Getting the Mass of the First Construction Climber under 900 kg

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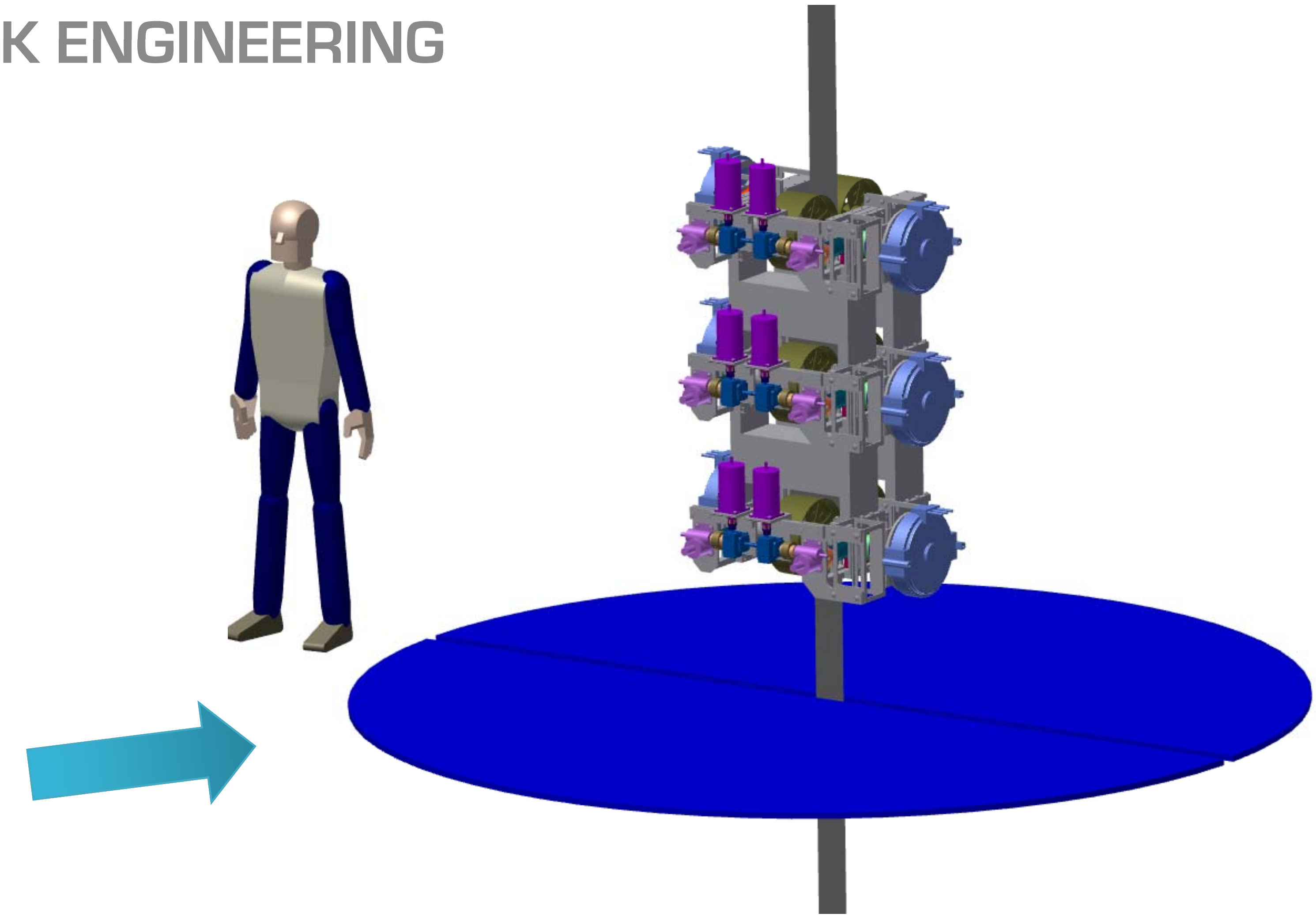


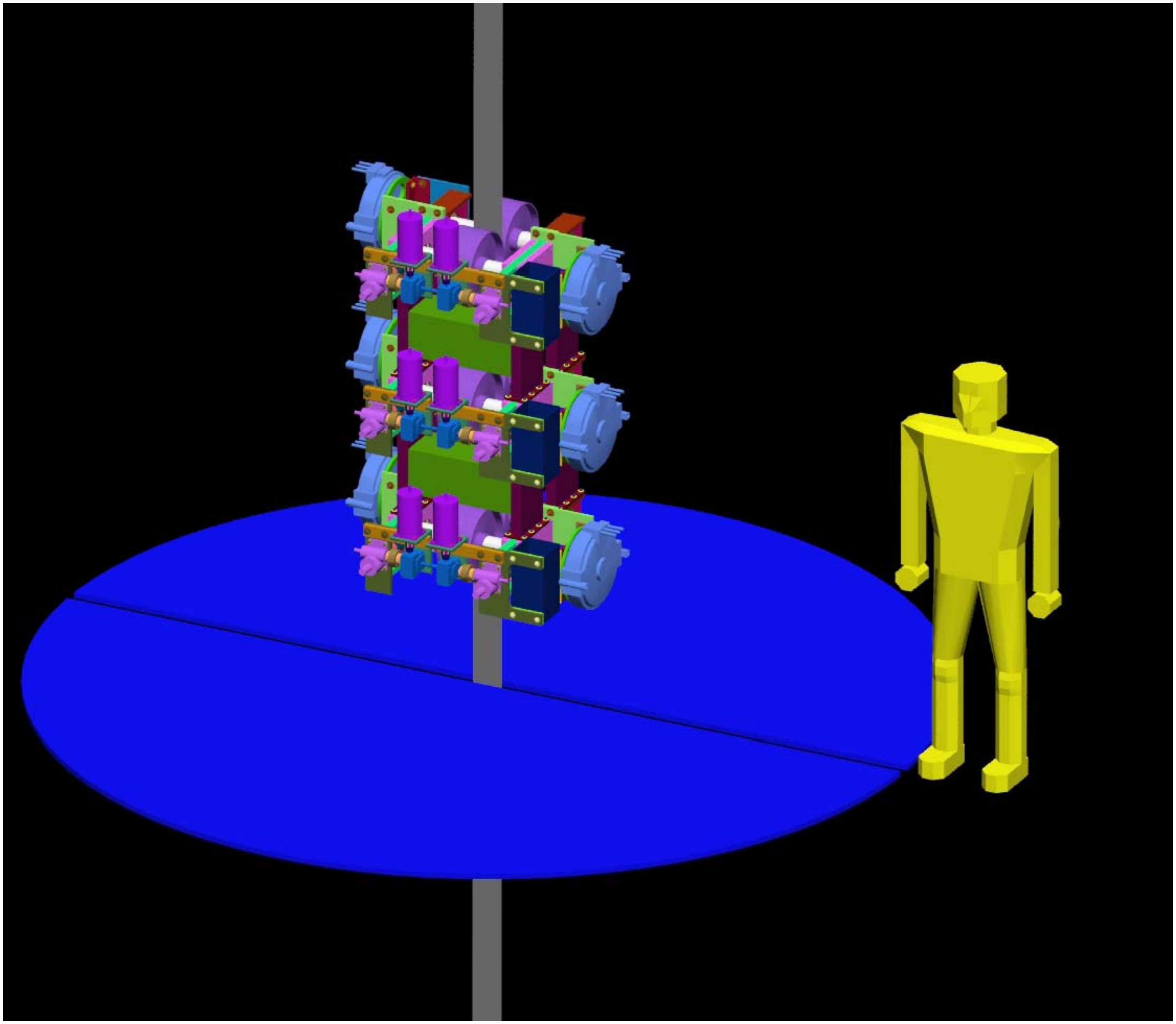
Introduction

- In 2004 I presented a design for the first construction climber using non-spaceworthy parts to see how close I could come to the mass budget in Edwards’ and Westling’s book
  - 900 kg for the first construction climber
  - 20-100 kW motors
  - I focused on the traction drive only
- I eliminated the Edwards track after analysis
- **The design was ~3X too heavy**
  - Not too bad for a first try but…
- The paper I wrote for this conference is a primer on climber design and shows some interesting things

Outline

- Brief description of 2004 conceptual design
- A free body diagram and some theory
  – Constant velocity climbing
  – Constant power climbing
- Discussion of the climber mass budget
- FEA of the wheels and dynamic stress
- FEA of the structure and static stress
- Motor mass problem
- Comments on the Center of Mass of the climber

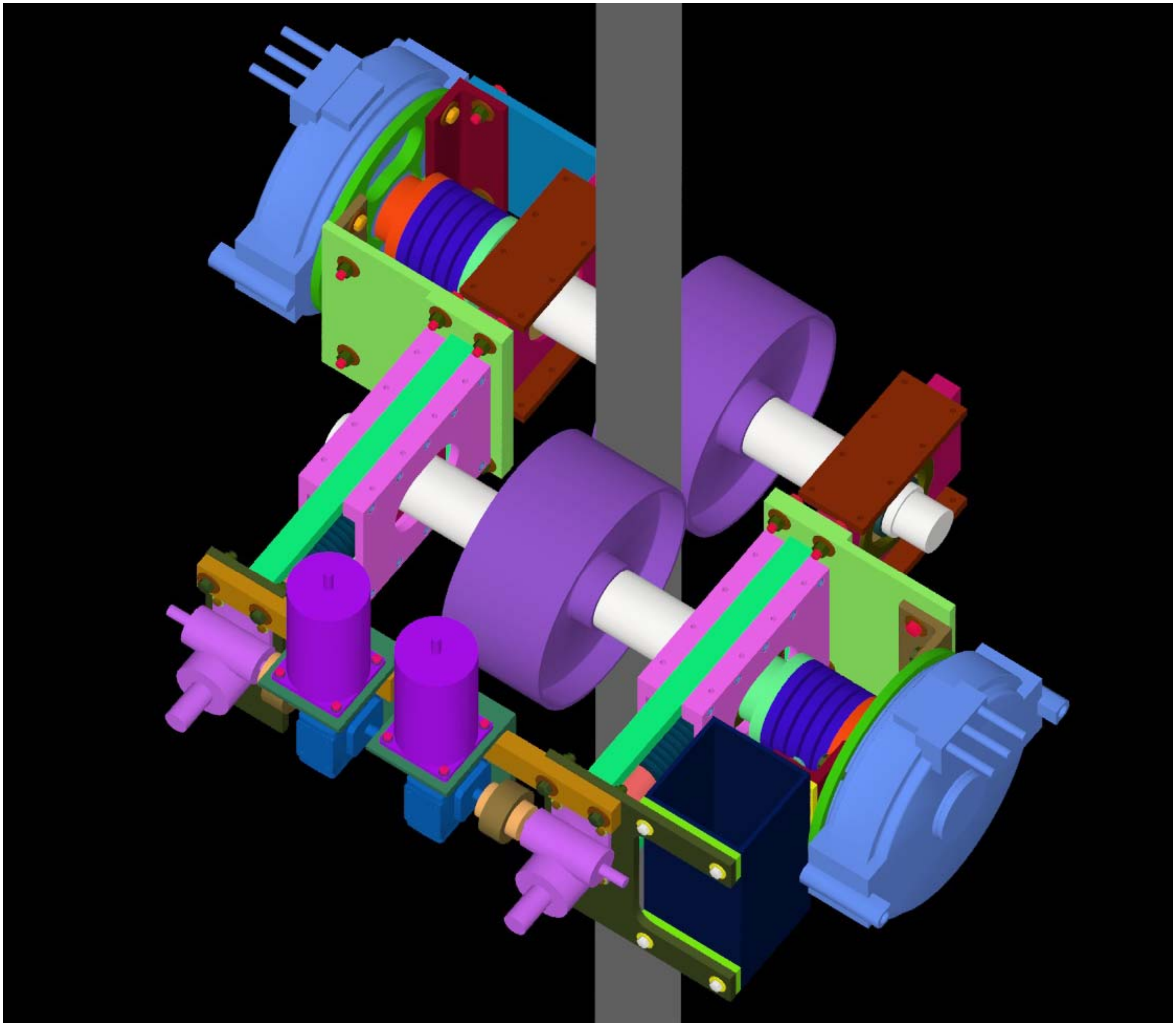
2004 Conceptual design



Pinched wheel design with no track

This is an incomplete scale model of the first climber. The PV array (blue disk) is 4 m in diameter

Two wheels clamped onto the ribbon

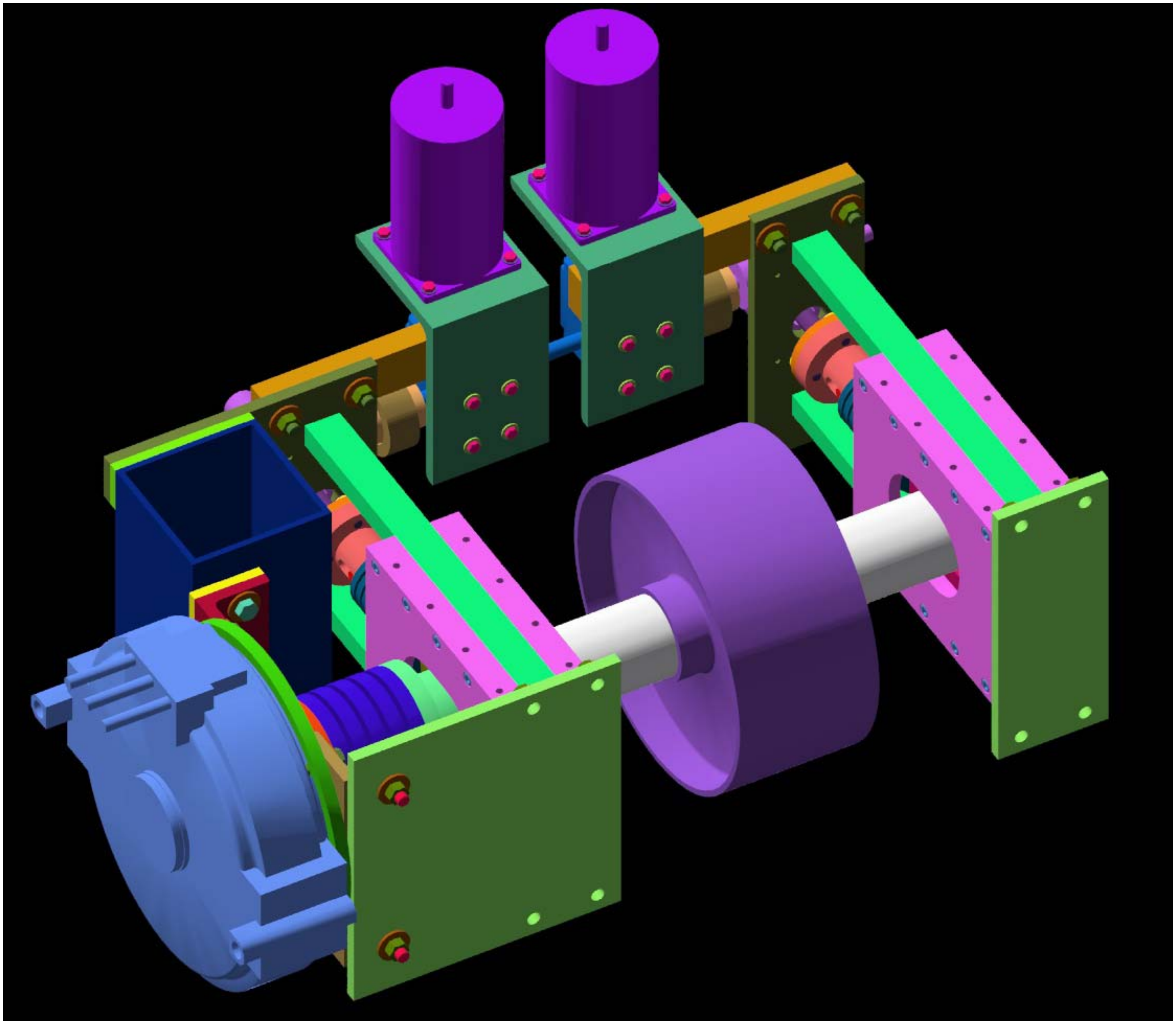


The axle on the far side of the ribbon is fixed to the frame of the climber through self-aligning bearings.

On the near side of the ribbon, the axle is mounted on a linear slide so the wheel can be pressed against the ribbon or retracted away from it.

Motors are connected to the axles by Schmidt couplings to absorb any angular or lateral offsets.

Floating axle traction module



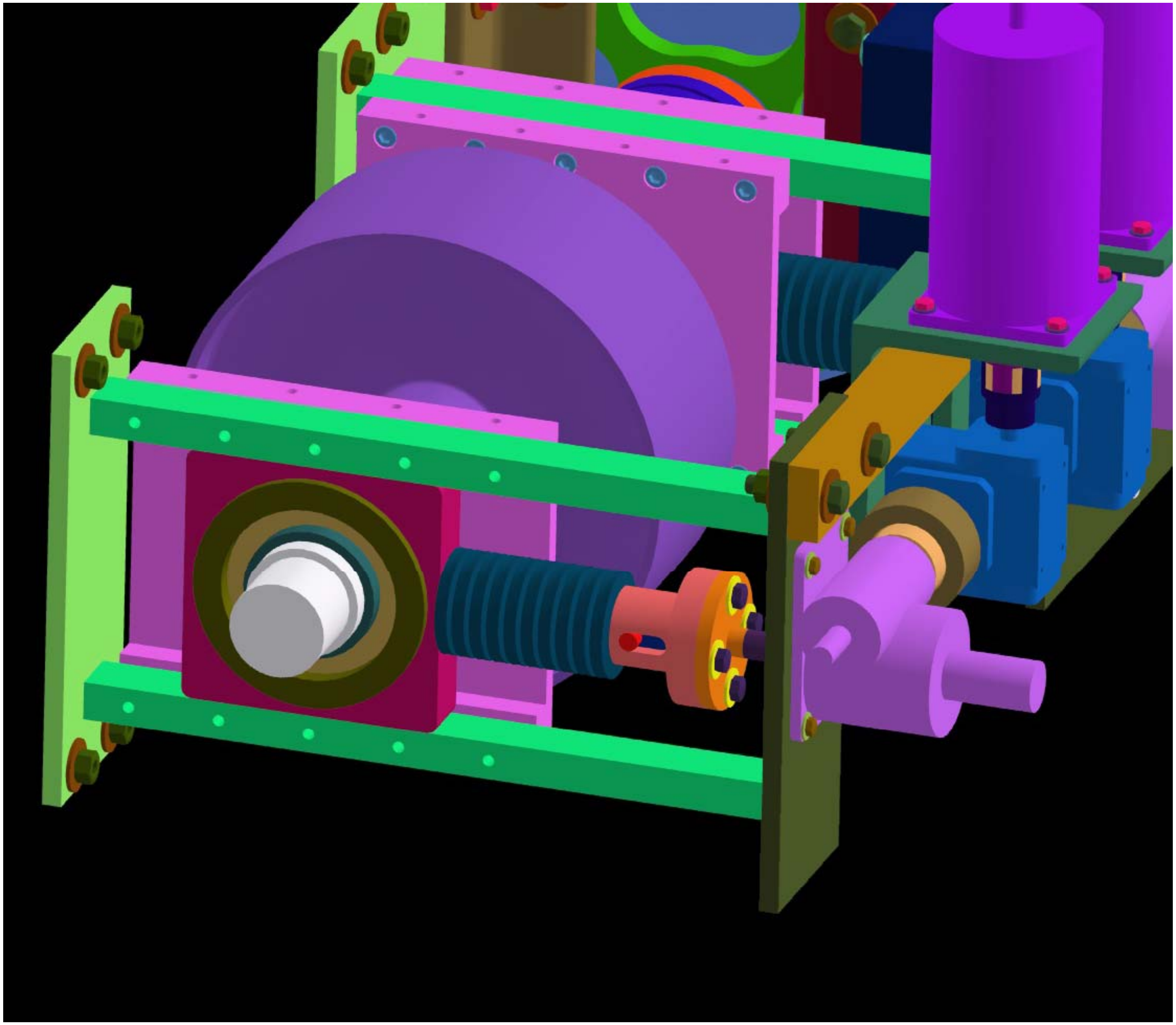
The two sides of this module are not stable to torsion without the interface structures between modules

Wheel pinch forces are transmitted through the light green plates on either side of the wheel.

Forces coming from the rest of the climber are connected through the bearing housing slides

Every wheel is motorized.

The wheel compression mechanism

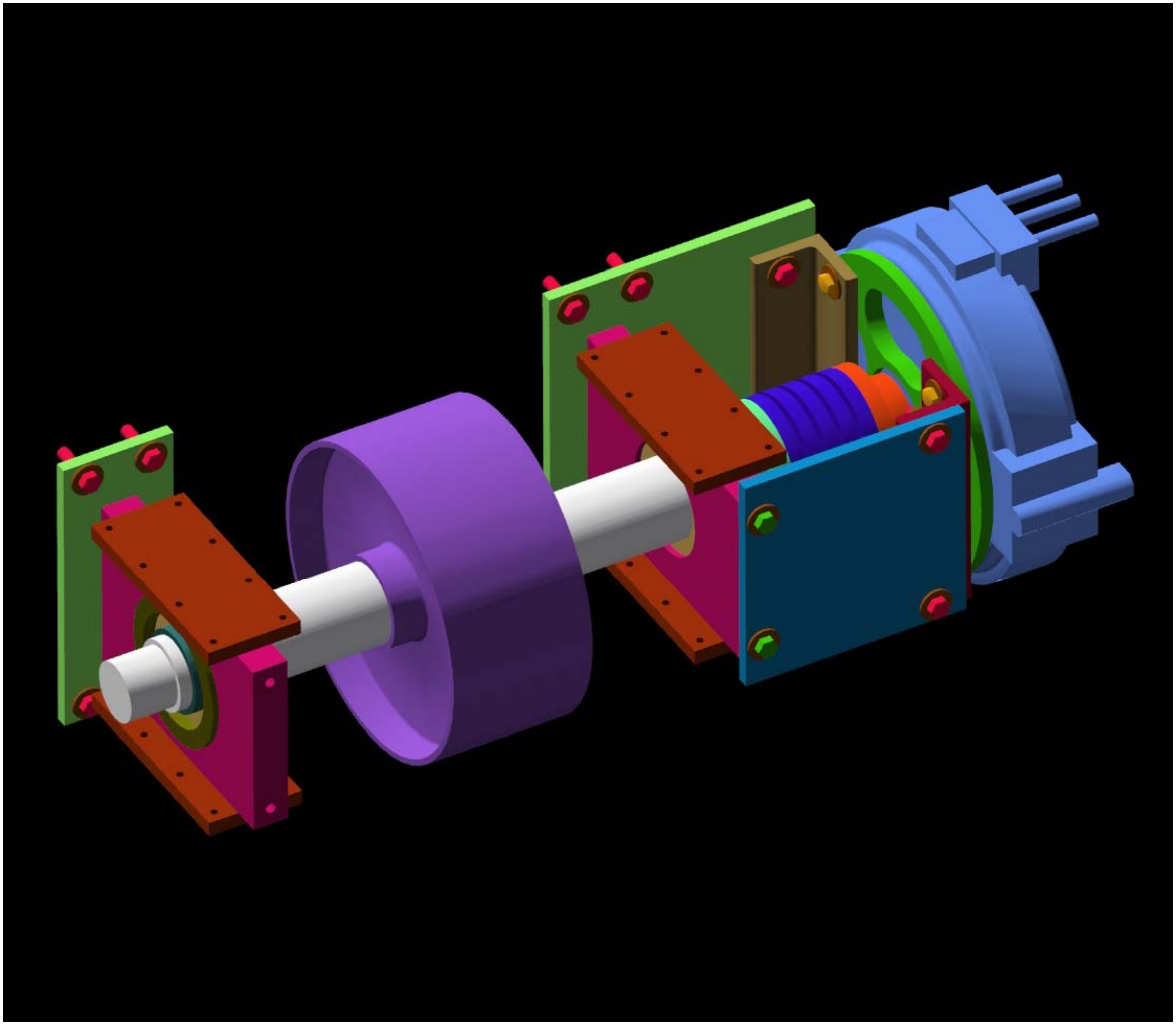


One ton screw jacks compress a stack of belleville washers

This concept allows great resolution in the application of force to the axle

The components were all sized to take the loads but are not space-worthy. A concern is whether space-worthy components are even larger.

Fixed axle traction module

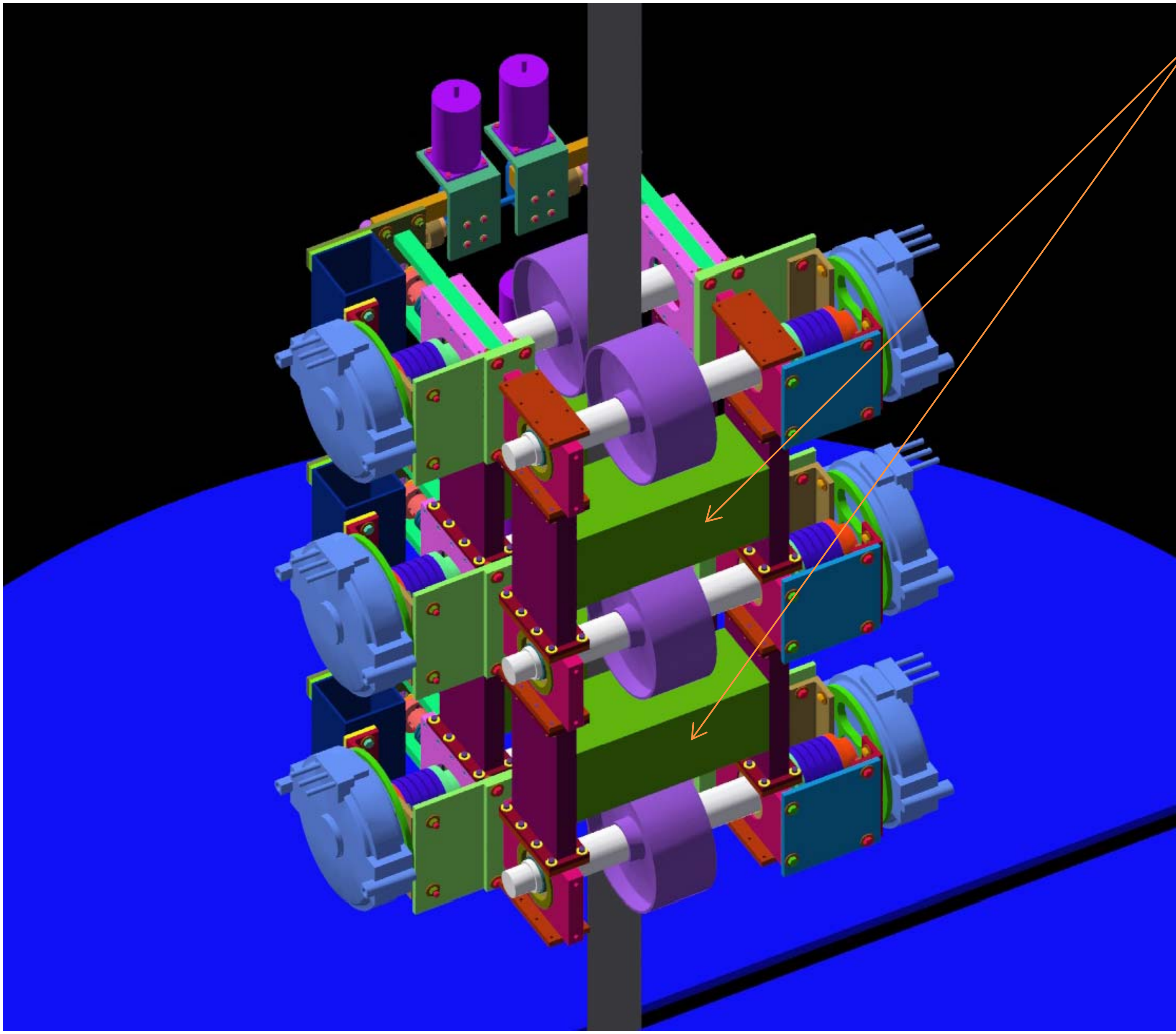


This module drives a wheel and absorbs the compressive force coming from the wheel on the other side of the ribbon.

This module is lighter than the one on the other side so balancing a climber to force the CG to lie within the ribbon is an issue.

Motors shown are 50kW axial gap models from Precision Magnetic Bearings.

Interface structures



The structural modules in between the traction modules give torsional stiffness to the traction modules and allow loads from the rest of the climber to be coupled to the drive train.

This drive design (not including the PV arrays) weighs 1625 lbs, or 737 kg. This is about 3.16X the allowed 233 kg for the drive train. 20kW motors reduce it to 647 kg, or 2.77X.

What was learned back in 2004

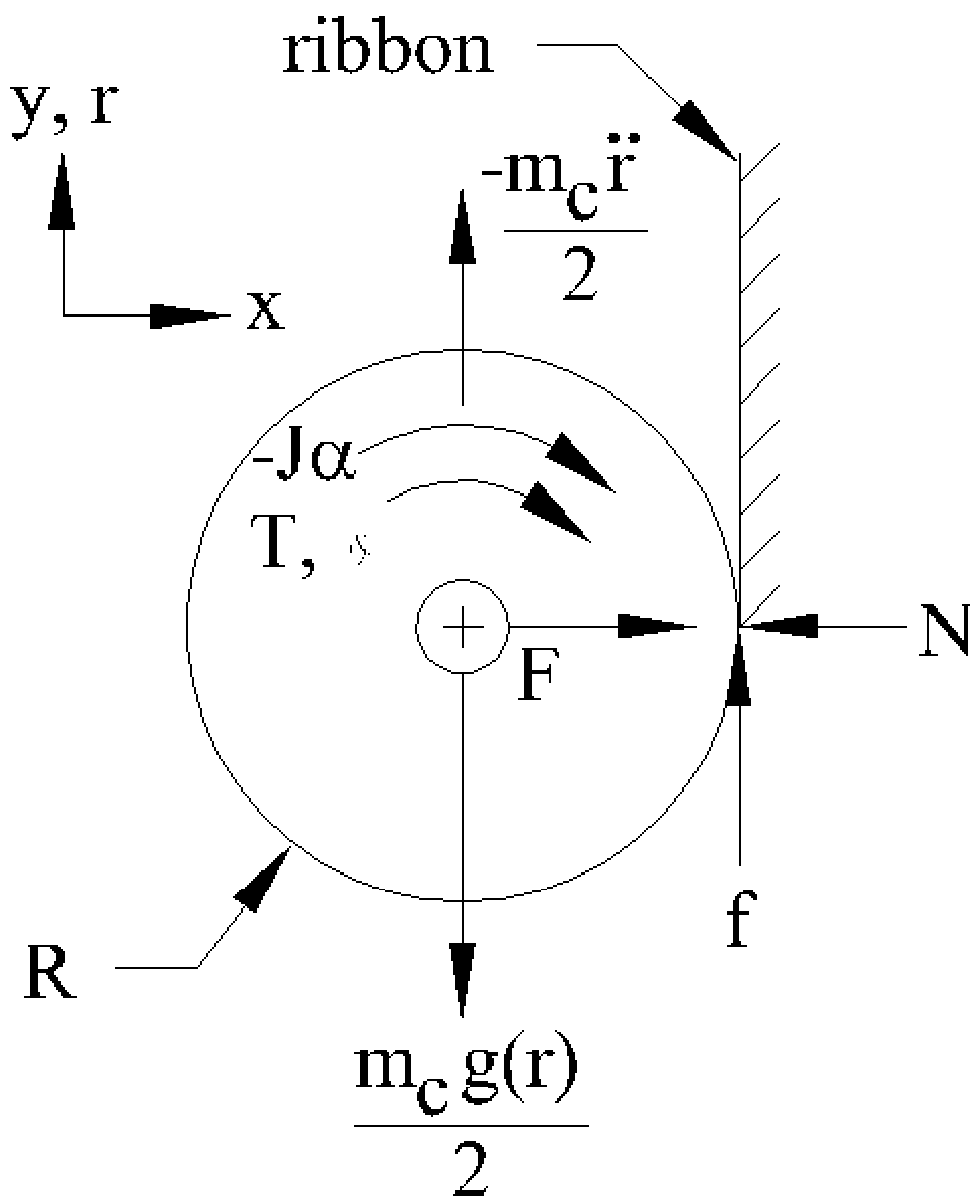
- The friction between the wheels and the ribbon determines how hard the wheels have to be compressed against each other to develop traction
- The wheel compression force and the rotation speed determines the *dynamic stress* in the rotating parts
  - Dynamic stress allowables are governed by fatigue
- The compression force also determines the **static stress** in the non-rotating parts of the climber
  - Static stresses are governed by yield stress divided by the safety factor
- Three pairs of wheels is the optimum number

What I learned recently

- Small wheels (d<4 inches) rotate too many times to get to the end of the ribbon
  - Can’t satisfy allowable fatigue stress
  - Can’t rotate them fast enough to get an acceptable speed out of the climber
- Large wheels (d>13 inches) require too much torque to keep the climber from rolling backwards down the ribbon
  - This is a limitation of the holding torque of the motors
- The power required by the climber is higher than stated in the book for reasonable speeds near Earth
- The climber cannot satisfy either constant power or constant velocity scenarios

Where it all starts:

Free Body Diagram of a Wheel



This picture models a single wheel on a climber with just two wheels, or it can be used to model the tracked drive as well by considering J as the rotary moment of inertia of the track plus its drive wheels.

f = friction force from ribbon

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

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:

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

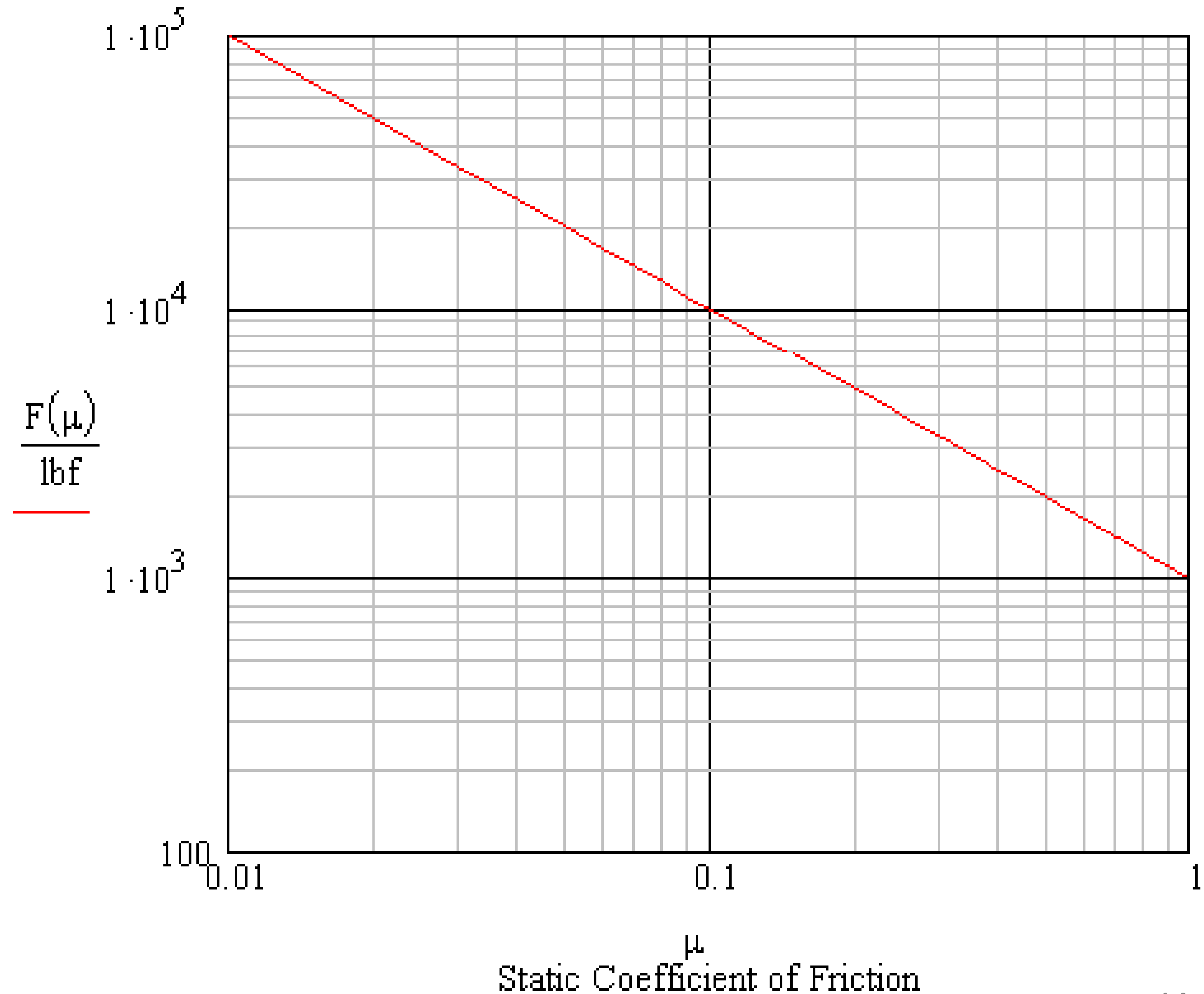
J is the rotary moment of inertia of the drive train.

(This equation shows why the track hurts the acceleration of the climber. We want J to be as small as possible. The track also cannot produce traction between wheels.)

Summing the forces in x and y determines the wheel pinch force as a function of μ, the coefficient of friction

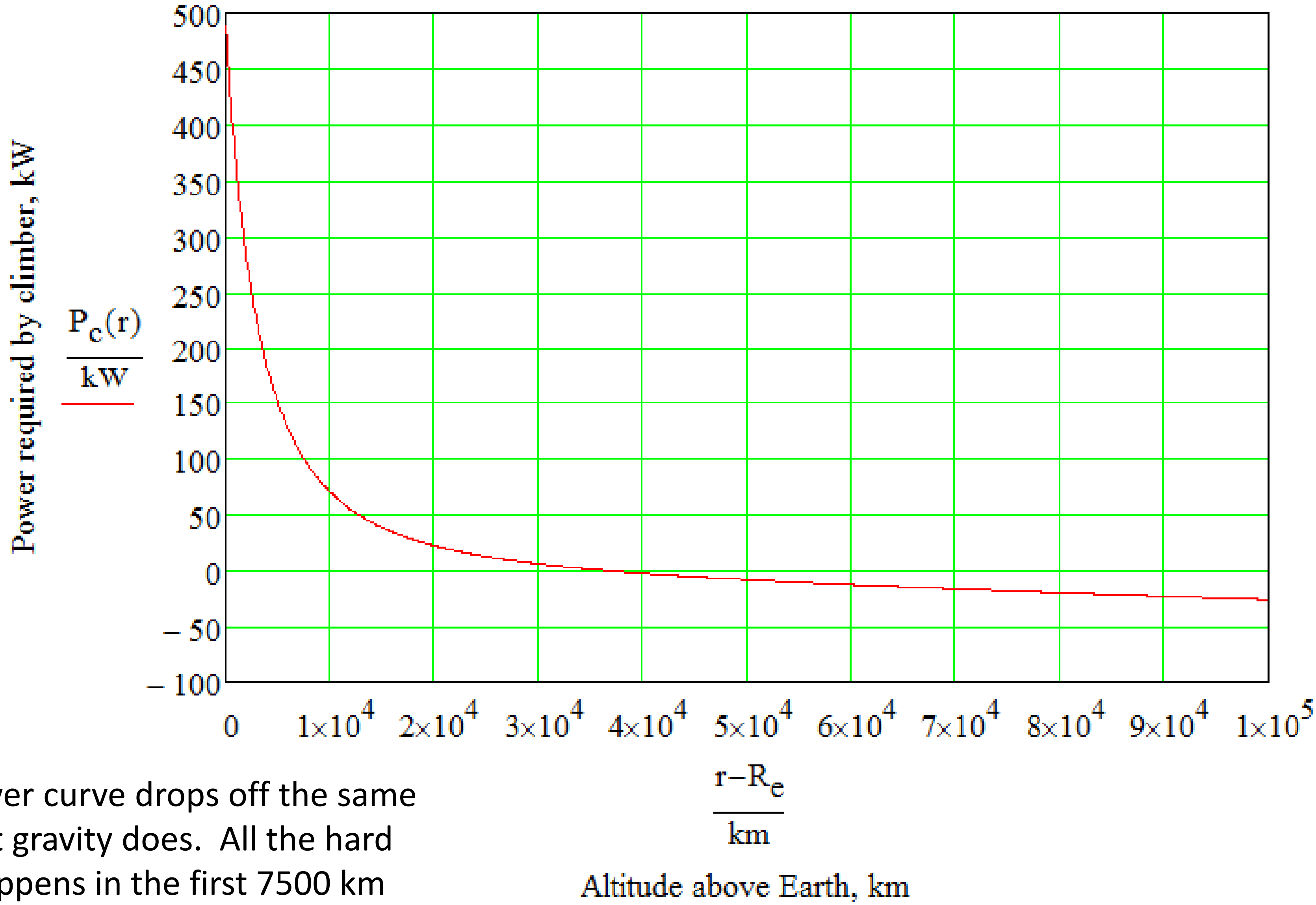
$$F(\mu) = \frac{m_c g(r)}{2\mu}$$

This graph and equation gives the total force required to pinch the wheels together around the ribbon to just keep a 900 kg climber from sliding down the ribbon. It takes almost 10,000 lbs, (5 tons) for μ = 0.1



Calculating the amount of power it takes a 900 kg climber to climb at a constant 200 km/hr

$$P_c(r) := m_c \cdot a_c(r) \cdot v_c$$



The power curve drops off the same way that gravity does. All the hard work happens in the first 7500 km above Earth.

Constant Velocity Power conclusions

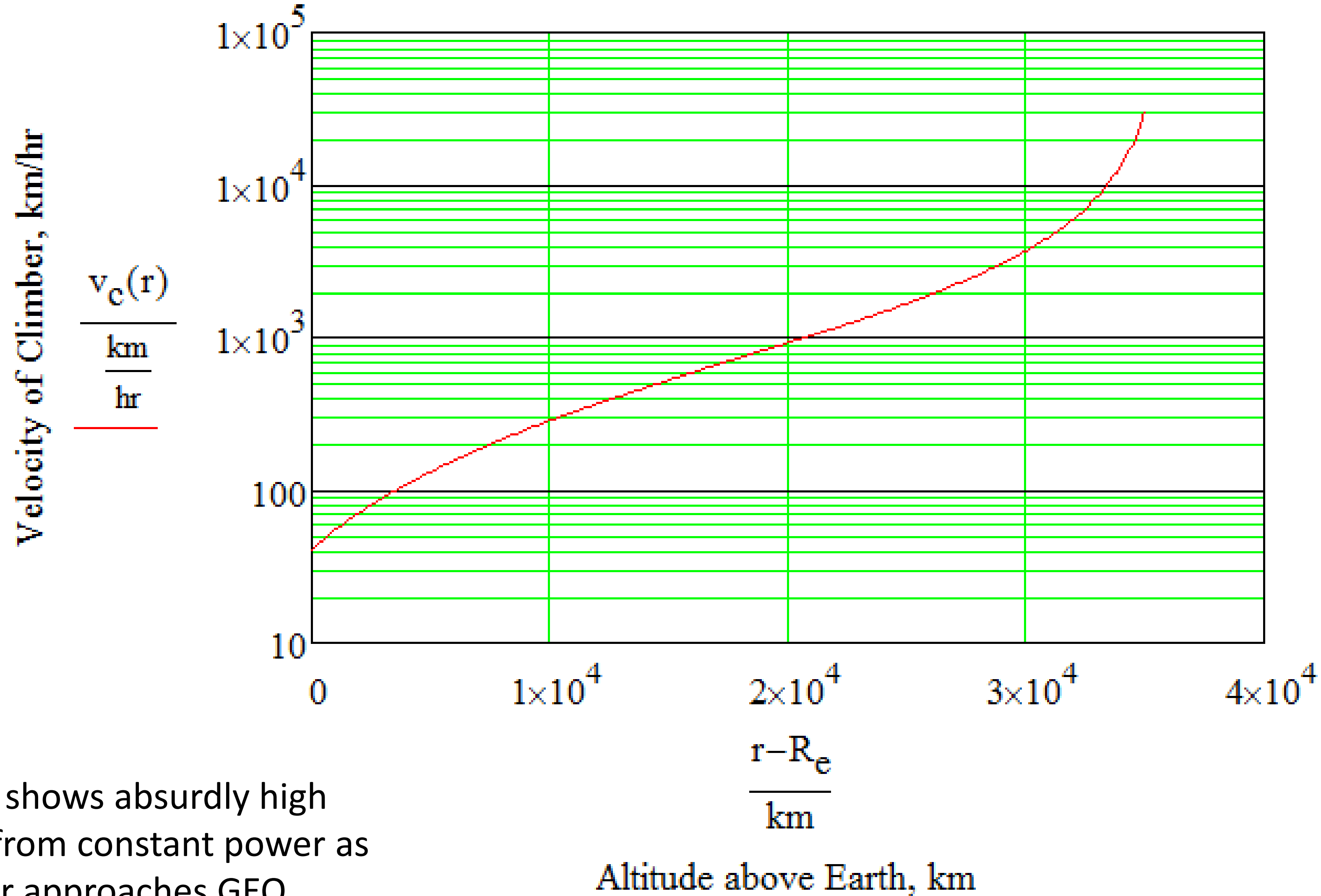
- The climber requires more than 100 kW to climb at 200 km/hr near the Earth's surface
- The power requirement for 200 km/hr climbing does not drop below 100 kW until an altitude of 7,500 km (4,660 miles) above the surface of the Earth
  - For comparison, the altitude of the International Space Station is 370 km (230 miles) up
  - The Space Shuttle's maximum altitude was 960 km (600 miles)

Calculating the velocity of the climber if the power is held constant at 100 kW

$$v_c(r) := \frac{P}{m_c \cdot a_c(r)}$$

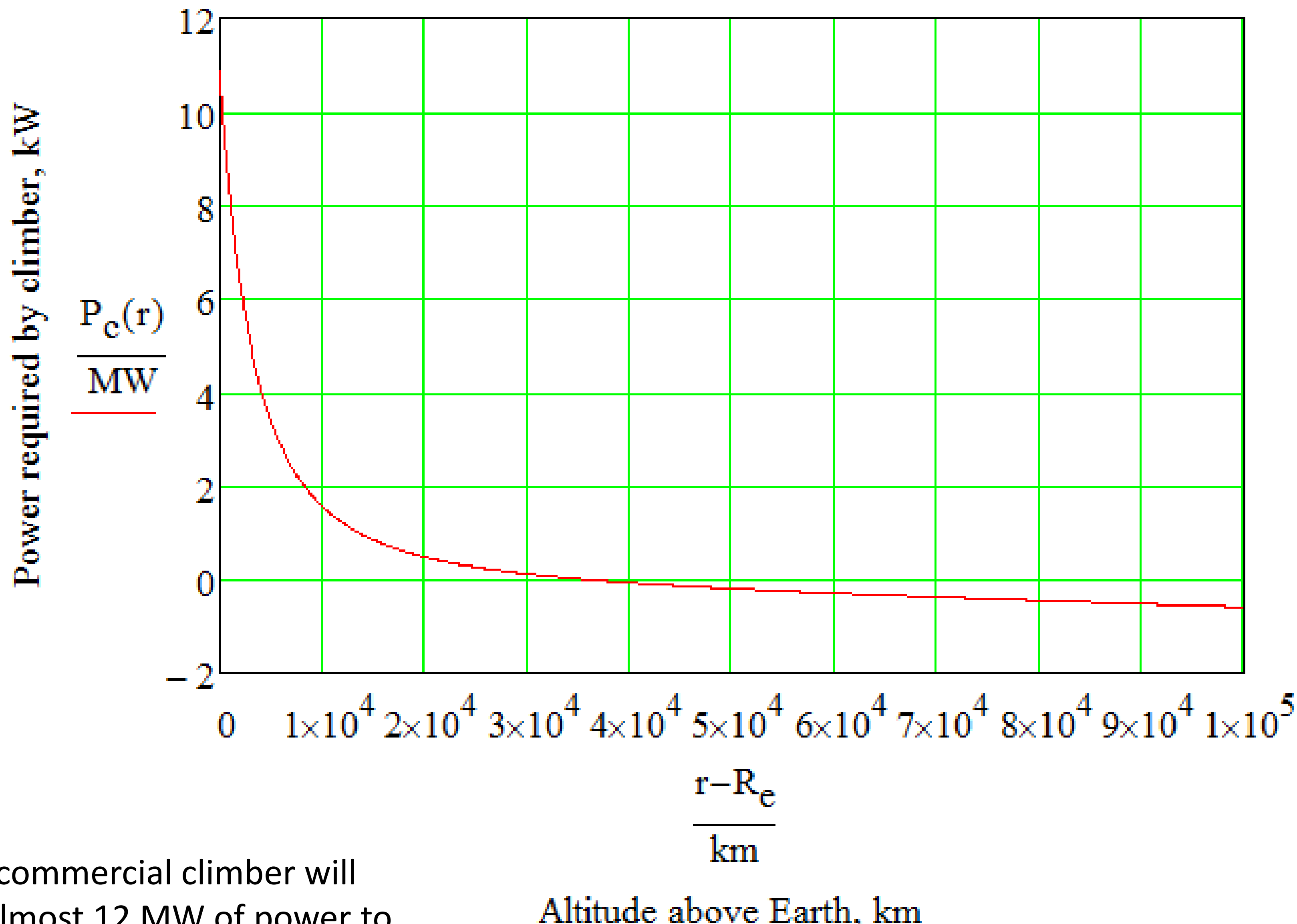
Rearranging the previous equation

This graph has to be truncated below GEO because the climber does not need power above GEO, it has to dissipate power.



This graph shows absurdly high velocities from constant power as the climber approaches GEO.

Calculating the amount of power a 20 tonne commercial climber needs to climb at a constant 200 km/hr



The first commercial climber will require almost 12 MW of power to climb at 200 km/hr close to Earth.

More Power Conclusions

- The velocity profile of the climber will be a programmed curve of high torque/lower speed at lower altitudes and lower torque/higher speed at higher altitudes
- It is not clear that time at higher speed can make up for the reduced speed close to Earth
  - Trips up the ribbon will be longer than calculated for constant velocity
- The construction climber's purpose is to add more ribbon to the pilot ribbon
  - This process will have to be designed with variable speed climbing in mind

The Mass Budget Problem

- Edwards and Westling laid out a mass budget for the first 900 kg construction climber
- Since I was only looking at the traction drive, I was only using 3 items from the budget
- The calculated compression force sized the motor, gear box and screw jack in the floating axle module
- I reduced the masses of the wheels, axles and aluminum structure from the 2004 design to try to satisfy the mass budget (without success)

Table 1: Climber Mass distribution from *The Space Elevator* by Edwards and Westling

Table 3.2: Mass Breakdown for the first climber (from the book)

Component	Mass (kg)
Ribbon	520
Attitude Control	18
Command	18
Structure	64
Thermal Control	36
Ribbon Splicing	27
Power Control	27
Photovoltaic Arrays (12 m^2, 100 kW)	21
Motors (100 kW)	127
Track and Rollers	42
TOTAL	900

Design constraint of <233 kg comes from adding the red numbers in the table.

Not all of the structure can be dedicated to the drive system.

Table 2: Mass Breakdown of components in 2004 design

Description of climber components:	Climber with six 20 kW motors	Climber with six 50 kW motors
Mass of 12 self-aligning bearings, kg	16	16
Mass of 6 axles, kg	32	32
Interface structural material, kg	51	51
Mass of 6 wheels, kg	53	53
Mass of 6 Schmidt couplings	63	63
Mass of structure in 3 fixed axle modules, kg	71	71
Mass of 6 motors, kg	84	174
Mass of 3 pairs of compression mechanisms, kg	136	136
Mass of structure in 3 floating axle modules, kg	141	141
Total mass of climber traction drive only, kg:	647	737
Required drive system mass, kg:	<233	<233

Motor masses courtesy of Rick Halstead, Empire Magnetics

Colored numbers are the 2004 masses of components later reduced by FEA analysis

Things to note about the mass distribution

- From Table 1, the motors represented almost 56% of the 233 kg budget for the drive train.
- Table 2 shows that the mass of the motors I found in 2004 made up only 13% of the total mass of the design, and were two thirds of the allowed budget in Table 1.
- The fact that the motors I used were lighter than the budget meant that **the structure was the problem** in reducing the mass of the drive.
- The mass of the conceptual design without the motors was 562.5 kg and the budget for this mass was less than 106 kg. The structure needed to be reduced in mass by a factor of 5.3.
- Is there a better material than aluminum to make the structure from?

Table 3: Comparing various engineering materials to aluminum.

One of the ways to reduce the mass of the structure is to use a lighter material with the same (or higher) strength as aluminum. This table shows that none of these engineering materials is 1/5 the density of aluminum and Aerographite is nowhere near the strength of Al. None of them are 5X stronger than Al either.

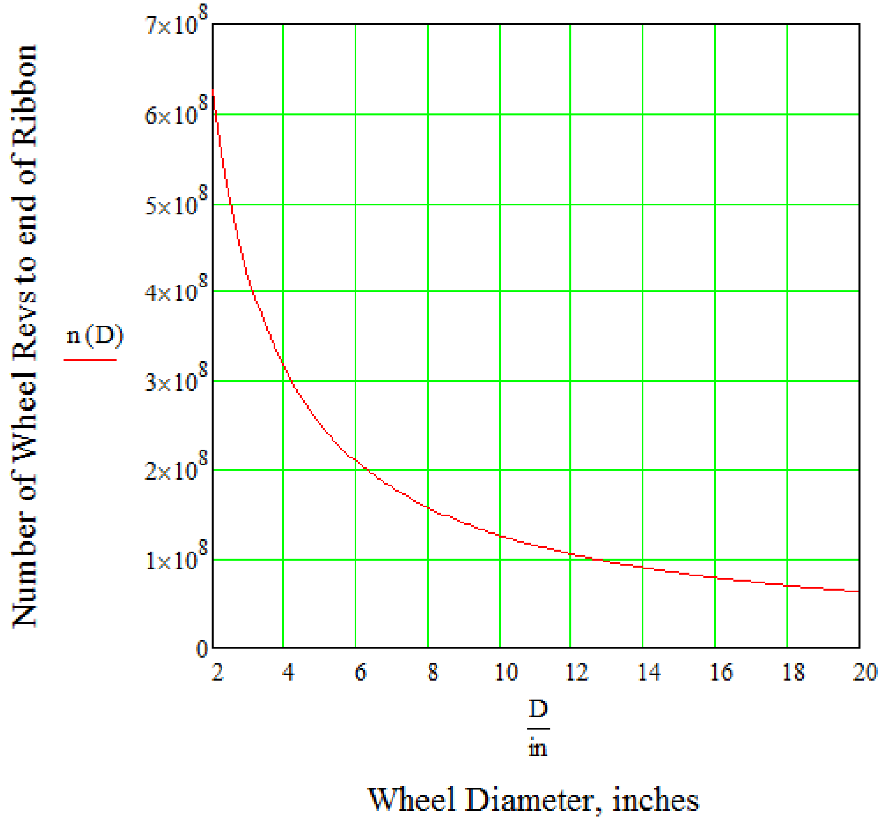
Material	**Density, lb/in^3**	**Ratio of density to Al**
Aerographite	3.07E-04	0.003
Carbon composite	0.058	0.592
Magnesium AZ80A-T5	0.065	0.663
Beryllium	0.067	0.682
Al 6061-T6	0.098	1.000
Titanium, Ti-8Al-1Mo-1V	0.158	1.612
Titanium, Ti-6Al-4V	0.160	1.633
SS 321	0.290	2.959

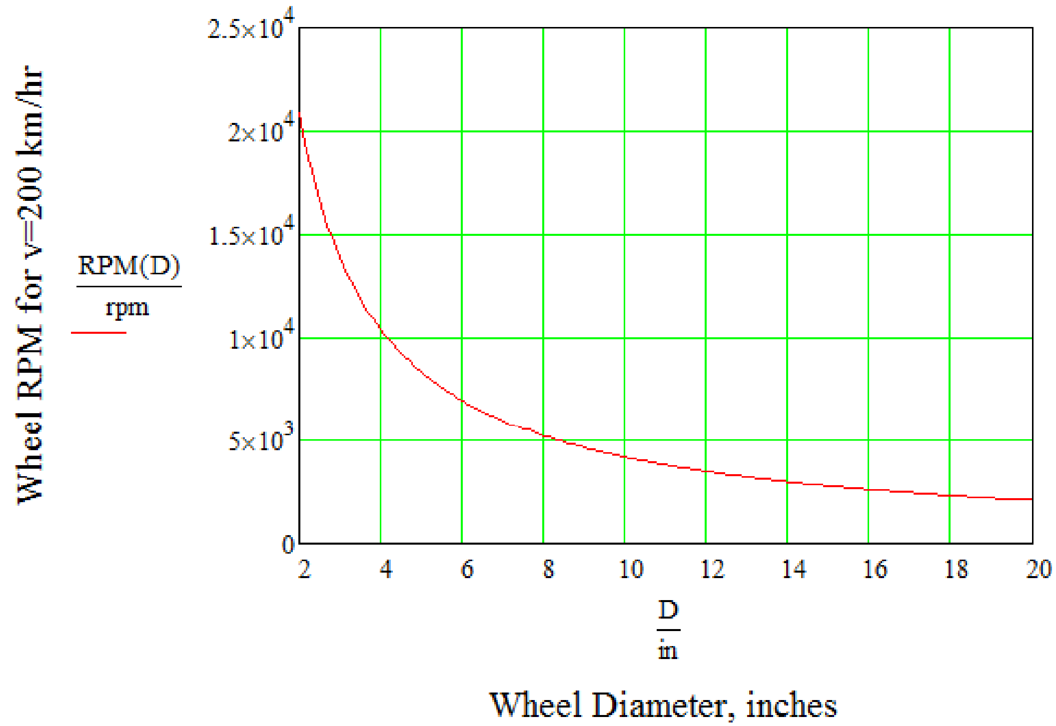
Conclusions on the mass budget

- The simple answer to the substitution of a better material than aluminum is “no”.
  - The design will have to be carefully reworked to reduce the cross-section of material wherever possible
  - Higher strength materials may help in some places
  - All of the alternatives to aluminum are more expensive
  - I didn’t want to consider CNTs as a structural material because nothing is known about using them to build structures yet
- The problem is serious because if I made every colored number in Table 2 go to ***zero*** I would still be over the mass budget
  - All of the numbers in the table need to be reduced by a lot

Wheel Analysis

- Analyzing the wheels with compressive stress and dynamic rolling stress demonstrates that getting ~12" wheels to rotate faster than 10,000 RPM may not be possible
- Almost everything you can think of (except for dentist's drills) runs at a few thousand RPMs (or less)
- Dynamic stress increases as the **square** of the rotation speed!

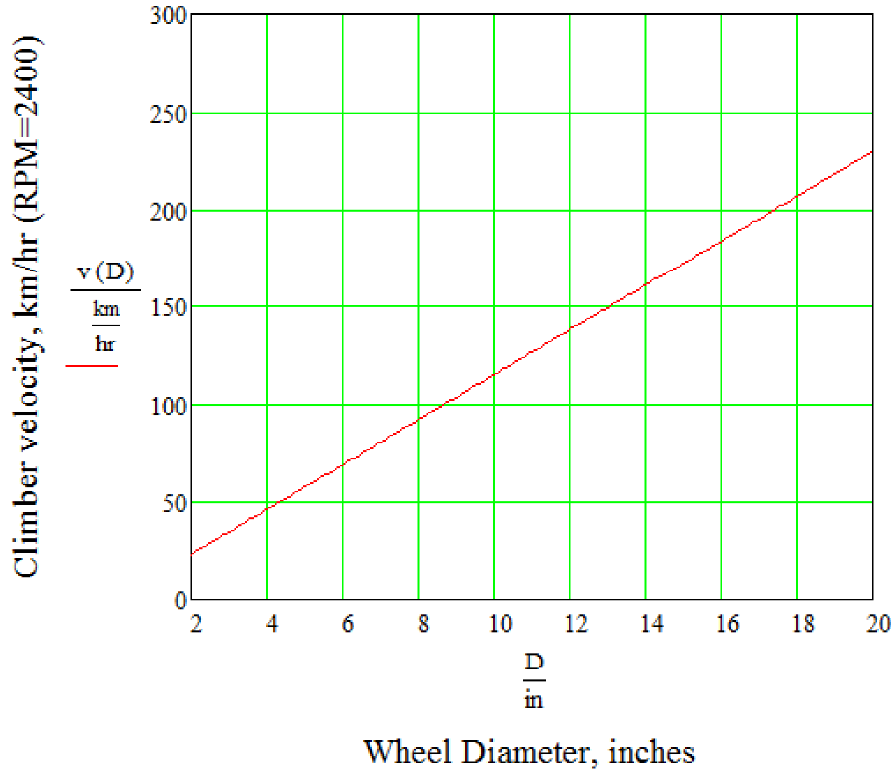




This graph shows how many times a wheel has to rotate to get to the end of a 100,000 km long ribbon as a function of the wheel diameter. Wheels below 12” in diameter are in the very high cycle fatigue range.

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. (The motors would have to be larger in diameter to develop the torque required.)

From Shigley's *Mechanical Engineering Design*, 5th ed:



This graph shows what the climber velocity would be as a function of wheel diameter if the rotation speed is limited to 2400 RPM. We want the climber to climb at least 200 km/hr.

The motors will need to be able to rotate faster than 2400 RPM at higher altitudes.

$$\sigma_t(r) = \rho \cdot \omega^2 \cdot \left(\frac{3+\nu}{8}\right) \cdot \left(r_i^2 + r_o^2 + \frac{r_i^2 \cdot r_o^2}{r^2} - \frac{1+3\cdot\nu}{3+\nu} \cdot r^2\right)$$

$$\sigma_r(r) = \rho \cdot \omega^2 \cdot \left(\frac{3+\nu}{8}\right) \cdot \left(r_i^2 + r_o^2 - \frac{r_i^2 \cdot r_o^2}{r^2} - r^2\right)$$

These equations give the radial and tangential stress in a thin rotating ring.

$\sigma_t(r)$ = tangential stress in the ring as a function of radius, r

$\sigma_r(r)$ = radial stress in the ring as a function of radius, r

ν = Poisson's ratio for the material of the ring

ρ = density of the ring material

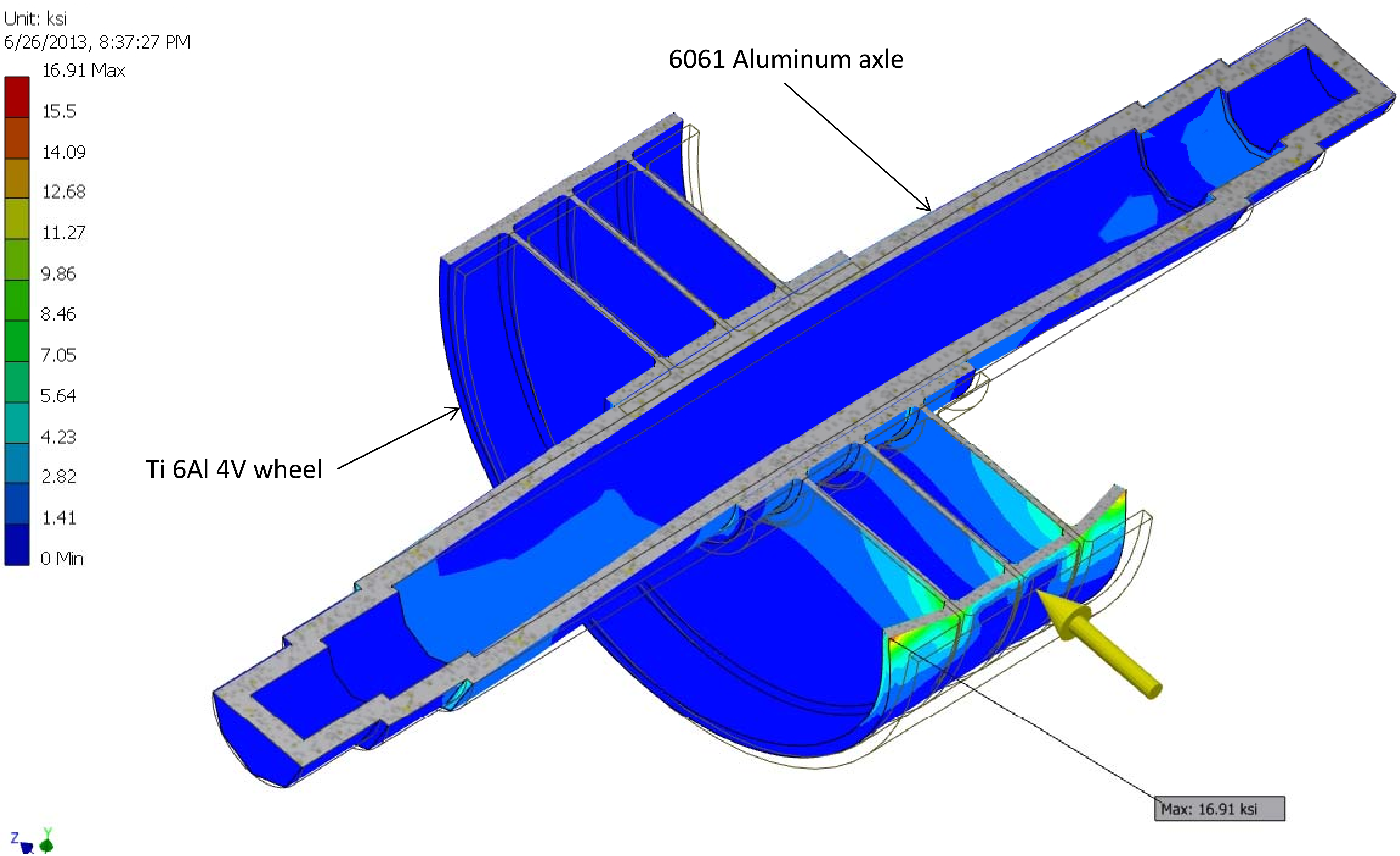
ω = rotational speed of the ring in radians per second

r_i = inner radius of ring

r_o = outer radius of ring

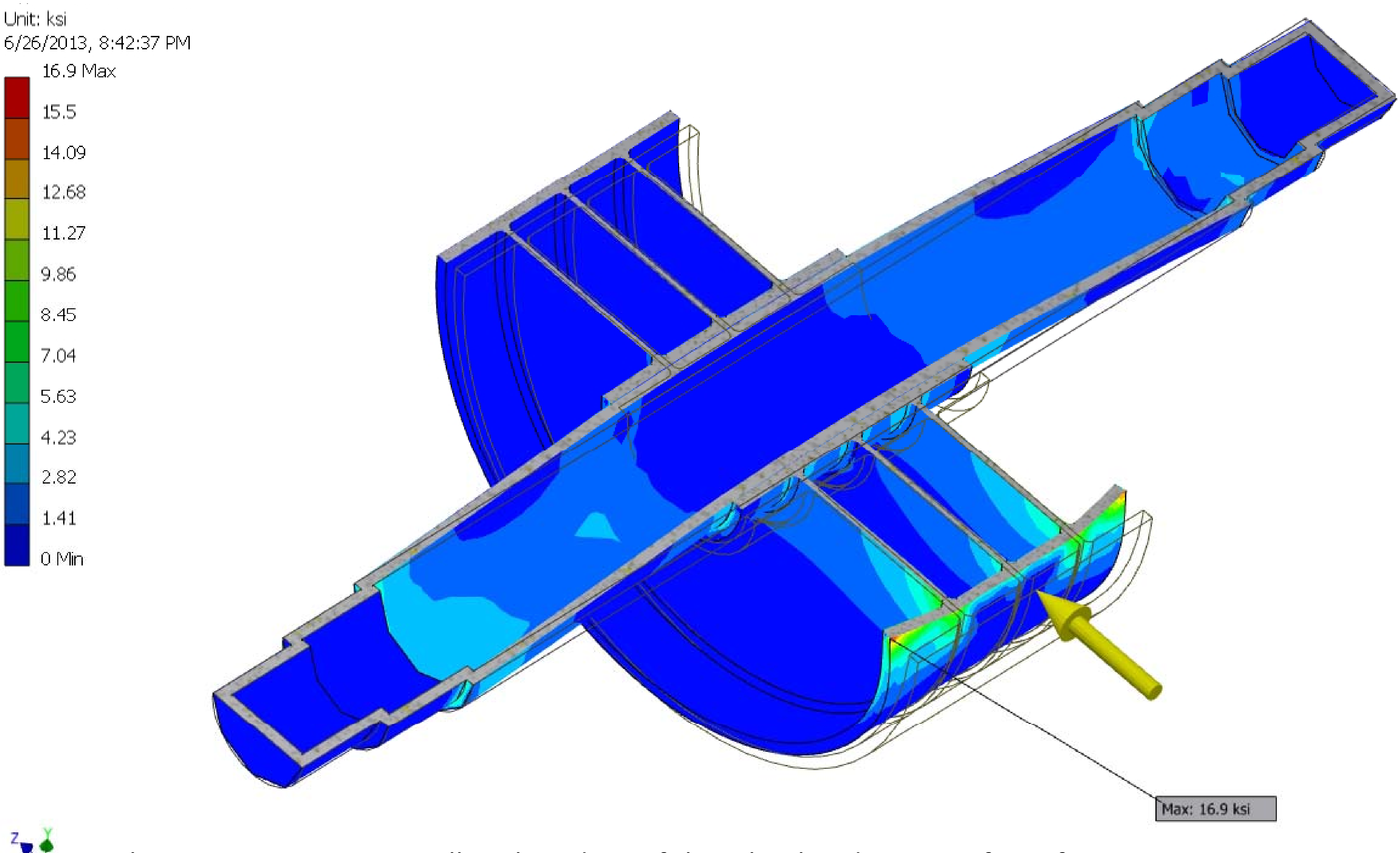
The red circles above highlight the squared rotational velocity terms.

FEA of 2004 wheel and axle design, compressive force of 3333 lbs only, no rotation



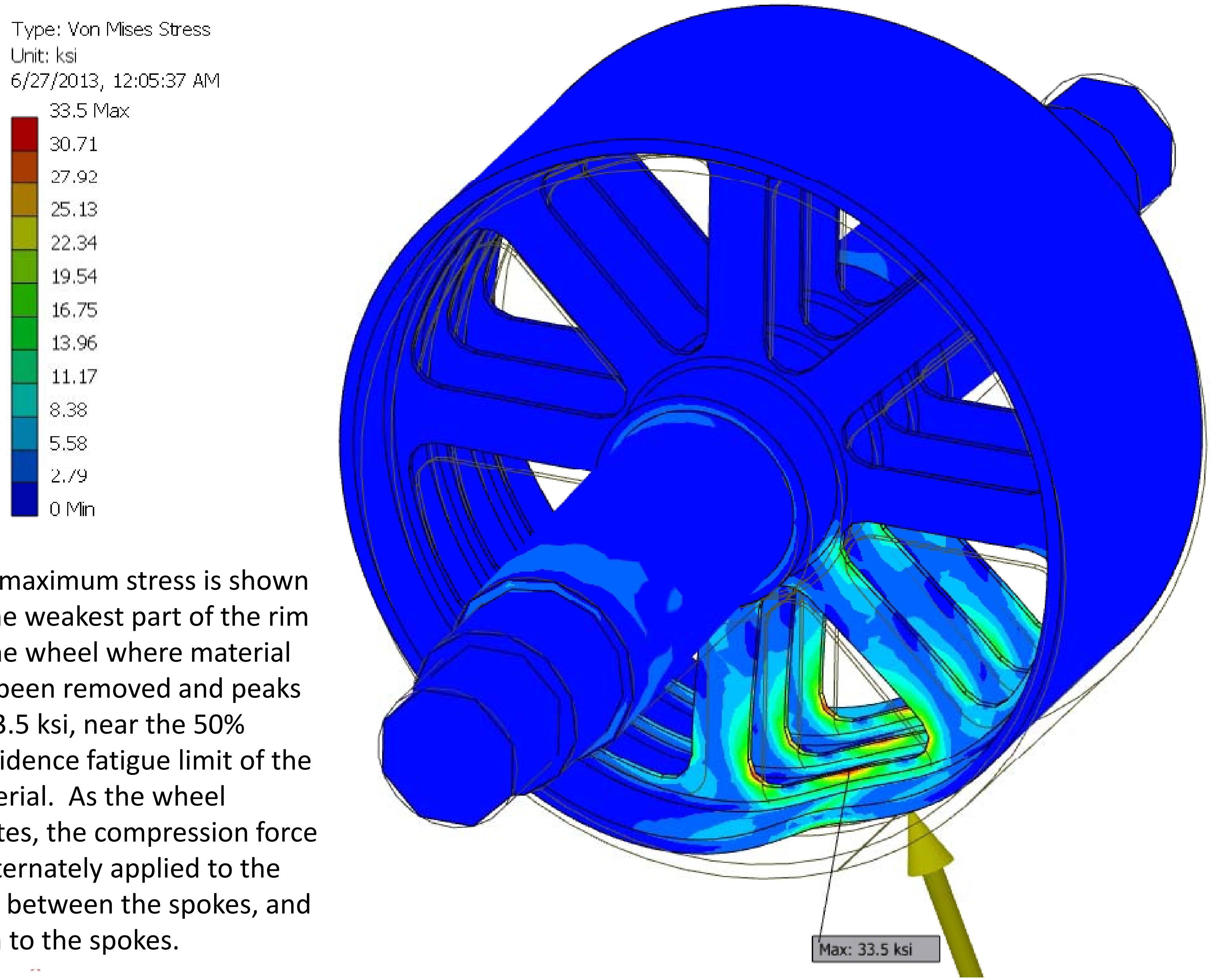
Axle is hollow with a 0.50" thick wall. The fatigue allowable for Al 6061-T6 is 6.5 ksi at 1.5E8 cycles of reversed bending.

FEA of thinned axle, 0.25" wall, compressive force of 3333 lbs only, no rotation



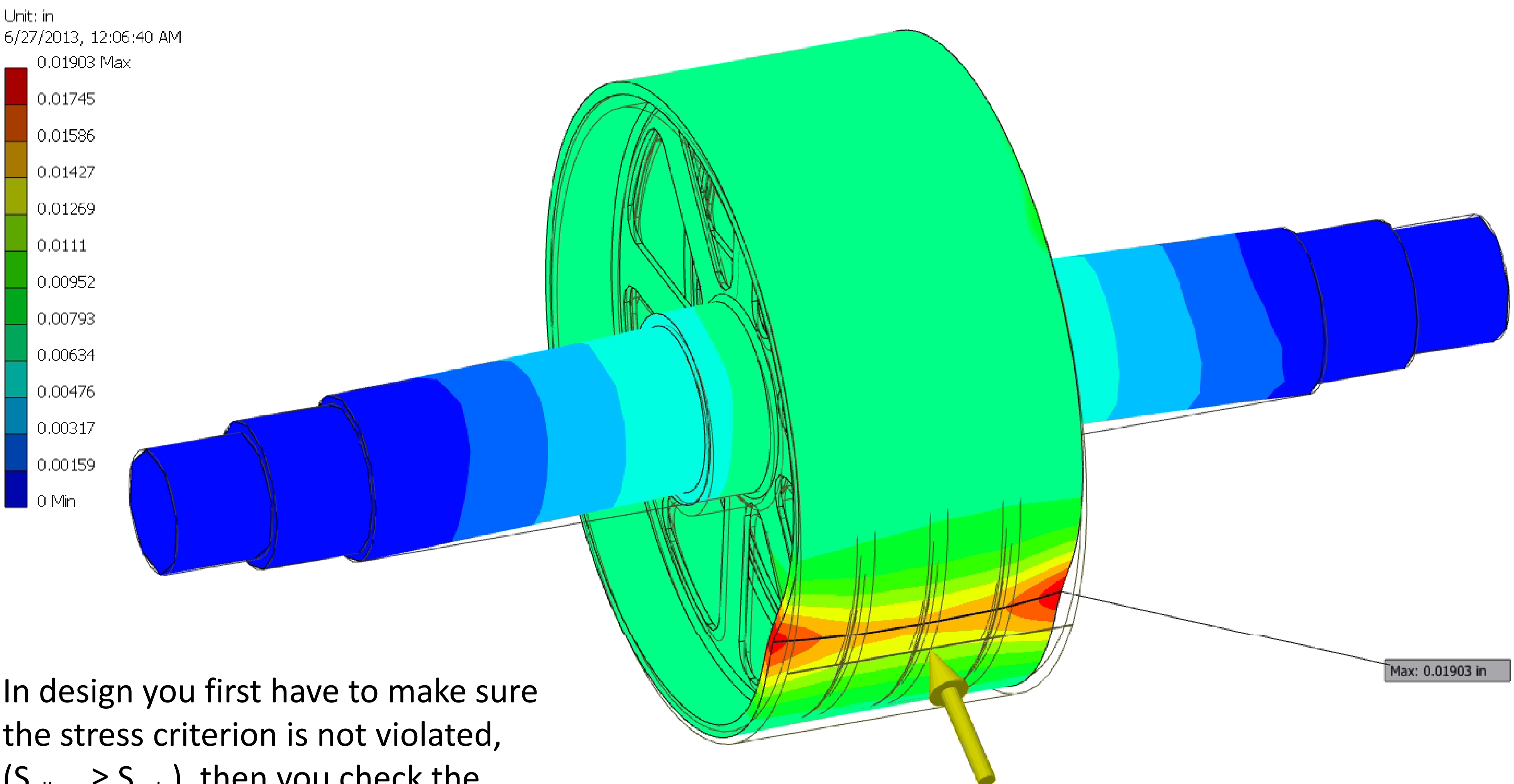
The maximum stress is still at the edges of the wheel and is an artifact of modeling. Stress in the axle has increased to about 5 ksi maximum near the bearings. Reducing the shaft wall more would violate the stress criterion.

Reducing the weight by cutting holes in the wheel web



The maximum stress is shown at the weakest part of the rim of the wheel where material has been removed and peaks at 33.5 ksi, near the 50% confidence fatigue limit of the material. As the wheel rotates, the compression force is alternately applied to the area between the spokes, and then to the spokes.

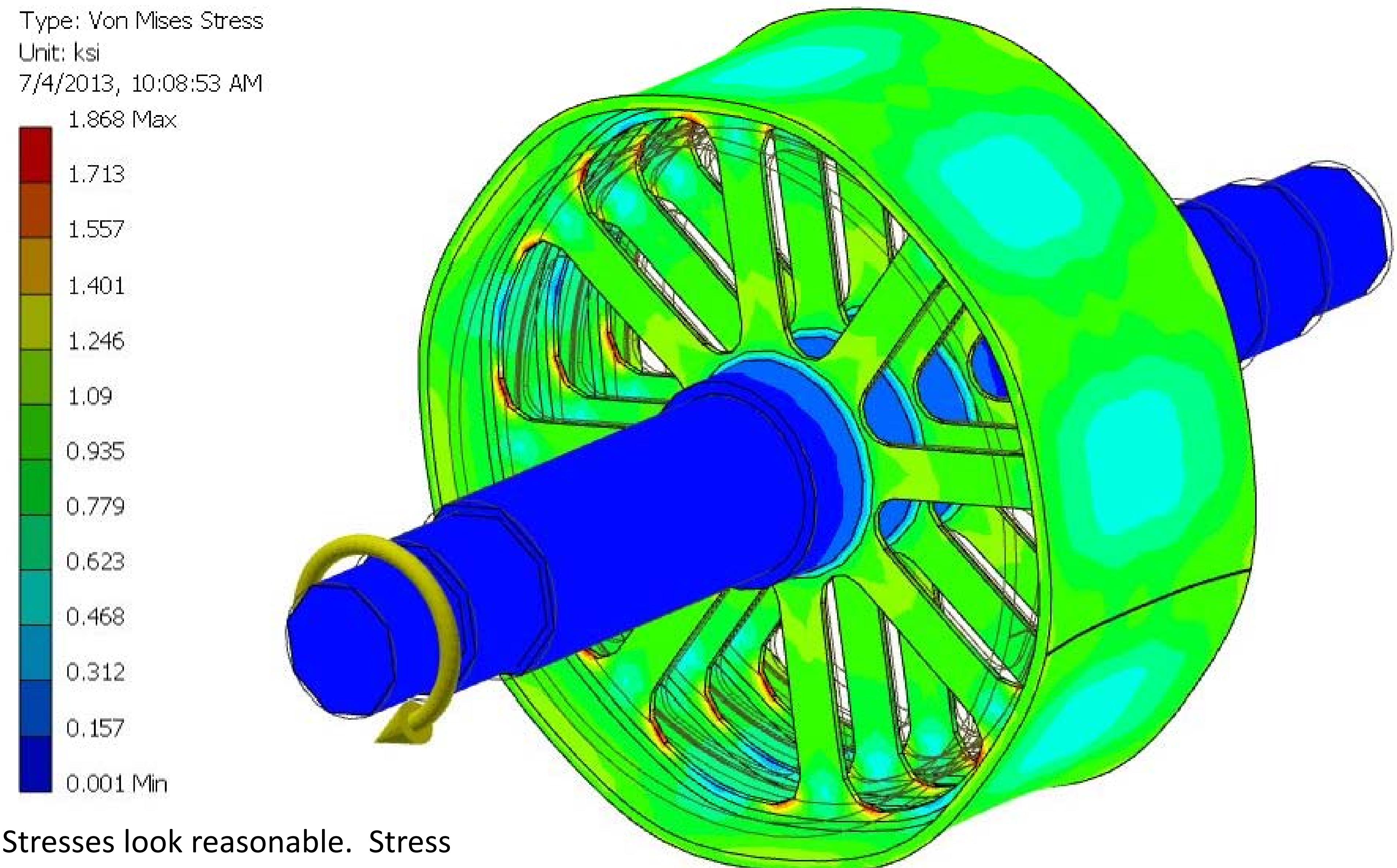
Deflection of the axle from the compressive load on the wheel



In design you first have to make sure the stress criterion is not violated, ($S_{allow} \geq S_{calc}$), then you check the deflection. Some designs are controlled by deflection instead of stress. Stress is low in such designs because the stiffness must be high.

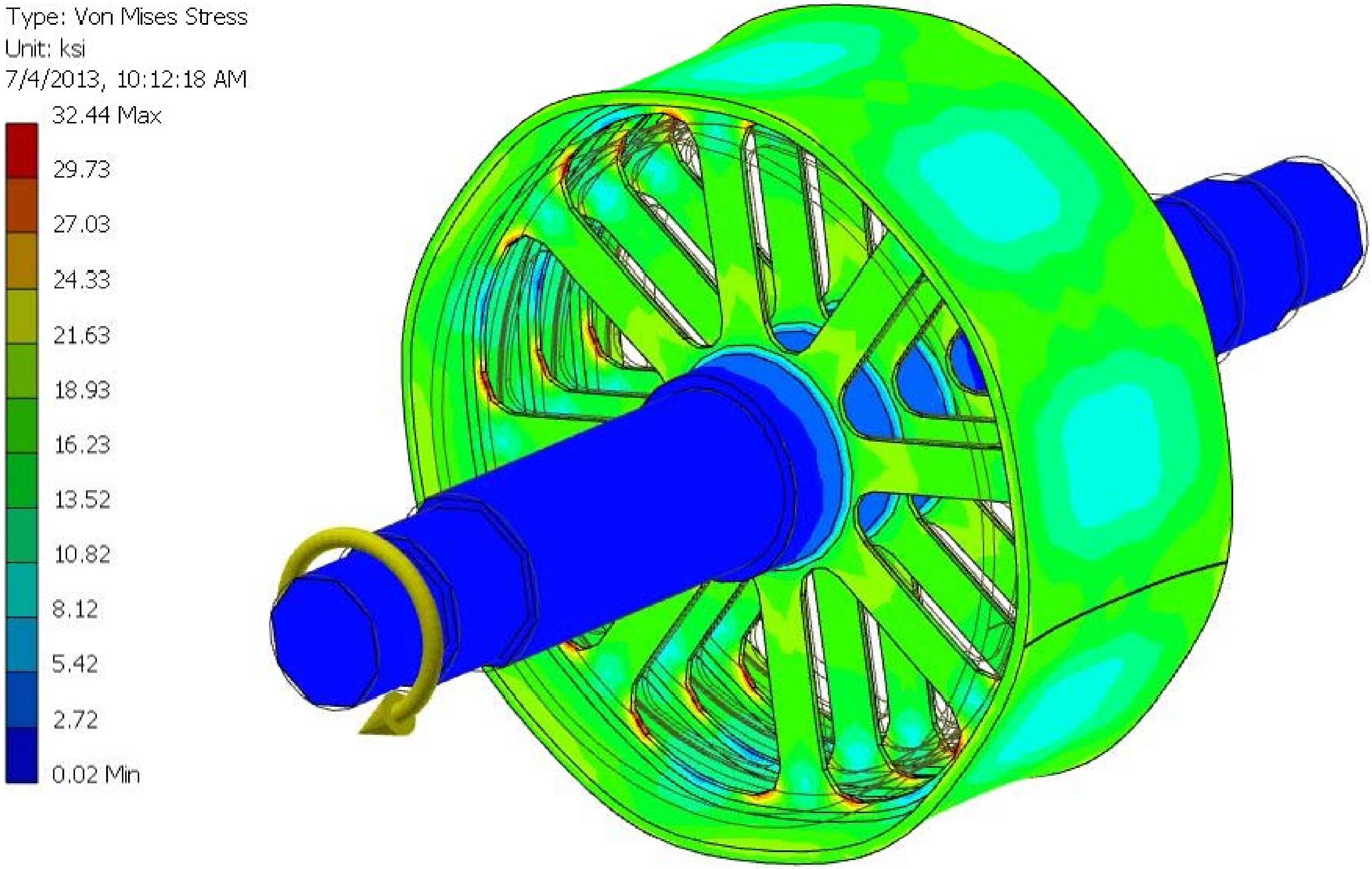
The green color of the wheel indicates the axle is bending by about 0.008 inches.

Von Mises stresses in the wheel and axle spinning at 2,400 RPM with no compressive load



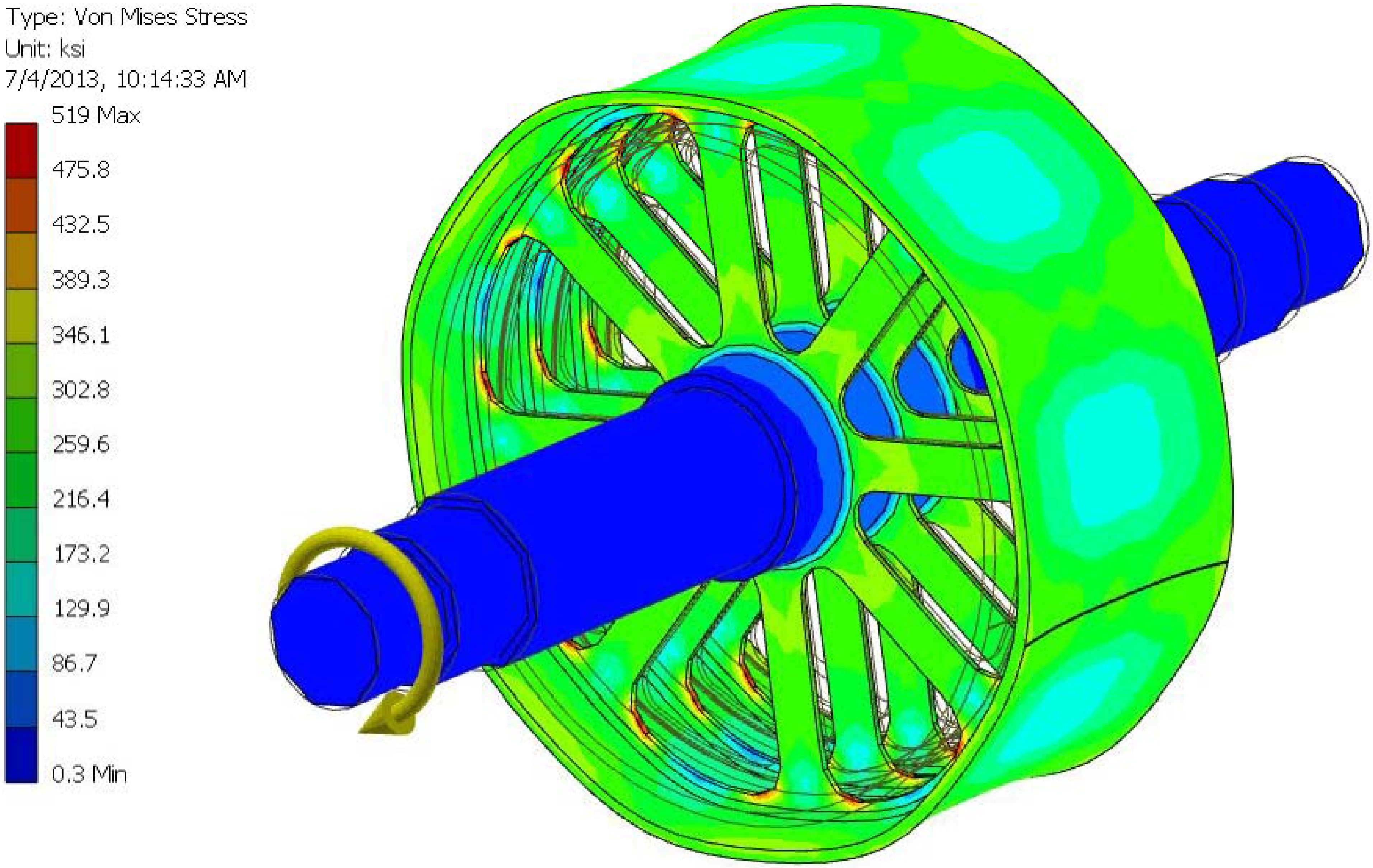
Stresses look reasonable. Stress concentrations in the web must be treated more carefully because of fatigue.

Plot of Von Mises stress for a wheel under no load spinning at 10,000 RPM



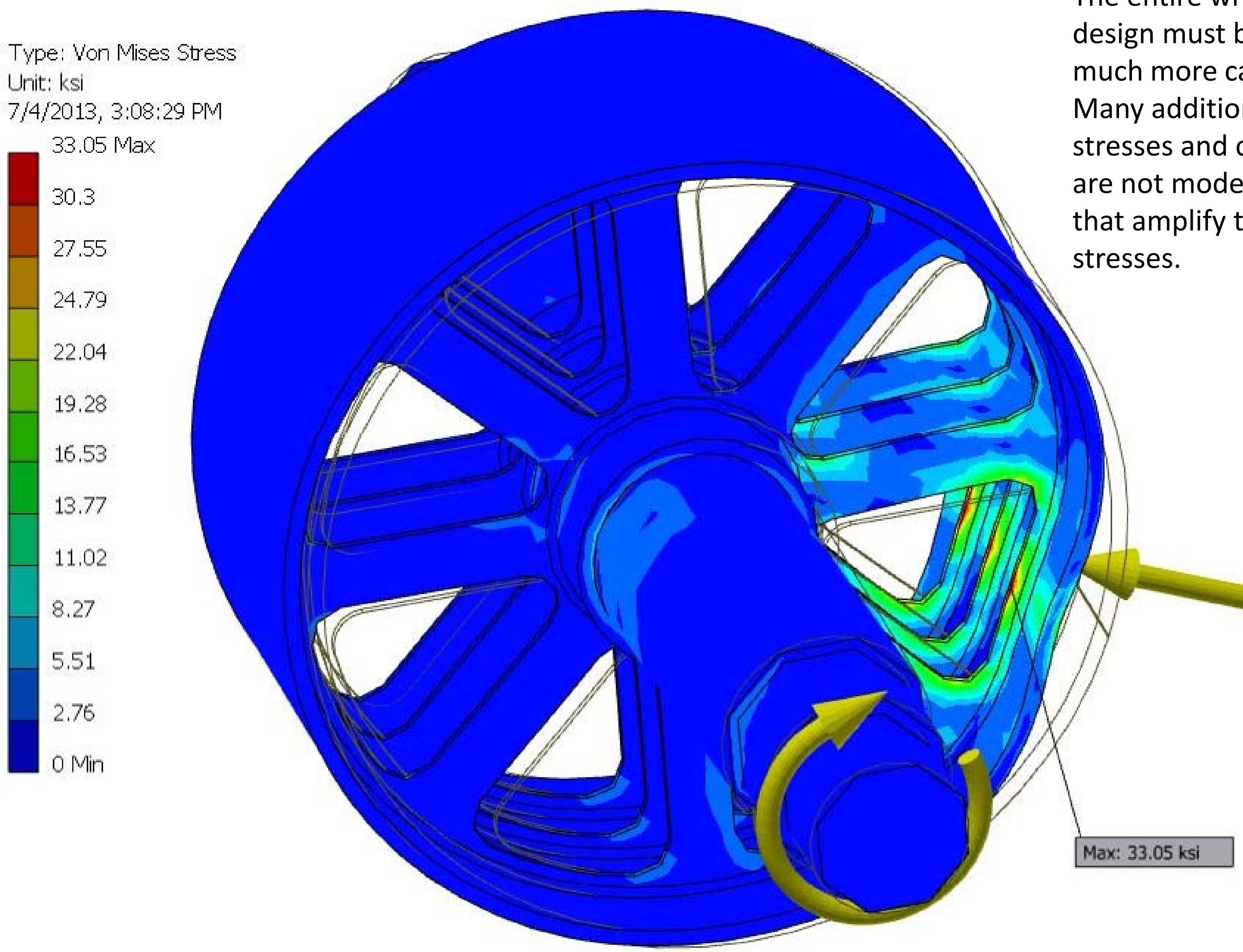
The maximum dynamic stress is still in the fillets of the web cutout but is now 32.44 ksi, close to the 50% confidence fatigue allowable for titanium. It is not known if it exceeds the 97.5% confidence level.

Von Mises stress for a wheel with no load spinning at 40,000 RPM



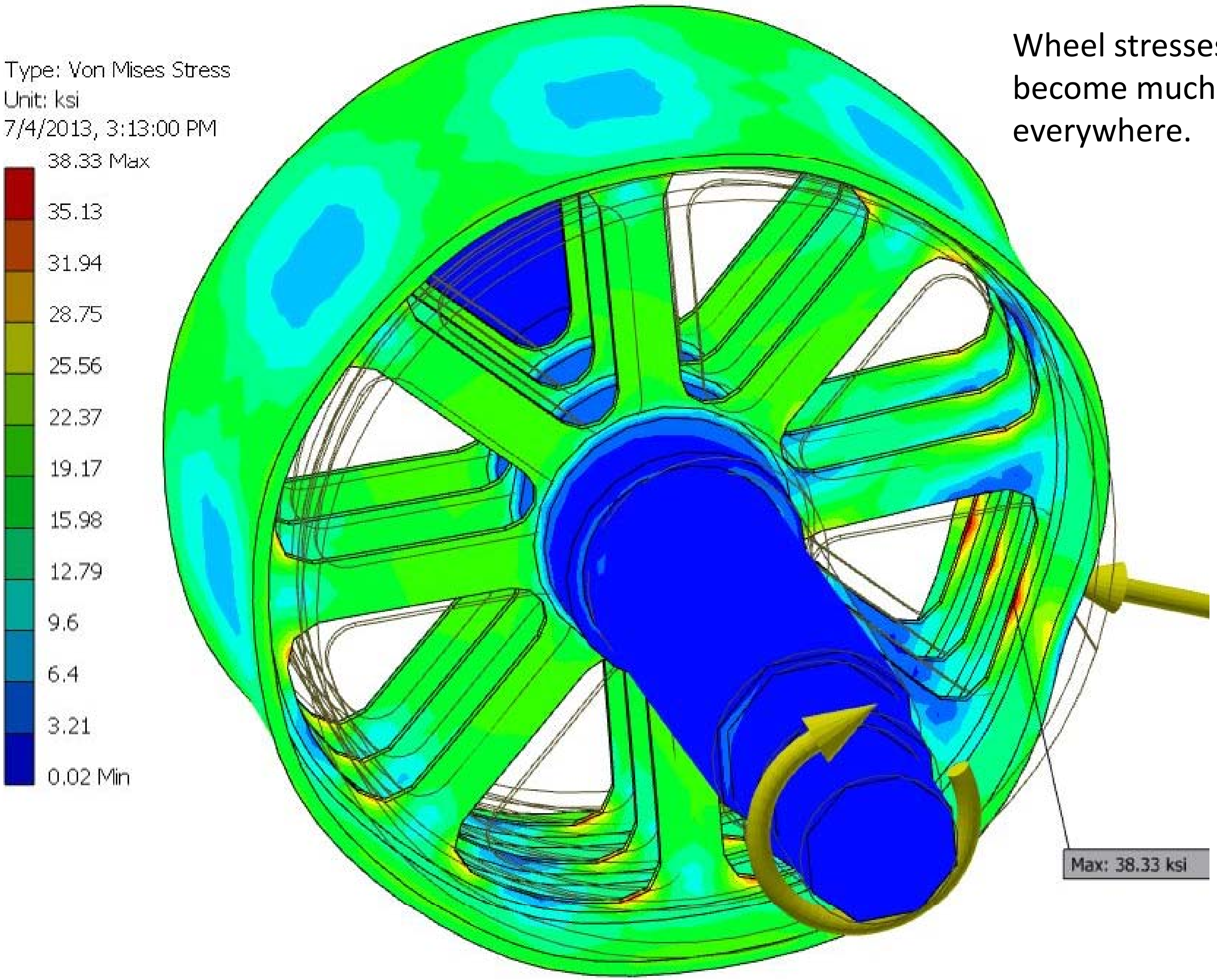
There is no engineering material that can handle the maximum stress here

Plot of Von Mises stress at 2,400 RPM with the compressive load of 3,333 lb



The entire wheel design must be done much more carefully. Many additional stresses and conditions are not modeled here that amplify the stresses.

Plot of Von Mises stress at 10,000 RPM with the compressive load of 3,333 lb



Wheel stresses have become much higher everywhere.

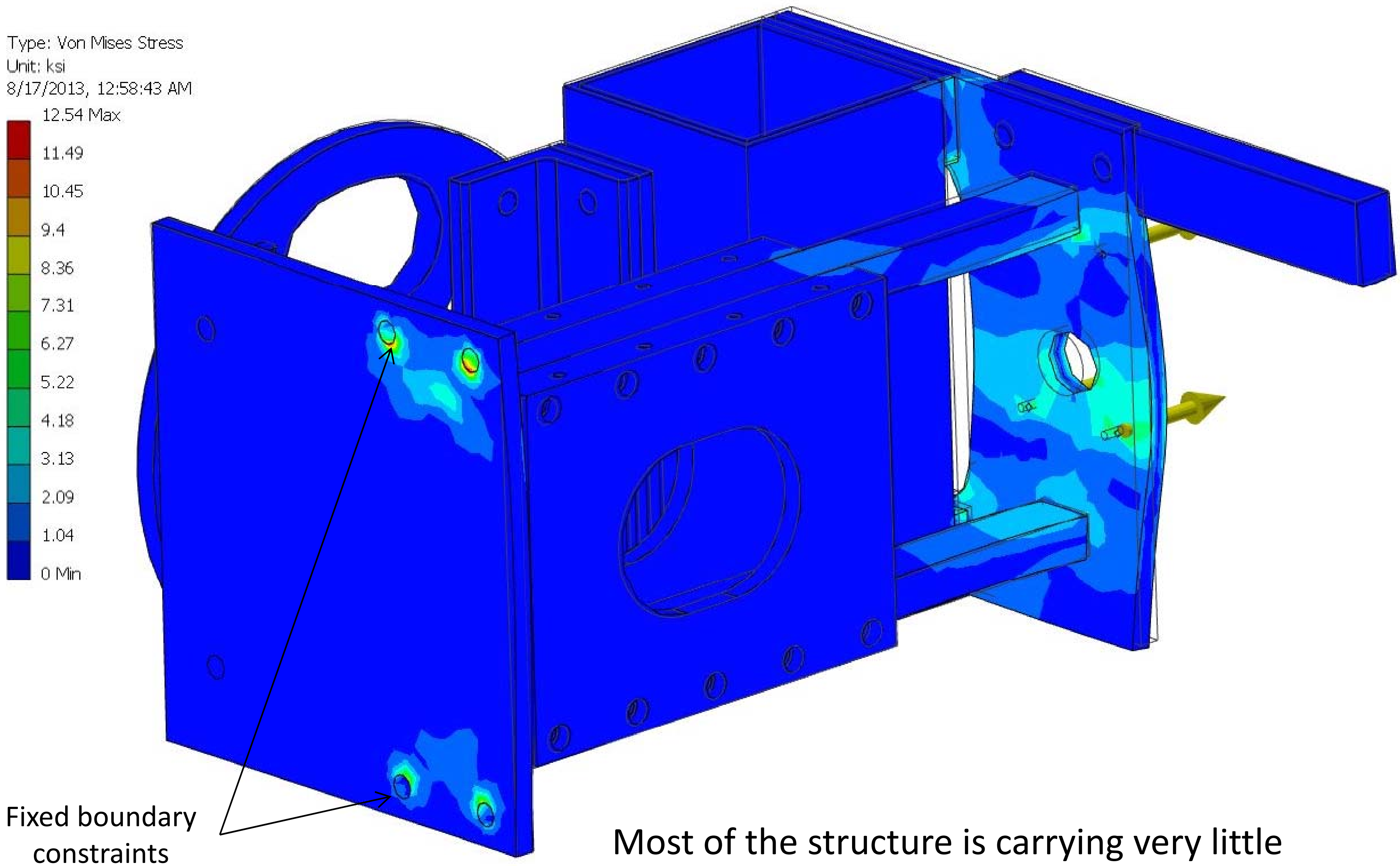
Conclusion from wheel FEA

- The 2004 design of the aluminum axle and titanium wheel weighed 31.0 lbs.
- The new lighter design weighs 21.9 lbs.
- This is a reduction of 29.4% on components that represented only 13.11% of the weight of the traction drive system.
- Even if I pushed very hard on trimming this mass it would not be enough.

FEA of the structure

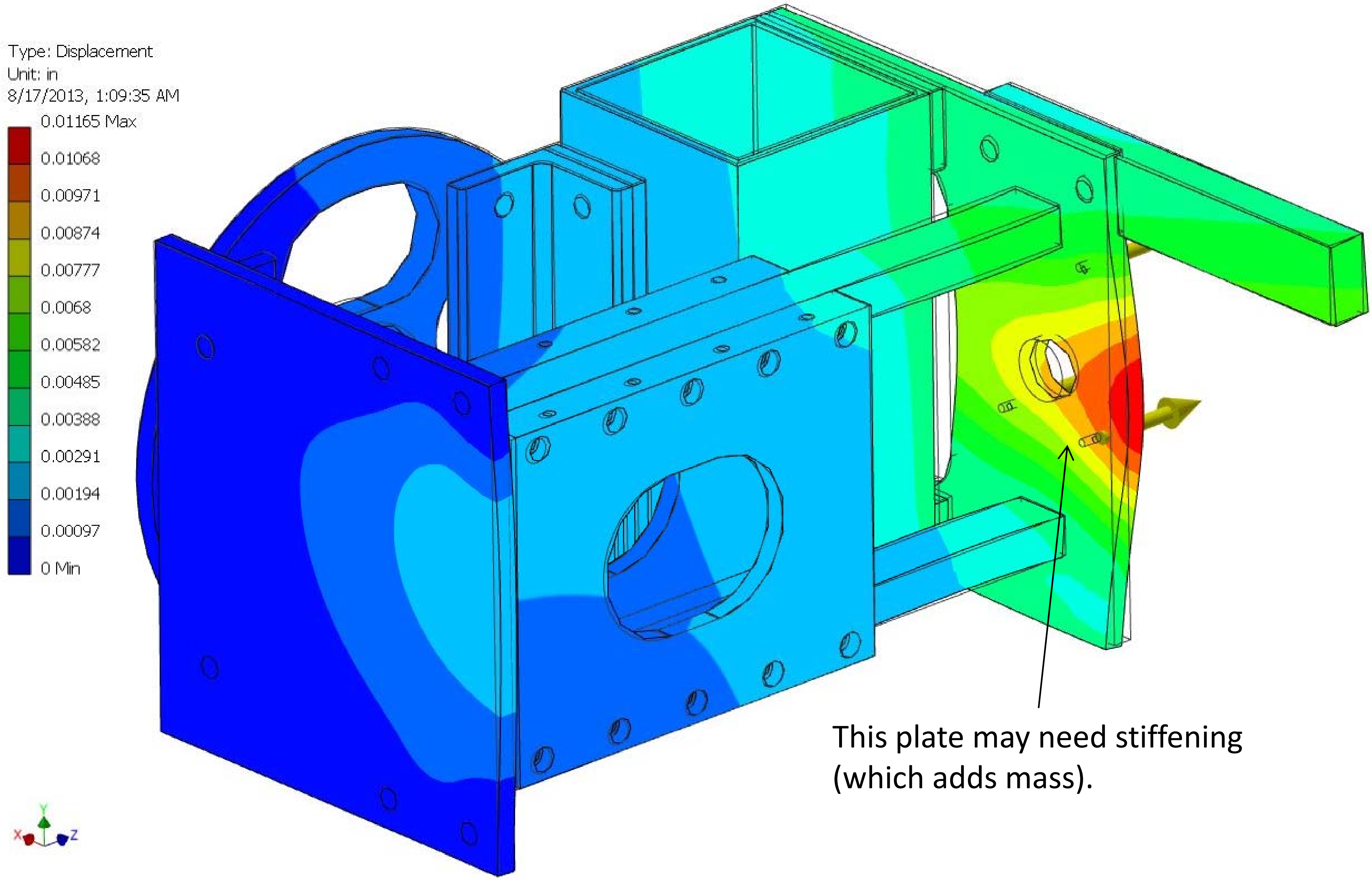
- I made structural members hollow that were solid in the 2004 design.
- I cut holes everywhere that the stress was low to increase the efficiency of the structure.
- It wasn’t enough and lots more work needs to be done.

Von Mises stress in half the floating axle module with 1 ton of tension from the screw jack



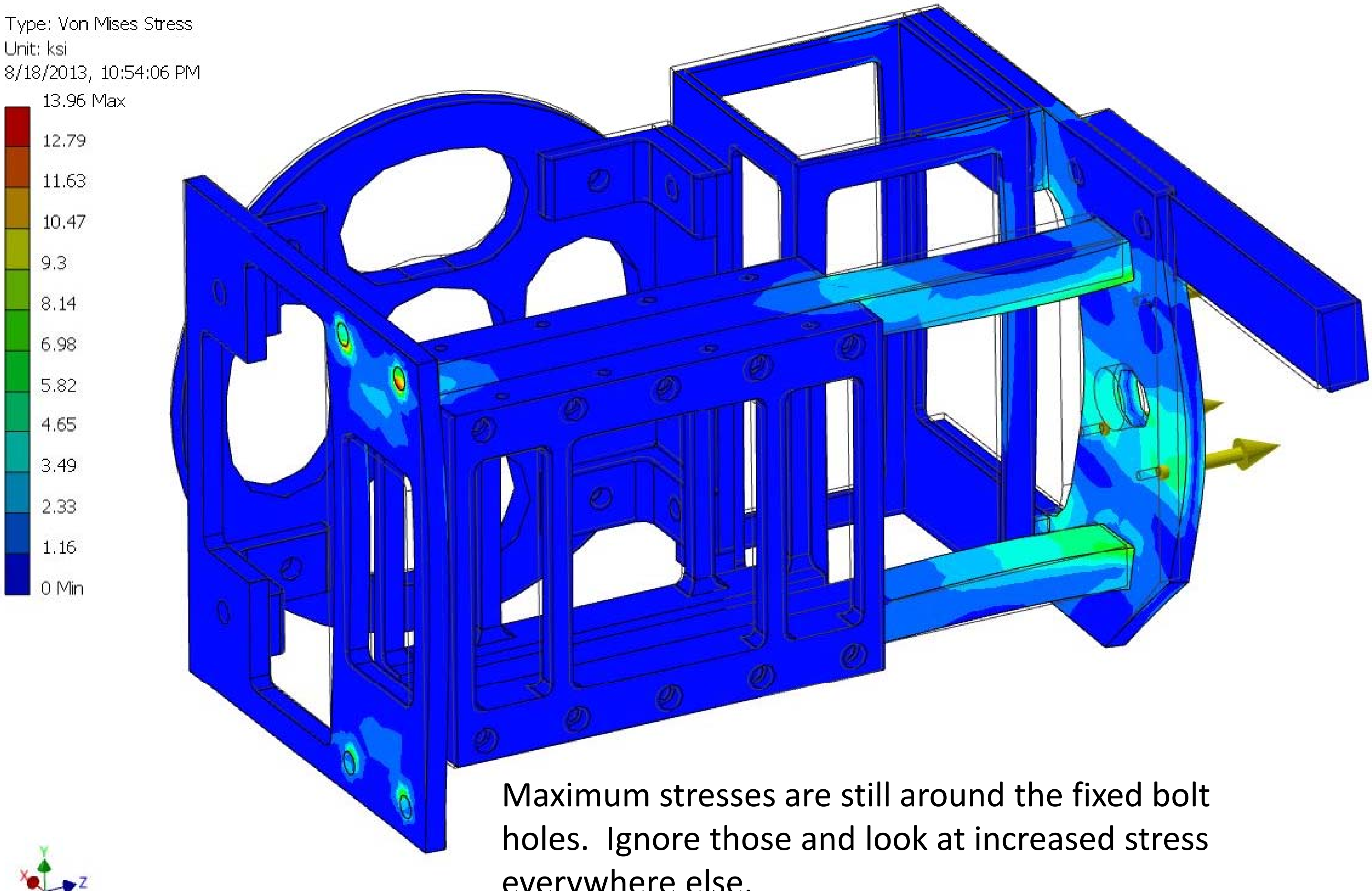
Most of the structure is carrying very little load. The only two forces on the structure are gravity and the wheel compression load.

Deflection in half the floating axle module with 1 ton of tension from the screw jack



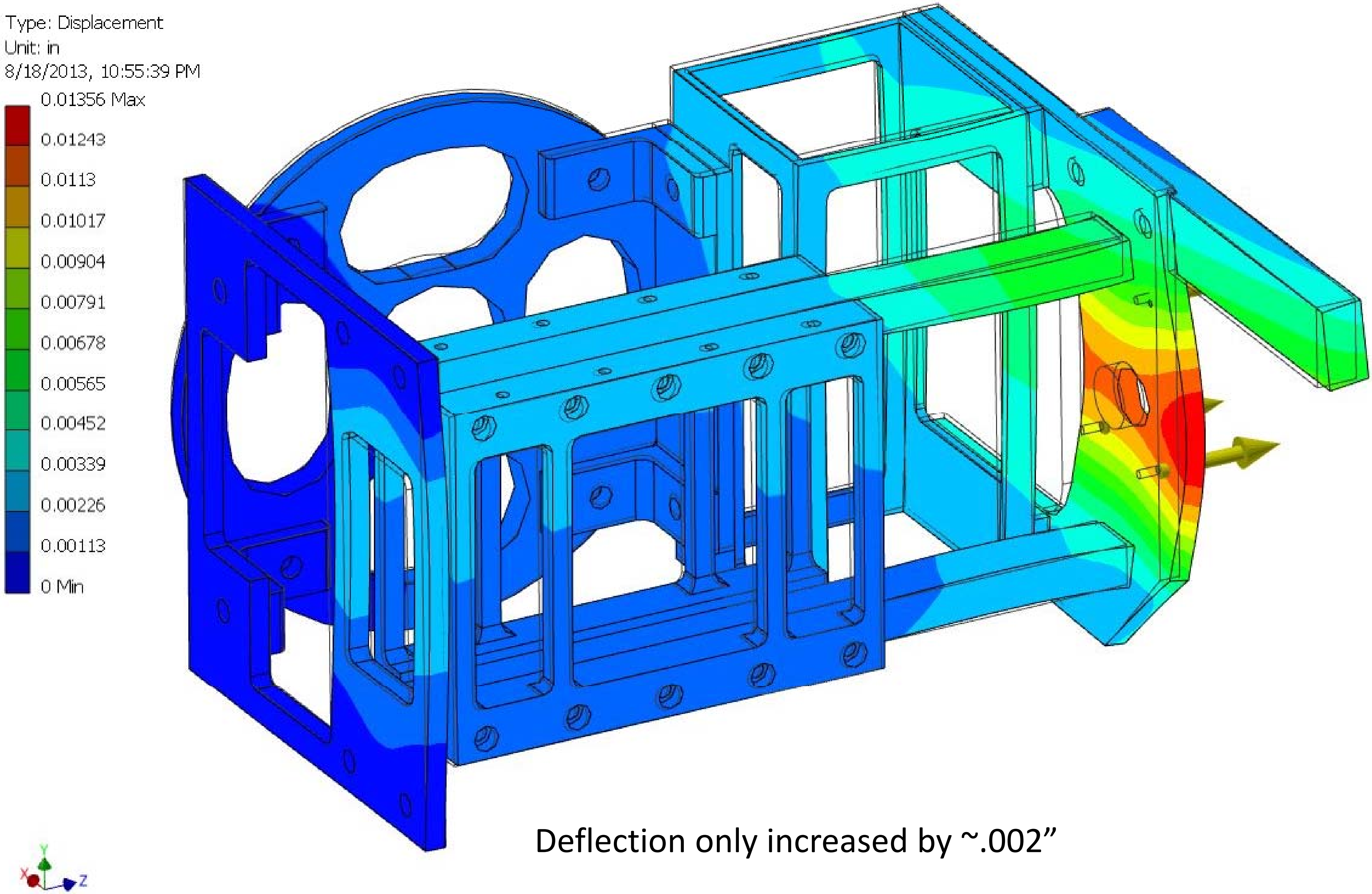
This plate may need stiffening (which adds mass).

Von Mises stress in the stripped down structure, same load as before



Maximum stresses are still around the fixed bolt holes. Ignore those and look at increased stress everywhere else.

Deflection in the stripped down structure, same load as before



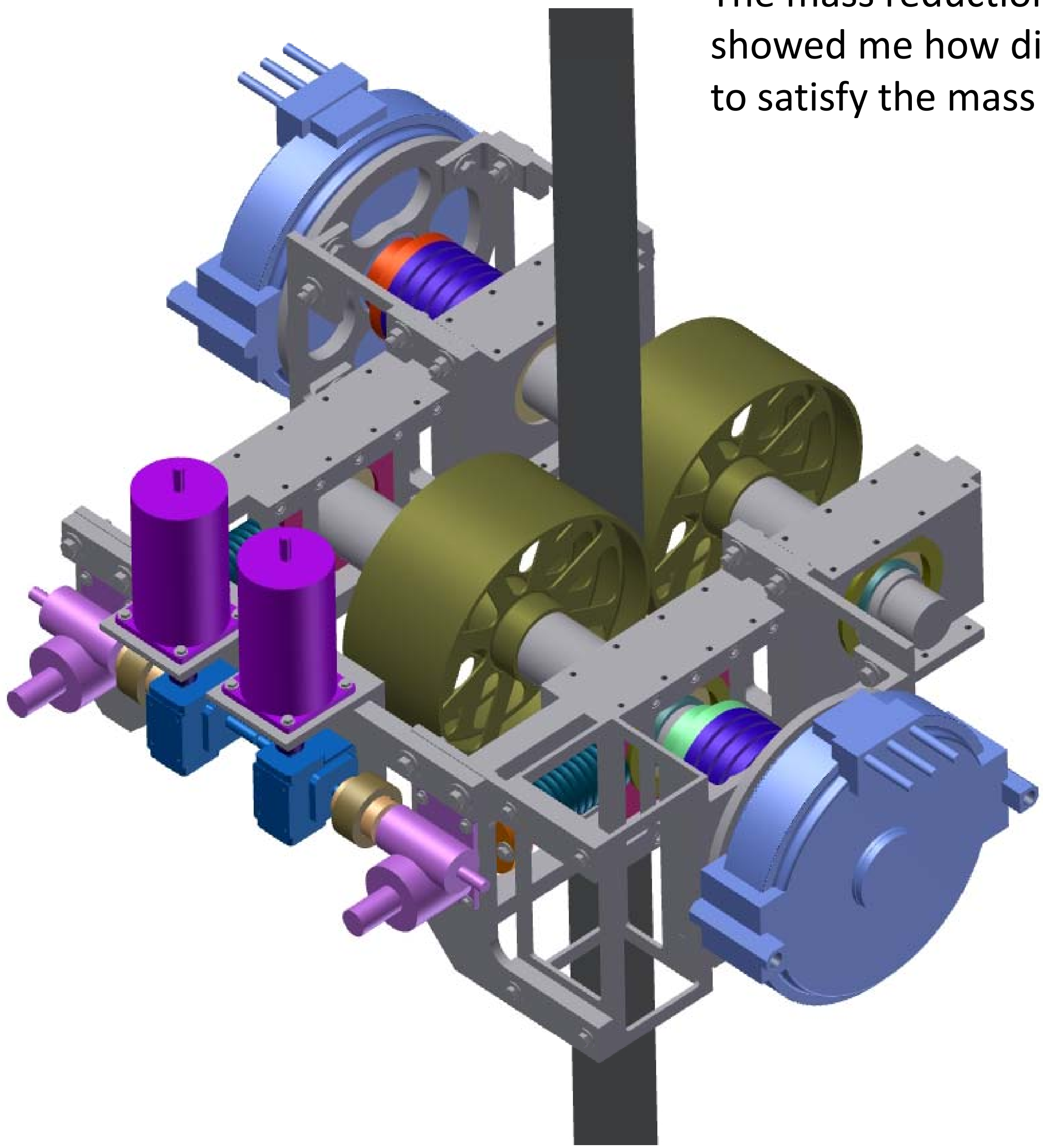
Deflection only increased by ~.002”

Some conclusions from structure FEA

- The mass of the 2004 floating axle module structure was 38.8 kg
- The reduced mass is 24.8 kg
- The percent reduced is -36.1%
  - A mass reduction ratio of 1.56
- I needed a reduction factor of 5.3!
- The non-structural elements of the module (gear boxes, shaft couplings, etc) need reducing too
- An important component was left out of the 2004 design: BRAKES
  - This will add more mass

Rendering of a reduced-mass wheel pair

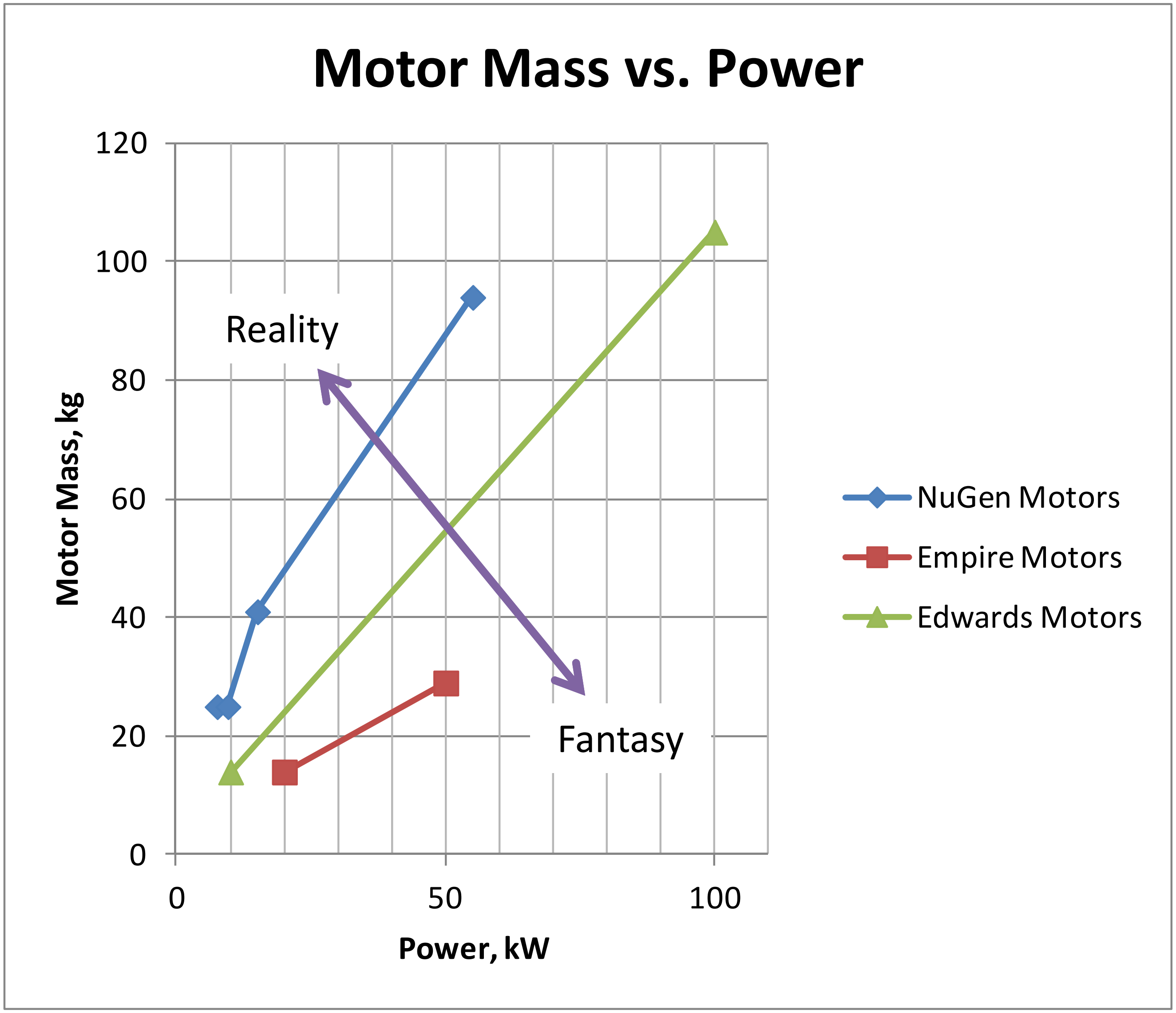
The mass reduction exercise showed me how difficult it will be to satisfy the mass budget



The Axial Gap Motor Problem

- Edwards and Westling concluded that axial gap motors were the most efficient motor for the climber
- They will have to be custom designs
- There are no good commercial examples with the right characteristics to get a mass baseline
- The controller is an integral part of the motor and can greatly modify the motor’s behavior
- They are inherently large in diameter, so hard to increase the top speed because of dynamic stress

Graph of Motor Mass vs Power for several different sources of Axial Gap motors



The NuGen motors shown are for electric vehicle applications on Earth. They are available, but are the heaviest of the motors considered. (The two equal mass but differently powered points on the graph show the effect of changing the motor's voltage.)

The middle curve is the estimate from Edwards and Westling.

The bottom curve may be an over-optimistic estimate based on insufficient design effort.

Without real motors it is impossible to know whether or not they satisfy the mass budget.

Climber Center of Mass Issues

- This picture shows a model climber from the 2011 toy climber competition
- The Center of Mass is off to one side of the climber causing the climber to rotate and distort the ribbon
- If real climbers are not balanced around the ribbon, the ribbon will be subjected to local higher tensile stress which reduces the safety margin of the ribbon
- Also, the Center of Mass of the climber must be below the traction drive when the climber is below GEO. Otherwise, the climber is metastable and can try to flip 180°
- It is not clear if the CM must change above GEO to avoid this condition

Conclusions and final questions

- It will take a lot of work to design light enough components to satisfy the mass budget
  - It probably doesn’t make sense for me to go any further without real spaceworthy components to design with
- Is the 230 construction climber/2year/ribbon augmentation reasonable?
  - Could the money and time be better spent developing a heavy launch vehicle to send up a heavier ribbon?
- Will the mass of components increase too quickly with load capacity making larger climbers even harder to satisfy their mass budgets?

Acknowledgements

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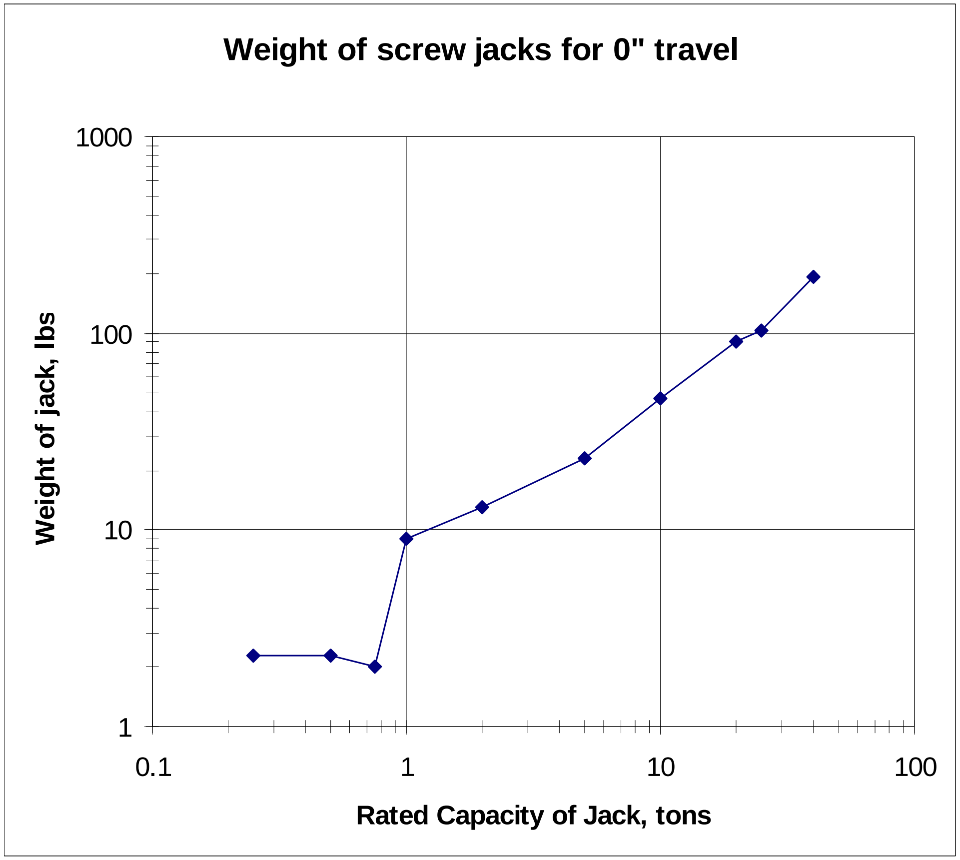
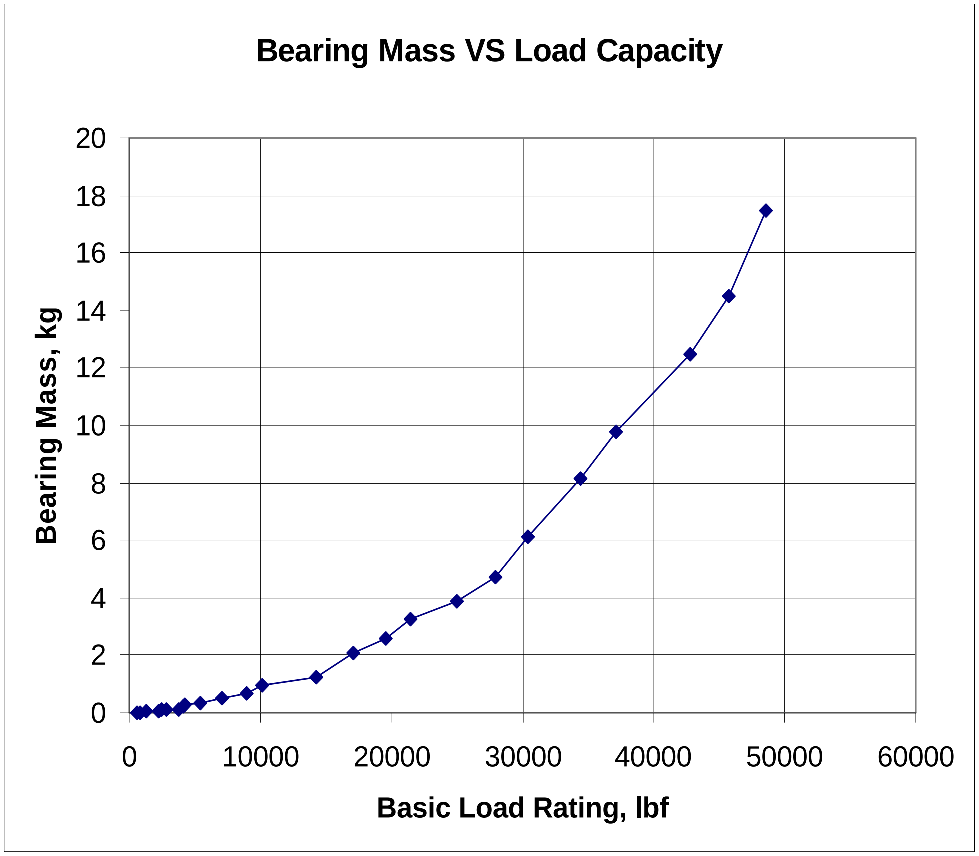
BACKUP SLIDES

How components scale with capacity

Templeton-Kenly Uni-Lift Screw Jacks



SKF Self-Aligning Ball Bearings



The implication of these graphs is that there is a “threshold” mass for components at the low end of capacity and that mass increases rapidly with capacity