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# Advanced Rover Chassis

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# ADVANCED ROVER CHASSIS

by

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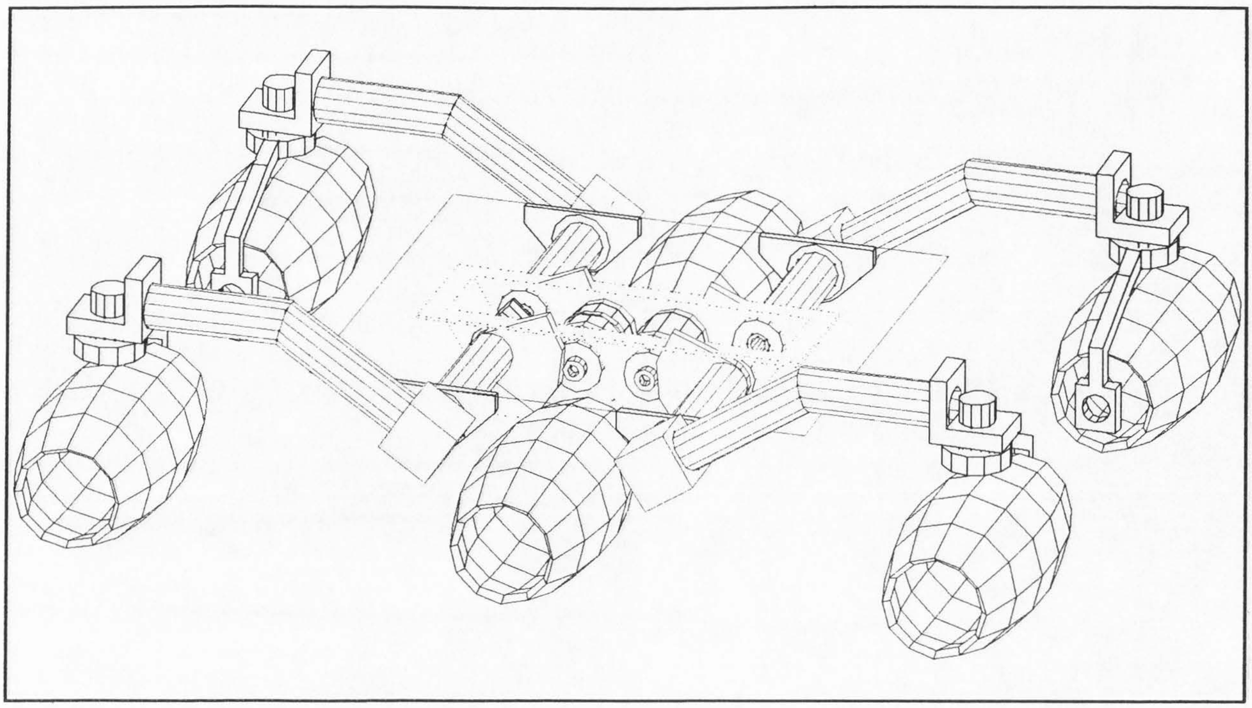

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# Design Requirements and Objectives:

## Background:

The six wheeled rover vehicle detailed in this design is intended as an upgrade test bed for the sensor array and autonomous navigation algorithms in use by Utah State University's Center for Self-Organizing and Intelligent Systems (CSOIS). The CSOIS's sensor suite can successfully detect and avoid unnavigable obstacles up to five vehicle lengths in front of the vehicle. The center presently uses a modified RC type chassis and only supports two wheel drive. This chassis was adequate to bring the CSOIS's algorithms to a proof-of-principles state, but in order to place the system in any practical application, a full mobility chassis must be implemented. Although the purpose of the sensor is to detect obstacles, the chassis must still be able to crawl over small obstacles since the navigation system will indicate a best route to goal, not a perfectly smooth path.

## Parameters:

The rover proposed by CSOIS is six wheeled in nature. The chassis must fit inside a 35 x 45 x 7.5 cm envelope. The total mass can not be any more than 2.5 kg and must be strong enough to support a 2.0 kg payload. The payload is designed to fit on a 20 x 22 cm platform centered over the chassis. Each of the six wheels must be individually driven. The rover msut be able to carry the payload up a 20° slope. The chassis must be capable of Ackerman steering (like on a car) and slip/skid steering (like on a tank). It must have a turning radius of 35 cm radius and have a total budget of $2000.00 or less.

# Final Design

The format of the final design discussion in this report will precede by discussing the wheel and hub design first (section 1), followed by the steering and drive train (section 2), and finally the frame and suspension (section 3).

Discussions of the system drivers, failure modes, maufacturing schedule, and cost are contained in section 4.

## Section 1.0 Wheel and Hub Design:

Each of the six wheels will be cylindrical in shape with a circular profile. This shape was chosen to maximize the variable bouyancy and the interior volume. This profile was originally designed to be elliptical to give more room for the motor. This original motor was bulky and inefficient. However, we found a smaller more powerful motor that was better suited for our needs. This allowed us to modify our design to a circular-arc profile. This change gave us a more dramatic bouancy change in our wheel. The variable bouyancy in these wheels is important in minimizing the friction on hard surfaces and maximizing bouancy on soft surfaces.

| **Materials Selection** | **Chart** | | | | | | |
|---|---|---|---|---|---|---|---|
| | | | | | | | |
| Material | Strength | Unit Wt. | Machinability | Abrasion | Cost | Availability | Total |
| All. Aluminum | 6 | 8 | 4 | 7 | 8 | 8 | 41 |
| Titianium | 9 | 9 | 0 | 8 | 0 | 0 | 26 |
| Magnesium | 5 | 8 | 3 | 7 | 5 | 5 | 33 |
| Stainless Steel | 8 | 5 | 1 | 9 | 4 | 6 | 33 |
| Carbon Steel | 9 | 5 | 2 | 9 | 8 | 9 | 42 |
| Carbon Fiber/Epoxy | 6 | 10 | 9 | 2 | 6 | 9 | 42 |

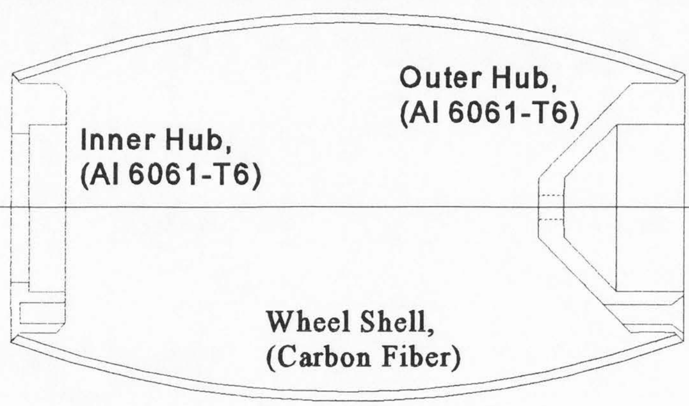



Figure 1.1 Composite Wheel Cross Section

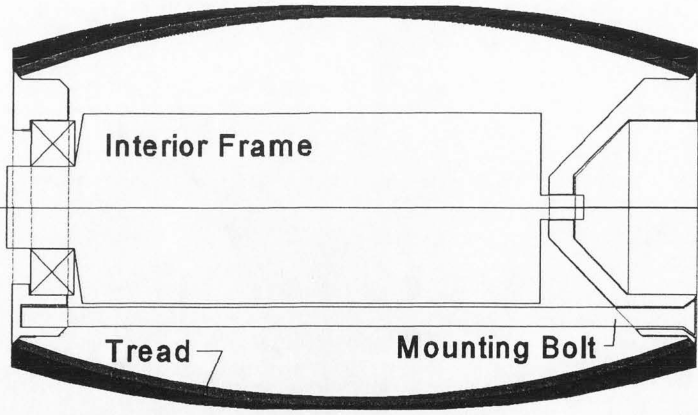



Figure 1.2 Composite Wheel Crossection

The hubs are designed to support the inner frame that contains the drive motor (See figure 1.2). The hub-wheel shell interface utilizes a friction fit to prevent realative motion. The normal force for this fit is supplied by 5 #632 bolts equally spaced around the wheel hubs (see demsioned drawings in appendix A or figure 1.5). The bolts extend through the inner shell of the wheel. The clearance between the interior frame and the mounting bolts is sufficient to allow the outer wheel to turn around the interior frame and motor. The inner and outer hubs will be turned from aluminum round bar stock to the profile shown in figures 1.3 and 1.4 (see appendix A).

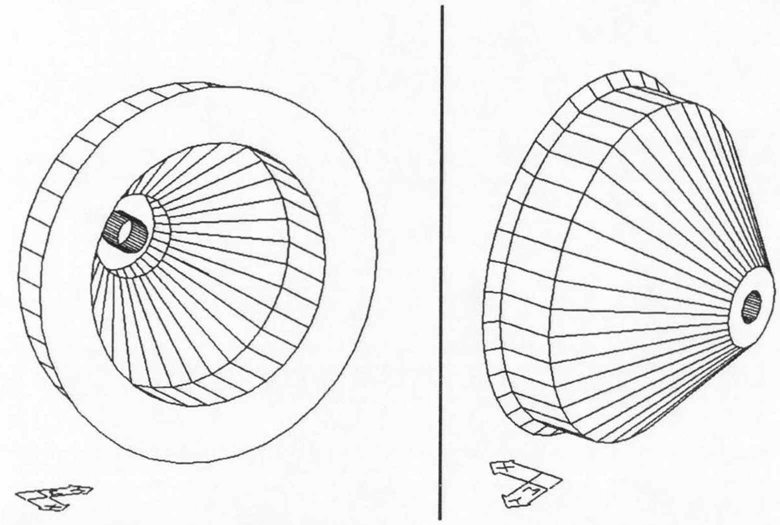


Figure 1.3 Outer Wheel Hub

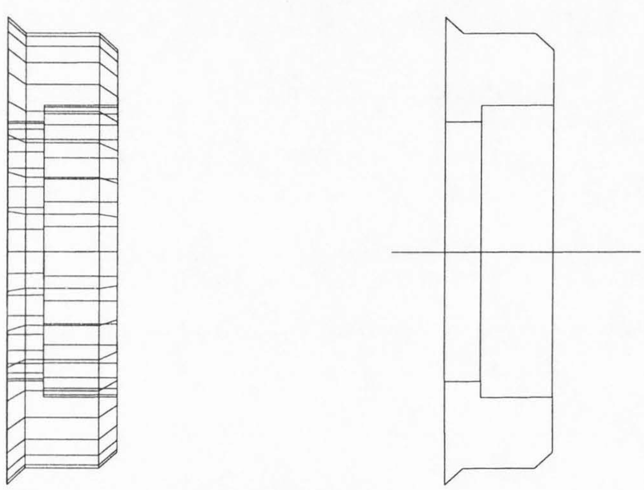


Figure 1.4 Inner Wheel Hub

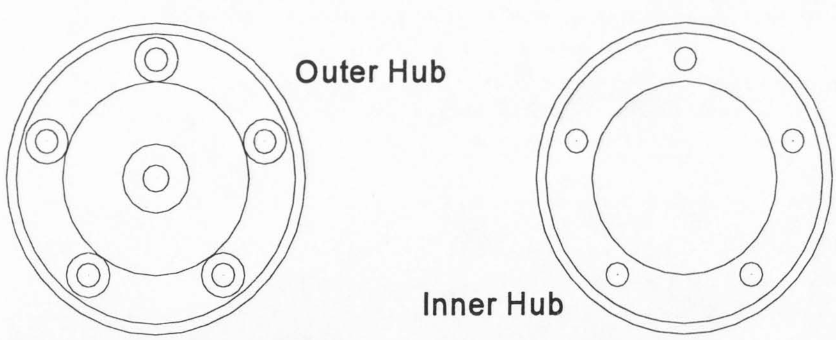



Figure1.5 Bolt Postions

and 1.4 (see appendix A). Aluminum 6061-T6 was chosen because of its good strength, machinability, and availability. The outer hub is hollowed out for weight savings and a smaller moment of inertia. Each hub will be machined with a 45° lip to provide a thrust surface for the hub-wheel shell mating surfaces.

The wheel shell is constructed of continuous carbon fiber matrix in an epoxy resin. Carbon fiber was chosen primarily for its manufacturability and its strength to weight ratio. (see Material Selection Chart) The wheel shell will be wound on a destructible mold to a thickness of 3.2 mm. The tread design will be a 'tractor tread' style for maximum grip in loose soils and will tapper in thickness, being thinnest in the center of the wheel, from the center to the outside of the wheel (figure 1.6). The tread will be thinnest in the center of the wheel to minimize the turning torque on hard surfaces and thickest on the outside of the wheel to maximize traction in loose soil. The wheel shell and tread will be coated with a rubber compound to protect the carbon fiber matrix from excessive wear and create a greater coefficient of friction between the wheel and matting surfaces.

The analysis conducted on the wheel and hub produced significant factors of safety. The shear analysis on the inner and outer hubs produced stresses an order of magnitude lower than the 6061-T6 aluminum material is capable of maintaining. The slow speed of the rover allows us to neglect any impact loading.

The wheel shell analysis is a complex problem. The failure would occur in the

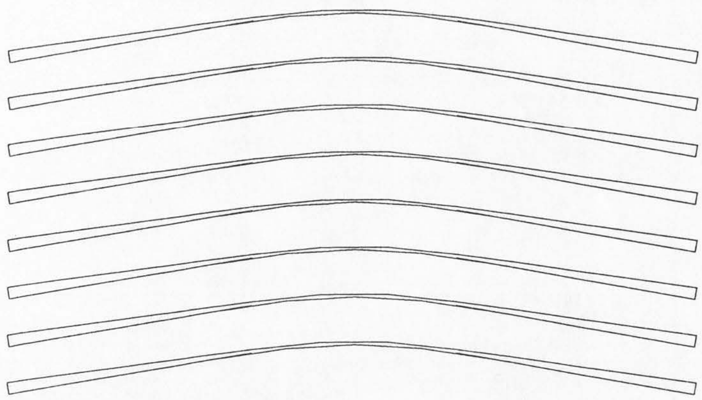


Figure 1.6 Tread Pattern

wheel shell if a sharp object were to break through the shell of the wheel. The stress field created by this type of failure is 3-D in nature and would require a finite element analysis. Due to time constraints, this analysis was not conducted. However, we consider this to be a minimal threat to failure. The rubber coating on the wheel would serve to distribute the localized stresses to a more general area. The five mounting bolts also put a positive pre-stress the wheel shell, which increases the fibers effective load capability. Stress estimations on the wheel shell matrix several orders of magnitude less than excepted stress values for the carbon fiber/epoxy matrix composites (see Appendix A for calculations).

The manufacture of the hubs and wheel shell was based on a production scale of less than ten units. However, large production runs of the wheels could be made economical by blow molding the wheel shell out of a plastic and casting the hubs.

## 2. Steering and Drive System Requirements

The design requirements in the R.F.P. that are applicable to the drive and steering systems are:

- Individual drive systems on each wheel
- Steering in Ackerman and slip/skid modes
- Turning radius < 35cm
- Sufficient torque to climb 20 degree slope

These requirements are the basis for quantified requirements that are included in the individual sections for "drive train" and "steering".

## Steering and Drive Train Systems Overview

The drive and steering systems will interface to the suspension system and wheel shell. A sketch of the combined drive and steering systems is shown in figure 2.1.

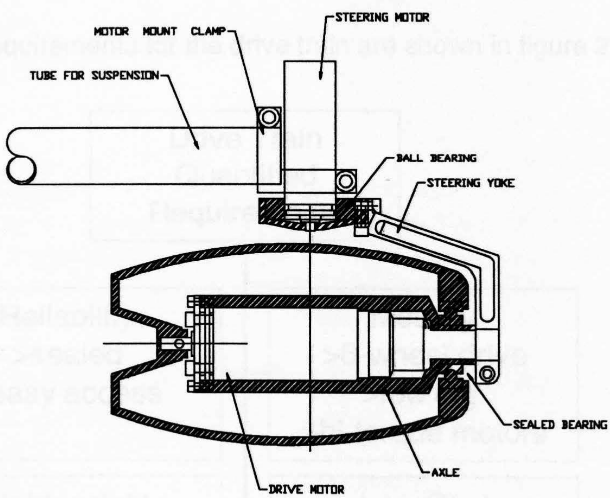



Figure 2.1

The horizontal tube near the top of the system joins the suspension system with the steering system. The steering motor is linked to a steering yoke via ball bearing interface. The steering yoke attaches to the axle that contains the motor inside the wheel. A number of especially machined parts are needed to make the drive and steering systems. These will be machined from aluminum 6061-T6. A decision matrix was used to help select the most feasible material. (Fig. 2.2)

| SDP MATERIAL MATRIX FOR DRIVE AND STEERING SYSTEMS | | | | | | |
|---|---|---|---|---|---|---|
| BEST: 5 WORST: 1 | | | | | | |
| FACTOR | 1 | 1.5 | 2 | 3 | 2 | |
| MATERIAL | COST | MANUFACT. MACHINABILITY | STRENGTH | WEAR , RELIABILITY | MASS | TOTAL |
| 1020 CRS | 5 | 2 | 5 | 3 | 3 | |
| weighted | 5 | 3 | 10 | 9 | 6 | 33 |
| G-10 | 1 | 1 | 3 | 1 | 5 | |
| weighted | 1 | 1.5 | 6 | 3 | 10 | 21.5 |
| DELRIN | 3 | 3 | 1 | 4 | 2 | |
| weighted | 3 | 6 | 4 | 15 | 2 | 30 |
| 6061 T-6 AL | 4 | 5 | 4 | 2 | 4 | |
| weighted | 4 | 7.5 | 8 | 6 | 8 | 33.5 |

Figure 2.2

**The Drive System**

Quantified design requirements for the drive train are shown in figure 2.3.

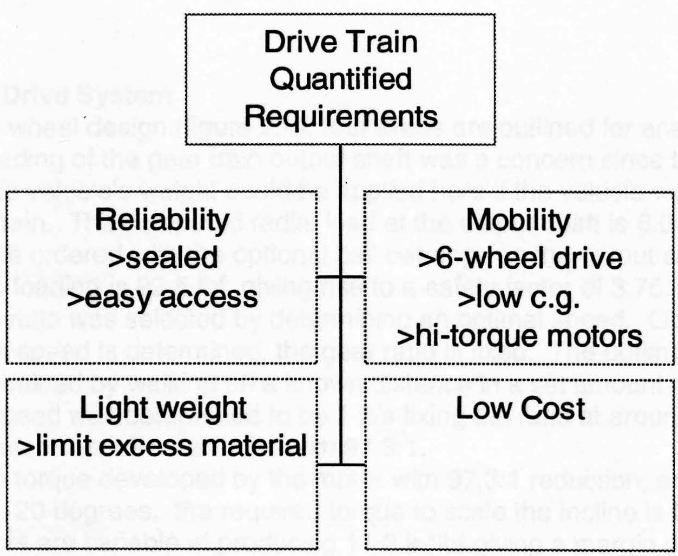



Figure 2.3

Figure 2.4 shows the drive system. It consists of a hollow axle that houses the drive motor, gear reduction, and magnetic encoder. A clamp joint fixes the axle to the steering yoke. The in-wheel motor mounting location was selected to lower the center of gravity of the vehicle and to afford the drive mechanism protection from dirt and moisture. Wires for control and feedback enter through the hollow axle center. The axle is supported by a double sealed ball bearing on the inboard side and by the gear train output shaft on the outboard side.

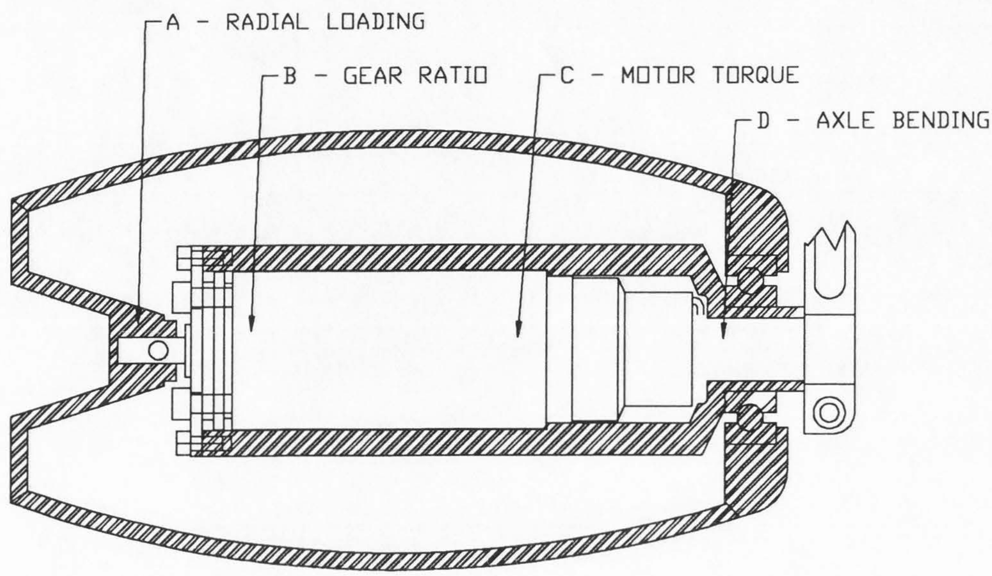



Figure 2.4

**Analysis of Drive System**

In the wheel design (figure 2.4), four areas are outlined for analysis:

A. Radial loading of the gear train output shaft was a concern since the moment load of the vehicle's weight could be applied here if the vehicle were on rough terrain. The computed radial load at the output shaft is 6.0 lbf. If the motors are ordered with the optional ball bearings on the output shaft, the allowable loading is 22.5 lbf, giving rise to a safety factor of 3.75.

B. The gear ratio was selected by determining an optimal speed. Once the maximum speed is determined, the gear ratio is fixed. The optimal speed was determined by walking off a known distance in a set amount of time. The optimal speed was determined to be 1 ft/s fixing the ratio at around 100:1. The manufacturer offers a motor with 97.3:1.

C. Using the torque developed by the motor with 97.3:1 reduction, and a slope incline of 20 degrees, the required torque to scale the incline is 3.0 in*lb. The motors are capable of producing 11.3 in*lbf giving a margin of safety at 3.75. The gear train however, can sustain 1.17 times the maximum expected torque.

D. The bending of the axle under the applied moment load was also analyzed. Maximum bending stress was identified to be at 15.1 ksi yielding a safety factor of 2.3 for the 6061 T-6 aluminum.

## The Steering System

The six-wheel drive rover platform will be capable of Ackerman and slip/skid steering by using four steering motors installed on the drive wheels at the corners of the vehicle. This configuration will also allow the vehicle to spin resulting in a zero radius turn.

The steering system (figure 2.5) consists of a motor attachment clamp, steering motor, ball bearing pivot ring, and steering yoke. The motor will be glued inside the inner diameter of a bearing whose outer diameter will be press fit into an aluminum ball-bearing pivot ring that will connect to the steering yoke.

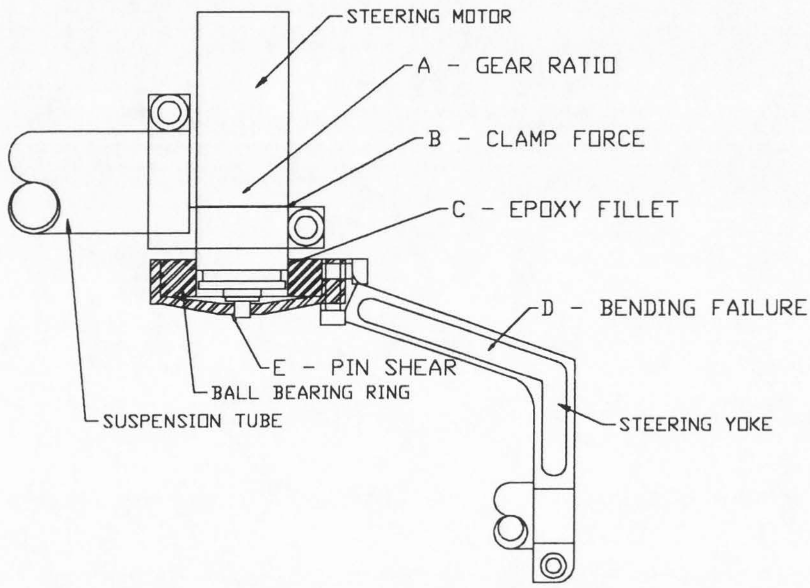



Figure 2.5

Dimensioned sketches of each of the components can be found in the appendix. Engineering requirements for the steering system are shown in figure 2.6.

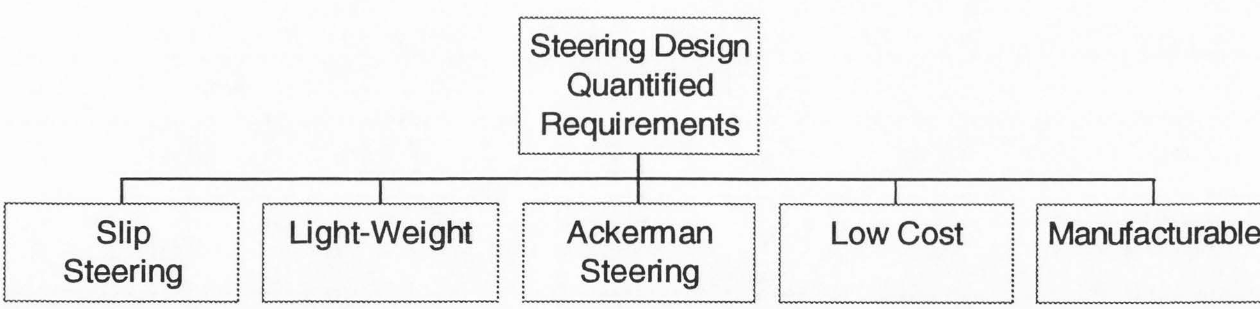



Figure 2.6

**Steering System Analysis**

A. The selection of the gear ratio for the steering system, like the drive system was an empirical test. A reduction of 989:1 (available reduction) was chosen to yield 1.8 seconds lock to lock on the steering. Lock to lock on this steering system is defined and -90 degrees to +90 degrees. The maximum torque that will be developed by this motor and gear train is 4 in*lbf.

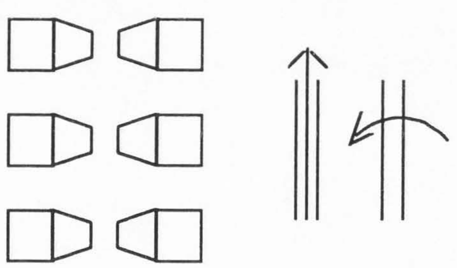
B. Since clamping joints are used at three points in the steering system, an analysis was performed to verify that they would not slip. The lowest of these torques was computed to be 150 in*lbf. This is an order of magnitude greater than anything the joints will experience.
C. The glue joint holding the motor in place will be epoxy with 4.0 ksi shear strength. Maximum expected stress will only reach 87 psi.
D. Bending failure of the beam was analyzed using a program to optimize the I/c ratio. Worst case stress is 8.7 ksi. Safety against bending is 4.0. Safety against shear failure at the central axis is 6.4.
E. Pin shear at the drive pin is prevented by a safety factor of 1.45 with a computed maximum stress of 13.7 ksi.

**More About the Drive and Steering Systems**

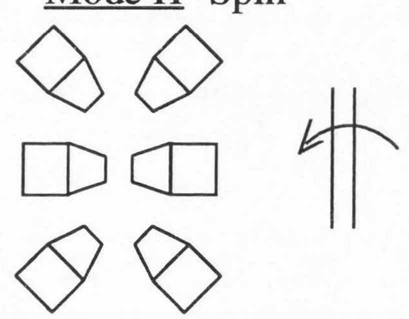
The motors used in this design to power drive and steering units are precision units from MicroMo Electronics. Both units include 12VDC motor, low back-lash gear train and 16 pulse-per-revolution magnetic encoder. Sketches of each motor are included in the appendix-B p8.

Adding Ackerman steering capabilities and active suspension to a chassis already capable of skid/slip steering opens up some additional possibilities. Some of the steering modes supported by the chassis are illustrated here.

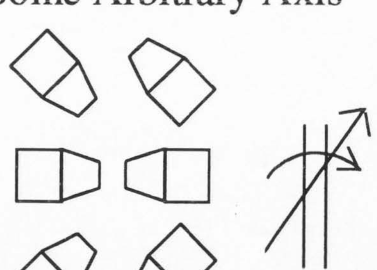
Mode I -Skid Steering



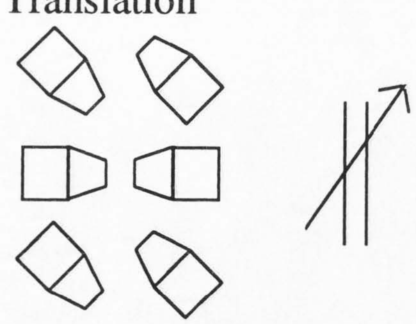
Mode II -Spin



Mode III -Rotation About Some Arbitrary Axis



Mode IV -Hovercraft Style Translation



- Mode I is army-tank style skid steering requiring high torque and power.
- Mode II is a spin mode about the central axis of the vehicle. This capability takes advantage of the fact that the corner wheels can achieve+-90 degrees of travel.
- Mode IV shows the vehicle rotating about some arbitrary axis off the vehicle.
- Mode V occurs when the suspension raises the center two wheels off the ground and hovercraft-style motion in any direction with any rotation can be achieved.

**Meeting Drive and Steering System Requirements**

In summary, the drive and steering design will meet the following requirements:

1. Drive motors inside each wheel provide sufficient torque to pull the vehicle up a 20+ degree slope.
2. Steering and drive mechanisms are compact enough to fall inside the 35 x 45 x 7.5 cm envelope.
3. Full range of motion on the four corner steering motors supplies more than just Ackerman and slip/skid steering modes.
4. Structural strength of parts is sufficient to support the 2 kg payload.

# SECTION 3

## SUSPENSION AND FRAME SUBSYSTEM

### REQUIREMENTS

The system requirements applicable to the suspension and frame subsystem are:

- hold payload mounting plate 7.5 cm above the ground
- minimize tipping of the payload
- support steering mechanism
- negotiate loose soil
- negotiate 8 cm tall obstacles

From these system requirements, the following set of quantified system requirements was developed:

The suspension mechanism must interface with the steering mechanism for each of the front and back wheels. The attachment point for each wheel above the wheel center. The suspension mechanism must also allow 360° rotation of the wheel and steering mechanism about the vertical axis.

The suspension mechanism must operate outside the payload volume. The payload must be kept near a height of 7.5 cm, and the front and back wheels must not interfere with the payload corners at any combination of steering and suspension positions. (This requirement has been relaxed to allow intrusion of suspension gears into the payload volume.)

The attachment of the frame and suspension the payload must be simple, versatile, and easy to assemble and disassemble.

The rover is required to move in a variety of surface conditions, including loose soil and large obstacles. As shown in the figure 3.1, these requirements indicate a suspension system which at minimum distributes the load between the wheels on uneven ground and lifts the front wheels to go over obstacles. We determined that load distribution can be accomplished passively, but that to climb onto an 8 cm overhang, an active system is needed.

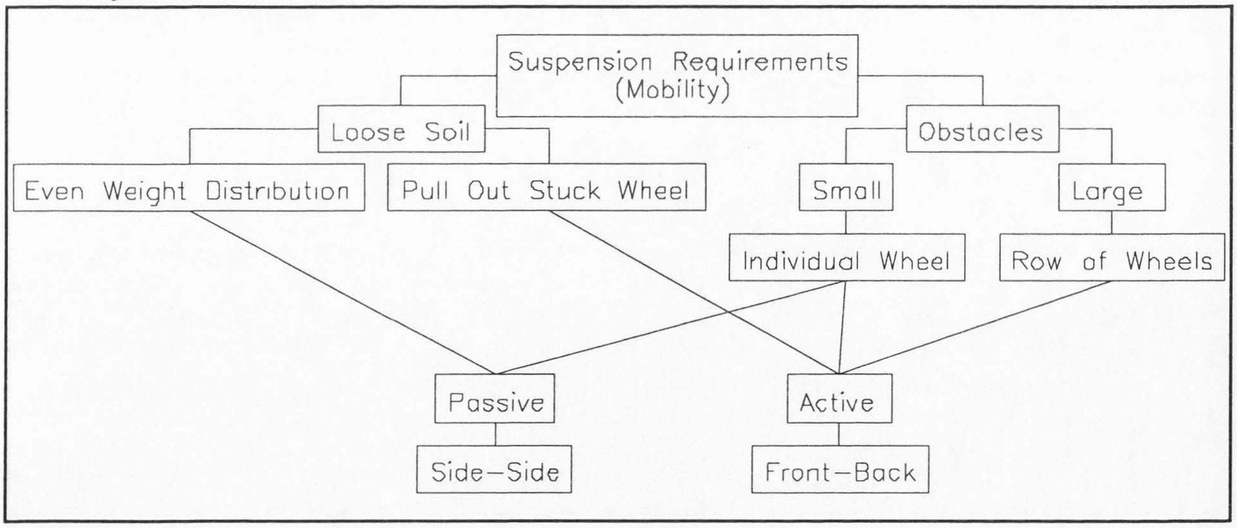



Figure 3.1

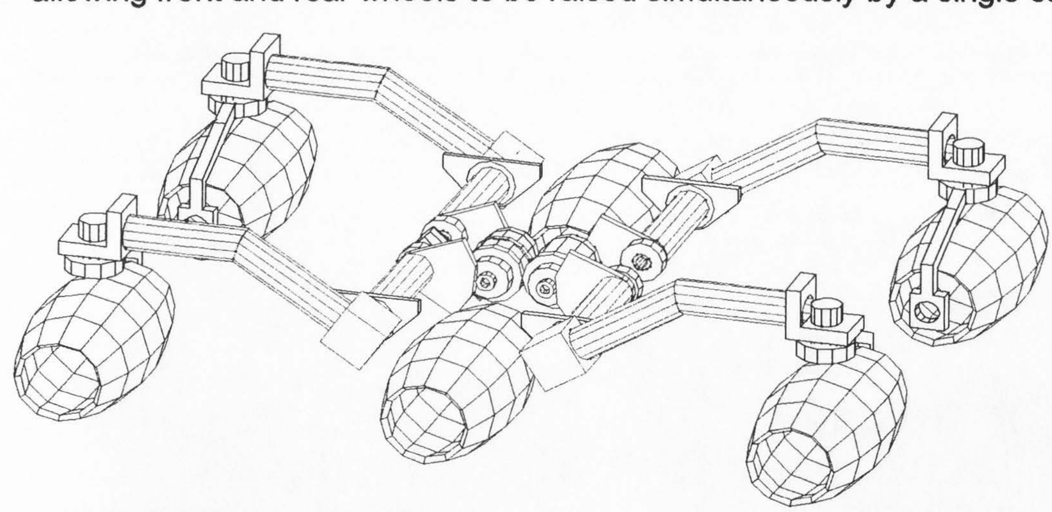
The suspension system must have enough active wheel travel to climb over both fence shaped as well as step shaped obstacles of 8 cm height. The passive system must distribute the load evenly in order to avoid pushing any one wheel too far into loose soil, and must keep the payload relatively steady on uneven ground.

The frame and suspension members must withstand all likely loads with a reasonable safety factor, and must not impose extraordinary loads on other system components.

## SUSPENSION

We considered numerous suspension configurations for our design. One design would be to have each wheel move up and down independently with a separate active control mechanism for each, but this design would be complex and difficult to control. This design could be simplified by fixing two of the wheels relative to the payload, reducing the number of control variables to four. One way to incorporate passive load distribution would be to have a common support beam for both front wheels (and another for the rear wheels), and allow it to rotate about the long axis of the vehicle. This design could be combined with a mechanism to make the frame flex in the center, or to move the center wheels down to lift the front ones.

Because the steering design does not lend itself to the conventional axle to support the wheels, and to allow large vertical wheel movement without interference from the payload, we elected to support the front and rear wheels with longitudinal arms. The arms are beside the payload, and connect to the suspension mechanism under the center of the vehicle. Model car differentials for the front and rear allow even load distribution side to side, and lifting of the wheels can be accomplished by driving the differential ring gears. Each arm has an elbow, located a short distance from the fore and aft vehicle center and just under the payload. The arms are each connected to a pie-shaped partial gear, which mates with a small gear on an output shaft of the differential. The ring gears of the two differentials mesh in the center of the vehicle, allowing front and rear wheels to be raised simultaneously by a single control motor. This feature effectively doubles the wheel travel, because the vehicle will tip back to keep four wheels on the ground. This also reduces the number of suspension control variables to one.



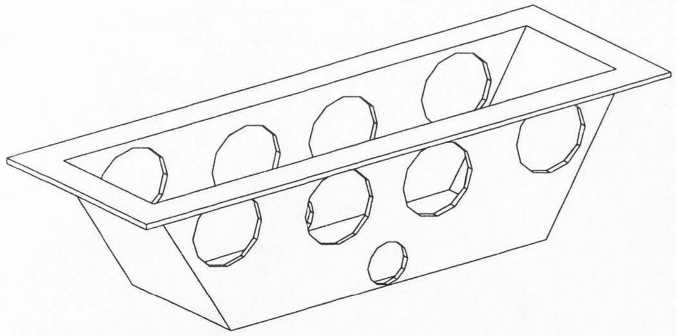
Assembly with frame removed
Figure 3.2

The combination of the active and passive suspension components allows any combination of wheels to lift up. This provides several additional capabilities. All four suspended wheels can be pushed to their lowest position, lifting the center wheels off the ground. Since the four wheels remaining on the ground can be turned to any position by the steering mechanism, the vehicle could travel in any direction, and change direction without rotating. (scramble steering)

All suspension components are designed with a safety factor of four against static failure. For the arms, round tubing was chosen. This provides good strength in the longitudinal plane to resist drive and weight loads as well as in the transverse plane to resist steering loads. Aluminum was the chosen material because of its low cost, ease of manufacture, high strength, and low weight. Steel was rejected because a tube of equal strength and weight as aluminum would be too thin-walled to resist local buckling. The differentials were chosen on the basis of cost, strength, and availability. They are mass produced for use in model cars, are among the strongest available of comparable size, and are kept in stock by major hobby suppliers. If greater suspension precision is desired, the differentials could be replaced with precision machined bevel gears. This could significantly decrease suspension backlash, but would add considerable cost. Appropriate gears and bearings were chosen from standard parts catalogs for cost and availability. If it is desired to cut initial costs at the expense of durability and suspension precision, the outboard suspension arm ball bearings could be replaced with machined plastic bearings. This could bring a cost savings of approximately $200. The suspension motor is the same as the steering motors.

FRAME

The frame will be constructed of an aluminum sheet, bent into a trapezoidal box shape. The axles of the center wheels will extend through the sides of the box at the bottom. Flat headed bolts will attach them to the bottom of the box. The bearings which support the differentials and the inboard ends of the arms will be mounted in the sides of the box, and the top will attach to the payload. The suspension control motor will be mounted inside the box, just to the rear of the differentials. Bearings to support the arms near the elbows will be attached to the payload at the outer edges. The frame is designed to protect the differentials and active suspension motor while maximizing ground clearance. A closed box design was chosen to seal out contaminants and provide maximum rigidity. The material thickness was chosen after subjective testing of similar structures. If greater confidence is required in the frame stiffness and strength, Finite Element Analysis could be performed and the design optimized. The additional design cost would be significant.



Frame
Figure 3.3

## Section 4.0 System Drivers

The major system driver in the design of ARC is the strength of its frame. This parameter significantly effects the cost of the vehicle in several ways. If the payload for the chassis was increased it might be necessary to find stronger materials for the structural members in order keep the chassis within the mass constraints. An increase in the payload would also increase the stresses on the suspension differentials, bearings, and gear teeth. The current chassis uses off the shelf parts and these increases in stress might force us to fabricate our own, driving the part cost through the roof. The increase in payload might also merit the selection of new motors and batteries in order to carry the extra weight.

The mass of the chassis is also a significant driver. Relaxing the mass constraint might merit the use of steel for the structural members instead of aluminum alloys. This would make the material cost less without significantly affecting the manufacturing costs. Conversely, if the mass constraints were tightened, the cost would go up significantly.

## Failure Modes Analysis

Possible failure modes of the chassis are many but unlikely. They would include wheel shell failure, motor failure (navigational or drive), battery failure, differential failure, gear failure, and suspension arm failure.

Perhaps the most significant failure mode is that of the suspension arm. We feel confident that the arms would not fail due to the loads produced in the slip steering mode. Neither the friction forces nor the torques produced in the steering would could produce stresses significant enough to fail the arm. Should this arm fail due to a large object falling on it, the rover might be totally crippled. The likelihood of this ever happening is remote at best. A failure of this type would require the total replacement of the suspension arm.

If one of the differentials should fail it would probably be due to some type of dirt or grit locking the differential planetary gears in place. The inner gears are sealed and turn while immersed in oil. It would be nearly impossible for the internal gears to fail due to foreign objects and the differential motor does not produce enough stall torque to shear the differential gears. However, a failure in the planetary gear would cause the suspension arms to loose their active movement. The arms would still retain there passive motion and the rover would still be able to proceed step over individual bumps.

A battery failure would be catastrophic to the entire rover. The rover would not be able to move in any manner at all. This problem would have to be addressed by the controls development team.

Should any of the motors fail a number of things could happen. First, if a drive motor should fail, the rover should be able to move around with the remaining five motors with only minor restrictions. If the failure was on a motor in the center, the chassis could be lifted up by driving the suspension arms down using the planetary gears and rover could move around completely unrestricted. If the planetary gear motor failed the suspension movement would be limited to the passive component once again. Failure of a steering motor would result in the loss of the Ackerman, spin, and scramble steering modes. This leaves the rover with a tank-type slip/skid steering. The rover would still be able to maneuver but the chassis would have to drag a wheel assembly around in a fixed position.

It should be noted that any motor failure is highly unlikely. They are a sturdy motor with low internal friction. The motors are equipped with 93:1 gear reductions and have been proven reliable in other applications.

The final failure mode is in the wheel shell. This is an insignificant mode due to the fact that should the wheel shell crack or crush in a localized area no real harm would be done to the chassis. Internal clearances are such that the wheel would still be able to turn. The motor is doubly sealed. (It is self contained as well as sealed inside an internal frame.) We consider this failure mode insignificant in the operation of the chassis.

## Manufacturing Schedule

A detailed manufacturing schedule is contained in appendix D. It illustrates start finish dates and time allotted to each phase of the manufacturing process.

## Mass and Cost Budgets

The budgets for this project are summarized in the table below. The total project cost is $1970 assuming a student labor rate for machining and assembly. The total mass of the ARC chassis is 2341 grams which does not included the mass of any of the motors. A detailed budget break down is given in appendix D.

| **Cost** | **Summary** | | | | | | |
|---|---|---|---|---|---|---|---|
| | | | | | | | |
| STEERING AND DRIVE SYSTEMS | | | | | | | |
| | TOTAL MASS (g) | | 830.62 | | TOTAL COST ($) | | 433 |
| | | | | | | | |
| SUSPENSION AND LINKAGE SYSTEMS | | | | | | | |
| | TOTAL MASS (g) | | 740 | | TOTAL COST ($) | | 724 |
| | | | | | | | |
| WHEEL AND HUB SYSTEMS | | | | | | | |
| | TOTAL MASS (g) | | 770.4 | | TOTAL COST ($) | | 613 |
| | | | | | | | |
| ASSEMBLY COSTS | | | | | | | |
| | | | | | TOTAL COST ($) | | 200 |
| | | | | | | | |
| **Project** | **Totals** | | | | | | |
| | **MASS (g)** | | **2341** | | **COST ($)** | | **1970** |