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The Space Elevator Ribbon and Climber From a Machine Design Perspective

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Abstract. This paper will look at the Space Elevator ribbon and the ribbon climber from a machine design perspective. In particular, the difficulties encountered in a conceptual design for the first construction climber will be discussed in the context of the space elevator construction scenario as laid out in *The Space Elevator–A revolutionary Earth-to-space transportation system* by Edwards and Westling. Many crucial pieces of data necessary to make a complete design of a space elevator system are not known at this time. This paper will present a conceptual design of the traction drive of the first construction climber and discuss the following items of necessary further research. First, the effect of the coefficient of friction between the elevator ribbon and the climber's traction drive. Second is the lack of a commercially available axial gap motor in the 20 kW and up range. Third is the difficulty of designing within the mass budget laid out in Edwards' and Westling's book.

1. Introduction

The design described in Edwards' and Westling's book strives to be the cheapest and most practical way to construct a space elevator the moment a strong enough carbon nanotube fabric can be made. The construction of the elevator uses the absolute minimum number of Space Shuttle flights to deploy a pilot ribbon 100,000 km long. Once the pilot ribbon is anchored to the base station on the equator, 230 construction climbers ascend the pilot ribbon as fast as they can. Each construction climber adds additional ribbon material to the pilot ribbon increasing the strength of the ribbon by 1.3% with each pass. Each construction climber ends its days at the end of the ribbon as counterweight for the additional ribbon it added. At the end of two years of construction, the ribbon is strong enough to support a 20 ton climber with a 13 ton payload, and the elevator is ready to enter the space cargo business.

The size of the pilot ribbon was determined by the maximum size of 100,000 km long spools of fabric that could still fit in the Shuttle's cargo bay. Because of this limitation, the cross-section of the pilot ribbon is about six inches wide, less than .001 inches thick, and only strong enough to support a 900 kg climber, including the 520 kg load of the next spool of CNT fabric. That leaves only 380 kg to fit a traction drive, connecting structure, controls, heat dissipation system, and a photovoltaic array.

In the summer of 2004, this author attempted to create a rigorous design for the first construction climber. The design focused on the traction drive. The presentation of the design for the Third International Space Elevator Conference can be found at http://www.isr.us/spaceelevatorconference/2004presentations.html.

The process of doing a rigorous mechanical design uncovered several critical areas where basic and applied research needs to be done to have enough information to complete the design. Before discussing these research areas, the features of the climber will be presented.

2. Summary of the Design Features of the Proposed Construction Climber

The basic design of the construction climber traction drive is sets of wheels on each side of the ribbon being squeezed together with the ribbon in between them. There is no belt or track in this design as shown in the climbers in Edwards' and Westling's book. Analysis showed that a track design would create more problems than it would solve and that simpler wheel pairs should be lighter and easier to accelerate. The climber is separated into modules that bolt together around the ribbon. There are two different module designs. The axle of the wheel in one module design is fixed to the climber's structure (free to rotate, but not to move off-axis with respect to the driving motor shaft.) The wheel/axle in the other module design floats horizontally to allow it to be compressed against the fixed wheel. The compression mechanism is a screw jack compressing a belleville spring stack to achieve greater resolution in the applied force. The components shown in Figure 1 are sized for the appropriate loads, but they are not space-worthy motors and gear boxes. They were sized to begin to understand the mass budget of the climber. One concern is that actual space-worthy components may be larger and heavier than the ones shown.

Each module contains one wheel and each wheel is motorized. Pairs of fixed and floating wheel modules bolt around the ribbon and then the pairs are attached to each other through structural modules. The structural modules are designed to transfer any load from the rest of the climber to the axles of the wheels, the shortest load path to the ribbon. The floating wheels are designed to have two independent compression mechanisms on opposite sides of the wheel. This allows differential squeezing of the wheel pair to control drifting of the climber across the ribbon. A small amount of differential pressure will cause the wheels to deform from cylindrical to conical shapes. This will be part of an active feed-back loop that maintains the centering of the climber during ascent.

3. The Coefficient of Friction Between the Ribbon and the Climber Wheels

The exact configuration of the carbon nanotube fabric that the ribbon is made from is not known. Currently, the technology to make CNT fibers 100,000 km long does not exist. Edwards and Westling propose a taping technique to connect long strands of CNT fibers into a ribbon that looks promising once the fibers are available. The lack of a real ribbon composite fabric means that the coefficient of friction between the wheels of the climber and the ribbon cannot be known. To get traction to keep the climber from sliding down the ribbon, the wheels on opposite sides of the ribbon must pinch together so that there is a normal force on the ribbon.

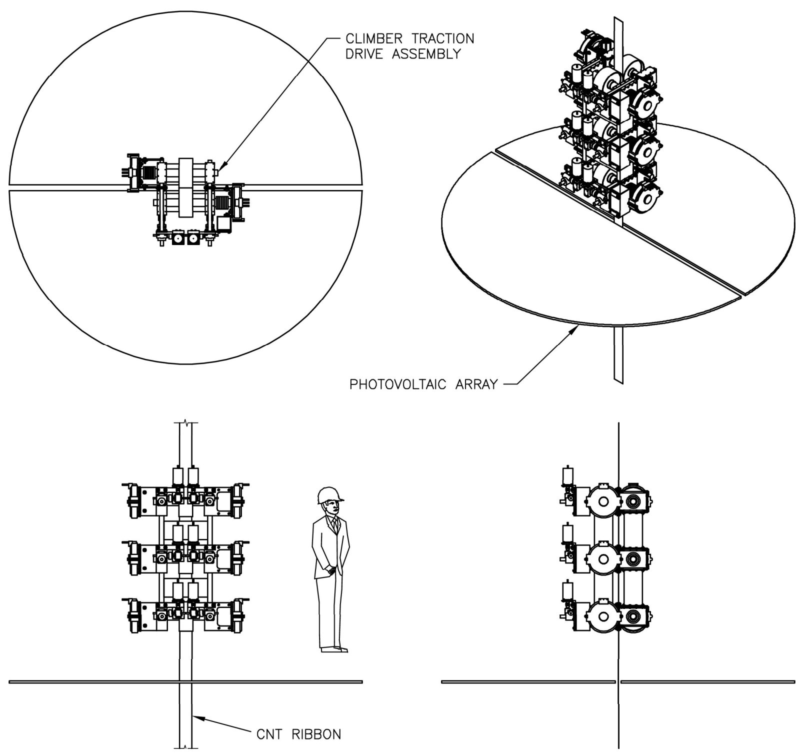

Figure 1. This is a conceptual design of the first construction climber proposed by Bartoszek Engineering. The pilot space elevator ribbon is only about six inches wide at the base station. As shown in the following pictures, the climber must be assembled in halves around the ribbon. The design focuses on elements of the traction drive mechanism. Many necessary sub-systems of the final climber are not shown here and have not been designed yet. The structure as discussed in the section on the mass budget refers only to the material shown in this picture. For example, no structure is shown connecting the PV array to the drive train and no estimate was made of the mass of that undesigned structure.

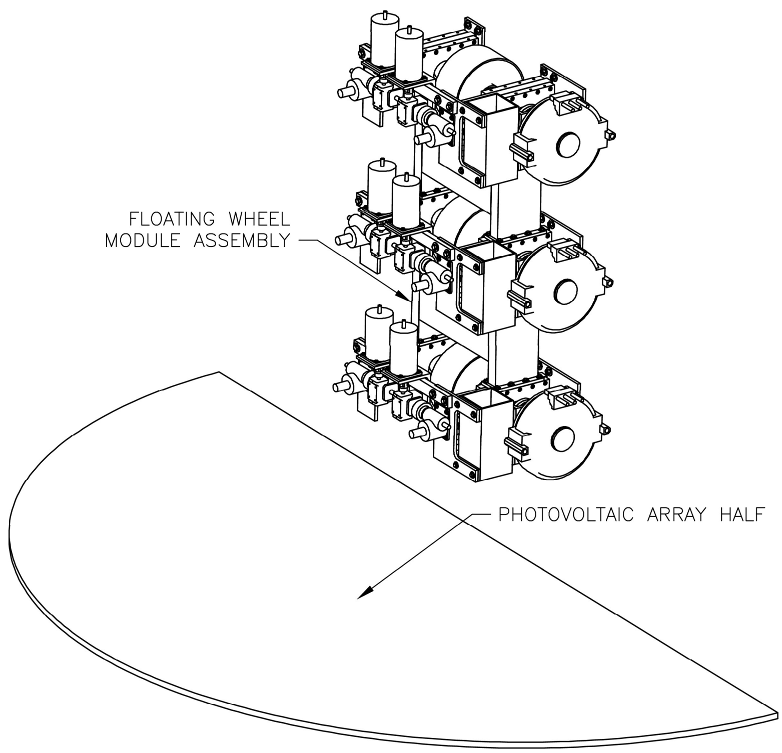

Figure 2. The space elevator ribbon poses unique assembly challenges because it effectively has no ends. The climber must be assembled in halves on opposite sides of the ribbon and the halves bolted together. This picture shows what the floating wheel module assemblies on one side of the ribbon would look like. No structure is shown connecting the traction drive to the PV array.

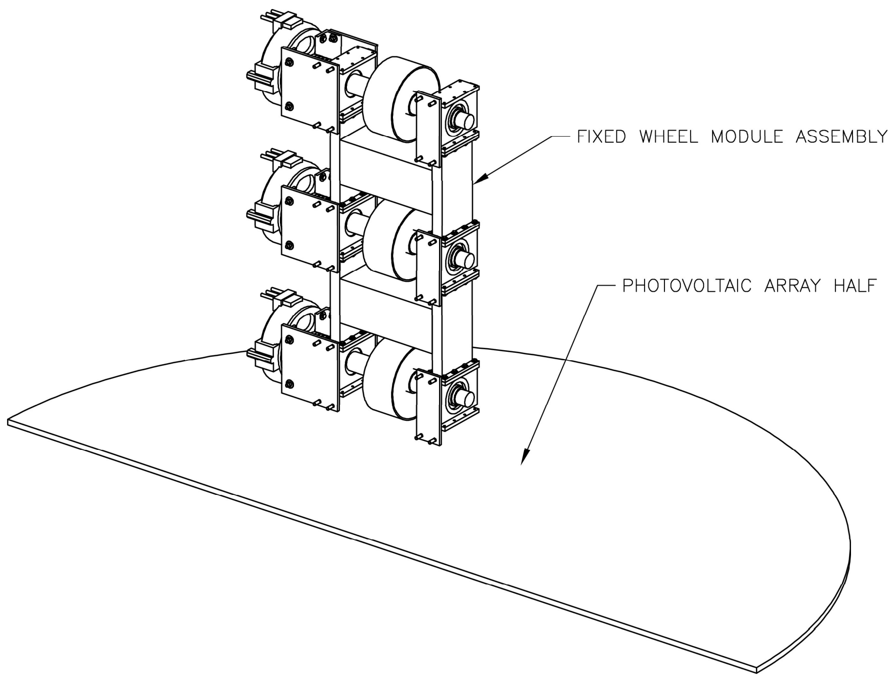

Figure 3. This picture shows the assembly of the fixed wheel modules. The bearings that support the axle are mounted in blocks that are fixed to the climber structure. Without the structural interface modules, the bearing assemblies on opposite sides of the wheel are free to rotate with respect to each other in both styles of modules.

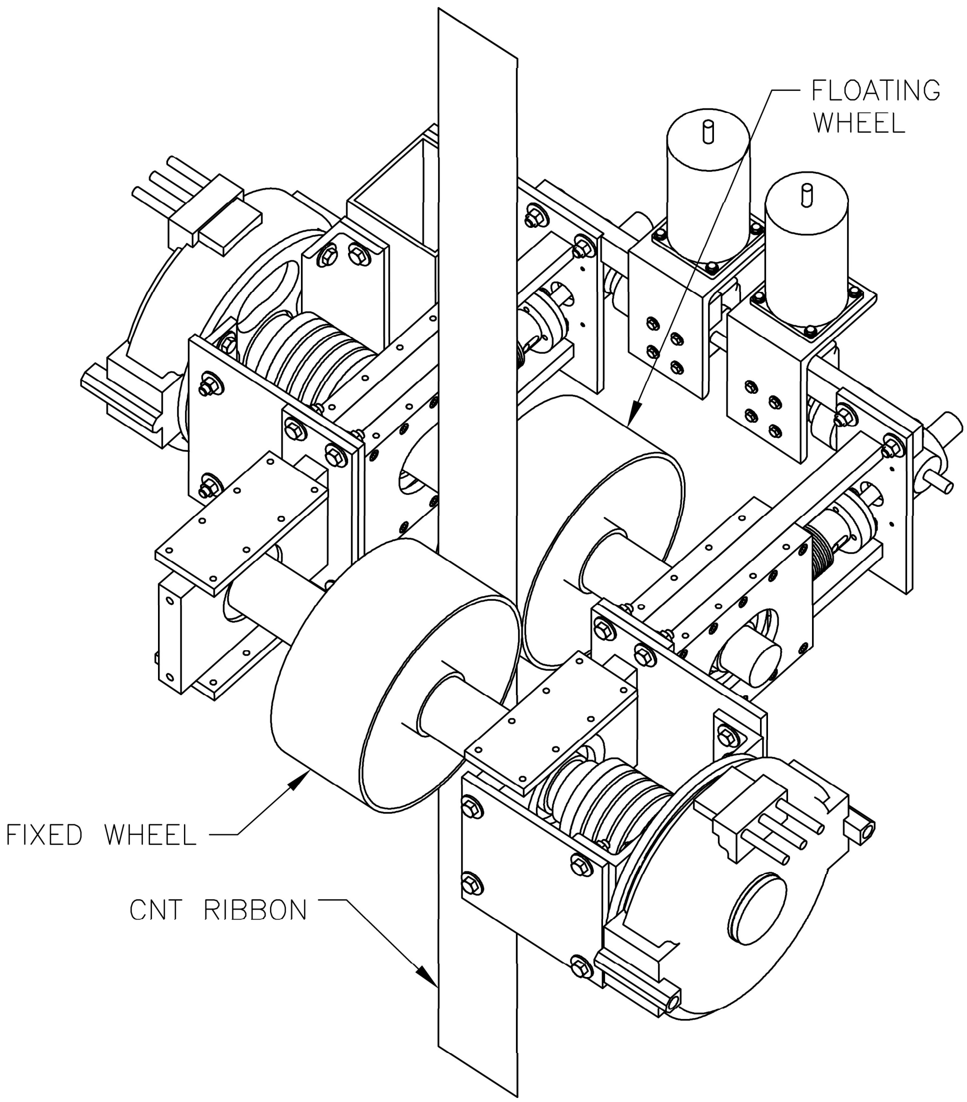

Figure 4. This shows a simplified assembly of one fixed wheel module and one floating wheel module to see how they interact to pinch the ribbon between them. The pinch force is distributed through the climber frame as efficiently as possible by light members in tension and fastener patterns that are concentric to the lines of action of the forces. Structural interface to the rest of the climber happens right at the bearing housings.

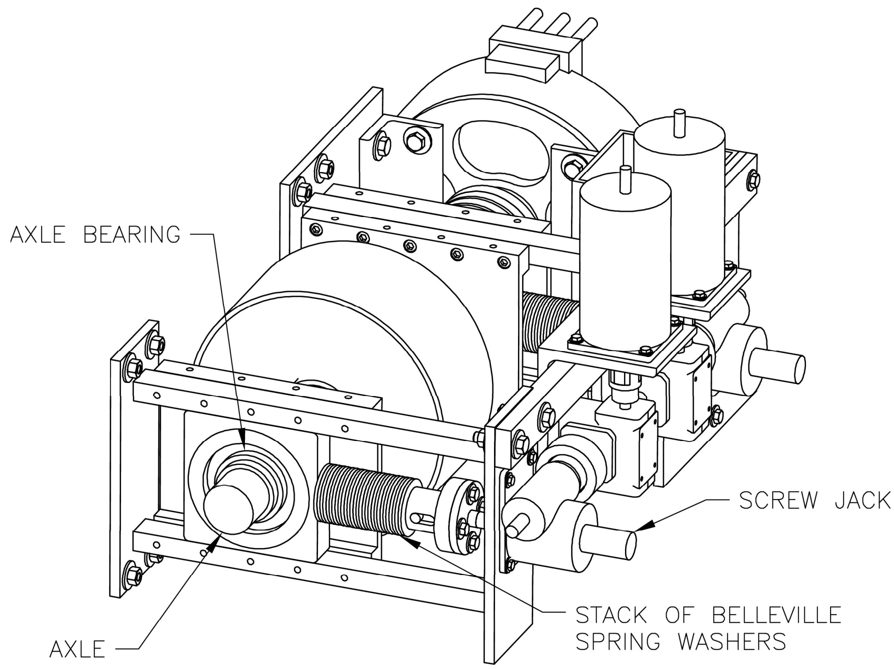

Figure 5. This picture shows the conceptual design of the compression mechanism that forces the floating wheel against the fixed wheel. The bearings that support the axle are mounted in blocks that are free to slide along the climber structural frame. The motor is fixed to the climber frame while the axle is allowed to move with respect to the shaft of the motor. The motor coupling shown is a Schmidt coupling which allows the driven shaft to move off the axis of the motor shaft. Schmidt couplings do not cause the motor shaft to be side-loaded by the offset, and the amount of offset is variable and can change during rotation of the coupling. All bearings and sliding contact surfaces on spacecraft pose unique and difficult design challenges as heat is much harder to remove in the vacuum of space. Vacuum welding of metals and lubrication are also space mechanism design challenges. The belleville washer spring stack increases the force resolution of the screw jack by spreading out the distance over which the wheel goes from no compression to full compression. No instrumentation such as position or force sensors are shown.

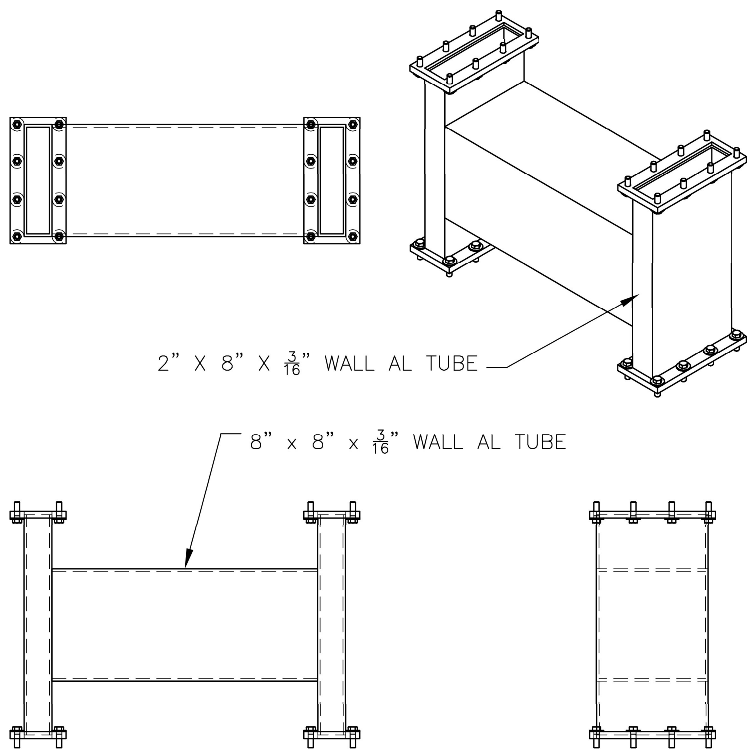

Figure 6. The interface modules connect the traction modules on one side of the ribbon together and transmit forces from the rest of the climber to the wheels. The goal of these modules is to make them as light as possible, yet strong enough to support the loads. These modules are shown conceptually designed with very thin wall standard aluminum tubing. In a final design they would carry fluid plumbing for the heat dissipation system and cables for power and instrumentation.

Using the Coulomb theory of dry friction, the normal force required to prevent slipping is directly proportional to the weight of the climber and indirectly proportional to the coefficient of friction, μ. A reasonable guess of μ=.1 and a climber weight of 900 kg means that the force between a single wheel pair supporting the whole climber must be 44,145 Newtons, or about 10,000 lbf (5 tons). Analysis of simple wheel designs paying attention to fatigue allowables showed that that force on a single wheel pair was high enough that the wheels and axles were unlikely to survive the long climb to the end of the ribbon.

The allowable stress on a material decreases as the material experiences a series of cyclic loads (a process of material degradation called "fatigue".) The number of times the wheels rotate in going to the end of the ribbon must be limited to about 150 million rotations. Beyond that there is very little fatigue data for any wheel material. The size of the wheel is controlled by the maximum stress it sees during rolling. Larger wheels with the same contact force have a lower maximum stress than smaller wheels, but they weigh more than smaller wheels. As the fatigue allowable of the wheel increases (controlled by material selection,) the diameter of the wheel can be decreased. Smaller wheels mean a lighter climber. There is an optimum combination of numbers of wheel pairs and diameter of the wheels. The wheels cannot be below a certain size (8.4 inches in diameter), or they will rotate too many times during the trip to the end of the ribbon. Adding additional wheel pairs can lower the force on each pair, but beyond three pairs the total weight of the climber rises again.

A variety of materials and designs were studied, with the conclusion that three pairs of wheels able to share the weight of the climber equally would lower the force on each pair enough that the fatigue allowables for $> 99\%$ confidence could be satisfied. The conceptual design shown gives the smallest wheel diameter for stainless steel 321 wheels, and three pairs are used to lower the force sufficiently on each pair.

If the material of the ribbon is such that the coefficient of friction is much less than .1, the stress in the climber goes up because the pinch force must be increased to provide the same level of traction. Within one order of magnitude lower, the ribbon becomes too slippery and the forces on the climber become too high for a practical climber.

4. The Challenge of the Traction Drive Motors

Edwards and Westling estimated that the laser power available to the first construction climber would be about 100 kW. The climber outlined in their book has five 20 kW motors. They did a motor study and concluded that a permanent magnet (Neodymium-Iron-Boron) brushless motor with a liquid cooling system could run at the torques, power efficiency and speeds necessary. To quote the book on page 50: "A 10 kW motor of this design would have a mass of 14 kg, require 5 kg of control electronics and could be produced in quantity for under \$9K. A 100 kW motor of this design would have a mass of 105 kg, require 20 kg of control electronics and could be produced in quantity for under \$50K." Using these numbers, the power-to-mass ratio for the 10 kW motor is 714 W/kg, and the ratio for the 100 kW motor is 952 W/kg.

The author performed an independent motor study and concluded that the most efficient, highest torque motor for the application is probably an axial gap motor. I contacted Rick Halstead, the president of Empire Magnetics, to enquire about purchasing a motor in this power range. Empire was recommended by other motor vendors as one of the few companies familiar with the design of motors for operation in a vacuum. Empire has been studying axial gap motor technology for the last few years, but currently does not offer this technology of motor as an off-the-shelf item. Rick was kind enough to provide a spreadsheet that compared the dimensions and weights of theoretical 20 kW and 50 kW axial gap motors. These weights were used to develop the mass budget of the proposed climber. From the numbers in the spreadsheet, it was calculated that the power-to-mass ratio of the 20 kW motor was 5714 W/kg, and 7246 W/kg for the 50 kW motor. These numbers are almost eight times larger than the power densities of motors found by Edwards and Westling. It is not known how physically realizable the motor parameters in the Empire spreadsheet are.

During the search Precision Magnetic Bearings (PMB) turned up as a company that had designed axial gap motors for use in electric vehicles under a grant from the DOE (No. DE-FG02-98ER82647). The Principal Investigator, Dantam Rao, was kind enough to provide a CAD model of a 50 kW liquid cooled axial gap motor. The main website for Precision Magnetic Bearings no longer exists, so I am uncertain what the status of the company is. This motor model was used in the CAD model of the climber shown here. This 50 kW motor was similar in size to the dimensions of the theoretical motor from Empire so it was considered a reasonable place holder. I do not know if this motor was ever built. A request for the speed/torque curve and the mass for the motor went unanswered. The volume of this motor from the CAD model was measured as 610.5 in^3. The volume of the 50 kW motor from the Empire spreadsheet was 350 in^3. The volume in the CAD model includes the entire casing, not just the magnetic volume, so the PMB volume and the Empire volume may not be directly comparable. Empire did not clarify exactly what volume was being calculated on their spreadsheet.

If one assumes that the volumes are comparable, it may be the case that the PMB motor weighs more than the mass estimated by the Empire spreadsheet. If so, the power density for the PMB motor could be reduced by a factor of 1.7, making its power density still 4.7 times larger than that of the Edwards and Westling motors. The uncertainties in the material provided made the design effort very frustrating.

Lynx Motion Technology has also been a pioneer in this style of motor, but a request for an evaluation of the application went unanswered. No catalog of motors was available from their web site.

A Google search for axial gap motors found a number of papers analyzing this technology by Metin Aydin, PhD. Metin described the process of designing a new motor. Because the technology is not mature like that of radial gap induction motors, no one has a catalog of commercially available motors in this power range. The organization building the space elevator will have to specify the motors they want. According to Metin, there are only a few companies and individuals with the capability of doing the complex magnetic finite element analysis that must be done to optimize the motor. Once the motor magnetic structure is known, and cooling has been designed, the motor can be prototyped by a vendor who specializes in motor prototyping. The design of the motor can cost much more than the first prototype. It is also expected that costs per motor will go down as production ramps up, but that initial costs will be high.

The conclusions reached from this motor study are that power densities could be higher than found by Edwards and Westling leading to lighter motors. Unfortunately, as described below, the reduction in motor weight is not enough to offset the overage in weight from the surrounding structures and hardware.

5. The Mass Budget of the Climber

The mass budget for the first construction climber appears on page 54, Table 3.2 of *The Space Elevator*. It is reproduced here.

Table 1. The mass budget for the first climber

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

To get the mass budget for the conceptual design of the traction drive, I added the masses of the structure, the motors and the track and roller components to get a maximum drive mass of 233 kg. It was hoped that the traction drive would come in substantially less than this because not all of the structure mass could be allocated to the drive system. After assigning materials to all the components in the CAD model and taking catalog values for the masses of off-the-shelf components, the mass of the climber with six 20 kW motors turned out to be 647 kg, or at least 2.8 times larger than the required budget allotment. Considering that the conceptual design has not had a rigorous finite element optimization to minimize the weight of the structure, a factor of three is not too far off from the goal. If the Empire 20 kW motors can be made at 14 kg each, then six motors would total 84 kg instead of the 127 allotted in the Edwards and Westling design, a savings of 43 kg. It is hard to imagine reducing the wall thickness of the structural interface module aluminum tubing from .19 inches thick to .06 inches, but it may be possible to reduce the structural framework around the wheels and motors to compensate.

In summary, the conceptual design with non-space-worthy components was almost a factor of three heavier than the Edwards and Westling mass budget

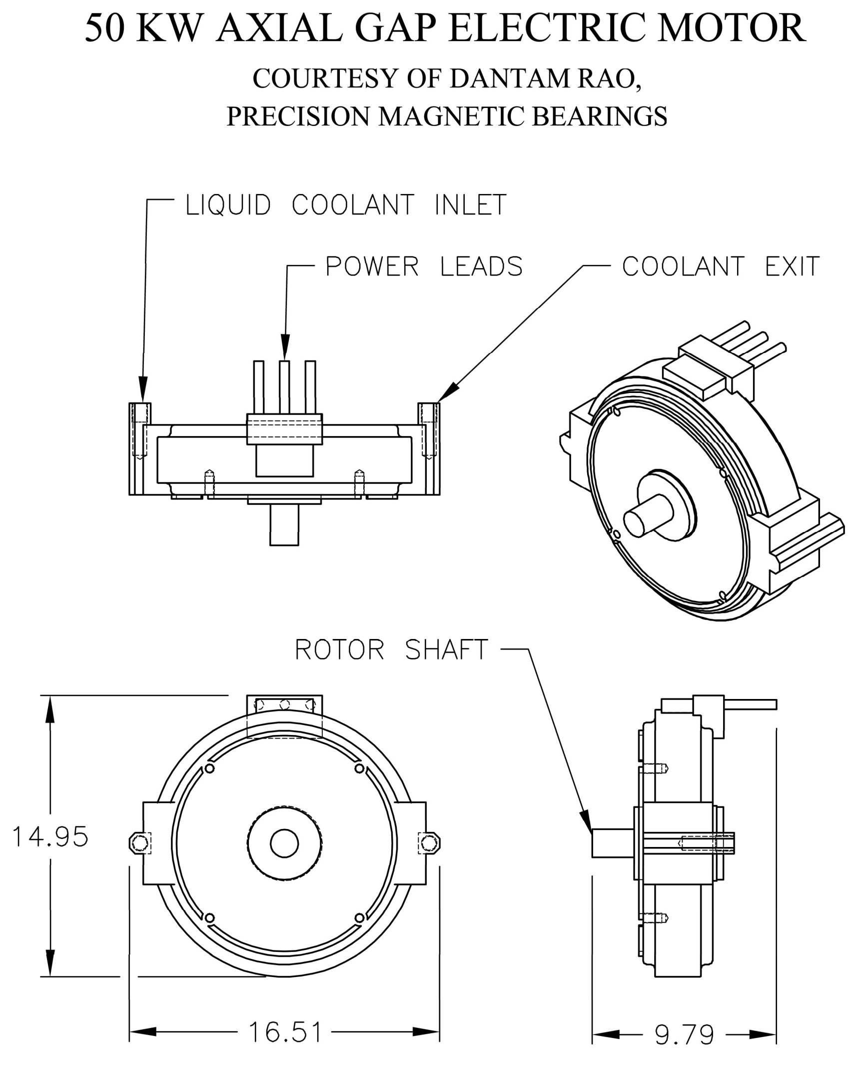

Figure 7. This picture shows the 50kW axial gap motor provided by Dantam Rao of Precision Magnetic Bearings. It is a liquid cooled motor. Torque specs for this motor were unavailable.

allows it to be. A discussion of space bearings and tribology in the reference by Fortescue implies that space-rated components would likely be heavier than their earth-bound analogs, making the design problem even tougher. The fact that the overage was not more than a factor of 10 gives this author hope that the design challenge can be met with real components.

6. Conclusion

The design of the first few construction climbers is very challenging to stay within the required mass budget. It is expected that as the ribbon gets stronger the design of later climbers becomes easier. Before the design can be attempted in earnest, the material properties of the CNT ribbon fabric must be known. A coefficient of friction too low can move the design outside the realm of practicality as the ribbon becomes too slippery for traction drives. The details of the climber structure cannot be determined without a real motor design. No commercially available motors in the desired power range were found, but even if they did exist they would not be space-worthy. The motors will have to be custom designed to make them space-worthy, an expensive proposition. Without a known motor with a given torque/speed curve, it is impossible to verify that the drive can attain the speed necessary to complete the ribbon construction in the time estimated. The design cannot be taken much further without these prerequisites in place.

References

Boyer, H. (1986) *Atlas of Fatigue Curves*, ASM International, Ohio.

Edwards, B. and Westling, E. (2003) *The Space Elevator-a revolutionary Earth-to-space transportation system*, BC Edwards, Houston

Fortescue, P. and Stark, J. (1995) *Spacecraft Systems Engineering*, Wiley, Chichester.

Johnson, K. (1985) *Contact Mechanics*, Cambridge University Press, Cambridge

Lampman, S. (1996) *ASM Handbook Volume 19 Fatigue and Fracture*, ASM International, Ohio.

Shigley, J. (1977) *Mechanical Engineering Design*, McGraw-Hill, New York.

Shigley, J. and Mischke, C. (1986) *Standard Handbook of Machine Design*, McGraw-Hill, New York.

Weichel, H. (1990) *Laser Beam Propagation in the Atmosphere*, SPIE, Washington.