Payload mostly consist of proof mass Nb (niobium):

$\rho = 8,57 \text{ g/cm}^3$

1 block : $12\text{cm} \times 12\text{cm} \times 4,5\text{cm}$ Volume = 648cm^3

Mass = 1 block = $5,55\text{kg}$, 2 blocks = 11kg -> round up to **15 kg** (electronics, subsystems etc)

Requirements: (~launcher Ariane 64)

Vibration tolerance during launch

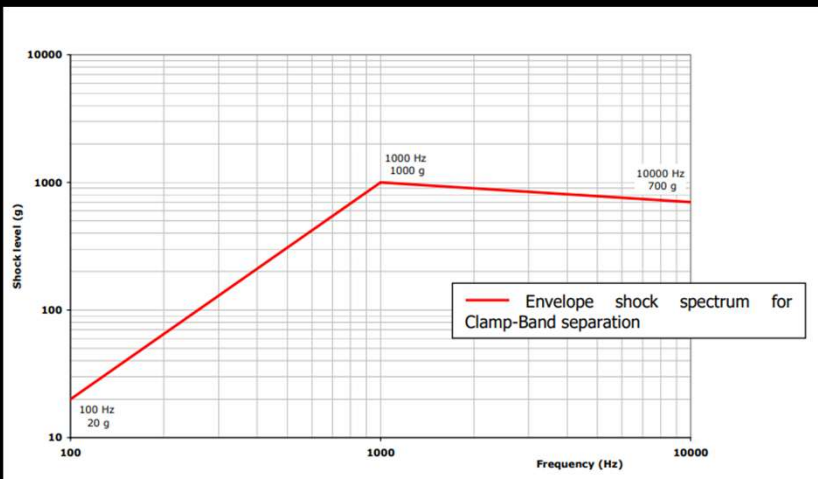
Mass: Trajectory to the Moon (TLI / Trans-Lunar Injection) up to **8,000–8,500 kg** (depending on adapter parameters and the specific lunar mission profile).

Important: This mass includes not only the spacecraft itself but **also the interface systems-the payload adapter (PAF) or dispenser.**

Dimensions: In its stowed configuration, the spacecraft must strictly fit within the available static and dynamic envelope. Maximum usable **diameter** for the satellite: approximately **4.57 meters** (accounting for clearances for thermal insulation and dynamic oscillations during flight).

Vibration/Shock Tolerance / Launch Environment:

The leveling platform structure shall withstand launch-induced shock and vibration loads without damage or performance degradation.



Peak shock inputs of up to 1,500 g at the spacecraft interface are attenuated to 100–200 g at the device housing via elastomeric or metal-mesh shock isolators.

The structural assembly is most vulnerable in the **100–300 Hz** band, where sustained overloads of **20–100g** are expected; first eigenfrequency of the assembly shall be kept outside this band with adequate margin



Requirements: (~launcher LVM3)

Vibration tolerance during launch

Solid rocket boosters (S200) **LVM3** typically produce a more **resonance-heavy vibration** environment than a purely liquid-engine launch due to combustion instability and pressure oscillations in the solid propellant.

This may mean more **stringent low-frequency vibration requirements (10-50Hz)** than Ariane, even if peak shock levels are comparable.

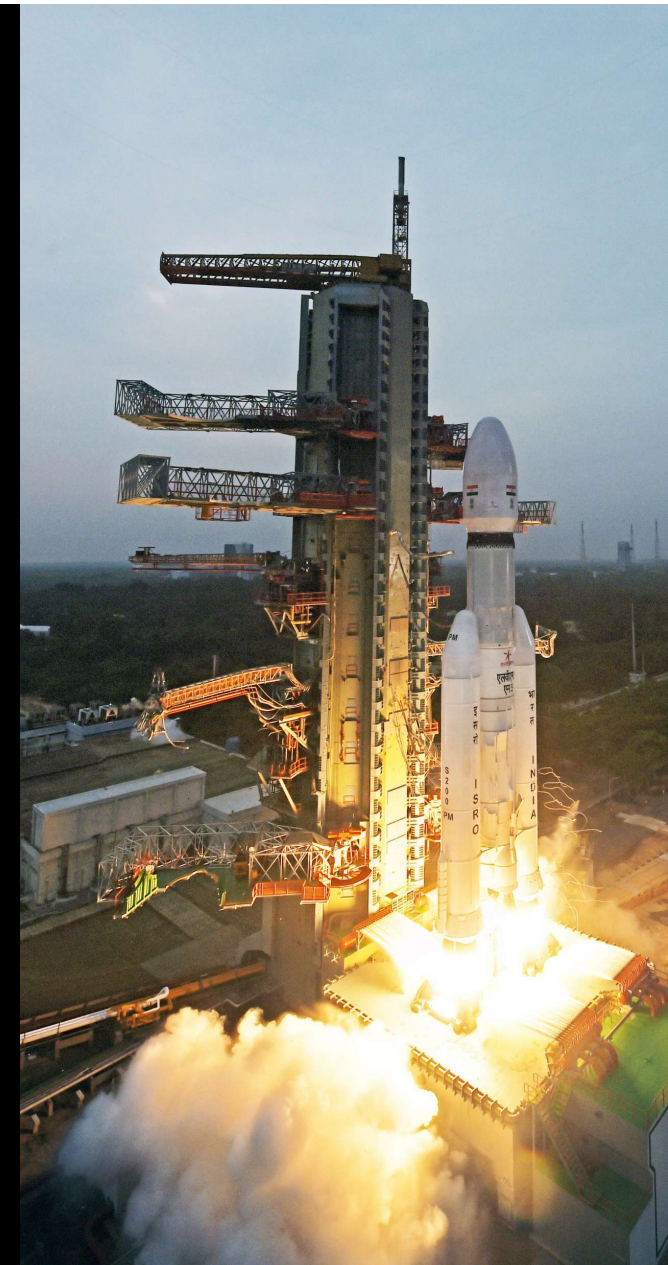
This means that for LVM3, two separate sounds need to be tested separately:

- **10-50 Hz** (solid rocket booster thrust oscillations, typical for LVM3)
- **100-300 Hz** FEA critical range, a universal risk zone⁶ relevant for both launchers of the same type

Generic/enveloping environment

Vibration Tolerance: 100 g (quasi-static equivalent design load, component/instrument level, generic conservative estimate - consistent with **3 σ** random-vibration envelope per GEVS-class baseline)

Source: NASA GSFC-STD-7000A (GEVS),
https://soma.larc.nasa.gov/lws/pdf_files/3.12%20GSFC-STD-7000A.pdf



Requirements:

Vacuum compatibility



Operating Conditions:

Vacuum compatibility **10^{-9} Torr**

The Moon has virtually no atmosphere. In permanently shadowed regions (PSRs, $<40\text{K}$), volatiles freeze onto the surface and become trapped as cold traps, producing a naturally occurring ultra-high vacuum environment.

Structural materials must be UHV-compatible:

- Extremely low outgassing rate
- No release of volatile contaminants from polymers, lubricants, or coatings
- Long-term stability over the 10-year mission duration
- Critical for reliable operation of the cryogenic sensors within the LGWA stations