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**ME4182**  
**MECHANICAL DESIGN ENGINEERING**  
**ENABLER PROJECT**  
**WHEEL AND WHEEL DRIVES**  
**Group 3**

**FACULTY COORDINATOR**  
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**FINAL REPORT**

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(NASA-CR-192060) MECHANICAL DESIGN  
ENGINEERING ENABLER PROJECT WHEEL  
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## **Executive Summary**

Our group was assigned the responsibility of designing the wheel and wheel drive system for a proof-of-concept model of the lunar-based ENABLER. ENABLER is a multi-purpose, six wheeled vehicle designed to lift and transport heavy objects associated with the construction of a lunar base. The resulting design was based on the performance criteria of the ENABLER.

The drive system was designed to enable the vehicle to achieve a speed of 7 mph on a level surface, climb a 30% grade, and surpass a one meter high object and one meter wide crevice. The wheel assemblies were designed to support the entire weight of the vehicle on two wheels. The wheels were designed to serve as the main component of the vehicle's suspension and will provide suitable traction for lunar-typed surfaces.

The expected performance of the drive system for the ENABLER was influenced by many mechanical factors. The expected top speed on a level sandy surface is 4 mph instead of the desired 7 mph. This is due to a lack of necessary power at the wheels. The lack of power resulted from dimension considerations that allowed only an eight horsepower engine and also from mechanical inefficiencies of the hydraulic system. However, the vehicle will be able to climb a 30% grade, surpass a one meter high object and one meter wide crevice. The wheel assemblies will be able to support the entire weight of the vehicle on two wheels. The wheels will also provide adequate suspension for the vehicle and sufficient traction for lunar-type surfaces.

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## **Introduction**

Our group was assigned the responsibility of designing the wheel and wheel drive system for a proof-of-concept model of the lunar-based ENABLER. ENABLER is a multi-purpose, six wheeled vehicle designed to lift and transport heavy objects associated with the construction of a lunar base. The resulting design was based on the performance criteria of the ENABLER.

The performance criteria of the ENABLER associated with the wheel and wheel drive system are as follows. The vehicle needs to achieve a speed of 7 mph on a level surface, climb a 30% grade, surpass a one meter high obstacle and one meter wide crevice. The wheel assemblies need to be strong enough to support the entire weight of the vehicle on two wheels. The wheels will also serve as the main component of the vehicle's suspension and must provide suitable traction for lunar-type surfaces.

The design approach for the drive was a hydraulic system powered by a propane fueled engine. The components of the drive system consist of a four cycle engine, hydraulic pump, one hydraulic motor per wheel, and one speed reducer per wheel. The design approach for the wheel was a cone-shaped wheel made from a carbon fiber / epoxy resin composite.

# **THE ENGINE AND PROPANE SYSTEM**

## **1.1 Introduction**

The function of the engine and propane system is to supply power to the hydraulic pump in order to drive the wheels. It is composed of three parts:

1. A combustible engine
2. Propane conversion kit
3. Fuel supply

### **1.1.1 Performance Requirements**

To supply power for the three ENABLER functions:

1. Achieve a top speed of 7 mph on level surfaces.
2. Climb a 30% grade from dead stop.
3. Surpass a one meter high obstacle.

## **1.2 Engine**

### **1.2.1 Introduction**

The combustion engine is used to convert chemical fuel to shaft power for the hydraulic pump. It is to mount in a two feet chassis extension off the back T section of the ENABLER.

### **1.2.2 Design Considerations**

1. Deliver required power to the hydraulic pump.
2. Constant running speed at 3600 RPM before propane conversion.

3. Easily converted to propane.
4. Small enough to mount into 18" diameter chassis tube.

Because of the design considerations of the vehicle, the engine had to be able to supply at least 0.75 Hp to each wheel. From the criteria it was suggested that a 5 hp or greater engine was needed.

Selected Engine: brand: Briggs & Stratton  
model: 8 Hp Engine Series 190400

Address: Briggs & Stratton Corp.  
12301-T W. Wirth St.  
Wauwatosa, WI 53201

Through the hydraulic system this motor will be able to supply 0.92 hp at each wheel.

### **1.3 Propane Conversion Kit**

The propane conversion kit serves to convert a normally fuel operated engine to operation on propane. Propane conversion was necessary inside the chassis since propane emits a far smaller volume of harmful exhaust than gasoline. However the drawbacks to using propane are a 10% loss in power from the engine and a 10% loss in operation time of the fuel (see Appendix A-1). It was suggested that an off-the-shelf propane conversion kit would be best for our purpose.

Selected Kit: brand: Garretson Small Engine Conversion Kit  
model: #650-132

Address: Garretson Equipment Co., Inc.  
P.O. Box 111 West Industrial Park  
Mount Pleasant, IA 52641

## **1.4 Propane Fuel Supply**

### **1.4.1 Introduction**

The propane fuel supply was used to supply fuel through the conversion kit to the engine's carburetor.

### **1.4.2 Design Considerations**

1. Supply sufficient amount of fuel for a half hour of continuous operation.
2. Had to be vapor fuel.
3. Must be small enough to mount into 18" chassis.
4. Had to be easily refueled in the chassis.

Since the engine operates at 0.87 gallons/hr at full throttle, it was found that 3.5 pints of fuel would be sufficient to operate the vehicle for a half hour of continuous running in a worst case situation(see Appendix A-2).

Selected Kit: brand: Sears Propane Bottle

Address: Sears, Roebuck and Co.

Sears Tower

Chicago, IL 60684

Considering duty cycle of the engine three of these propane bottles would be sufficient for continuous operation.

# **THE HYDRAULIC SYSTEM**

## **2.1 Introduction**

The function of the hydraulic power system is to deliver power from the combustion engine to the wheels. It is composed of three main parts:

1. A hydraulic pump
2. One hydraulic motor per wheel
3. Hydraulic lines

### **2.1.1 Performance Requirements**

To supply power for the three ENABLER functions:

1. Achieve a top speed of 7 mph on level surface
2. Climb a 30% grade from dead stop
3. Surpass a one meter high obstacle

## **2.2 Hydraulic Pump**

### **2.2.1 Introduction**

The hydraulic pump serves to convert shaft power (from the combustion engine) to fluid power for the hydraulic motors. It is mounted in the rear T-section of the ENABLER.

### **2.2.2 Design Considerations**

1. Required power deliveries to hydraulic motors
2. Constant operating pump speed of 3600 RPM
3. Constant operating pressure of 2400 psi
4. Must be small enough to mount in 18" diameter chassis tube

Because of the constant pressure and speed, practicality required the use of a variable displacement pump equipped with a pressure compensator. The appropriate size displacement was determined from the required power deliveries (see Appendix C).

Selected Pump : brand: Vickers  
model: PVB5

Address: Vickers Inc.  
P.O. Drawer 302  
Troy, MI 48007

This pump was also found to be much lighter than other pumps of equivalent power. A three quart fluid reservoir will also be required.

## **2.3 Hydraulic Motors**

### **2.3.1 Introduction**

The hydraulic motors serve to convert fluid power (from the pump) into shaft power for the wheels. There is a total of six motors (one per wheel) with each being mounted in a wheel.

### **2.3.2 Design Considerations**

1. Required wheel torques at corresponding speeds
2. Must be small enough to mount in 18" wheel tube

The main consideration involved with the motor selection was that it had to provide sufficient torques at the corresponding speeds. Therefore, the motor whose torque and speed ranges best correlated with the desired torque

and speed ranges was chosen (see Appendix B). A gear ratio was also added to improve this correlation.

Selected Motor: brand: Char-Lynn  
model: H-Plus Series, 2.2 in<sup>3</sup>/rev displacement  
gear ratio: 8:1 between motor and wheel

Address: Eaton Corp., Hydraulics Div.  
15151 Hwy.5  
Eden Prairie, MN 55344

## **2.4 Hydraulic Lines**

The hydraulic lines connect the pump to the motors.

### **2.4.1 Design Considerations**

1. Required flow rates and flow speeds
2. Flexibility for ease of installation and maintenance

Selected Line: 3/8" diameter hydraulic hose

## **2.5 Speed Reducer**

### **2.5.1 Introduction**

The purpose of the speed reducer is to reduce the torque which the wheel requires to an acceptable speed and torque range for common hydraulic motors.

## 2.5.2 Design Considerations

1. Between 7:1 and 15:1 ratio.
2. Light weight.
3. Fit in a minimum distance in axial direction.

Selection of the gear reduction and motor are interdependent. The ratio between our high and low torque requirements were approximately 5 to 1 and approximately a 7 to 1 ratio between our speed requirements. Therefore a gear ratio had to be selected which would help match motor characteristics to our system requirements.

Selection: Browning Roller Chain Drive, 8:1 ratio.

Part No. 3512      12 Tooth Sprocket

Part No. 35B96    96 Tooth Sprocket

ANSI 35            Standard Chain; 111 " (295 pitches)

Address: Browning Mfg., Emerson Power Transmission Corp.

1935 Browning Dr.

Maysville, KY    41056

The large sprocket will be welded to an extension pipe which is welded to the bearing plate (See Drawing I-8). (Appendix E. contains load calculations for the sprocket arrangement.) The drive sprocket will attach to the motor shaft with a standard Woodruff Key.

# **WHEEL DESIGN**

## **3.1 Introduction**

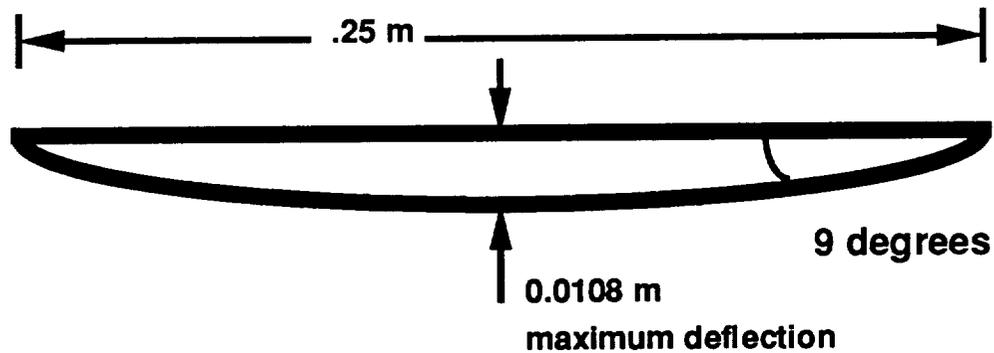
To meet the stated design criteria for the wheels, a conical, carbon fiber epoxy resin composite, non-pneumatic tire was utilized. These requirements included:

1. A strong, light-weight construction.
2. Support the weight of the vehicle plus load carried by the boom.
3. Wheel should provide sufficient shock absorption and traction.
4. Each wheel assembly should be modular.
5. The wheel / chassis interface was designed for ease of removal by a single astronaut wearing a space suit.
6. Conform to restrictions on the outer dimensions and weight.

## **3.2 Bending Moment and Stress Analysis**

To calculate the maximum bending moments and stress experienced by the wheel, a model the wheel deformation was developed. The modeled deflection of a 0.25 m span of composite material 7.5 mm thick was calculated to be 0.01 m.

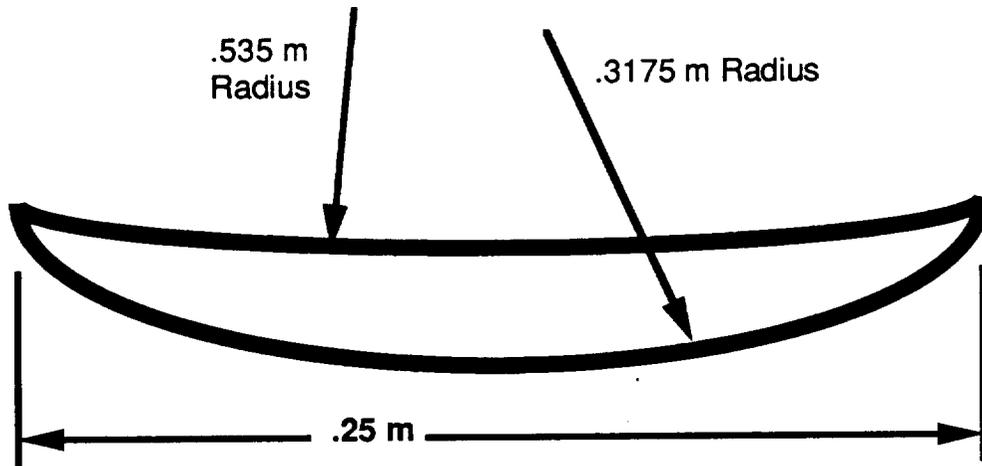
Figure 3.1  
**Deformable Bodies Model**



Making assumptions that the end angle made by the deflected beam is 9 degrees, the calculated maximum deflection of this beam was 0.02 m. Even though this is slightly larger than the model, it is probably more accurate than the model. This is because the modeled deflection is a circle, while the mathematical model is a curve following actual deflection. Actual deflection is expected to be slightly greater than that modeled. The value of the bending moment necessary to deflect the tire this amount is approximately 10,000 N-m. The calculated maximum stress on the outer fibers of the wheel was  $2.07 * 10^9$  Pa (see Appendix F for formulas)

Figure 3.2

### Change in tire Radius



### 3.3 Component Design

Within the wheel assembly, there are several components that needed to be designed.

#### 3.3.1 Wheel

The wheel design is a modification of the U.S. Patent 4,705,087 by Edward G. Markow (Nov. 10, 1987). The patent design is a convoluted cone constructed out of an elastic material. The modification is to simplify the design so that it is easier to build by students while providing the same performance characteristics. Instead of a convoluted cone, the wheel is a section of cone that makes a 20 degree angle with the central axis. The new design also allowed the wheel to extend back over the wheel cylinder, providing protection to the vehicle chassis while transversing obstacles.

### **3.3.2 Wheel Hub**

The wheel hub is a convoluted sheet of aluminum that connects the wheel to the wheel cylinder. The hub connects directly to the inner race of the bearing with eight bolts. The hub then extends out and around the cylinder in a "top hat" design leaving 0.5 inch clearance from the outer surface of the cylinder. The hub retreats to approximately 8 cm of the back edge of the wheel. Once again, the hub curves out and forward running adjacent to the inner surface of the wheel. Along this parallel region, the wheel is connected to the hub with forty rivets. This convoluted shape was designed to sufficiently support the wheel while allowing for some flexibility. It also gives the tire the maximum amount of freedom to deform to absorb shocks.

### **3.3.3 Sprocket Assembly**

To transfer the torque output of the motor to the wheel, a sprocket assembly was designed. Two sprockets were chosen to provide an eight to one ratio. The larger of the two sprockets is attached to the central axis of a sprocket plate. This plate is bolted directly onto the inner race of the ring bearing. To provide sufficient clearance between the sprocket chain and chassis bolts, the sprocket is welded to a 3 inch tubular spacer. This spacer is then welded to the sprocket plate.

### **3.3.4 Ring Ball Bearing**

A ring ball bearing was selected to provide reliable, smooth rotation of the hub and wheel. The ring bearing also resists the bending moment

exerted by the wheel. The open design allows additional room inside the wheel cylinder for the motor and sprocket assembly.

### **3.3.5 Wheel Assembly / Chassis Interface**

The wheel cylinder interfaces with the chassis through a V-band clamp. The V-band clamp works by squeezing together two angled flanges. The contacting surfaces are milled with a radial groove and tongue. A spring loaded pin prevents axial rotation of the wheel cylinder to the chassis. A 10 cm clearance between the chassis body and the back edge of the tire to allow for a gloved hand to disengage the V-band belt latch.

Selected V-band Clamp:            Brand: Clampco  
Part no.: V0916300N-1800-S4

Address:    Clampco Products, Inc.  
1745 Wall Rd.  
Wadsworth, OH 44281

### **3.4 Conclusions**

The amount of bending moment and stress needed to deform the wheel to maximum deflection is very large. This value of bending should not be attained from the normal range of activities assumed to be performed by the vehicle. Accordingly, the wheel is designed to adequately support the vehicle.

The end of the conical wheel in contact with the ground is elastic and not supported directly. This will give enough deformation to absorb the majority of minor shocks. Because of this, sufficient shock absorption is provided and no other suspension system is required.

The one inch high ridges that run along the exterior surface of the wheel should give adequate traction for the required performance of the vehicle on a lunar type surface.

The V-band clamp interface between the wheel cylinder and the chassis has a simple pull-over latch that can be easily disengaged by a single astronaut. Sufficient clearance was allowed for this task.

## **Conclusion**

The accomplishments of our group during the quarter are as follows. We designed a hydraulic drive system for the concept vehicle. Secondly, we redesigned the wheel and its assembly. These designs are expected to produce certain results. The top speed of the vehicle on a level surface is anticipated to be 4 mph instead of the desired 7 mph. The vehicle will be able to climb a 30% grade as well as surpass a one meter high object and one meter wide crevice. The wheel assembly will be strong enough to support the vehicle on two wheels and provide adequate suspension and traction.

The components necessary for the hydraulic system design consist of propane fueled engine, a hydraulic pump, one hydraulic motor per wheel, one speed reducer per wheel, two four way valves and six servo valves. The components of the wheel design consist of a composite wheel, an aluminum hub, V band clamp, wheel cylinder, bearing and bearing/sprocket plate.

We feel that we have accomplished our objectives for this quarter. By following the design set forth in this report, future design students will have sufficient groundwork to conclude the ENABLER Project wheel and wheel drive system.

# APPENDIX A

## THE ENGINE AND PROPANE SYSTEM

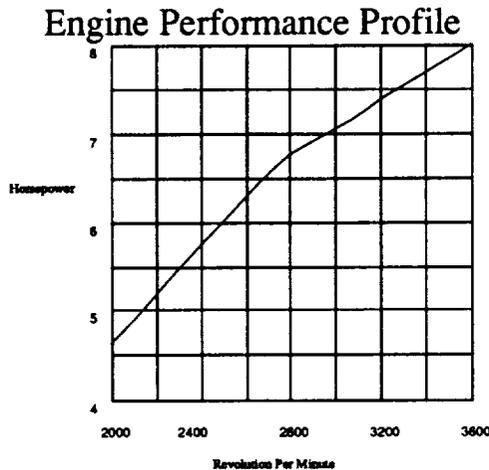
### Introduction

Before a combustion engine can be used inside a enclosed area it has to be converted to use a clean fuel system. The clean fuel systems available on the market today are propane, butane, and natural gas. Propane was selected because of its wide use in conversion of gasoline engines. The drawback to using propane is the 10% loss in power due to the characteristics of propane fuel.

### A-1 Propane Conversion

Using the performance characteristics, described in the manual by the manufacturer, for the engine powered using gasoline, the loss of power from propane conversion had to be considered in the operation of the vehicle. Shown below is a graph of the performance profile of the engine from the 2000 to 3600 RPM.

Figure A-1



From the performance profile, the power at 3600 RPM was established at 8 hp for the engine used. With conversion to propane, the

engine can only produce 7.2 hp because of the 10% power loss due to conversion.

## **A-2 Fuel Supply**

Using fuel consumption characteristics of the engine, described in the manual by the manufacturer, for the engine powered by gasoline, the fuel consumption is tabulated as 0.87 gallons/hour at optimal speed. The required fuel supply is calculated below for the worst case situation of continuous full throttle.

$$0.87/2 * 8 \text{ pints/gallon} = 3.5 \text{ pints of propane fuel}$$

Since the selected bottles contain more than 1 pint of vapor propane fuel each, the amount of bottles needed for continuous running will be 3.

# **APPENDIX B**

## **HYDRAULIC MOTOR SELECTION**

### **Introduction**

Before a hydraulic motor could be selected, the required performance of the motor had to be determined. This performance consists of required torques and at what speeds these torques should be applied. These criteria were then used to select a motor.

### **B.1 Equations used to Calculate Tractive Effort**

The first step in determining the required torques was to determine the tractive force that the driven wheels needed to exert on the ground. This tractive force is called the Tractive Effort (TE). The Tractive Effort is determined from several different factors: rolling resistance (RR), grade resistance (GR), accelerating force (F), and drawbar pull (DP).

$$TE = ( RR + GR + F + DP ) \times 1.10$$

The Tractive Effort is the sum of the factors with an additional allowance of 10% to account for starting friction.

#### **B.1.1 Rolling Resistance**

The rolling resistance is defined as the product of the gross vehicle weight and the rolling resistance factor (R). The rolling resistance factor varies for different surface types. The rolling resistance factor for dune sand (0.3) was chosen as the appropriate value of R because dune sand is a comparable lunar-type surface.

$$RR = GVW \times R$$

### **B.1.2 Grade Resistance**

The grade resistance is a measure of the force required to move a vehicle up a grade at constant speed. It depends only on the gross vehicle weight and the percent of grade.

$$GR = GVW \times (\% \text{ grade})$$

### **B.1.3 Accelerating Force**

The accelerating force is the necessary force to increase vehicle speed to a desired speed in a certain period of time.

$$F = (V \times GVW) / (T \times 32.16)$$

V = desired change in speed

T = accelerating time period

### **B.1.4 Drawbar Pull**

The drawbar pull is the amount of force that the vehicle can exert on a load. The ENABLER will not be designed to pull a load.

$$DP = 0$$

## **B.2 Calculation of Tractive Efforts for the Three Requirements**

As stated before, the ENABLER drive system has three performance requirements:

1. Achieve a top speed of 7 mph on a level surface
2. Travel up a 30% grade from a dead stop
3. Surpass a one meter high obstacle

Naturally, the Tractive Efforts for each of the three requirements were different and were found separately as follows.

### **B.2.1 Seven mph Top Speed On Level Surface**

For the first performance requirement of 7 mph top speed, an accelerating time period of 15 seconds was chosen. A more sluggish value would not have been beneficial because the magnitude of T does not have a substantial impact on the final value of TE. The grade resistance is zero; therefore the Tractive Effort for this performance requirement ( $TE_{level}$ ) is a function of the rolling resistance and accelerating force.

$$F = [(10.27 \text{ ft/sec}) \times (1500 \text{ lb})] / [(15 \text{ sec}) \times 32.16] = 32 \text{ lb}$$

$$RR = (1500 \text{ lb}) \times (0.3) = 450 \text{ lb}$$

$$TE_{level} = (F + RR) \times 1.10 = (450 + 32) \times 1.10 = 531 \text{ lb}$$

### **B.2.2 30% Grade Climb**

There were no given speed or acceleration constraints for the grade climb. Those values were decided from the following design considerations. As stated in section A.2.1, the acceleration time is not an important factor; a value of  $T=15$  sec was assumed.

To find an appropriate value for the design speed of the climb, two different speed values were used to see the effect the speed has on the resulting Tractive Effort:

$$7 \text{ mph} : TE_{7\text{mph}} = 1025 \text{ lb}$$

$$3 \text{ mph} : TE_{3\text{mph}} = 1005 \text{ lb}$$

As can be seen, the speed of the climb has a negligible effect on the resulting Tractive Effort. In contrast though, the speed does have an effect on the required power from the motor. Since the required power is a function of the required torque and the wheel speed,

$$\text{Power} = \text{Torque} \times \text{wheel speed}$$

twice as much power is needed for the greater speed. Therefore, 3 mph was chosen as the design speed of the climb.

$$RR = (1500 \text{ lb}) \times 0.3 = 450 \text{ lb}$$

$$GR = (1500 \text{ lb}) \times (30\%) = 450 \text{ lb}$$

$$F = [(4.4 \text{ ft/sec}) \times (1500 \text{ lb})] / [(15 \text{ sec}) \times (32.16)] = 13.68 \text{ lb}$$

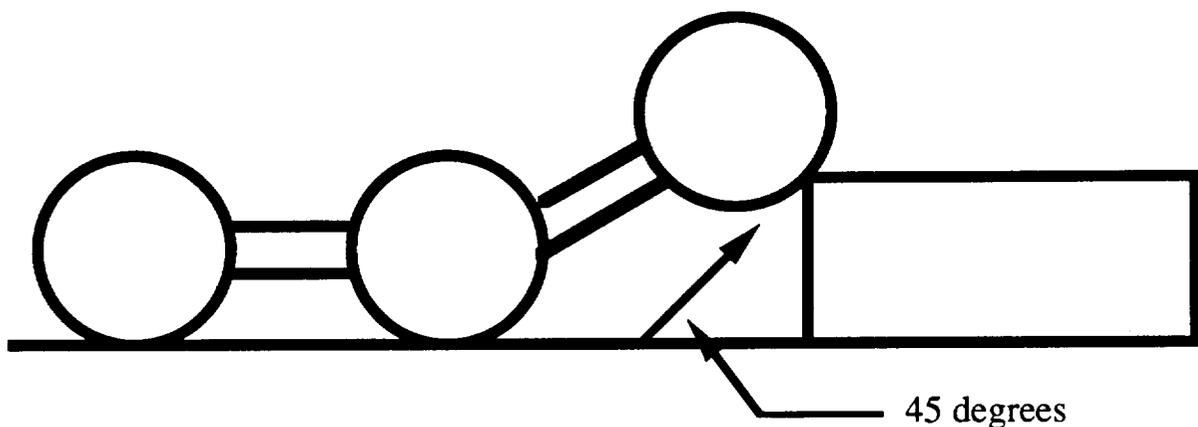
$$TE_{\text{grade}} = (RR + GR + F) = (450 + 450 + 13.68) = 1005 \text{ lb}$$

### B.2.3 One Meter High Obstacle

To overcome the one meter high obstacle, the following assumptions were made. The articulation joints would be used to pick up the front axle. This is a very useful luxury because the front wheels would then hit the obstacle at an angle instead of head on. A speed of one mph in an accelerating time period of 5 sec was also assumed.

Figure B.1

Surpass One Meter High Obstacle



The articulation joints would also be used to aid with the second and third axles in the same fashion.

As can be seen from the drawing, one axle is at a 45° angle (100% grade) climb while the other two are at no climb. For this reason, the TE for each individual axle was found, and then the three TE's were summed for a total TE.

$$TE = [4 \times (RR_W + F_W) \times 1.10] + [2 \times (RR_W + GR_W + F_W) \times 1.10]$$

$$RR_W = \text{rolling resistance per wheel} = (250 \text{ lb}) \times (0.3) = 75 \text{ lb}$$

$$F_W = \text{accelerating force per wheel}$$

$$= [(22/15 \text{ ft/sec}) \times (250 \text{ lb})] / [(5 \text{ sec}) \times 32.16] = 2.28 \text{ lb}$$

$$GR_W = \text{grade resistance per wheel for climbing axle}$$

$$= (250 \text{ lb}) \times (100\%) = 250 \text{ lb}$$

$$TE_{\text{obstacle}} = 1060 \text{ lb}$$

### **B.3 Calculation of Torques for the Three Requirements**

For useful analysis, the three performance criteria now needed to be converted from Tractive Efforts to torques. Tractive Effort is converted to Torque through the simple relationship:

$$T = (TE \times r) / (G \times N)$$

$$r = \text{radius of wheel} = 21.0625 \text{ inches}$$

$$G = \text{gear ratio between motor and wheel (to be decided)}$$

$$N = \text{number of motors} = 6$$

A proper gear ratio needed to be selected. To determine a suitable gear ratio, the resulting required torque and speed ranges were found for five different gear ratios. From these calculations, the most practical gear ratio could then be selected. The results of these calculations are shown in Table B.1. The gear ratios range from 1:1 (no gear ratio between motor and wheel) to 40:1. This range is broad enough to include all practical situations.

Table B.1

Resulting torques and motor speeds with different gear ratios

Torques (in-lb) at Corresponding Motor Speeds (RPM)

gear ratio:	1:1	10:1	20:1	30:1	40:1
T <sub>level</sub> (in-lb)	1860 at 56	186 at 559	90 at 1117	62 at 1677	47 at 2236
T <sub>grade</sub> (in-lb)	3530 at 24	353 at 240	177 at 479	118 at 720	89 at 960
T <sub>obstacle</sub> (in-lb)	3721 at 8	372 at 80	186 at 160	124 at 240	93 at 320

#### B.4 Motor Selection

Table B.1 shows the five different possible scenarios (for the five different gear ratios) of required torques. Now the motor selection process would seem to reduce down to finding a motor whose performance correlates to one of the scenarios. Unfortunately though, most motor catalogs do not give motor performance specifications in (in-lb); the catalogs rate motors by fluid displacement per motor revolution (in<sup>3</sup>/rev). Therefore, the required torques in Table B.1 were converted to fluid displacement per revolution with the relationship:

$$D = (T \times 6.2832) / P$$

T = required torque (in-lb)

P = operating motor pressure (psi)

D = fluid displacement per revolution  
(in<sup>3</sup>/rev)

The motors and pump will be operating at constant pressure. A pressure of P=2400 psi was assumed as the design pressure. Since the pressure is constant, the displacement becomes proportional to torque. The calculated required displacements can be seen in Table B.2.

Table B.2

Required Displacements and Motor Speeds for different gear ratios

Displacements (in<sup>3</sup>/rev) at Corresponding Motor Speeds (RPM)

gear ratio:	1:1	10:1	20:1	30:1	40:1
D <sub>level</sub>	4.87 at 56	.487 at 560	.236 at 1117	.162 at 1677	.123 at 2236
D <sub>grade</sub>	9.24 at 24	.924 at 240	.463 at 479	.309 at 720	.233 at 960
D <sub>obstacle</sub>	9.74 at 8	.974 at 80	.487 at 160	.325 at 240	.243 at 320

Now a motor could be selected by finding a motor whose fluid displacement and RPM specifications match closely with one of the five columns in Table B.2. After reviewing various catalogs, the selection was made:

Selected Motor : brand: Char-Lynn  
 model: H-Plus Series, 2.2 in<sup>3</sup>/rev displacement  
 gear ratio: 10:1

Our selected motor is a fixed displacement motor. A possible alternative was to use a variable displacement motor which could have been set for a low range and a high range. The low range would have accommodated the obstacle and grade climbs, and the high range would have accommodated the level top speed. A variable displacement motor was not selected because one small enough could not be found.

# APPENDIX C

## HYDRAULIC PUMP SELECTION

### Introduction

The hydraulic pump serves to power the six hydraulic motors. Therefore, pump selection was based on the flow requirements of the motors.

### C.1 Flow Requirements

The first step in determining what flow is required from the pump was to determine what flow each motor will require. The required motor flow is a function of the required fluid displacement per motor revolution (found in Appendix B, Table B.2) at the corresponding motor speed:

$$F = (D \times \text{RPM}) / 231$$

$$F = \text{required flow (GPM)}$$

$$D = \text{required displacement (in}^3\text{/rev)}$$

$$\text{RPM} = \text{corresponding motor speed}$$

The required motor flow was then calculated for the three performance criteria:

$$F_{\text{level}} = 1.18 \text{ GPM}$$

$$F_{\text{grade}} = 0.96 \text{ GPM}$$

$$F_{\text{obstacle}} = 0.338 \text{ GPM}$$

The total flow required from the pump was then found by multiplying the required flow per motor times the number of motors (6).

$$(\text{required total flow from pump})_{\text{level}} = 7.08 \text{ GPM}$$

$$(\text{required total flow from pump})_{\text{grade}} = 5.76 \text{ GPM}$$

$$(\text{required total flow from pump})_{\text{obstacle}} = 2.03 \text{ GPM}$$

## C.2 Pump Selection

As with the hydraulic motors, pump performance ratings in pump catalogs are given in fluid displacement per pump revolution. Therefore, the required pump flows found in Section C.1 had to be converted to pump displacements per pump revolution:

$$PD = (231 \times PF) / \text{RPM}$$

PD = pump displacement per pump  
revolution ( $\text{in}^3/\text{rev}$ )

PF = required pump flow (GPM)

RPM = constant pump speed = (3600)

$$PD_{\text{level}} = 0.511 \text{ (in}^3/\text{rev)}$$

$$PD_{\text{grade}} = 0.416 \text{ (in}^3/\text{rev)}$$

$$PD_{\text{obstacle}} = 0.147 \text{ (in}^3/\text{rev)}$$

Since the pump will be ran at constant speed and pressure, a variable displacement pump equipped with a pressure compensator is necessary.

Selected Pump : brand: Vickers

model: PVB5 pressure compensated for 2400 psi

## APPENDIX D

### HYDRAULIC LINE SELECTION

The hydraulic lines connect the hydraulic pump to the hydraulic motors. The only two design considerations for the line were size and flexibility. The line has to be large enough for the required flows. Flexibility is also desirable for ease of installation and maintenance. The required size of the line is calculated using the maximum required flow (found in Appendix C) and an assumed flow velocity.

$$A = F/V$$

$$A = \text{required cross sectional area (in}^2\text{)}$$

$$F = \text{maximum required flow} = 7.08 \text{ (GPM)}$$

$$= 27.26 \text{ (in}^3\text{/sec)}$$

$$V = \text{flow velocity} = 360 \text{ in/sec}$$

From the required cross sectional area, the required line diameter was found:

$$A = \pi d^2/4$$

Selected Line : 3/8 inch diameter hydraulic hose

# APPENDIX E

## SPEED REDUCER

### Introduction

Roller chains are designed and rated for continuous operation and many hours of service. Ratings are limited by a fatigue factor due to the cyclic stressing of the chain riding up and down the gear tooth when engaging the sprocket. Roller chains seldom fail because they lack tensile strength, rather, failure is usually in the form of the roller surface wearing out and elongation of the chain. (Shingley; p. 682)

### E.1 Selection Discussion

In our application we were designing for demonstration purposes and low life expectancy. Chain tension and available space were the only design consideration for selecting a chain.

Max torque required at wheel: 3,800 in-lbs

Radius of large sprocket: 5.725 in

Tension in chain: 665 lbs

The selected design of an single standard-ANSI 35 Chain has an average tensile strength of 2,100 lbs. Due to space limitations it was required to use a drive sprocket with only 12 teeth. For normal roller chain operation more drive sprocket teeth are desirable to minimize chain pulsation. Again, for our case of low speed, low duty cycle and low life expectancy, a drive sprocket with 12 teeth is adequate.

The bearing and sprocket assembly arrangement required an extension tube between the sprocket and bearing plate. The following analysis demonstrates that the design is adequate.

Max stress in pipe due to motor torque:

$$T = \frac{T c}{J}$$

T = Torque  
c = Pipe O.D.  
J = Polar Moment of Inertia

$$J = \frac{\pi \times c^4}{2} - \frac{\pi \times b^4}{2}$$

Where b = Pipe I.D.

$$T = 3800 \text{ lbs} \quad b = 3.0625 \text{ in}_{\text{max}}$$

$$c = 3.25 \text{ in}$$

$$J = 37.08 \text{ in}^4$$

$$T_{\text{max}} = 333.3 \text{ psi}$$

Max allowable stress for steel 14,500 psi (Popov; p. 554)

The selected pipe design can handle the torque requirements.

### E.3 Alternatives Explored

A flange mounted helical reducer was initially determined to be a suitable product for our design. The efficiency, general simplicity of mounting, and enclosed gear design of these units were considered desirable.

The helical reducer alternatives located did not match our size and weight criteria. The units designed for hydraulic motor interface were all too large and too heavy. Other units located were not readily adaptable to hydraulic motor flange mounts.

The roller chain reducer selected met our size and weight requirements well. The best advantage being the relatively short axial distance required. Although a roller chain was selected, it has certain drawbacks. Initial mounting and necessary adjustments associated with a roller chain require more attention than is desirable. Efficiency is in the low range of reducer efficiency at 85 to 90 percent.

A helical reducer may exist which would be more desirable than the selected roller chain design. An all inclusive search of reducers may locate such a reducer.

# APPENDIX F

## CONICAL WHEEL DESIGN

### Introduction

To perform satisfactorily, the carbon fiber epoxy resin composite wheel had to withstand the forces exerted on it. It also needed to deform to absorb any shocks experienced by the vehicle. Once the composite material was chosen, a deflection and force analysis was performed to verify that these criterion were met.

### F.1 Deflection Analysis

To determine the maximum stresses and bending moments experienced by the wheel, a deflection model was constructed. This was done by assuming a maximum allowable deflection of  $1/3$  of the wheel radius, approximately 0.18 m ( 7 in ). One sixth of the circumference of the wheel was assumed to deform to the ground contour by flattening. The upper third of the wheel does not deform. The rest of the wheel bulges out to accommodate the flattening of the bottom of the wheel. The bulge was modeled as a straight line running at a 60 degree angle tangential off the non-deformed upper portion of the wheel. The rest of the bulge was then modeled by a circle tangential to both of these lines.

For the bending moment analysis, the change in radius was converted into a deflection from a normal line, the undeformed radius of the tire. This corresponded to a maximum deflection of 0.02 m for a 0.25 m span of material. The deformable bodies theoretical model was a beam simply supported on both end under a single bending moment at one end. The deformation equations are as follows:

Figure F.1  
Deflection Analysis

$$v_{\max} = \frac{-M_0 L^2}{9\sqrt{3} E I} \quad \phi(0) = \frac{-M_0 L}{6 E I}$$

$$I = \frac{bh^3}{12} = 4.333 * 10^{-8} \text{ m}^4 \quad \phi_{\max} = \frac{-M_0 y}{I}$$

**F.2 Material Properties**

High-strength Graphite / epoxy unidirectional Composite

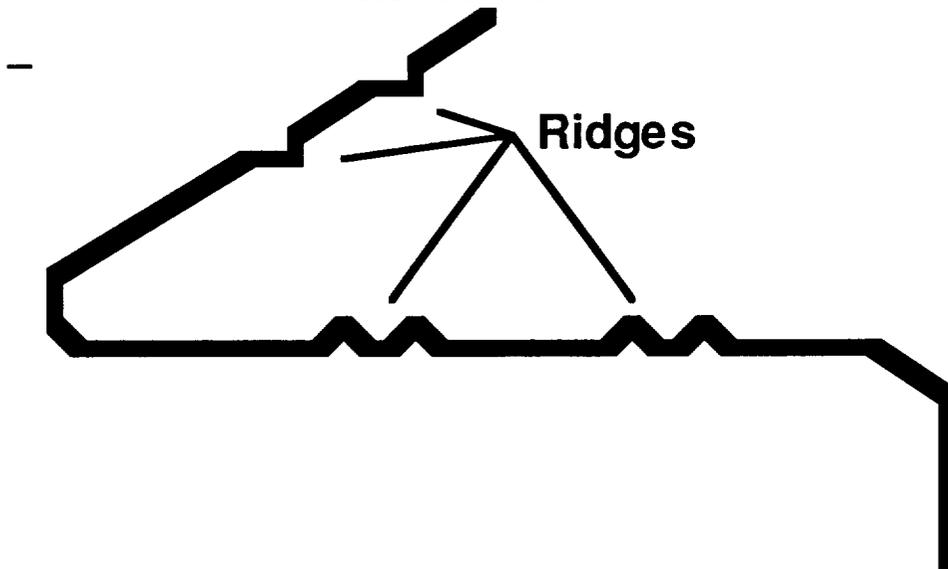
<u>Elastic Constants</u>	<u>G Pa</u>	<u>*10<sup>6</sup> psi</u>
Longitudinal Modulus, E <sub>L</sub>	145	21
Transverse Modulus, E <sub>T</sub>	10	1.5
Poisson's Ratio, ν	0.25	
 <u>Strength Properties</u>		
Longitudinal Tension, F <sub>L</sub>	1240	180
Transverse Tension, F <sub>T</sub>	41	6
Density, kg/m <sup>3</sup> ( lb/in <sup>3</sup> )	1.58*10 <sup>3</sup> ( 0.057 )	

### F.3 Wheel Hub Design Alternative

The current hub design is a 1/8 inch thick aluminum disk. This weighs approximately 12.41 kg (27.3 lbs). To cut this weight in half, the thickness of the hub can be cut in half. However, doing this will greatly compromise the strength of the hub. Therefore, the hub will have to be redesigned to include small circumferential ridges in the region parallel to the wheel cylinder and the region adjacent to the wheel. These ridges will increase the hub strength and make this lighter design plausible.

Figure F.2

Wheel Hub Alternative



# **APPENDIX G**

## **MOUNTS**

### **G.1 Engine Mount**

The engine mount will support the engine inside the rear T section of the vehicle and is shown in CAD Drawing I-5. The three curved stands on the bottom of the mount will be welded to the inside surface of the T section. The mount will be made from 1/4" stock aluminum plates. The bottom of the motor will be bolted onto the mount using six 5/16" diameter bolts.

### **G.2 Propane Bottle Mount**

The propane bottle mount will support the propane bottles inside the rear T section of the vehicle near the engine. The propane bottle mount is shown in CAD Drawing I-9 and will be made from 1/4" inch stock aluminum plates. Pipe clamps will hold the propane bottles in place.

### **G.3 Hydraulic Pump Mount**

The pump mount will support the pump inside the rear T section of the vehicle and is shown in CAD Drawing I-6. The three curved stands on the bottom of the mount will be welded to the inside surface of the T section. It will be a flange-type mount consisting of 1/4" stock aluminum plates, two 1/4" steel supporting rods, and a 1/2" thick rubber seat to support the rear end of the pump. Two 1/2" diameter grade 8 bolts connect the pump to the mount. The pump shaft will extend through the large center hole on the front face of the mount.

# APPENDIX H

## WEIGHT ANALYSIS AND PARTS LIST

### Introduction

Since weight is an important consideration for the ENABLER vehicle, a weight analysis was performed on the entire wheel and wheel drive system. The entire vehicle weight is specified to be around 1500 pounds when fully assembled. The weight corresponding to the wheel and wheel drive system came out to be 1113 pounds. This was much larger than had been hoped for in the design process due to the overdesign of the plates and mounts in the system. This could be lowered by approximately 20% if lighter metals, such as titanium, are used and more design considerations are made on each component.

### H.1 Weight Analysis

	<u>kg</u>	<u>lbs</u>
<u>Wheel Assembly</u>		
Carbon fiber/epoxy resin wheel (0.0075 m thick)	20.4	44.9
Aluminum Hub (1/8" thick)	12.4	27.3
Aluminum Sprocket Plate (1/4" thick)	4.9	10.9
Sprocket (large)	3.5	7.7
Motor	4.9	10.7
Sprocket (small)	0.1	0.2
Chain	1.0	2.2
Bearing (steel)	14.3	31.4
Chassis (1/8" Al skin)	8.5	18.7
V-band Clamp	3.5	7.8
Motor Mount (Al)	<u>2.5</u>	<u>5.4</u>
Wheel Assembly Total:	76	167.2
<u>Engine</u>		
Engine	20.5	45
Propane Bottle (3 at 1 lb each)	1.4	3
Mounting (Al)	<u>1.4</u>	<u>3</u>
Engine Total:	23.3	51
<u>Pump and Lines</u>		
Pump	7.3	16
Lines	4.5	10
Mount	2.3	5
Reservoir	<u>2.3</u>	<u>5</u>
Pump and Lines Total:	16.4	36

The Total Weight of the Wheel and Drive System: 496 kg or 1090 lbs.

# APPENDIX I

## LISTING OF CAD DRAWINGS

### **I-1 Hydraulic Schematic**

This drawing provides an overall schematic of the hydraulic system pertaining to the wheel and wheel drive. Shown are the major equipment, all associated valves and hydraulic lines.

### **I-2 Wheel Detail**

Provided in this detail are all pertinent wheel dimensions. A side and front view are given for visual understanding of the design.

### **I-3 Wheel Section**

The view shows the relative positioning of major equipment in the wheel tube. Also, included are the placements of all mounting apparatus within the wheel.

### **I-4 Rear Chassis Section**

This section gives one a conceptual understanding of the relative positioning of the equipment located in the rear portion of the chassis.

### **I-5 Engine Mount Detail**

This detail gives a visual interpretation of the engine mount with dimensions needed for fabrication and mounting.

### **I-6 Pump Mount Detail**

Furnished on this drawing are vital dimensions necessary to fabricate and install the pump mount.

### **I-7 Motor Mount Detail**

All needed dimensions of the motor mount are provided on this detail.

## **I-8 Bearing Connections**

All attachments to the wheel bearing are shown in this close up view of the wheel cylinder.

## **I-9 Propane Bottle Mount**

The dimensions of the propane mounting apparatus are shown in this particular detail.

