A Conceptual Design for a 20 tonne Space Elevator Climber

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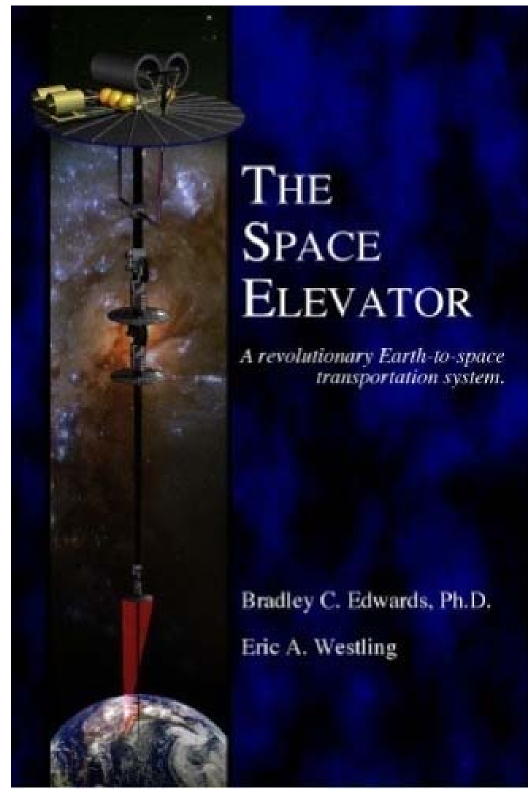
My hobby/obsession since 2004

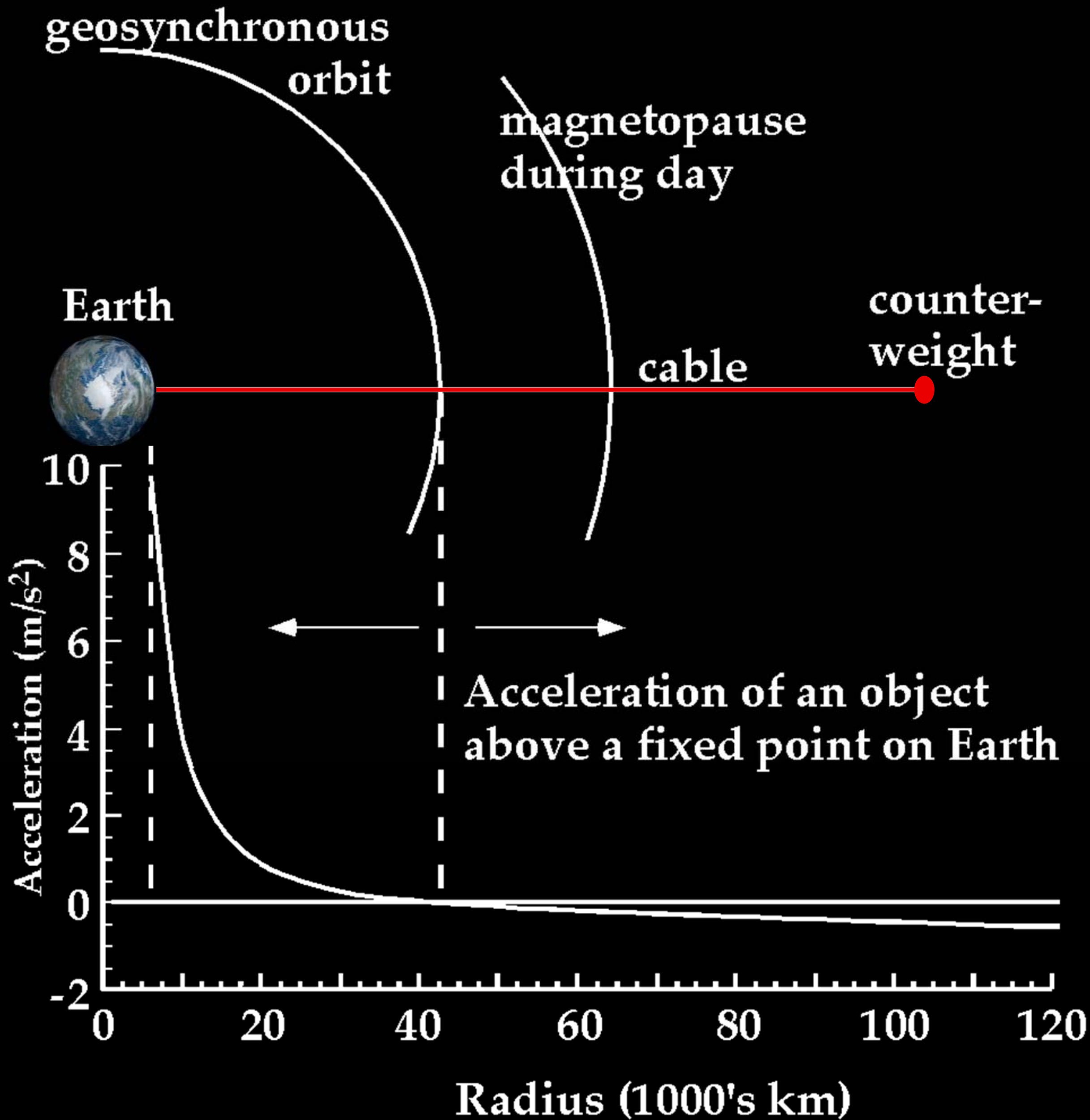
- It all started with reading Edwards and Westling's book, "The Space Elevator—A revolutionary Earth-to-Space transportation system
  - Still a must-read for SE afficionados
  - Edwards and Westling outlined most of the features of a complete SE business/transportation system



The basics of a space elevator

- The space elevator is like a string with a rock tied on one end and the other end tied to the surface of the Earth at the Equator
- I am the Earth in the picture to the right and I am spinning so the string stretches out
- An ant is walking out along the string to get into “space”

Space Elevator Basics



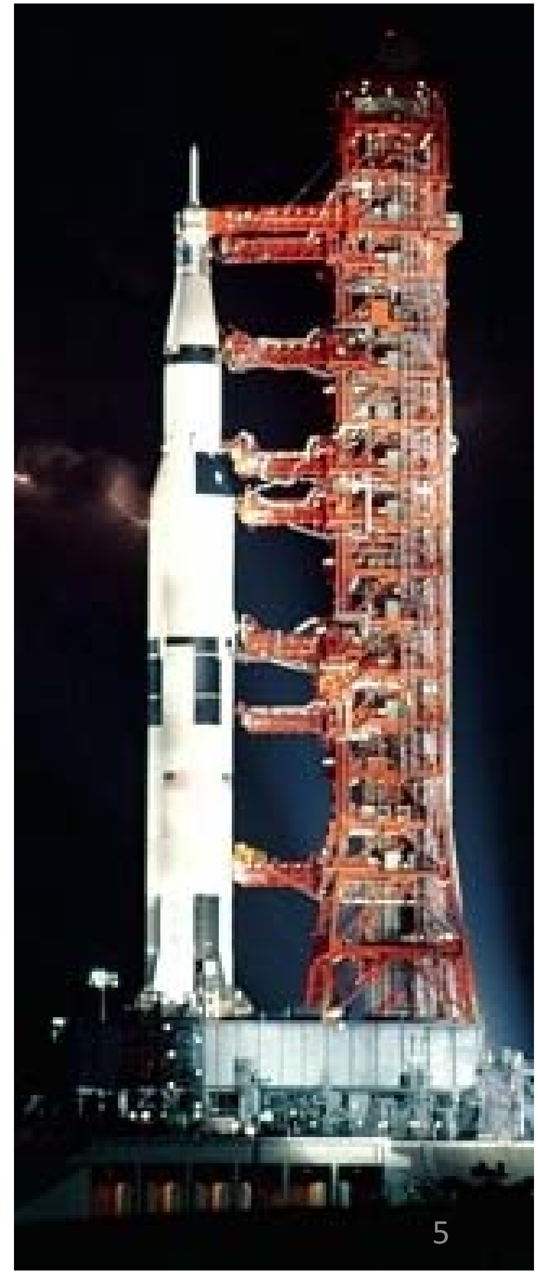
Gravity falls off really fast as $1/r^2$

This graph is scaled, so the elevator really is longer than the diameter of the Earth.

The only point on the elevator that is truly in orbit is the point at GEO

Why not just use rockets?

- Rockets have to carry their fuel with them
- Most of a rocket is fuel to get away from the Earth and go fast enough to be in orbit, so rocket payloads are a few percent of the rocket weight
  - You can't get better than this with chemical rockets
- It cost $25,000 per kilogram to put something in LEO from the Space Shuttle
  - SpaceX has lowered that to $2,720/kg
- The space elevator may be powered by lasers from the ground or space so it has no onboard fuel—it has a higher payload ratio
- The cost could go down to $250 per kilogram (or less) with the space elevator



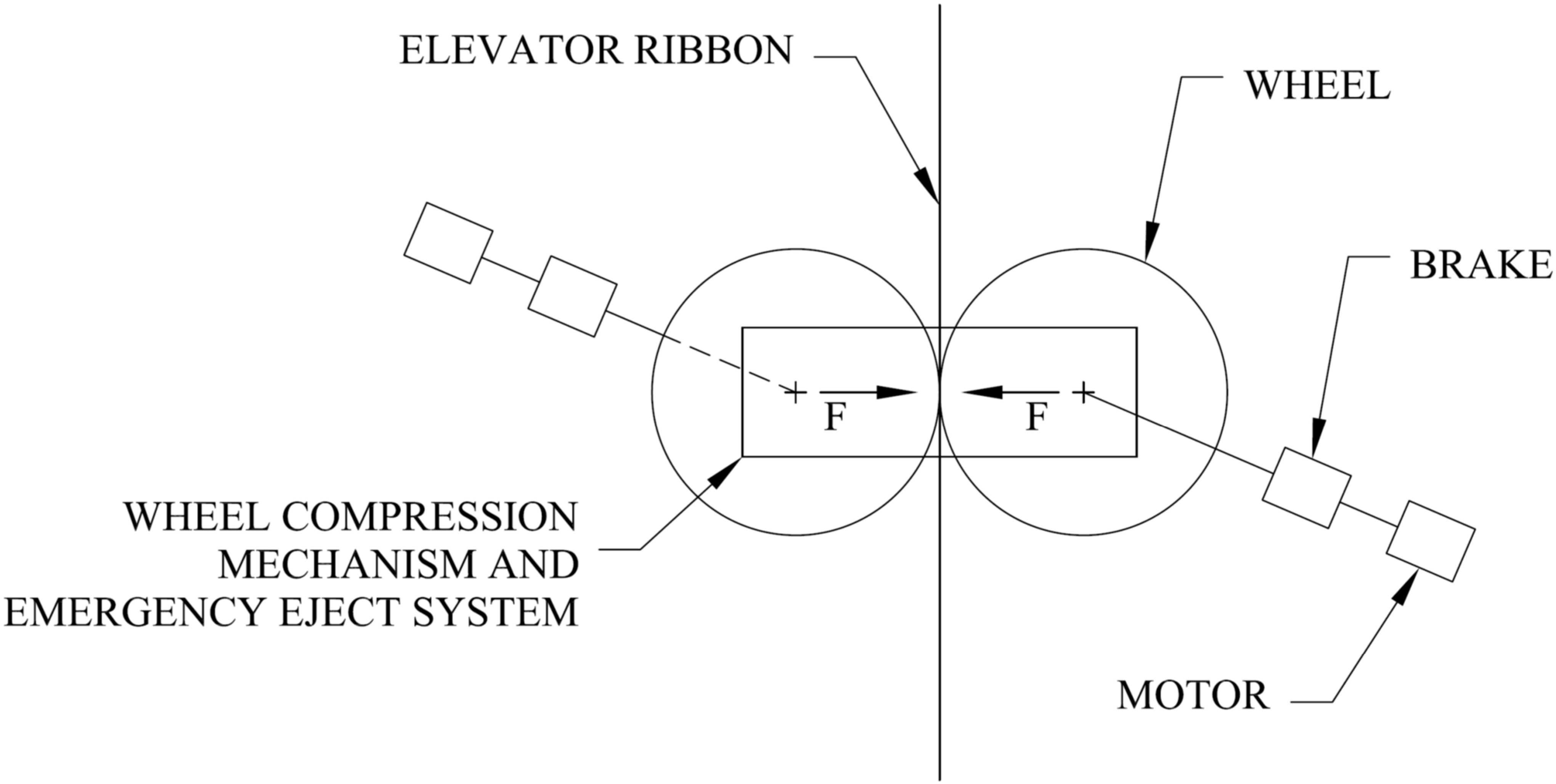
Every design starts with Requirements

1. Get to GEO in about a week
2. Carry as much payload as possible for a total climber mass of 20 tonnes
   - In his book, Edwards estimates 13 tonnes of payload and 7 tonnes of traction drive (*I'll show how that turned out.*)
   - 20 tonnes is considered the first commercial climber
3. The climber must not fail in a way that jeopardizes the tether

Assumptions for this design, #1

- This climber assumes a space elevator tether shaped like a ribbon with a constant thickness and a changing width as you get close to GEO, (determined by the *taper ratio*)
  - The width of the ribbon at Earth is .31 m
  - The width of the ribbon at GEO is 1.55 m
  - The thickness of the tether is 10 microns
- The design is a **pinched wheel concept** with no capstanning (see slide 8 for schematic)
  - I don't trust the traction forces I've seen calculated from capstan designs

What is meant by a "pinched wheel" climber



This is the schematic model of one wheel-pair of a pinched wheel climber design.

Assumptions, #2

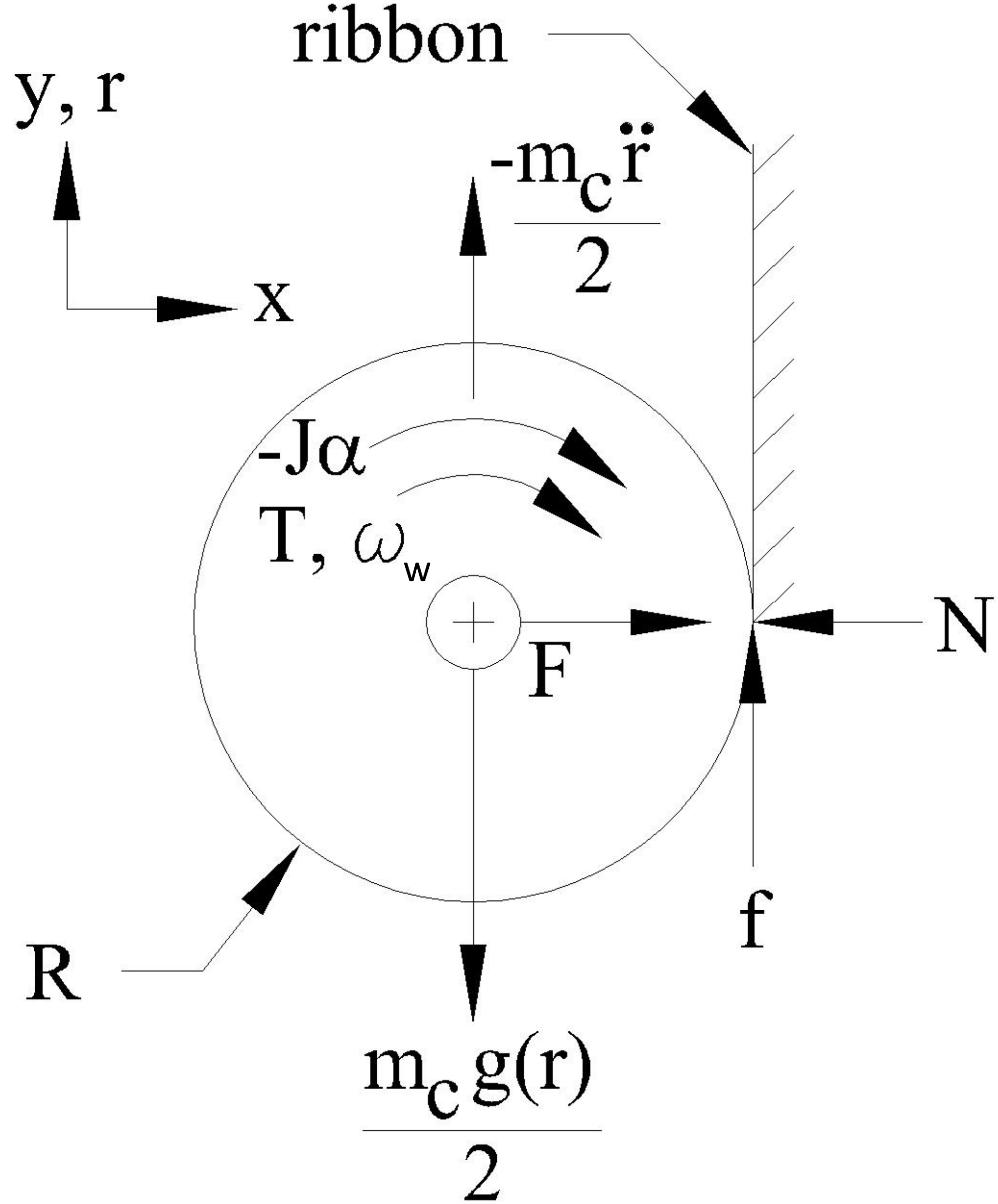
- The tether has a coefficient of friction with the wheels of μ~0.1
  - Carbon nanotubes or graphene are known to be slippery
  - If we can’t get friction to be at least this good we have big problems
- The design will only use commercially available products, if possible
  - The components may not be spaceworthy, but are existence proofs
- The assumptions come from 18 years of working on climber designs, and 39 years of being a mechanical engineer

Start with the basics

- The next slides show the mathematical model of the climber reduced down to the interaction between the wheel and the tether

- From the mathematical model, every component of the climber can be sized

Free Body Diagram of a Wheel



This picture models a single wheel on a climber with just two wheels

f = friction force from ribbon

F, N are compression and reaction forces pinching wheels on opposite sides of the ribbon together

This diagram allows us to write the equations of motion for the climber and determine all the forces acting on the climber

Defining the terms in the diagram

- R = radius of the wheel
- N = normal force between ribbon and wheel
- F = applied force compressing wheel to ribbon
- T = applied torque from motor
- m_c = mass of the climber
- f = friction force between ribbon and wheel
- g(r) = gravitational drag force expressed as a function of r, radius from the center of the Earth

$$g(r) = \frac{M_e \cdot G}{r^2} - r \cdot \omega^2$$

- ϑ = Angle of rotation around the axis of the wheel, radians

- G = Newton's gravitational constant

$$G = 6.67 \cdot 10^{-11} \cdot \frac{m^3}{\sec^2 \cdot kg}$$

- M_e = mass of the Earth
- M_e = 5.9788E24 kg
- ω = angular velocity of the Earth about its axis
- ω = 7.2929E-5 rad/sec
- J = rotary moment of inertia of wheel
- α = rotational acceleration of wheel, sec^{-2}
- $\ddot{r}$ = linear acceleration along ribbon
- x, y = Cartesian coordinates, y along ribbon, x perpendicular to face of ribbon
- $r = R\theta$,
- $\dot{r} = R\dot{\theta}$ and
- $\ddot{r} = R\ddot{\theta} = R\alpha$

Summing the moments

$$\sum M = T - \frac{m_c \ddot{r} R}{2} - \frac{m_c g(r) R}{2} - J\alpha = 0$$

Rearranging terms to get the torque required to accelerate the climber:

$$T = \ddot{r}\left(\frac{J}{R} + \frac{m_c R}{2}\right) + \frac{m_c g(r) R}{2}$$

The 2 in the denominators comes from having 2 wheels on opposite sides of the tether carrying the load equally. We can generalize by changing the 2 to n_w, the number of wheels in the climber.

If the climber is moving with constant velocity, the first term is zero. The second term never goes away. It is the torque required just to hold the weight of the climber up on the ribbon.

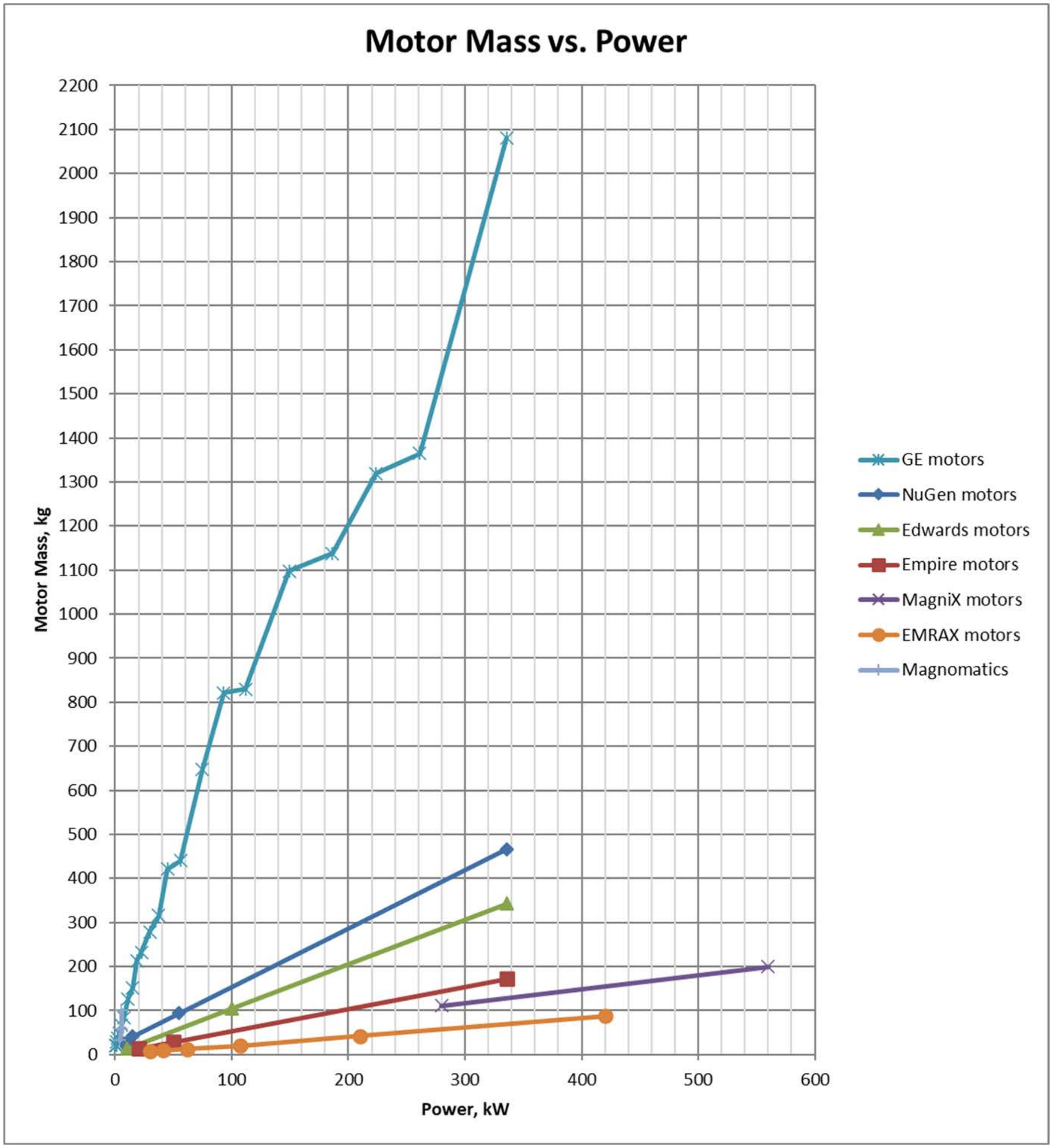
What do we have so far?

- The equations on the previous slide allow us to calculate the torque required of the motor/transmission that rotates the wheels
- Now let's look at the implications of the requirement to get to GEO in about a week
  - GEO is ~36,000 km above the surface of the Earth
  - If we take 36,000 km/(7 days)/(24 hrs/day), we get an average speed for the climber of 214.3 kph
  - For the drive train calculations, I assumed an average speed of the climber of 200 kph
  - The fastest electric vehicles on Earth can go >500 kph, but they don't do it for very long

Researching motors

- We did a lot of research on motors and it is still ongoing
- We need the lightest, most powerful motors ever built
- Our survey of the market showed that motors designed for electric aircraft combine the need for lightness and power
- The next slides show the results of this research

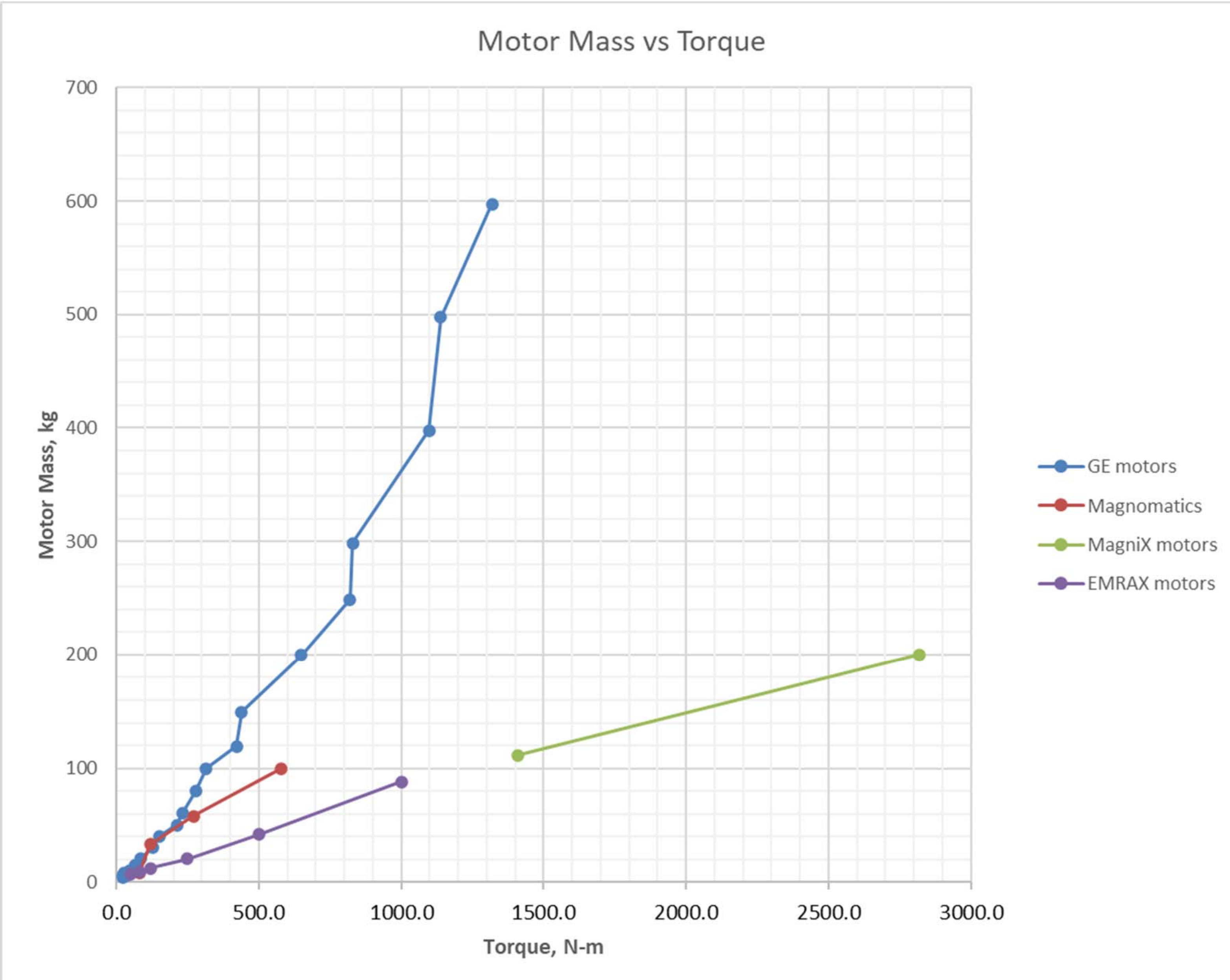
Motors have gotten much lighter



In 2013, I thought that any motors under the green line of the Edwards motors were fantasy. The electrification of aircraft have created new, much lighter and more powerful motors.

Modern drive electronics also simplifies motor design while allowing for more useful motor characteristics.

Motor mass as a function of torque

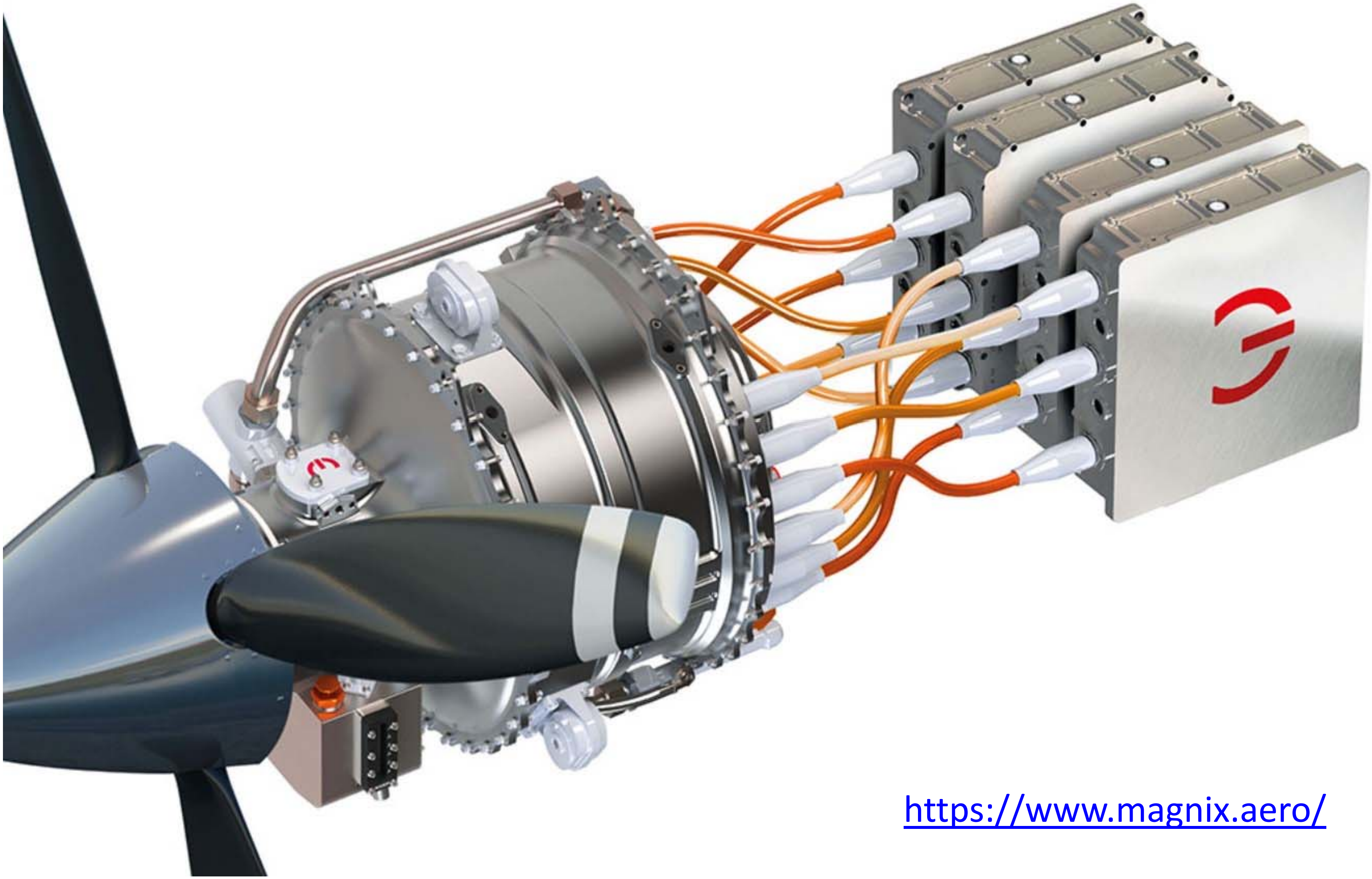


You can see from the graphs that the MagniX motors are in a class by themselves for power, torque and mass.

Emrax appears to be on a similar trajectory, they just don't make strong enough motors.

I based the design of the 20 tonne climber on the Magni 650 motor, delivering 2814 N-m of continuous torque at 1,900 RPM and weighing 200 kg.

The Magni 650 electric aircraft motor and its drivers



https://www.magnix.aero/

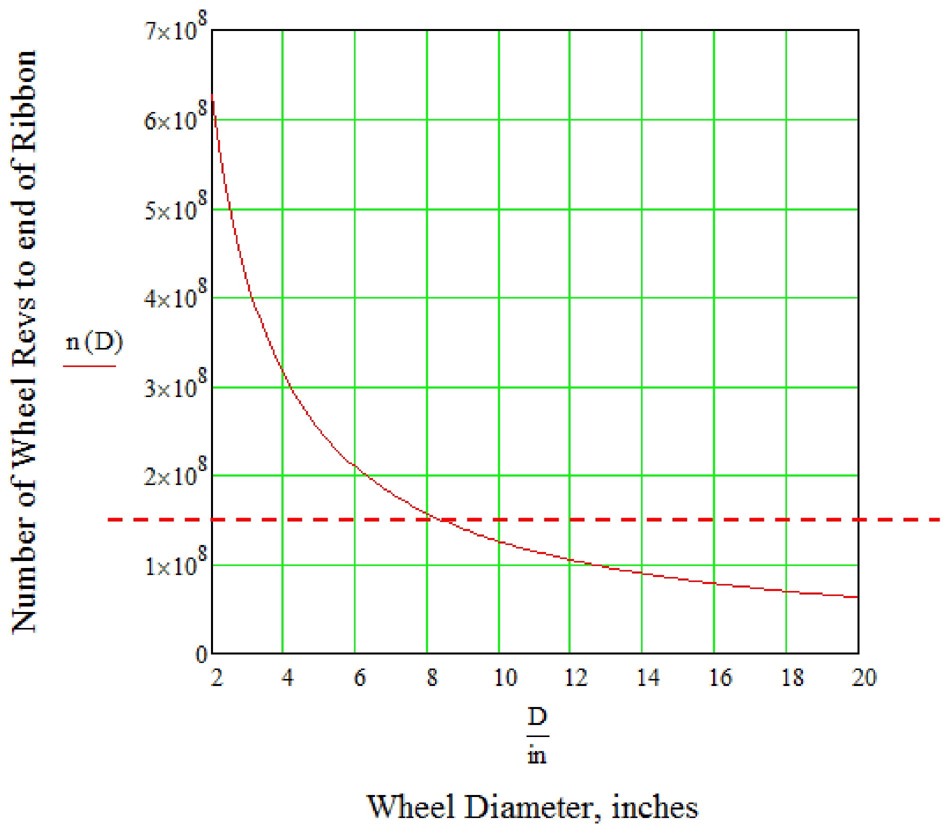
Back to the math model…

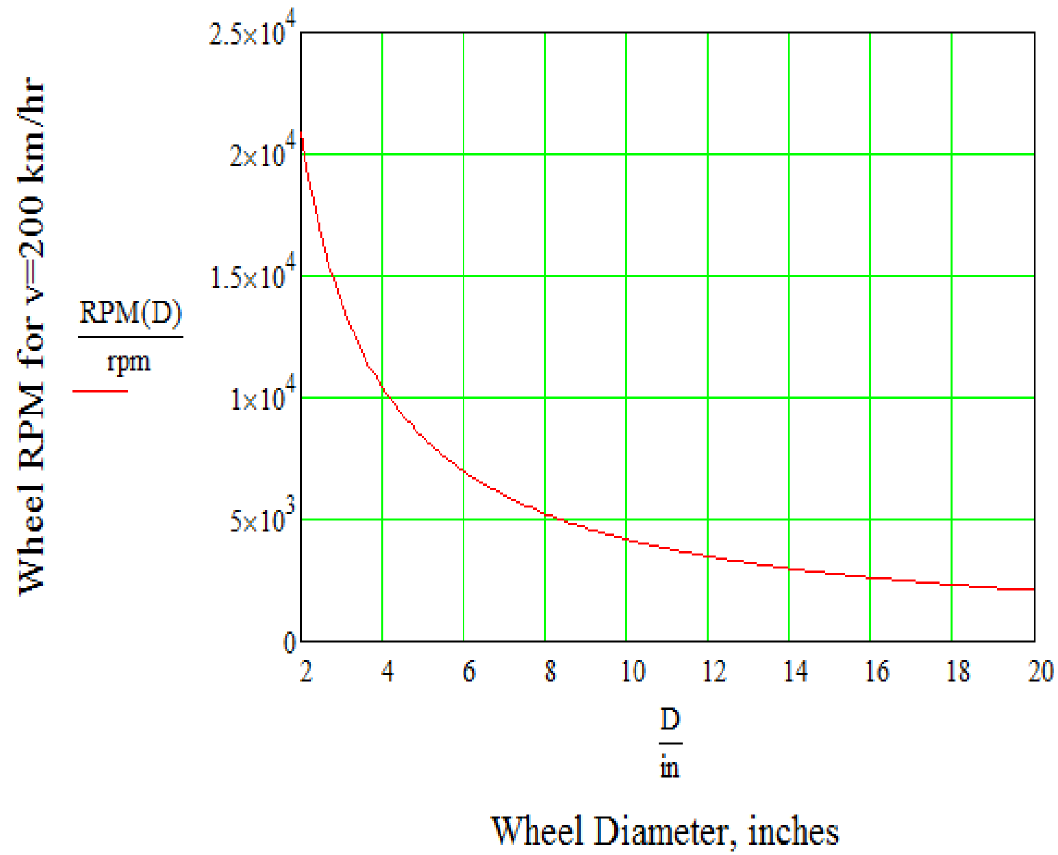
- Knowing the continuous torque of the motor is 2,814 N-m, and that it rotates at 1,900 RPM, we make the following assumptions:
  1. Every wheel is driven by a motor.
  2. Every wheel is directly driven. No gear boxes.
     - This is how locomotives work. An electric motor acts as a continuously variable transmission
     - This should be a minimum mass solution
- Now we have to specify the diameter of the wheel

Sizing the wheel

- We know the continuous rotation speed of the motor is 1,900 RPM = ω_w (max speed is 2,600 RPM)
  - Motor and wheel rotation rate are the same
- We want the average speed of the climber to be 200 kph
- We also know that we do not want the wheel so small that it rotates hundreds of millions of times to get to GEO (or farther,) because then metal fatigue might cause failure of the wheels or axles (see next slide)
- One revolution of a wheel covers $2\pi R$ distance
- v = velocity of the climber = $2\pi R\omega_w$
- Given v and $\omega_{w,}$ we calculate R = 279.22 mm
  - Because I live in the US, I rounded this to R = 11 inches, 279.4 mm

Some climbers have to go all the way to the end of the tether, 100,000 km up





This graph shows how many times a wheel has to rotate to get to the end of a 100,000 km long ribbon as a function of the wheel diameter. Wheels below 12” in diameter are in the very high cycle fatigue range. Fatigue data runs out for most metals above the red dotted line (150E6 cycles).

This graph shows how fast a wheel has to rotate to make the climber climb at 200 km/hr as a function of the wheel diameter. Wheels below 4” in diameter would rotate so fast that their motors would be destroyed.

Now we calculate the number of wheel pairs

- Generalizing the equation for torque on slide 13 and assuming the wheel is not rotating, the equation becomes:

  $T = m_c g R / n_w \leq 2{,}814$ N-m

  Or, rearranging:

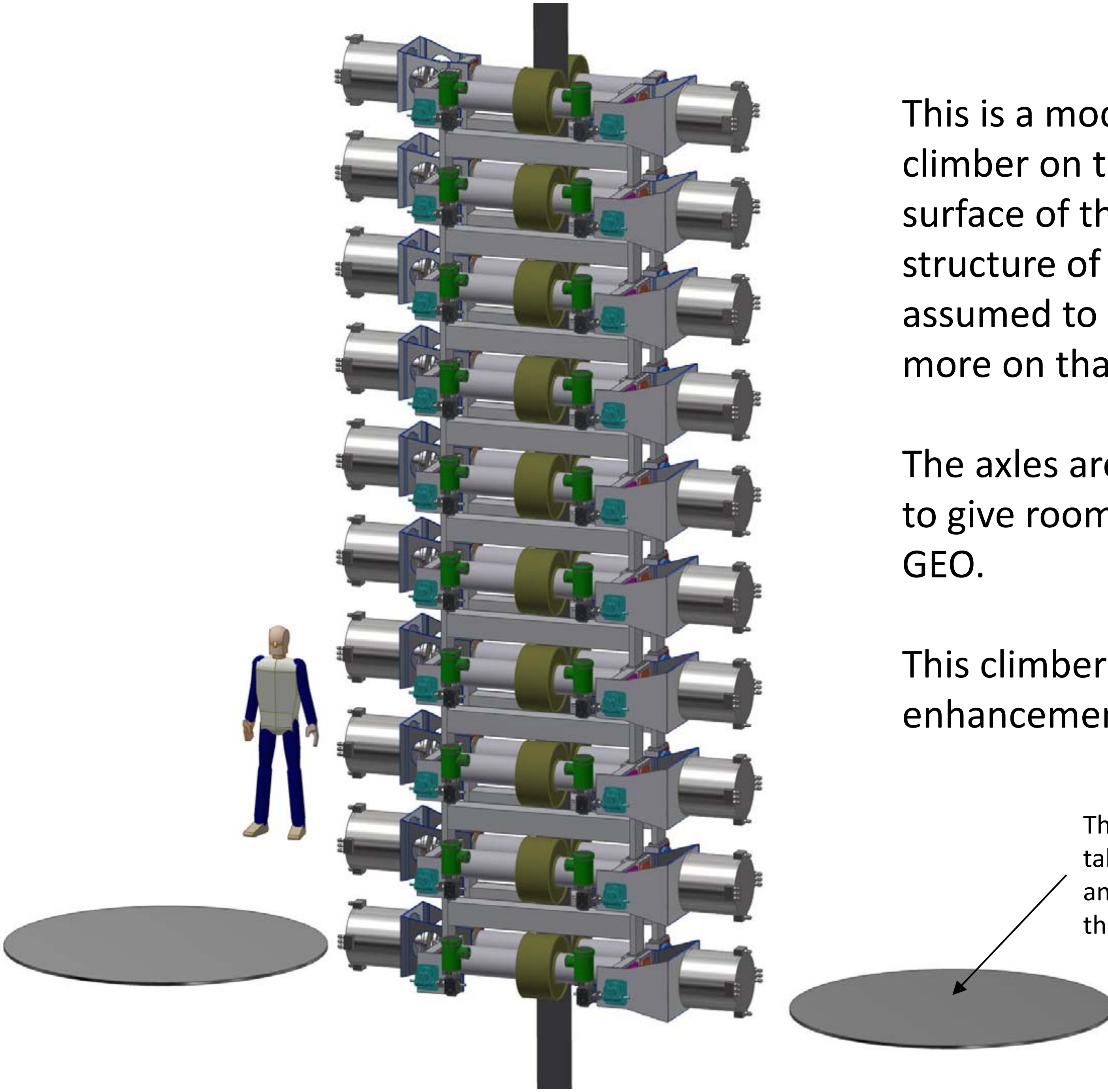
  $n_w \geq m_c g R / 2{,}814$ N-m

  $g = 9.8\ m/sec^2$

  $m_c = 20{,}000$ kg

  $n_w \geq 19.46$ The number of wheels has to be 20, 10 pairs

Early rendering of the solid model of a 10 wheel-pair climber



This is a model showing the climber on the tether at the surface of the Earth. Most of the structure of the climber is assumed to be aluminum here, more on that later.

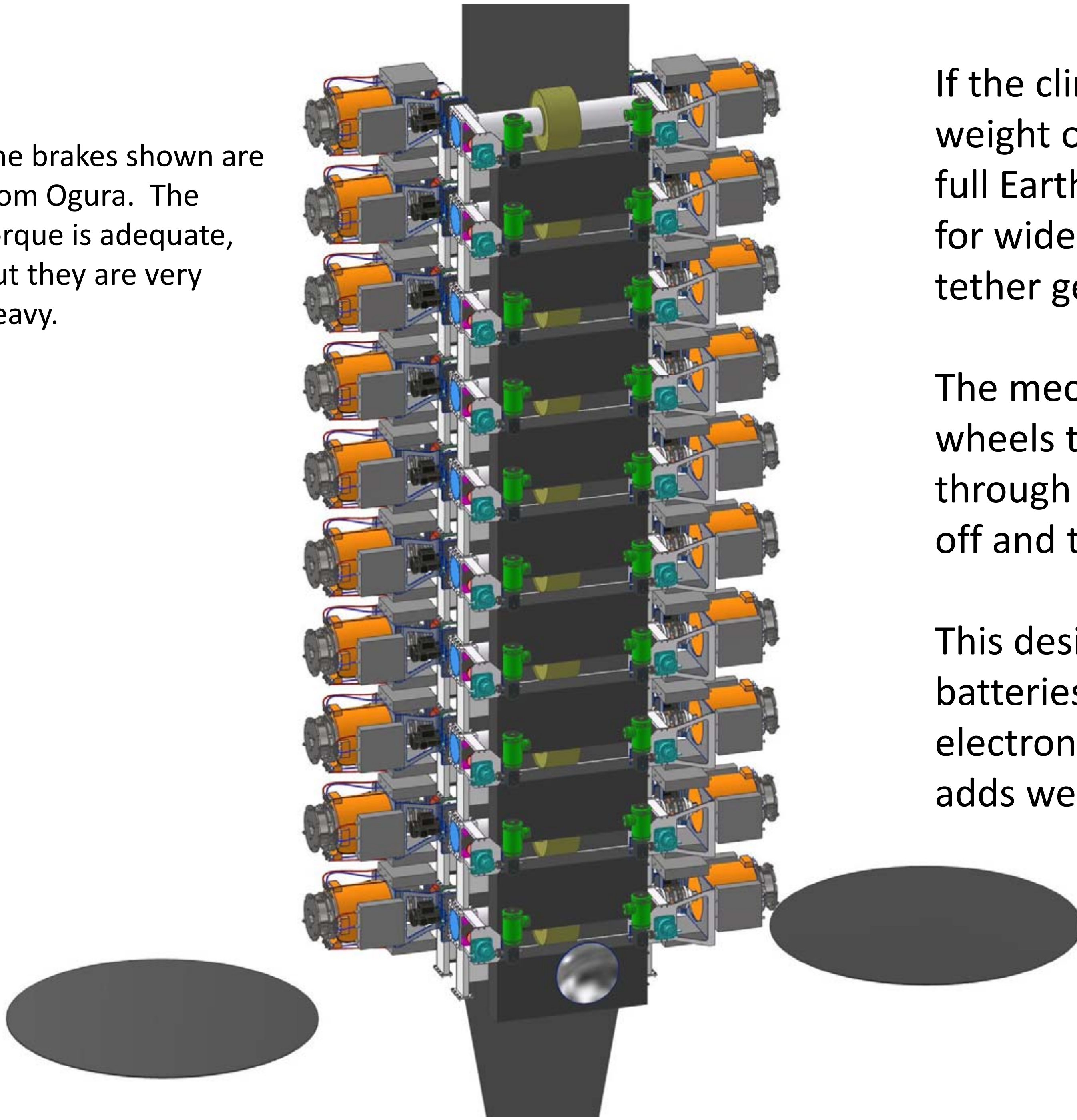
The axles are as long as they are to give room for the tether at GEO.

This climber is missing many later enhancements

These are photovoltaic arrays that take the laser light from the ground and convert it to electricity to power the climber. More on these later.

More developed model showing the climber near GEO where the tether is widest

The brakes shown are from Ogura. The torque is adequate, but they are very heavy.



If the climber can support its weight on the narrow tether at full Earth gravity, it has no need for wider wheels where the tether gets wider.

The mechanism compressing the wheels together can back off all through the climb as gravity falls off and the climber gets lighter.

This design shows lithium ion batteries and the motor drive electronics. Every system added adds weight.

Now let's talk about the pinch mechanism

- I designed a mechanism back in 2004 to press one wheel against its opposite on the other side of the tether
  - I have adapted this mechanism to the larger forces of the 20 tonne climber
- Let's go back to the free body diagram and see how to calculate the compression force required to hold up the climber
- The earlier calculation summed the moments around the point of contact between the tether and wheel to get torques
- The next calculation sums the forces in X and Y to get the conditions of static equilibrium

Summing the forces in X and Y from the Free Body Diagram

$$\sum F_y = f - \frac{m_c g(r)}{2} = 0$$ Eq. 1, summation of forces in Y

$$\sum F_x = F - N = 0$$ Eq. 2, summation of forces in X

Rearranging terms in Eq. 1 gives the following equation:

$$f = \frac{m_c g(r)}{2}$$ Eq. 3, the friction force on one wheel has to equal the weight carried by that wheel

Again, we can generalize Eq. 3 for the case of more wheel pairs than one by changing the 2 into n_w, the number of wheels.

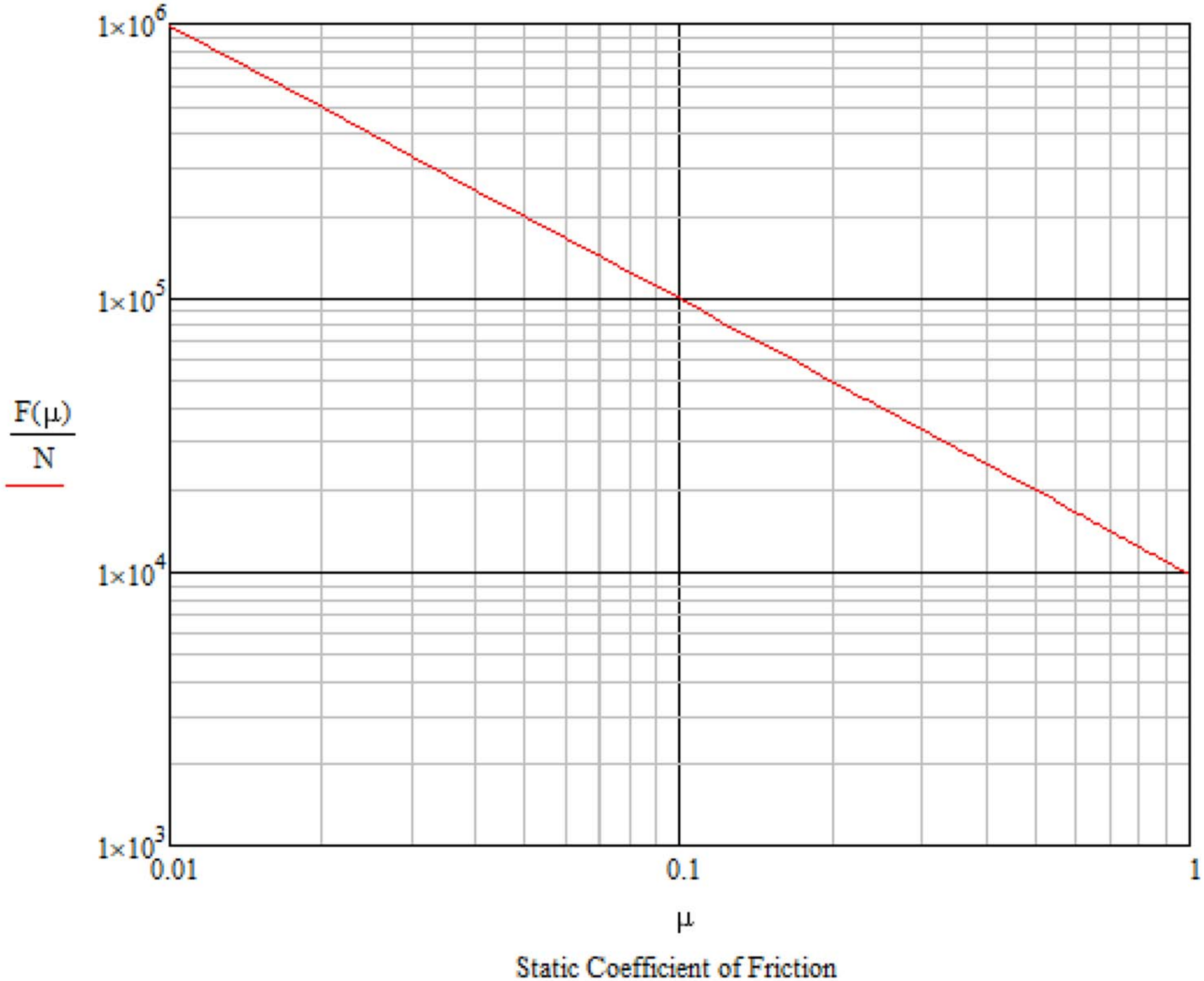
$$f = \frac{m_c g(r)}{n_w}$$ Eq. 4, the friction force on any wheel where all of them carry the climber weight equally

The friction model used here is Coulomb dry friction in which the traction does not depend on the area of contact, but only on the normal force and coefficient of friction as given by:

$$f = \mu N$$ Eq. 5, the friction force from the product of mu and N

$$F(\mu) = \frac{m_c g(r)}{n_w \mu}$$ Eq. 6, rearranging eq. 2 gives F=N. Substituting this into Eq. 5 gives f= μF. Substituting this into Eq. 4 and rearranging gives the compression force as a function of mu.

Log plot of the force pushing one wheel toward the opposite wheel



This graph shows the compression force (in Newtons) that one wheel exerts against the other to keep the climber from sliding down the tether, as a function of the coefficient of friction.

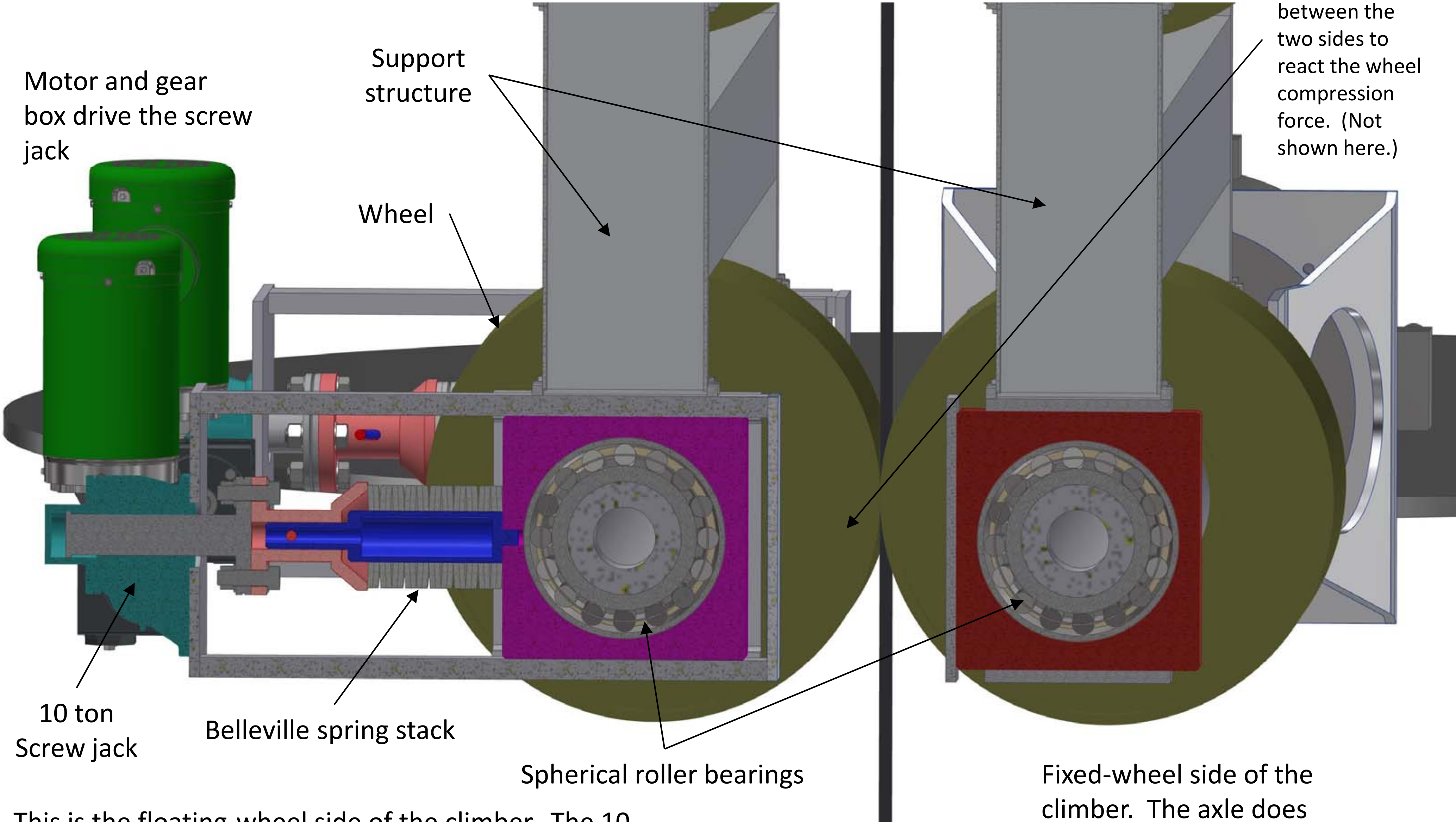
This graph is for a 20 tonne climber with 10 wheel pairs.

The load at μ = .1 is 9.81E4 Newtons, (11.03 tons)

This load is carried by two screw jacks, one on each side of the climber in the conceptual design.

Again, since I'm in the US, we buy screw jacks based on English tons. Each screw jack would need to be 5.515 tons, but they come in 5 and 10 tons. I chose 10 ton screw jacks for a safety margin.

Cross-section through the compression mechanism showing key features



This is the floating-wheel side of the climber. The 10 ton screw jacks push the axle on the left side toward the axle on the right. The axle bearing is mounted in a block that slides on the support structure.

Fixed-wheel side of the climber. The axle does not move with respect to the support structure.

Design choices made here

- The screw jack does not push directly on the axle bearing housing because if it did there would be a step function of force as the jack is operated
- The Belleville spring stack compresses over 0.75 inches from zero force to full 10 tons to allow a control system the resolution it needs to modulate the compression force
  - The load cell to measure the force is not shown in the model
- The wheel bearings were chosen to be spherical roller bearings to be able to handle the high load and angular offset of the axle caused by the bending of the long axles from the compression force
- The motor and gear box are designed to operate the compression force over about 18 seconds. To be faster than this would require a higher horsepower motor.
  - This is a future design problem to see what the control system would require of the speed of the compression jacks.
  - Modulating the compression force is conceived to be a possible steering mechanism for the climber to stay centered on the tether

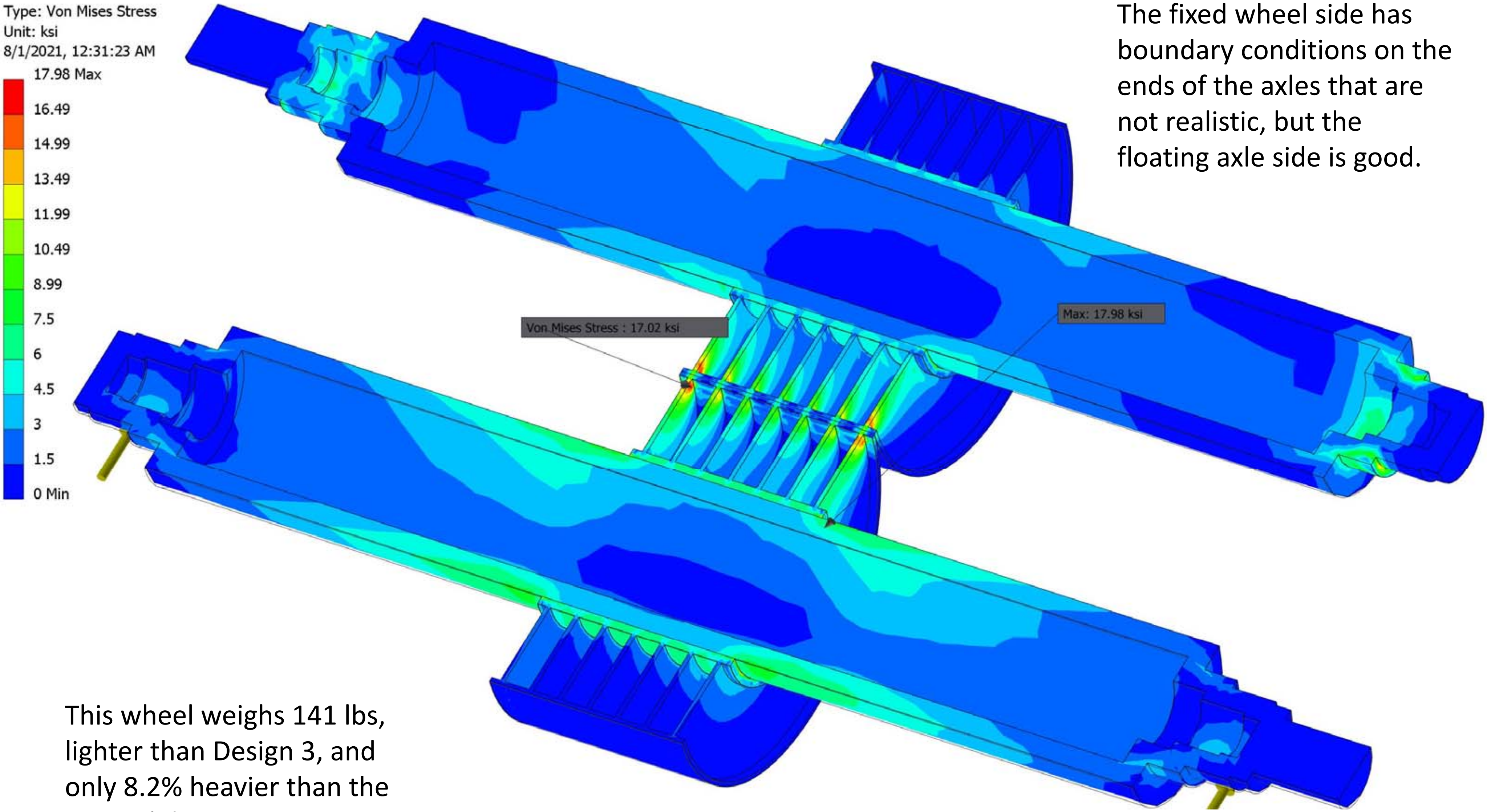
Sizing the axles

- I assumed the axles are made from aluminum and hollow to reduce weight
- I have experience designing aluminum for fatigue life
- The maximum stress the axles can be allowed to see for >100E6 revolutions is ~10 ksi (68.9 MPa)
- I ignored the reduction in stress with altitude

Wheel design #4 (.25” thick rim), von Mises stress plot

The wheels are Ti-6Al-4V
The axles are Al 6061-T6

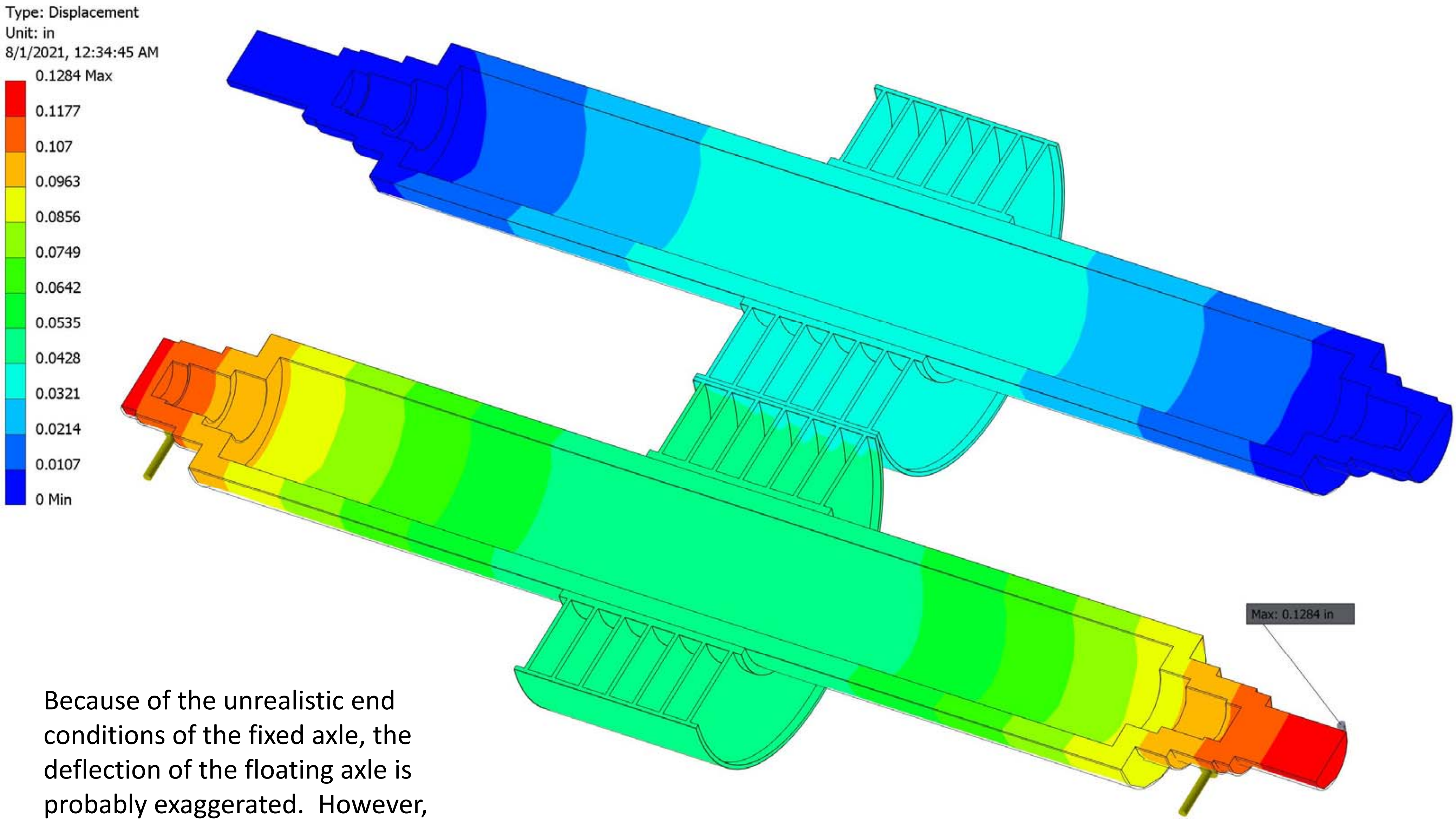
The fixed wheel side has boundary conditions on the ends of the axles that are not realistic, but the floating axle side is good.



This wheel weighs 141 lbs, lighter than Design 3, and only 8.2% heavier than the original design.

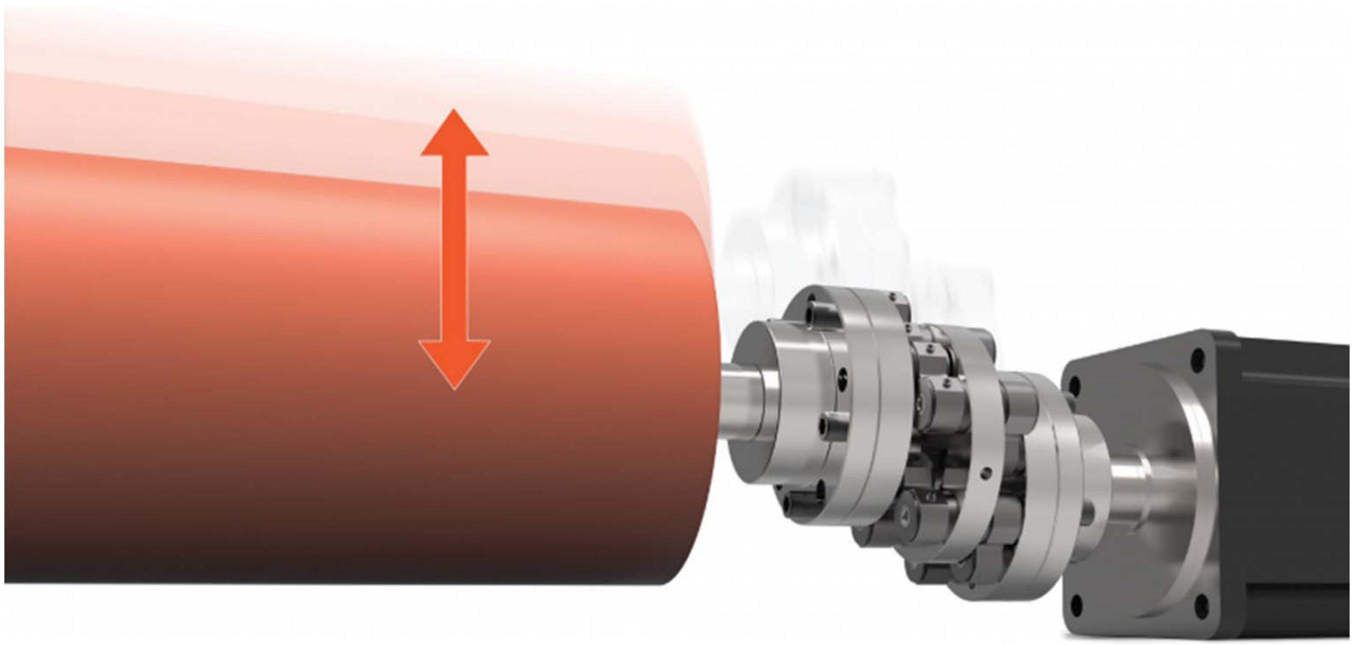
I experimented with wheels with cutouts to lighten them, but the stress concentrations were too high.

Wheel design #4 (.25" thick rim), actual deflection plot, 10 ton load on each end



Because of the unrealistic end conditions of the fixed axle, the deflection of the floating axle is probably exaggerated. However, the axles bend by a lot from the 10 ton compression force on each side.

How do you connect axles that move to a motor that does not? Why, a Schmidt coupling, of course.



If you've never seen a Schmidt coupling work, you're really missing something. This coupling transmits torque between a motor and axle that are not in the same centerline, with no overhung load. Zero-Max makes them with and without the ability to take an angular offset.

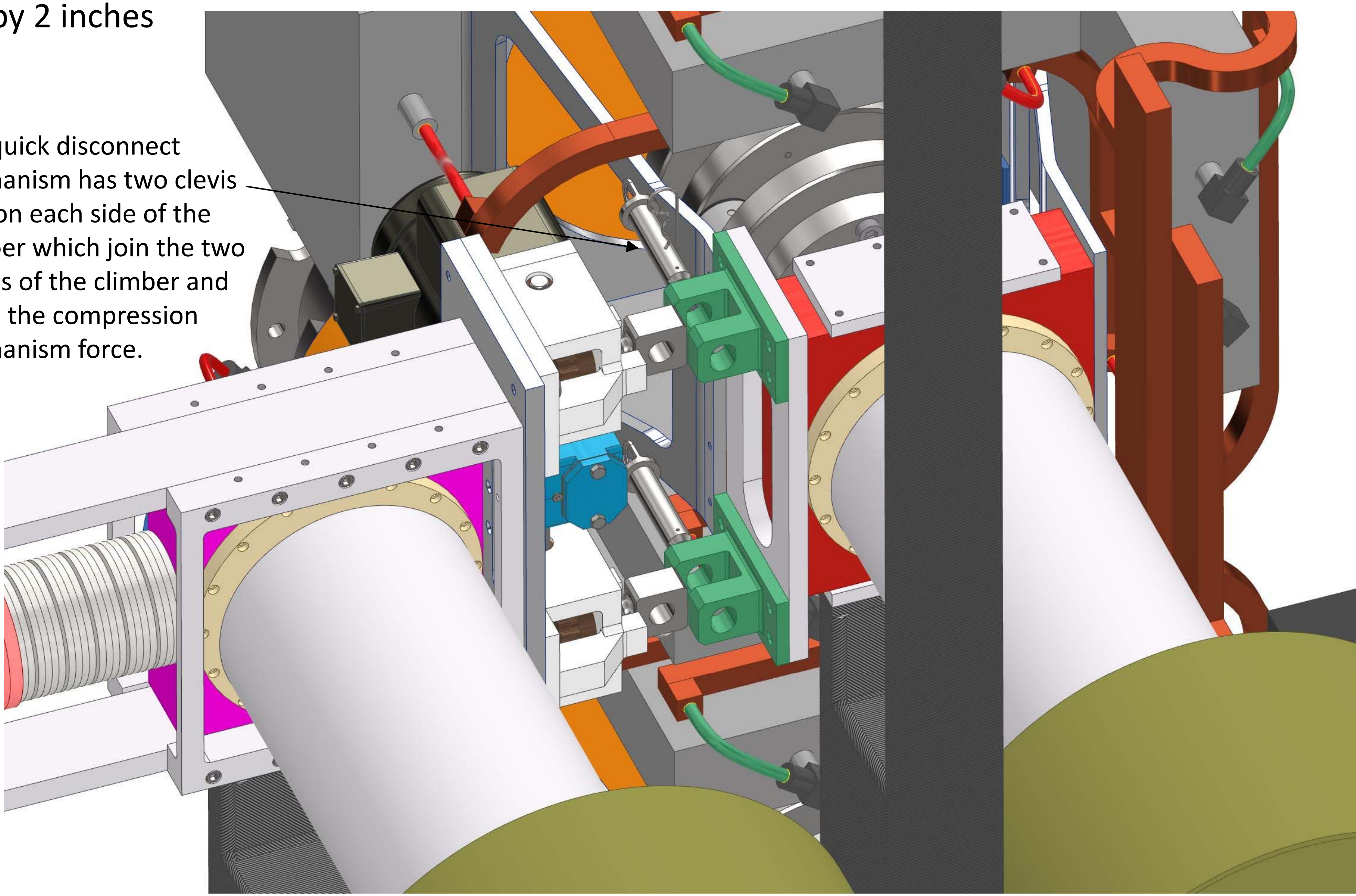
I put this style with angular offset on the floating side, and the angular offset only version on the fixed side. They work great, but are obviously heavy.

On to the quick disconnect mechanism

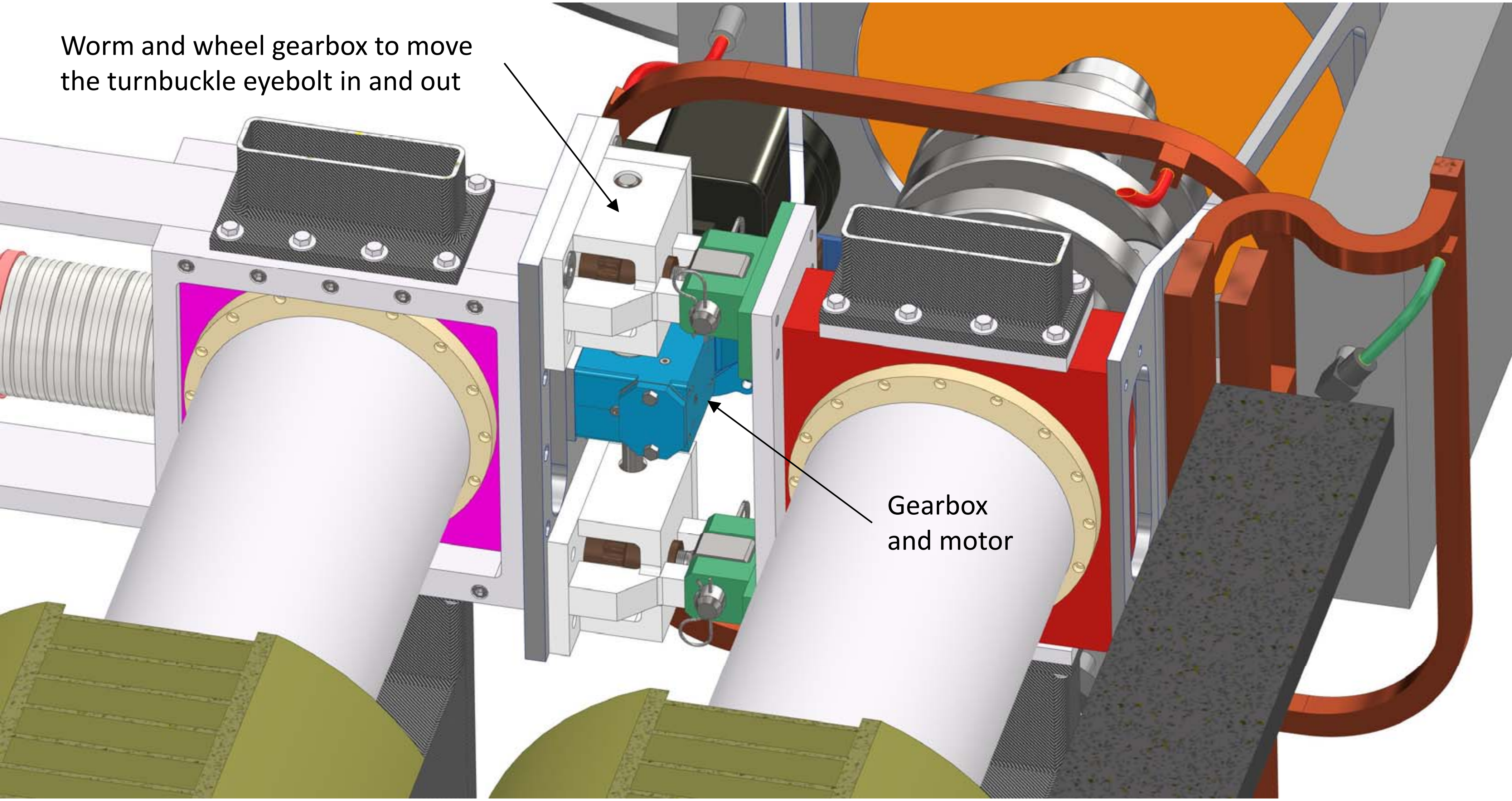
- A climber needs to be assembled at the Earth anchor station as two separate machines on opposite sides of the tether
- The connection between the two climber halves should be reasonably fast
- The climber may experience damage on its trip that requires it to be freed from the tether at any altitude
- In my 2004 design of a 900 kg construction climber, I just bolted the two halves together
  - That design does not allow for ejection
- For the 20 tonne climber I designed a quick-disconnect mechanism that is motor operated and can be activated at any altitude
  - An additional mechanism is needed (not shown) to push the climber halves away from the tether so that they will not hit it as they fall

Close-up exploded view of the Quick Disconnect mechanism, climber halves separated by 2 inches

The quick disconnect mechanism has two clevis pins on each side of the climber which join the two halves of the climber and resist the compression mechanism force.



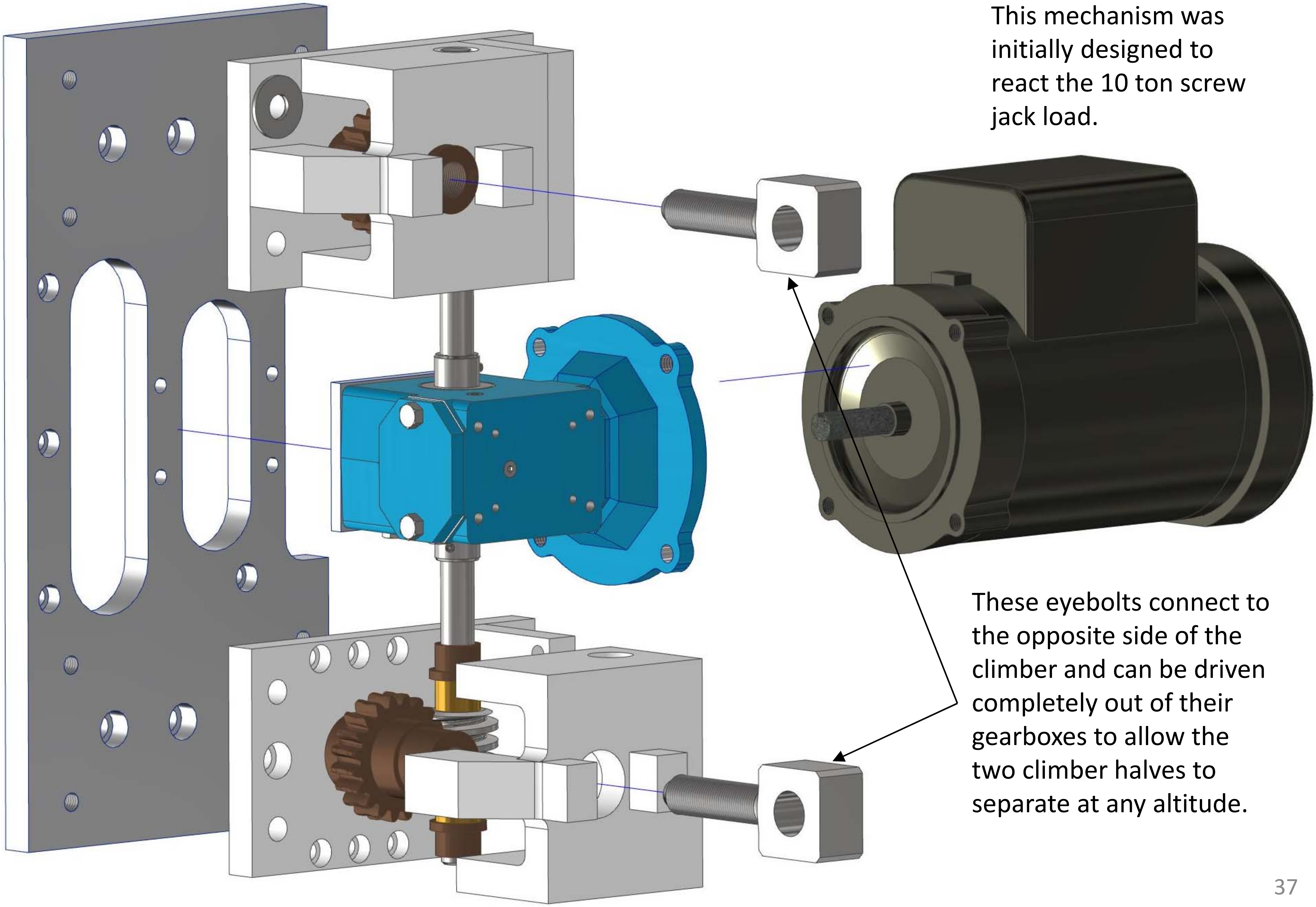
Section view close-up of the Quick Disconnect mechanism engaged



The mechanism is two single-ended turnbuckles driven by a single motor and gearbox. The turnbuckle eyebolt is pinned to the fixed axle climber half for easy assembly.

Fixed axle side of the climber

Exploded view of the motorized side of the quick disconnect mechanism



This mechanism was initially designed to react the 10 ton screw jack load.

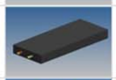
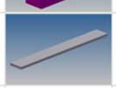
These eyebolts connect to the opposite side of the climber and can be driven completely out of their gearboxes to allow the two climber halves to separate at any altitude.

So where has this gotten us?

- I developed a number of the subsystems for the climber to get to a point where I could measure the mass of the traction drive and see how much payload it was capable of lifting up the tether
- Unfortunately, when I got almost everything in there* the traction drive weighed more than 20 tonnes and could not even lift itself up the tether!
- Back to the drawing board! ☹
  - I credit my colleague Martin Lades of ISEC with suggesting a different way of calculating the climber

*Except for the space radiator, control electronics, heat exchangers, PV arrays and supports, oil recirculation system, electronics support, busbar support…

Breakdown of highest mass components of the climber

Rank	Part Number	Thumbnail	QTY	Unit Mass, kg	Total mass, kg	Percent of total mass	
1	560 kW motor		20	200.25	4004.98	19.89%	
2	MNB_500J_modified		20	130.45	2608.98	12.96%	
3	SE-20t AXLE		20	109.20	2184.08	10.85%	
4	battery pack		20	84.02	1680.40	8.35%	
5	SE-20t WHEEL-2		20	63.96	1279.23	6.35%	
6	Motor driver electronics box		80	12.09	967.55	4.81%	63.21%
7	McM 6135K111 AC MOTOR		20	38.36	767.18	3.81%	
8	zero-max_l411c		10	68.69	686.90	3.41%	
9	McM 2721T24_SPHERICAL-ROLLER BEARING		40	12.66	506.58	2.52%	
10	PV Array		2	242.28	484.57	2.41%	
11	JD-WJ810I2S-002.00-STDX-STDX-X_Jack Housing - 10T_Inverted		20	22.44	448.76	2.23%	
12	zero-max_l582s		10	39.52	395.24	1.96%	
13	McM 3540N210 FACE-MOUNT AC MOTOR		20	16.63	332.59	1.65%	
14	0230-02568		20	15.20	303.99	1.51%	
15	Fixed BEARING HOUSING-2		20	11.33	226.63	1.13%	
16	Belleville Spring		440	0.50	220.90	1.10%	
17	SE-04 BEARING HOUSING		20	9.94	198.83	0.99%	85.92%
18	SE-04 SLIDE BAR 1		40	4.21	168.34	0.84%	
19	CFRP Interface structure		10	13.55	135.49	0.67%	

85.9% of the mass of the climber is in the first 17 components of the bill of materials (out of >100 components.)

The motors top out at #1, 20% of the weight of the climber.

The brakes are next in weight, then the axles.

The batteries have already been reduced in size and weight by making them lithium sulfur instead of lithium ion. Lithium sulfur has 3X the energy density as lithium ion.

The PV array is only a guess.

How to lighten the climber?

- I looked at the 10 wheel-pair climber and first thought, "What happens if I cut it down to a 5 wheel-pair climber?"
  - The compression mechanism needs to be strengthened because half the wheels are carrying the full 20 tonne load
  - The quick disconnect mechanism needs to be strengthened to counter the compression mechanism
  - The MagniX motor didn't have enough torque, so the next detour was through the land of gear reducers and transmissions

Problem statement

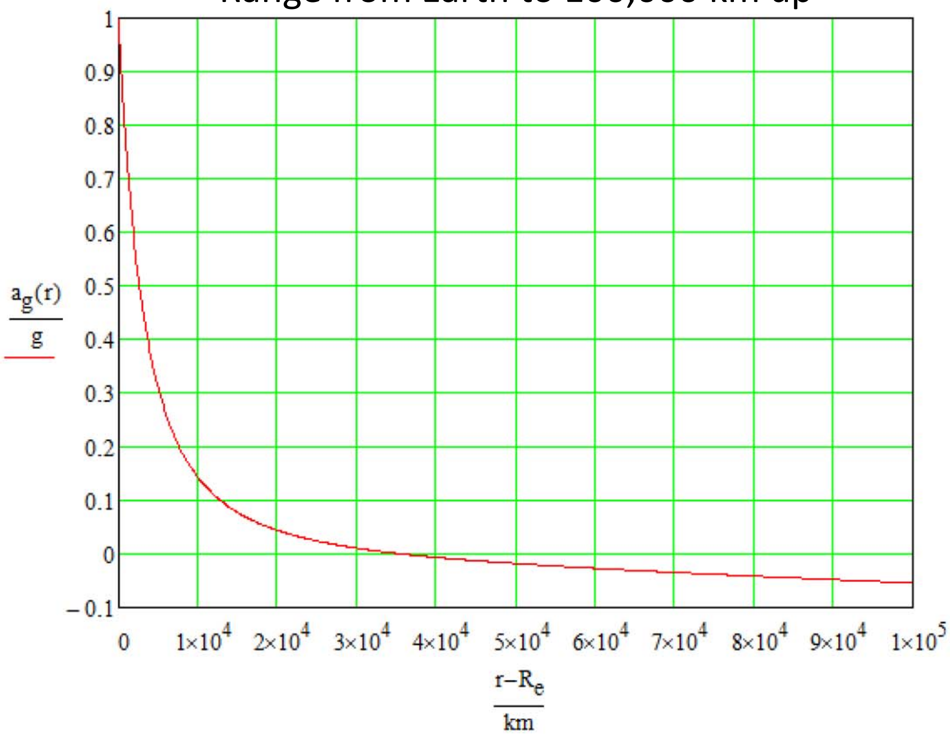
- 10 wheel pairs weighs >20 tonnes
  - The torque available from 20 motors cannot lift more than 20 tonnes in a direct drive with the current diameter wheels
  - Cannot reduce the diameter of the wheels significantly because of interferences with the rest of the climber
- Reduce to 5 wheel pairs
  - increase the clamping force on the ribbon >2X
  - Put at least a 2:1 reducer between the wheel and motor (probably need more) to increase the torque
  - Trip time increases by ≥2X.
- The only way to reduce the trip time is to change the gear ratio at different altitudes as the climber lightens

The Motor Torque problem

- At the surface of the Earth, the required motor torque is at a maximum and determines the number of motors required (for a given wheel radius)
- Motor torque required goes down as a function of altitude
  - At 2,600 km up the climber weighs .5X
  - At 6,207 km up the climber weighs .25X
  - See the curves on the next slide
- *This $1/r^2$ relationship is a natural for a speed-changing device between the wheels and fixed-torque motor*
- Torque and output speed inversely trade off in any transmission

Graph of gravity acceleration as a function of altitude, expressed as a ratio with g

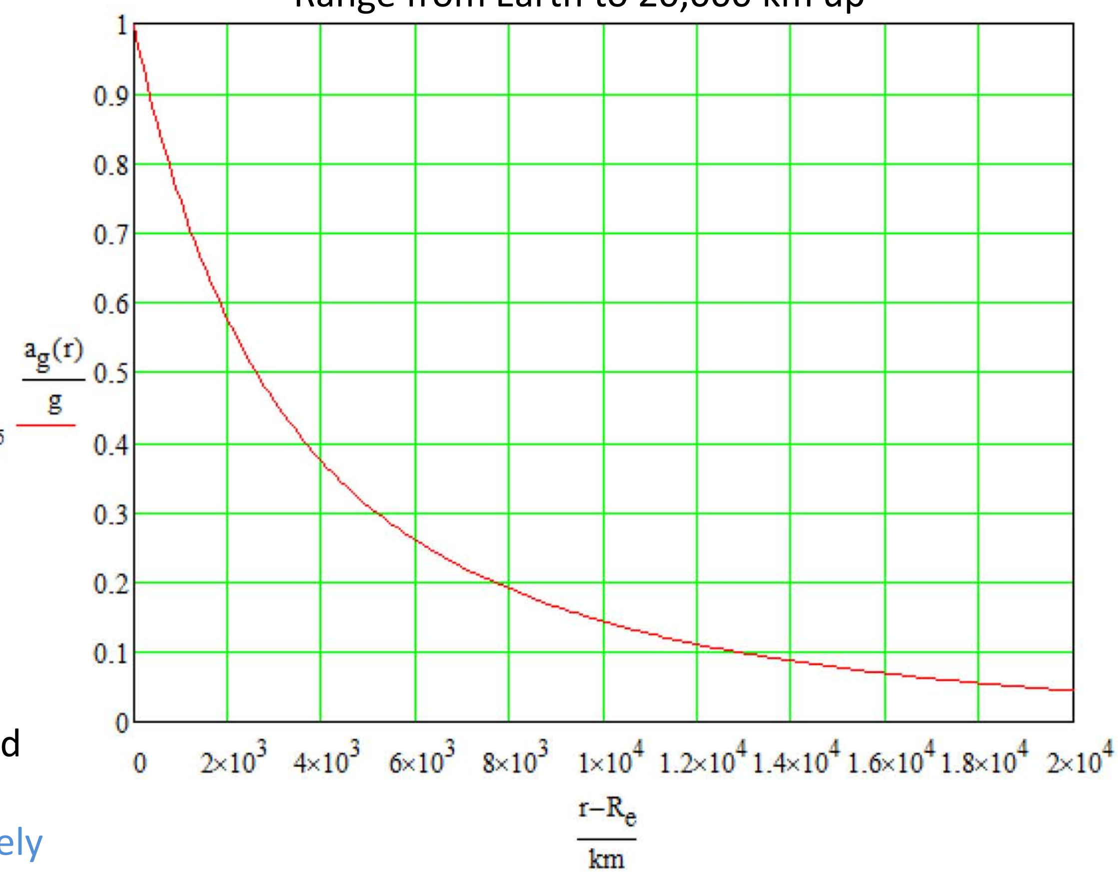
Range from Earth to 100,000 km up



The acceleration is expressed as a ratio of the acceleration to earth normal gravity at the poles.

The x-axis is altitude up the ribbon in km.

Range from Earth to 20,000 km up

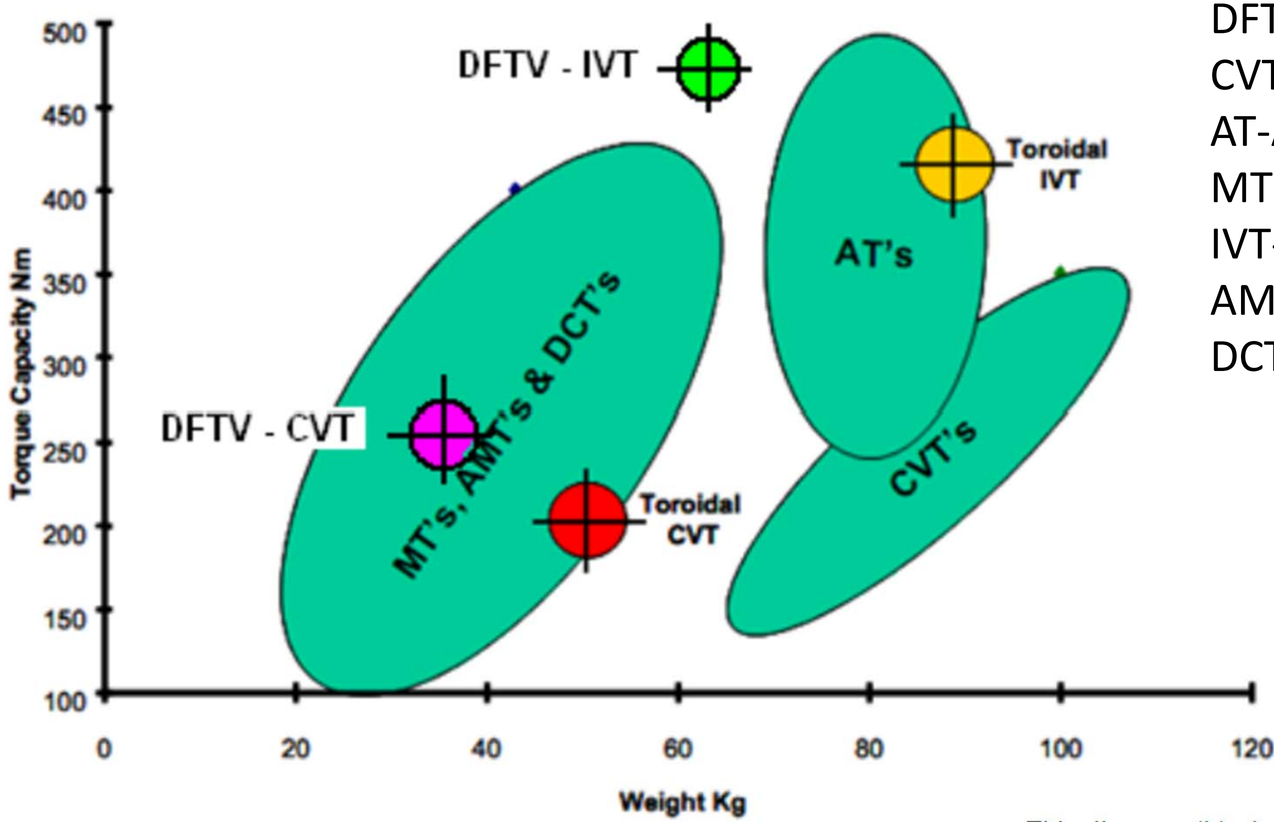


As the climber weighs less, less torque is required to support the weight on the ribbon.
This seems like a natural application of an infinitely variable transmission.
Above 13,000 km up required torque drops to less than 10% of that at Earth surface.

Speed increasers and reducers

- You can reduce the number of motors at the surface of the Earth by putting a speed reducing gear box in between the motor and wheel, to increase the torque and decrease the speed of the wheel.
  - If you can't change gears, the speed stays low and the trip takes longer. A 2:1 reducer makes the trip take twice as long
- Once the weight goes down, a transmission can switch to a speed increaser, reducing torque and increasing speed.
  - 2:1 $\rightarrow$ 1:1 $\rightarrow$ 1:2 (and numbers higher than 2) are possible in one device
- There are two basic types of transmissions:
  - Infinitely variable (or continuously variable (CVT))
  - Discretely variable (jumping from one gear ratio to another)
- There is an enormous landscape of transmissions to sort through and the end result was a transmission plus motor that weighed >2X the motor alone.

Summary of Torque Vs Mass for automotive transmissions from Ref. 4



DFTV-Double Roller Full Toroidal Variator
CVT-Continuously Variable Transmission
AT-Automatic Transmission
MT-Manual Transmission
IVT-Intelligent Variable Transmission
AMT-Automated Manual Transmission
DCT-Dual Clutch Transmission

I extrapolate the weight of units that could carry our torque on the next slide.

This is taken from Reference 4, caveats on the right.

One thing that can be concluded from the graph is that there is a linear relationship between torque capacity and mass for CVTs, AMTs and DCTs

This diagram (b) plots the position of a small DFTV CVT and a larger DFTV IVT showing the relative positions within this Mass and Torque relationship diagram created by Torotrak.

Ultimate Transmissions do not agree with Torotrak on where they have placed belt type CVT's as they are much lighter than this diagram purports

UT is also uncertain about where Torotrak places themselves at 50 kg and 200Nm. When they have stated in the same academic paper that a 150 Nm. Torotrak transmission will be 50 kg.

Exactly why the AT's are represented in such a way is also difficult to understand.

Extrapolations based on the previous chart for different transmissions

CVT performance from chart:
X_1= 66 kg, Y_1= 130 N-m
X_2= 108 kg, Y_2= 340 N-m
X_3= ?, Y_3= 2,814 N-m

$(Y_2-Y_1)/(X_2-X_1)$ = m   slope of the line

M= (340-130)/(108-66) = 5 N-m/kg

Y=mX+b   slope intercept form of a line

130=5*66+b

b= -200 N-m   (You would think that x=0, y=0)

X= (Y-b)/x

X_3= (2814+200)/5 = **602.8 kg**

602.8/200 = 3.01

A CVT transmission capable of delivering the torque of the Magni motor would weigh 3X as much as the Magni motor.

CVTs have a very bad reputation in the automotive world in general.

MT/AMT performance from chart:
X_1= 22 kg, Y_1= 110 N-m
X_2= 60 kg, Y_2= 420 N-m
X_3= ?, Y_3= 2,814 N-m

$(Y_2-Y_1)/(X_2-X_1)$ = m   slope of the line

M= (420-110)/(60-22) = 8.16 N-m/kg

Y=mX+b   slope intercept form of a line

110=8.16*22+b

b= -69.52 N-m   (You would think that x=0, y=0)

X= (Y-b)/x

X_3= (2814+69.52)/8.16 = **353.4 kg**

353.4/200 = 1.77

A MT/AMT transmission capable of delivering the torque of the Magni motor would weigh 1.77X as much as the Magni motor.

The new way of thinking

- *The previous slides show that adding a transmission was not going to help*
- I started the original design with the lightest, strongest motor I could find and calculated everything around that
- This motor doesn't have enough torque to drive the climber straight up the tether
- The new way of thinking was to imagine a motor that did have enough torque but was not on the market today
  - This is how we identified **future motor development** as crucial to climber design

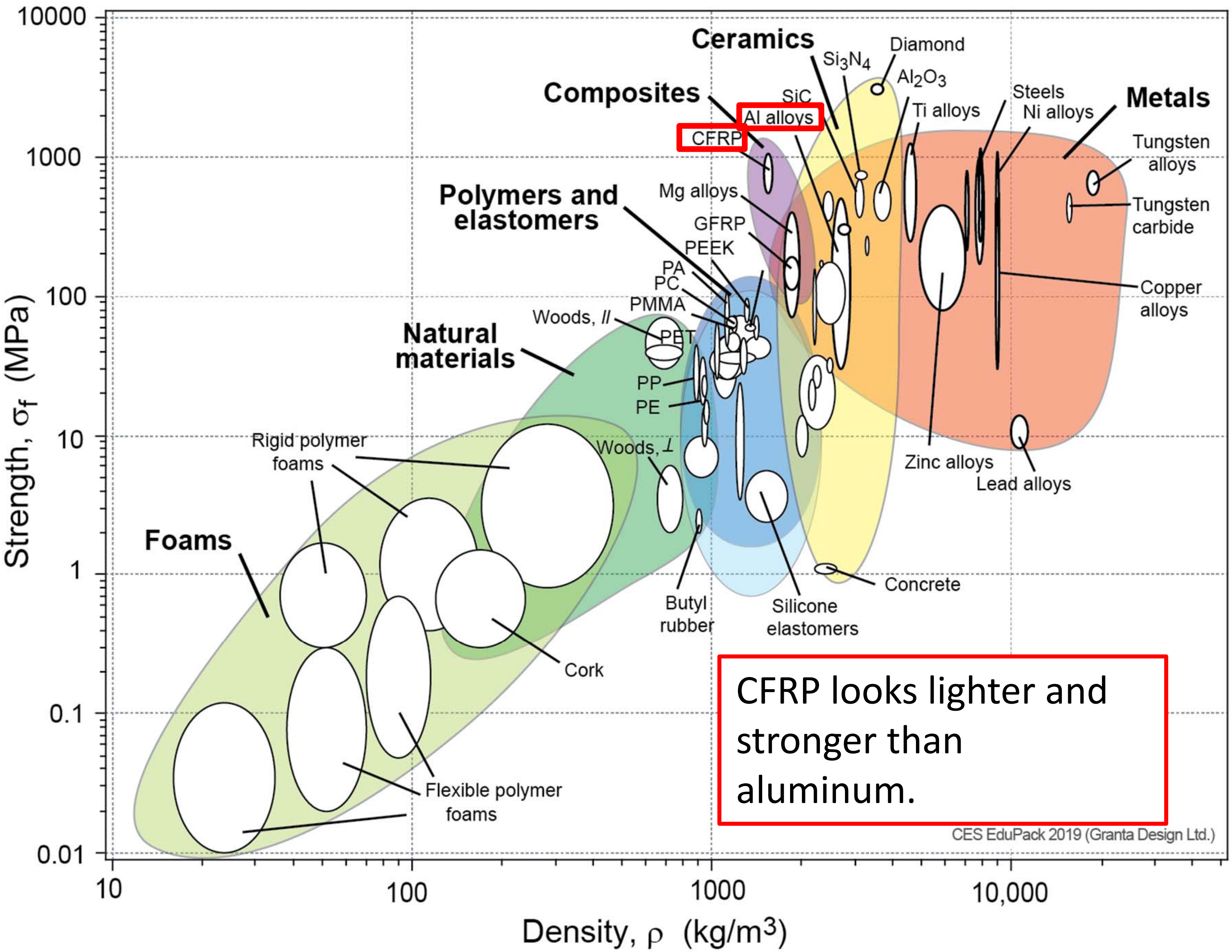
Possible new motor technologies

- Some lessons learned:
  - The torque of a motor is linearly dependent on the volume of the rotor
  - Most motors run at magnetic fields between 1.0 and 1.5 Tesla
  - The best magnetic materials saturate at 2 Tesla
- Getting twice the torque out of a motor with the same mass as the Magni 650 is not trivial
  - One possibility is that graphene wires will have the current carrying capacity to stuff more current into the rotor, hence more torque
  - High temperature superconducting motors might avoid the mass penalty of the iron core
    - But have the additional mass penalty of cryogenic systems
  - I did not think it reasonable to extrapolate beyond a factor of 2 in torque production for the foreseeable future
- To continue the design, the motor shown is the same weight as the Magni 650, but delivers twice the output torque

Other weight loss ideas

- I originally designed most of the structure of the climber out of **aluminum** because it is:
  - Lightweight
  - Strong
  - Cheap
- There is practically no other engineering material in use today that shares **all three** of those characteristics
- A material that is lighter and stronger is Carbon Fiber Reinforced Polymer, CFRP
- Another lesson learned is that aluminum busbars are about half the weight of copper bus bars, even though they are bigger in cross-section

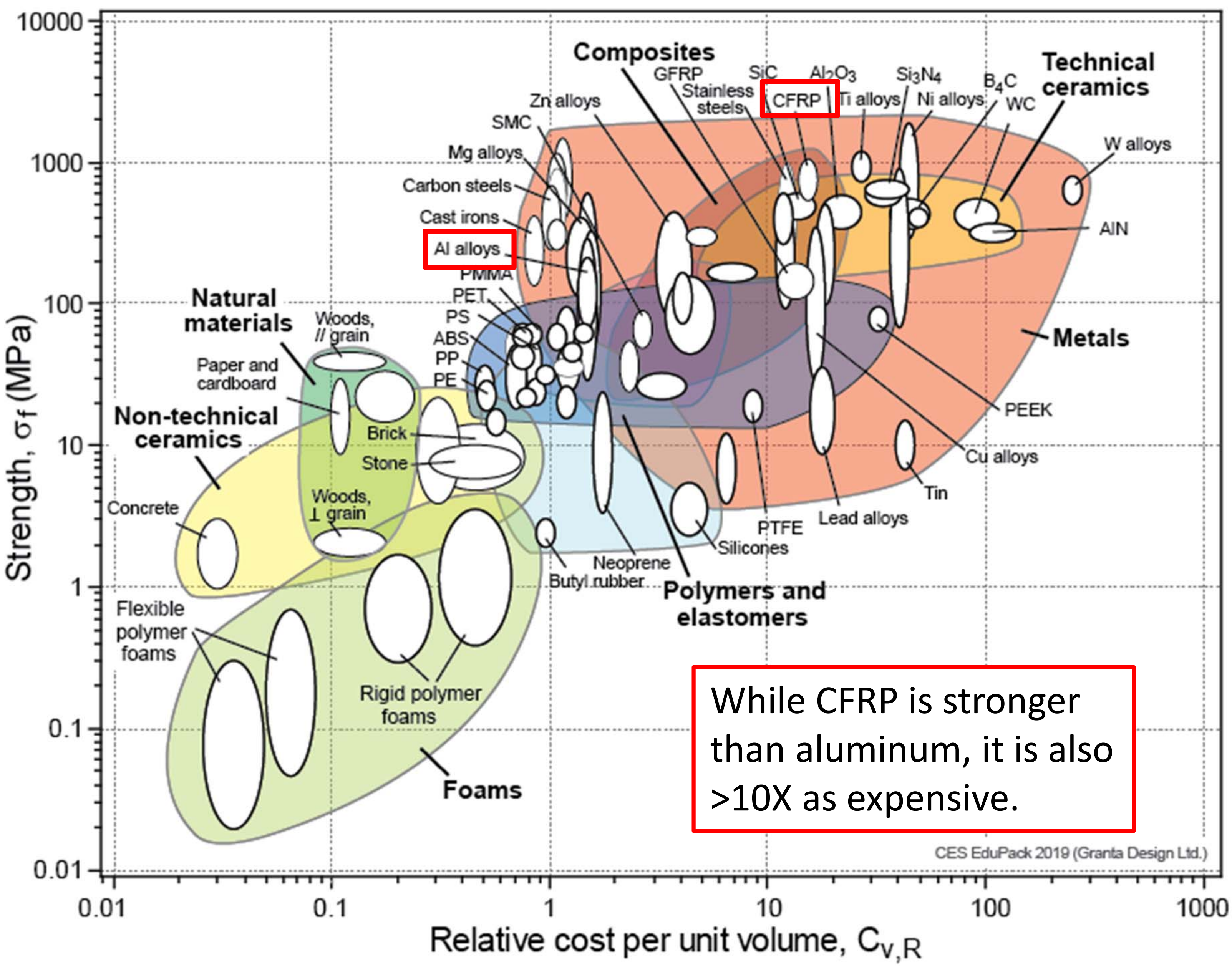
Choosing materials for the Climber



This is called an Ashby plot after the author. It is a convenient way to see where different materials fall in comparing their strengths and densities. *Note that it is a log-log plot.*

We need very light and very strong materials.

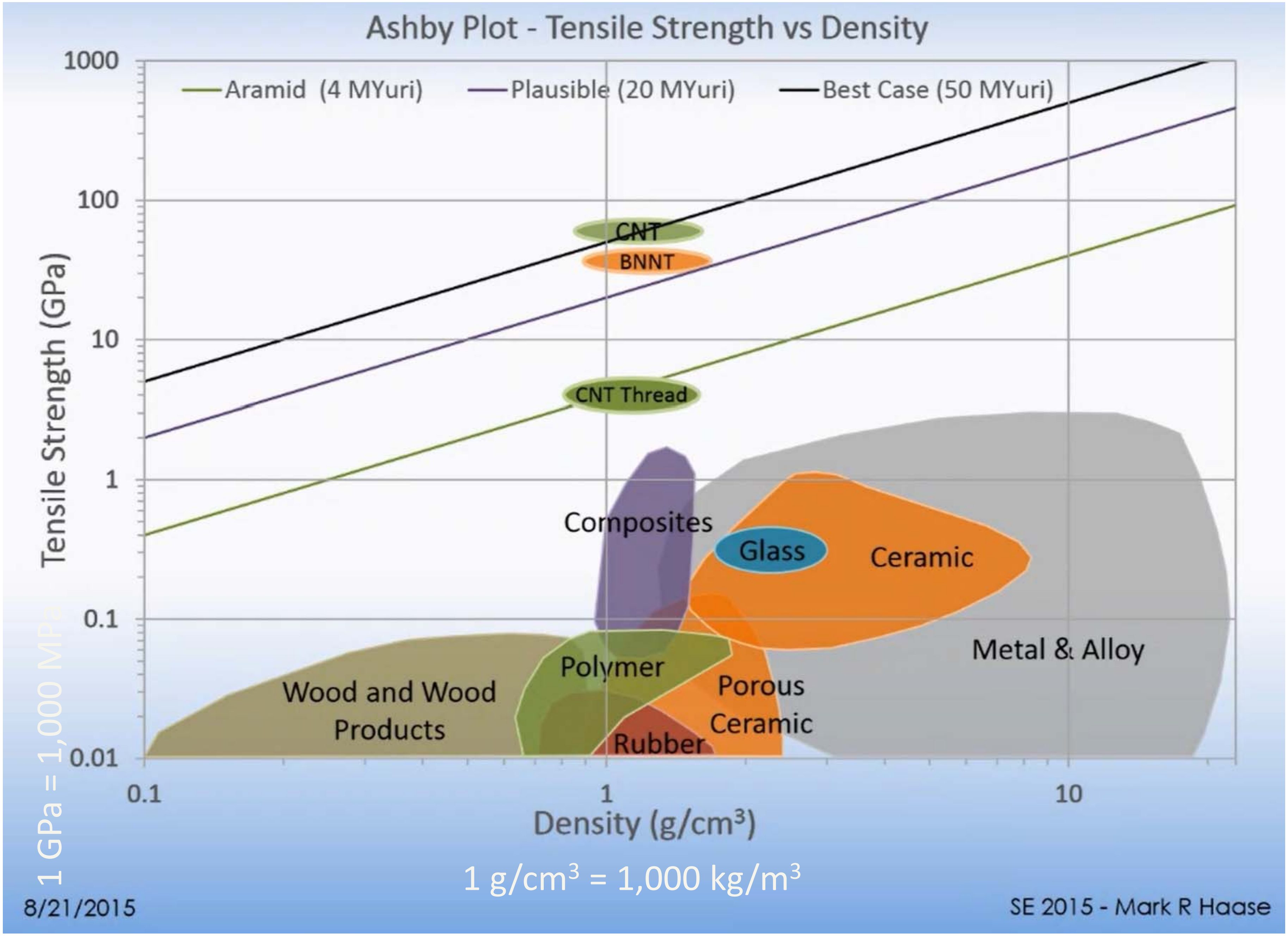
How does cost figure in?



I have tried to use only commercially available parts and materials in the reference conceptual design, to be able to see where the problems are.

I did redesign a number of parts to be made from CFRP

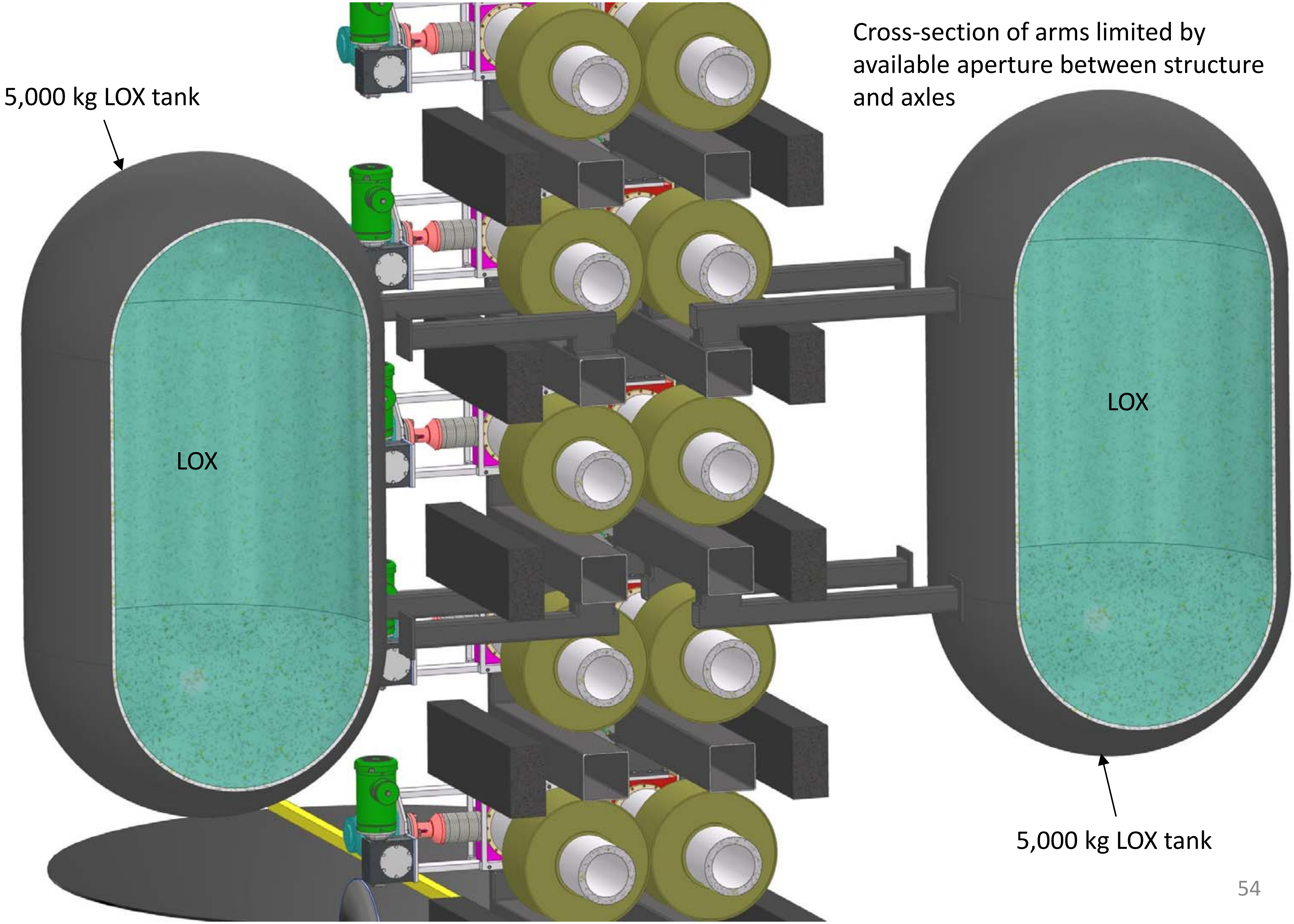
Where do CNTs fall on Ashby plots?



Next step: 5 wheel-pair climber

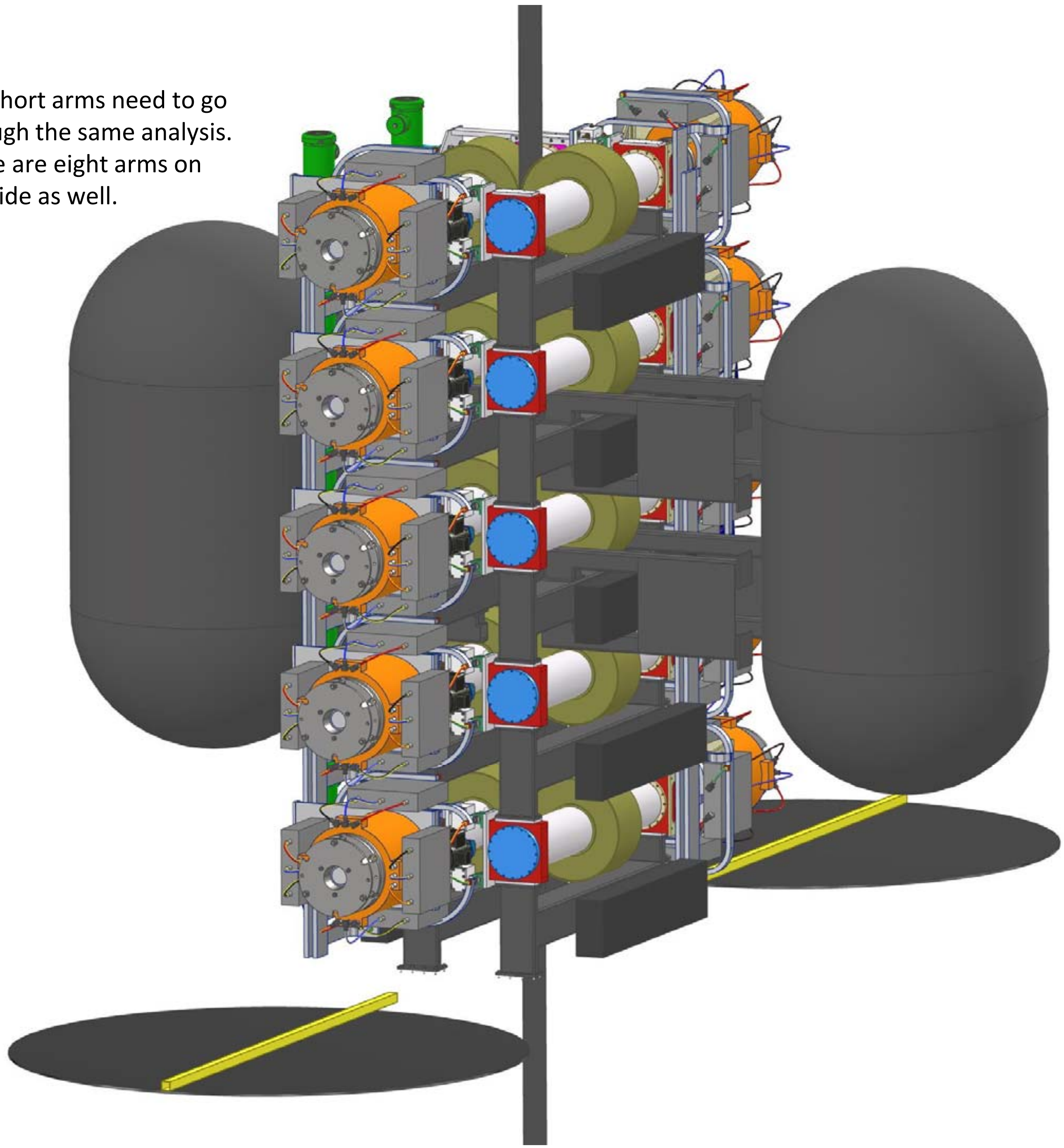
- After deciding to use an extrapolated motor, I redesigned the climber for 5 wheel pairs
  - The compression jacks went from 10 tons to 20 tons
  - The quick disconnect mechanism needs to be made from stronger materials, or made larger—not done yet
  - The shaft couplings between the motor and axles need to be replaced, but the next larger size is huge!
  - The aluminum structure was replaced with CFRP
  - The axles are being bent by twice the force
- I also looked at how to attach payload to a climber
  - I was inspired by a talk Elon Musk gave about how much fuel and LOX the Starship needs to go to Mars
  - I designed LOX tanks supported by the climber
  - The payload support arms became a non-trivial design exercise

Section view of first pass at arm design to attach LOX tanks to climber structure



Rendering of final arm design

The short arms need to go through the same analysis. There are eight arms on this side as well.



I did a finite element analysis of the longer arms and saw that there needed to be more arms and they needed a shear panel connecting them to stiffen them sufficiently to carry the load of the LOX tank. More work is needed on CFRP connections.

The PV arrays shown here were originally imagined (by me) to be able to deliver 1 MW/m^2, for a total of 12 MW. This is probably not practical in the near future. If power is limited to 4MW, it is imaginable that they could be capable of delivering 333 KW/m^2, but they may be too small to hit with a ground based laser array.

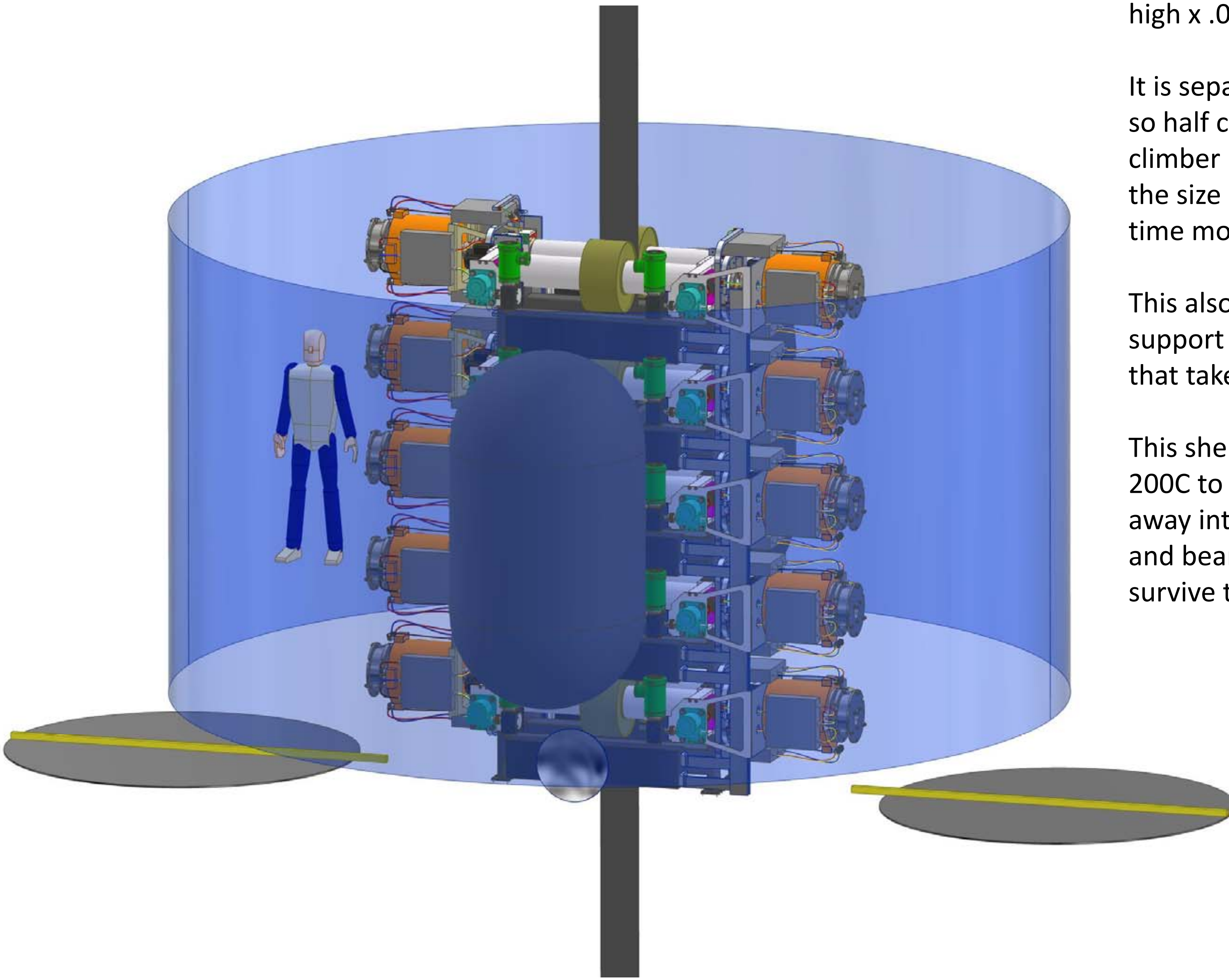
Lessons learned from payload arm design

- The two halves of the climber are not symmetrical about the tether because the wheel compression mechanism is only on one side
- This means the payload has to be placed asymmetrically to put the CG of the climber right on the tether
- There is a small access between axles and structure to attach payload arms
- 5,000 kg payload modules require significant structure to hold up

The waste heat problem

- The motors can be made 96% efficient, but that still means that >4% of 4 MW (160 KW) will need to be dissipated
- The only mechanism to dissipate heat in space is thermal radiation
- The Stefan-Boltzman law governs thermal radiation heat transfer, $q=\sigma\varepsilon A(T_2^4-T_1^4)$
  - We need ~83 m^2 of high emissivity material surface to radiate heat away at 200C
  - We need to avoid absorbing the energy of direct sunlight, which means special paint
- It's harder to radiate heat away when the temperature of the radiator is kept low (radiators become huge)

The 83 m² area shell of the space radiator



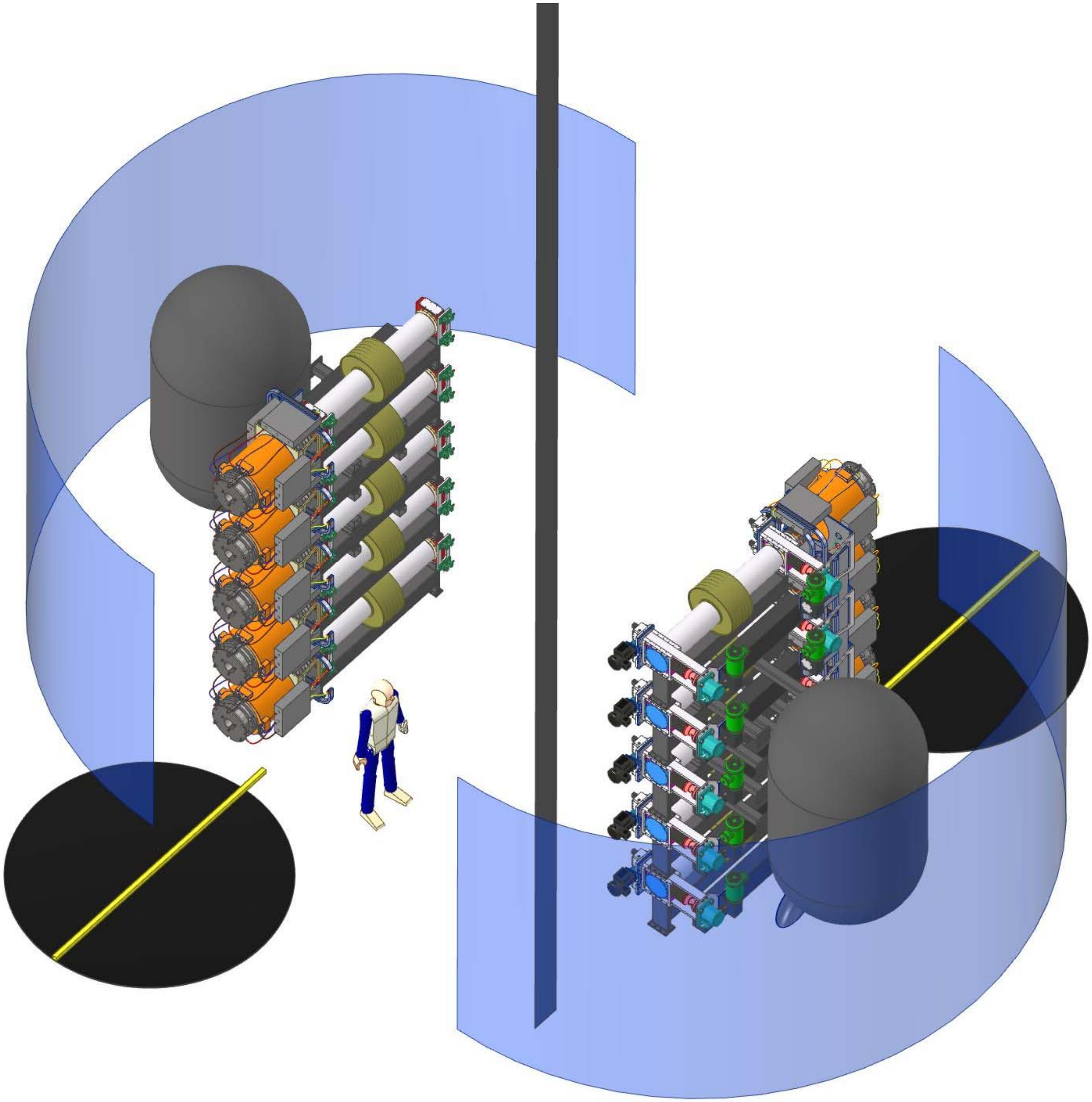
This shell is 7m OD x 3.774m high x .014 inches thick

It is separated into two halves so half can travel with a climber half. I wanted to see the size before I spent a lot of time modeling the plumbing.

This also needs structure to support it, which adds mass that takes away from payload.

This shell needs to heat up to 200C to radiate enough heat away into space. The motors and bearing lubricants need to survive this temperature.

The climber separated into two halves for assembly around the tether



The climber as two separate machines joined together around the tether at the surface of the Earth implies a host of fixtures to transport, support and position the climber halves.

Think of the gantries that support rockets before launch.

A sequence of assembly steps is necessary because payload is mounted inside the space radiator. A remote payload disconnect mechanism may be necessary to separate the payload at GEO.

Conclusions

- For years we have wanted 20 tonne climbers to carry 13 tonnes of payload, a 65% payload ratio
- The 5 wheel-pair climber shown can carry 10 tonnes of payload, so it has a 50% payload ratio to any altitude
- Rockets have payload ratios of a few percent and they get lower the farther the payload is delivered—*and chemical rockets cannot do better*

More conclusions

- We're trying to find/design motors with sufficient torque, but it's an open question if current motor technology is sufficient for direct drive climbers
- Once tether material is developed, other strong and light composite materials may become available to lighten the structure of the climber
- Much thought has to go into making the climber assemble-able around the tether
  - It is a multi-step process that will take preparation time for each lift-off

BACKUP SLIDES

The difference between the original couplings and the new one required to carry the load

A custom coupling may be made that is similar in size to the existing ones but made from stronger materials to handle the torque. The bearings may be the limiting factor in sizing the coupling.