MUSCLES

ManeUverable Self-Contained Launch and rEcovery System

Julie Hollenberg
Luke Lyndaker
Vincent Rizzo
Mike Spottswood
Daniel Shores

University of New Hampshire

Undergraduate Ocean Research Project

Dr. Gerald Sedor, Advisor

1994 - 1995 $7

UNHMP-AR-SG-95-7

Sea Grant
New Hampshire/Maine

FORWARD

"Imagination is more important than knowledge."

- Albert Einstein

TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

page

EXECUTIVE SUMMARY

The mission of the MUSCLE System is to transport a human powered submersible across roads and beaches, launch the submersible at a depth of four to five feet, and recover the submersible after its mission has been completed. This mission and the mission requirements are based on the UNH experience in three international competitions for human powered submersibles. The primary design goal was to achieve a high degree of safety and reliability during all modes of operation with only two operators.

The MUSCLE System is comprised of separate land and water platforms. The land platform is a gasoline powered vehicle which carries the water platform, or cart system. The gasoline powered vehicle was designed to transport the submersible and cart over various terrains. The cart cradles the submersible and is used to deploy the submersible from the land vehicle to an acceptable launching depth. This cart system, which is not powered, was designed to facilitate the submersible launch and recovery, as well as to support the submersible for maintenance.

The land vehicle consists of the frame, the drive system, and the maneuvering system. The drive system includes a gasoline engine, brakes, transmission, trans-axle, and rear axle. The frame system supports and houses all other systems while ensuring the stability of the land vehicle under all operating conditions during the mission. The maneuvering system incorporates a steering system and an independent front end suspension system. Although the cart is independent from the land vehicle, its design is

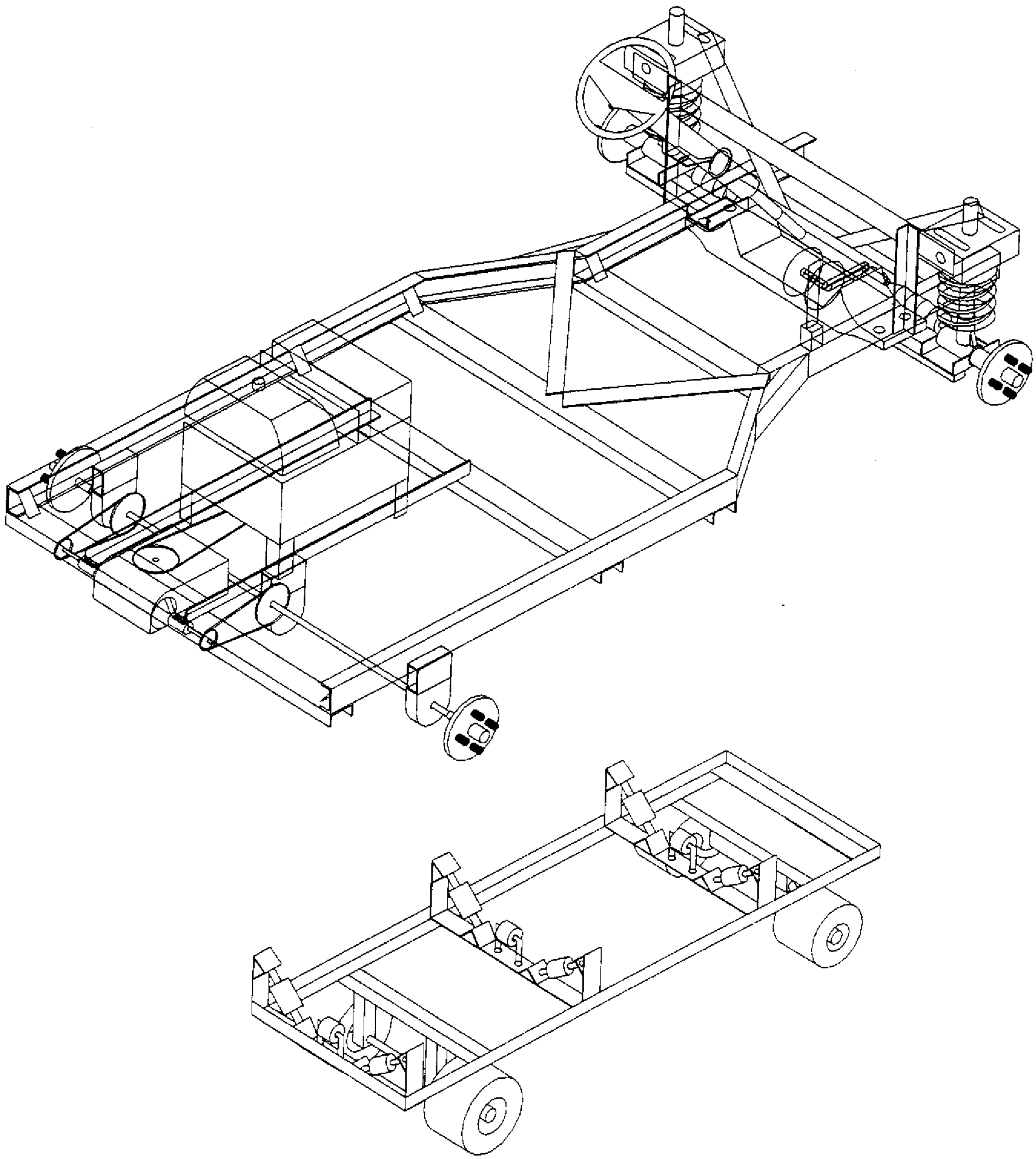
constrained by the dimensions of the land vehicle and the submersible.

The MUSCLES design effort involved the development of several alternative concepts to meet the mission requirements, as well as meet budgetary and time constraints. A mission scenario was developed, based on past experiences at the International Submarine Races. A prioritized design philosophy was developed to guide the design and construction process. A test program was also developed to verify system operability.

To meet the budgetary limitations, the MUSCLES team adapted components from a used garden tractor for the drive system and parts from a compact automobile for the maneuvering and suspension systems. Scrap steel obtained at no cost was used to fabricate most of the structural members of the frame and cart systems. Welding of the structural members was accomplished primarily by MUSCLES team members.

Adaptation of components from other systems resulted in some operational problems, which were identified during system testing and resolved by the MUSCLES team. The final product is a safe, reliable, and cost-effective system which meets the established mission requirements. The major subsystems of the MUSCLE System are shown in Figure 1.

Figure 1



MUSCLES

UNH University of New Hampshire Engineering Dept. Durham, NH

DESIGN M.U.S.C.L.E.S.	SCALE NTS
DRAWN Mike Spottswood	DATE 3-30-95

BACKGROUND

The International Submarine Races, originating in 1989, provide the opportunity for various industries, government agencies, and individuals to design, build, and compete with human powered, free flooding submersible vehicles. The competition is held every other year. Students from the University of New Hampshire have successfully competed in all three competitions held to date. One of the needs of the UNH team, which has not been met in the past, is a system capable of transporting, launching, and recovering the submarine in an efficient manner. This project was developed to meet this need.

The MUSCLE System mission design requirements and operational scenario were developed from the UNH experience at the first three International Submarine Races, which were held in Florida in shallow water ocean race sites adjacent to sandy beach areas. The difficulties encountered in transporting the submersible from a paved area, across soft sandy beaches, and then launching and recovering the submarine in ocean surf conditions clearly presented a design challenge. This demand had to be met by a system which was safe, reliable, adaptable to various submarine sizes and shapes, and capable of being built within the established budgetary limitations. This report describes how this challenge was met.

MISSION DESCRIPTION

The mission of the MUSCLE System is to provide an effective system for the transport, launch, and recovery of a human-powered submarine in its operating environment. This environment includes the ocean and beach areas associated with the International Submarine Races, and the lake side and pool side environments associated with the testing grounds in the close vicinity of the University of New Hampshire.

The following is a list of the mission requirements and criteria established by the MUSCLES project team.

* Transport submersible over paved and unpaved roads, and sandy beaches
* Transport submersible through surf to a depth of 4 to 5 feet
* Achieve a high degree of maneuverability on land and in water
* Maintain stability under all operating conditions
* Deploy and recover submersible in ocean environment
* Handle a submersible up to 3 feet in diameter and 16 feet in length
* Fit into a standard large commercial van
* Self-contained subsystems
* Reliable operations
* Capable of being operated by two persons
* Maximum personnel safety for all operations
* Materials compatible with the various operating environments
* Cost effective

MISSION SCENARIO

The MUSCLE System has been designed and built to support the transportation, launch, and recovery operations of a human-powered submarine in an ocean area with adjacent sandy beaches.

Once the human powered submarine is loaded on board the MUSCLE System and securely strapped to the cart, the entire system is transported to the site of the submarine operations via a large standard van. The cart is secured to the land vehicle by means of a pin and brace located on each end of the cart. The entire system, with the submarine, can propel itself up a ramp into the van and back down the ramp.

The MUSCLE System with the submarine is driven by one operator and one assistant from a paved area, where the van is located, to the water's edge. The MUSCLE System design provides for maneuverability over hilly terrain as well as sandy beach areas. Once at the water's edge, ramps are removed from their storage location within the MUSCLES frame and attached to the rear end of the frame to provide a path for the cart. The cart, loaded with the submarine, is then manually moved by the assistant. This process is controlled by the operator using the hand winch and cable attached to the cart.

When the cart and submarine are water borne and clear of the land vehicle, the straps securing the submarine to the cart are removed and the submarine is then prepared for operation by the submarine crew. The empty cart is retracted back to the land vehicle by means of the hand winch.

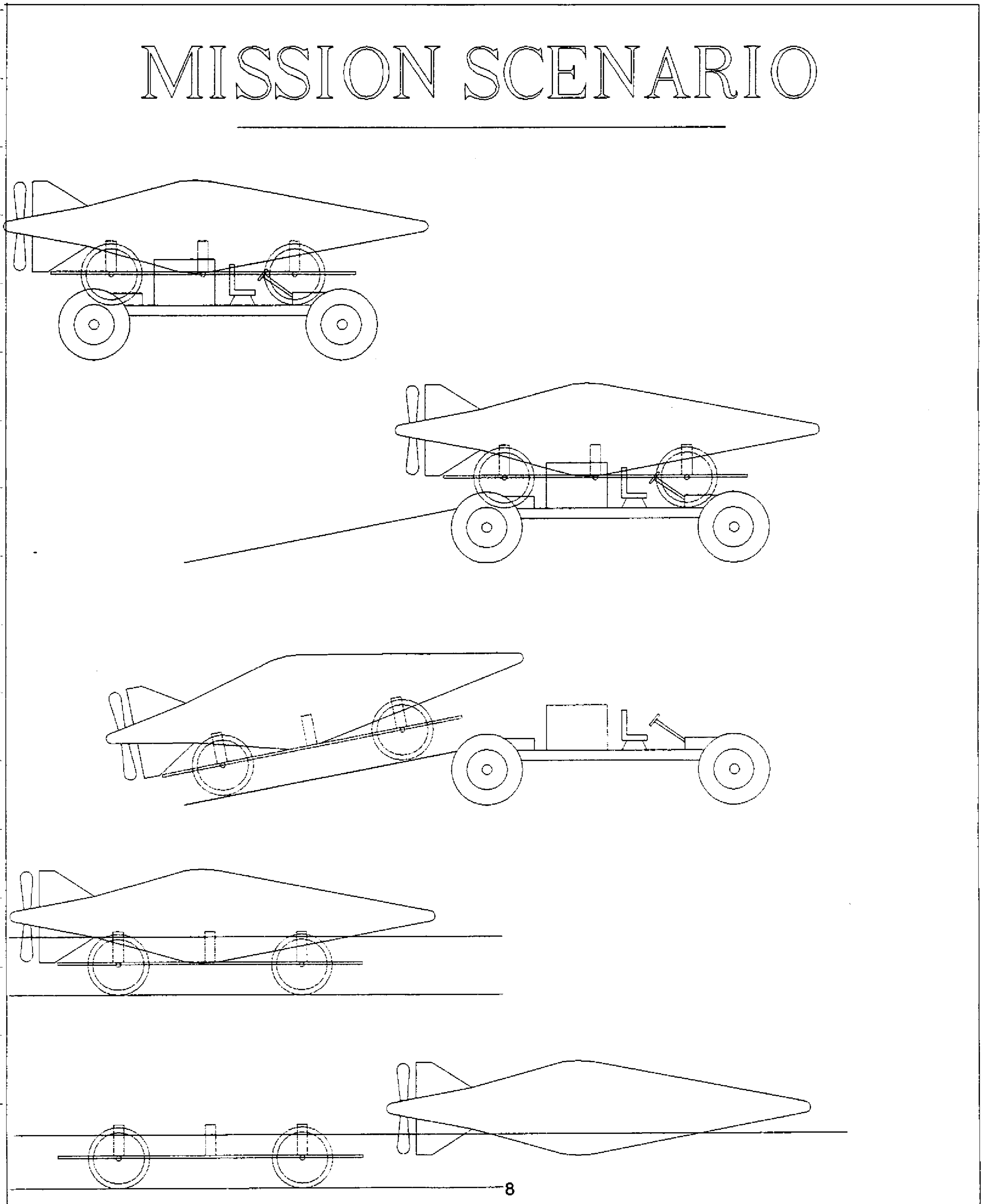
At the conclusion of submarine operations, the cart is repositioned to a predetermined recovery area and the submarine is loaded onto the cart by the submarine crew. The submarine is strapped to the cart and is then slowly hauled out of the water by means of the hand winch. Since the submarine is free-flooding, the entrapped water inside its hull will drain as the submarine is hauled out.

The winch operator will slowly haul the submarine and cart onto the loading ramps and up onto the land vehicle. To minimize strain on the winch and the overall MUSCLE System, this recovery operation must be carefully controlled to allow for natural dewatering of the submarine by gravity as it is being hauled out.

Once the dewatered submarine and cart are winched into the proper location on the land vehicle, the cart is secured to the vehicle frame to preclude motion of the cart during the transit back to the van area. The system is then relocated by means of the single operator located on the vehicle and an assistant walking alongside.

Figure 2 depicts some of the aspects of deploying a submarine using the MUSCLE System.

Figure 2



DESIGN PHILOSOPHY AND REQUIREMENTS

Numerous design approaches exist which are capable of satisfying the mission requirements and criteria. In selecting the optimal design to effectively accomplish the mission, the established design requirements were analyzed to determine the importance of each and the priority of each with respect to one another. The following design priorities were established as guidelines for the design process and provided a basis for evaluating various design options.

1 - Safety

2 - Size

3 - Maneuverability

4 - Simplicity and Reliability

5 - Cost/Material Requirements

INTERPRETATION OF DESIGN REQUIREMENTS

Safety - The MUSCLE System, with and without the submarine, must be safe and highly stable under all operating conditions. The risk of injury to operating personnel and bystanders is to be minimized at all times. The use of hazardous materials is also minimized to protect the environment and operating personnel.

Size - The MUSCLE System must be large enough to handle a submarine with a diameter up to 3 feet and a length up to 16 feet. The system must also be small enough to fit inside a large commercial van or truck.

Maneuverability - The land vehicle must achieve sufficient maneuverability to transport the submersible over roads and over sandy beach areas. The cart system must be easily guided by two persons to facilitate the launch and recovery process and the maneuvering of the submersible on land.

Simplicity and Reliability - The simplicity of the system design is maximized to accommodate ease of operation, repair, and optimize reliability. As the system is the sole means of retrieving the submersible, reliability is an essential concern.

Cost/Material Requirements - The system must be capable of being manufactured on a relatively small budget.

FRAME AND STRUCTURE SYSTEM

FUNCTIONAL REQUIREMENTS

The frame houses and supports all of the MUSCLES subsystems employed during the mission. The frame is required to withstand all external forces acting on the vehicle while the system is been driven across paved and unpaved roads, and sandy beach areas. The dimensions of the frame are constrained by the size of a commercial truck as the system must be transported long distances to the submarine testing grounds and competition areas. The frame must also be designed to allow for full steering range of the front wheels.

EVALUATION FACTORS

The frame of the land vehicle is required to sustain the weights of the submarine, the cart, and the various subsystems located on the land vehicle. During the transport of the submersible, the frame will encounter multidirectional forces. From these forces, the weight of the system, the longitudinal torsional forces, and the longitudinal shearing forces were concluded to be the limiting causes of stress and deformation on the frame.

MATERIAL SELECTION

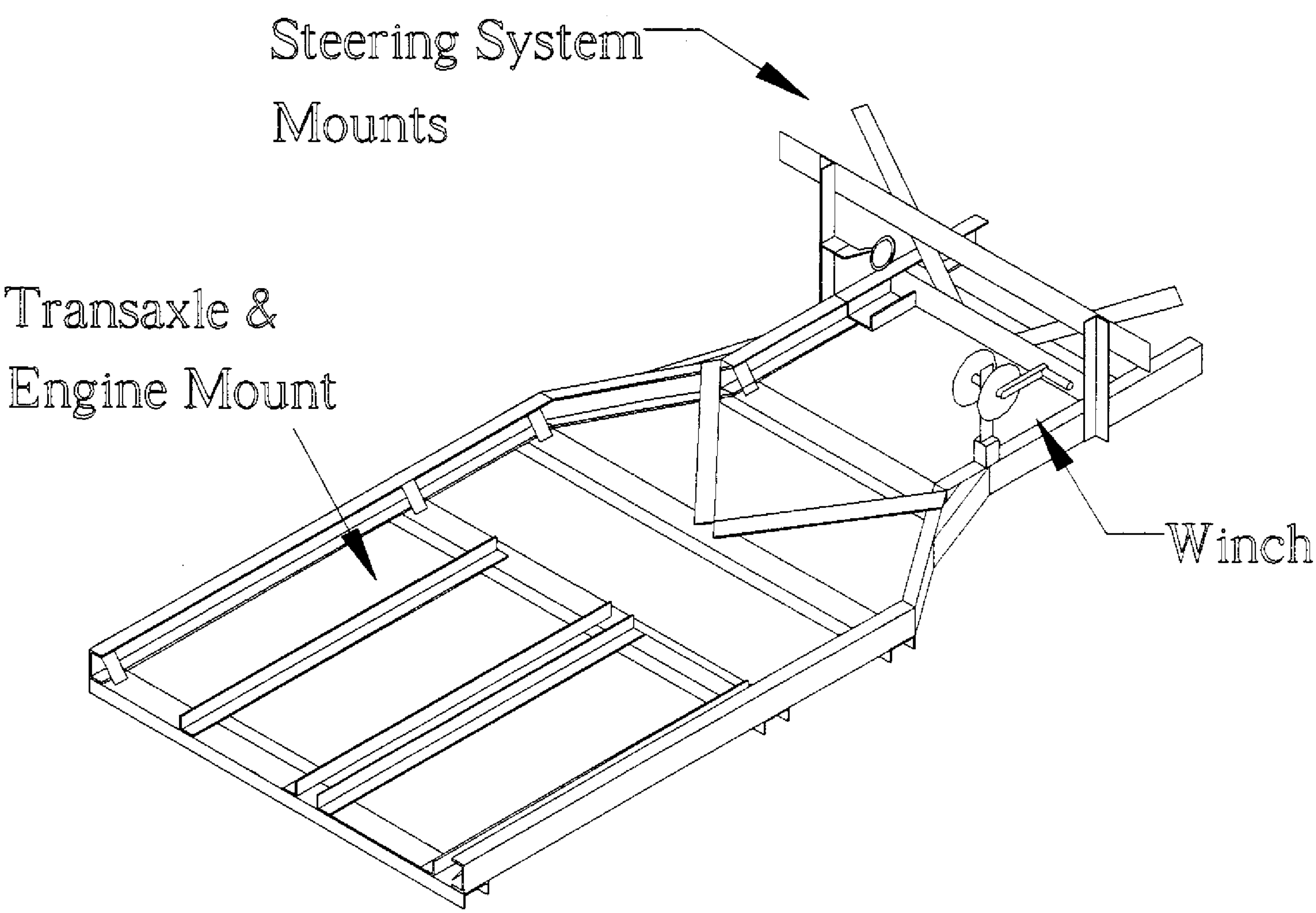
When the construction material for the frame was being selected, aluminum and steel were both under consideration. Steel was chosen based on the fact that steel is easier to weld and more workable than aluminum. Also, the cost of aluminum far outweighed the benefits of its light weight.

DESIGN ALTERNATIVES

Several designs were developed which satisfied the mission requirements. In an early concept, MUSCLES was designed to be positively buoyant and the submarine was to be launched from the midsection of the frame into the water. In this concept, the frame was a basic 'A' frame with all subsystems raised well above the water. Another alternative design using the 'A' frame configuration required MUSCLES to be negatively buoyant with all subsystems to be maintained above the water. These amphibious systems had the option of incorporating common land and water propulsion or separate land and water propulsion. The systems considered included those which could be operated by an on board operator, as well as those incorporating remote control. The amphibious systems offer greater launching and recovery capabilities beyond surf areas and a higher level of ocean maneuverability. However, these systems are very costly and complex. The final design concept selected for MUSCLES consists of separate land and water platforms. The land platform is a powered platform which remains on the shore, and the independent cart, or water platform, deploys the submersible from the shore to a desired launching depth and brings the submersible back to shore once it has completed its mission. The cart is controlled through a hand winch system mounted on the land platform. The advantages of this system include design flexibility, acceptable water maneuverability, and optimization of land and water functions at reduced cost and complexity. Figures 3 & 4 are schematics of the MUSCLES frame system.

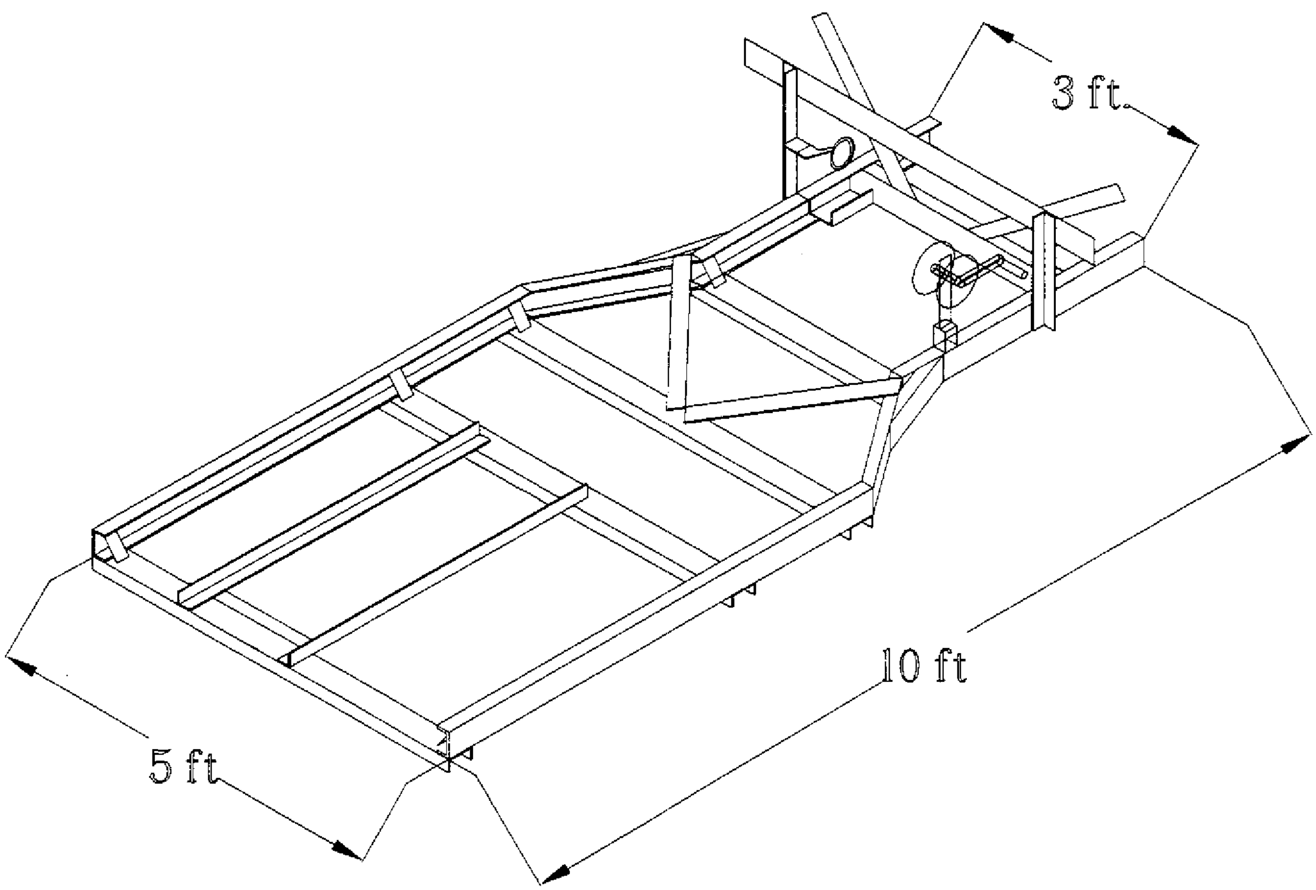
Figure 3

Frame



Frame Sides are 4 in. C-Channel
Cross Members are 4 in. C-Channel

Figure 4



MUSCLES - FRAME

UNH University of New Hampshire Engineering Dept. Durham, NH

DESIGN M.U.S.C.L.E.S. SCALE NTS

DRAWN Mike Spottswood DATE 3-30-95

DESIGN ANALYSIS

A three dimensional finite analysis was used to compute the deflection of the frame under uniformly distributed weight and torsional loads. The software package used for this analysis was Visual Analysis.

Visual Analysis requires the input of nodal coordinates for each frame member end point and member intersection. The user must specify the manner in which the nodes are fixed, in the x, y, or z direction. The frame members are generated by specifying a beginning node and an ending node. The program allows the user to select the member material (structural steel) and the standard shape (C-channel 4*7.25). Once the entire frame has been generated on Visual Analysis, the user is able to create member loads or nodal loads and apply these loads to selected members or nodes. Figure 5 displays the frame members with the selected loading. In the frame analysis, a uniformly distributed dead load of 2 Kips was applied over the entire frame. The deformation due to this loading can be seen in Figure 6. A torsional load of 750 ft*lbs was also applied to the nodes corresponding to the location of the tires (Figure 7). Results from this analysis can be seen in Figure 8. The maximum vertical deflection of the frame was 0.0132 inches which is considered negligible.

Figure 5

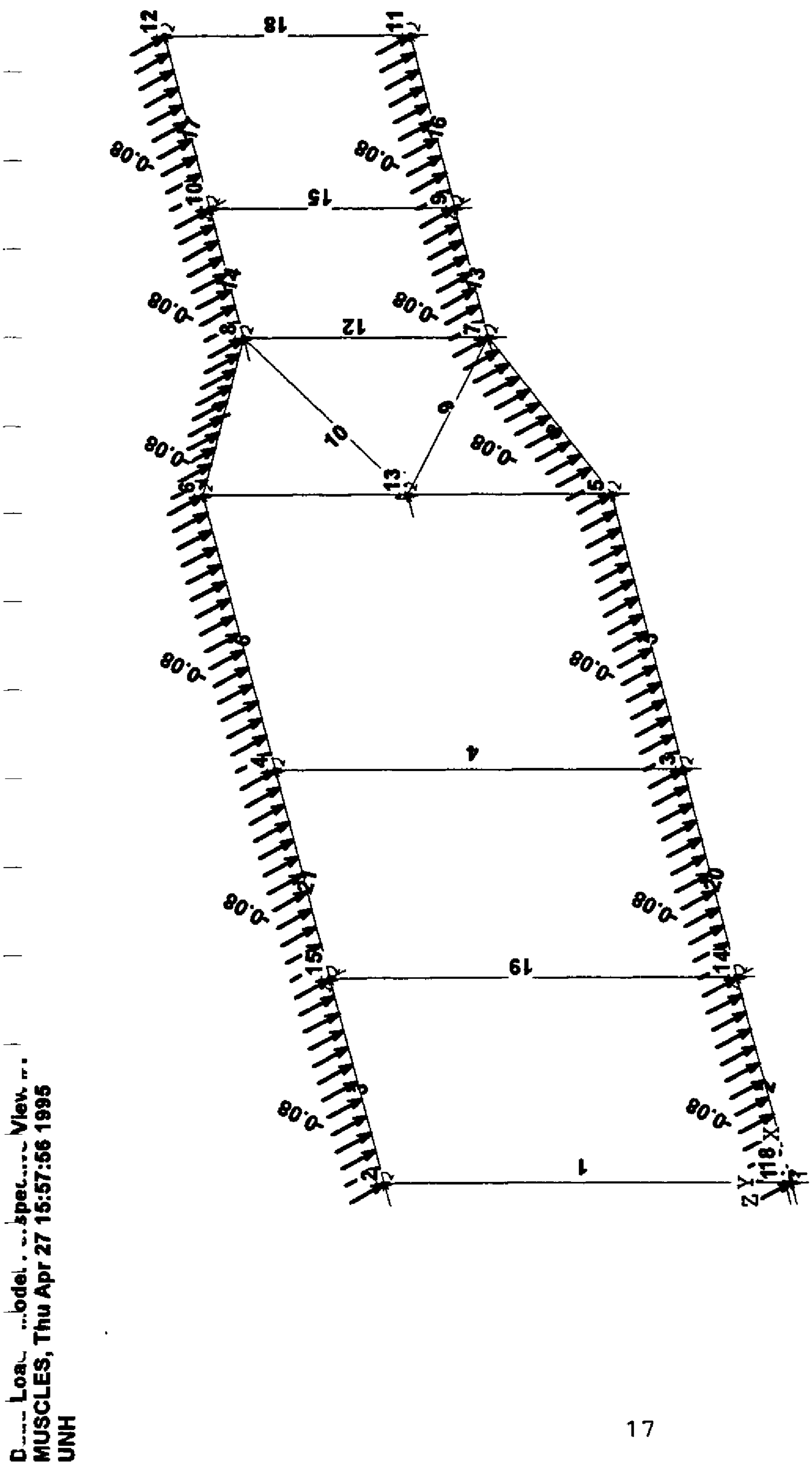


Figure 6

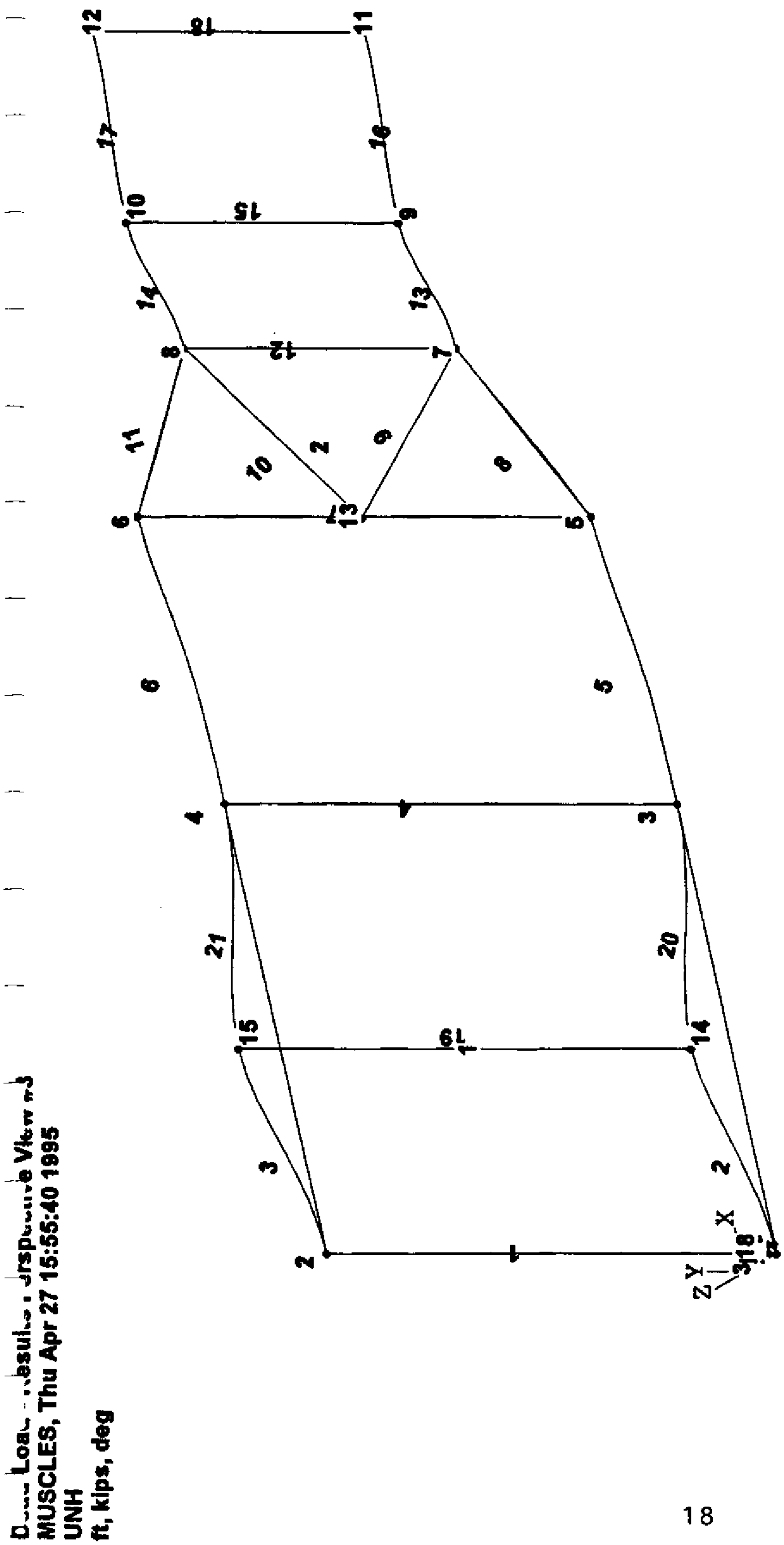


Figure 7

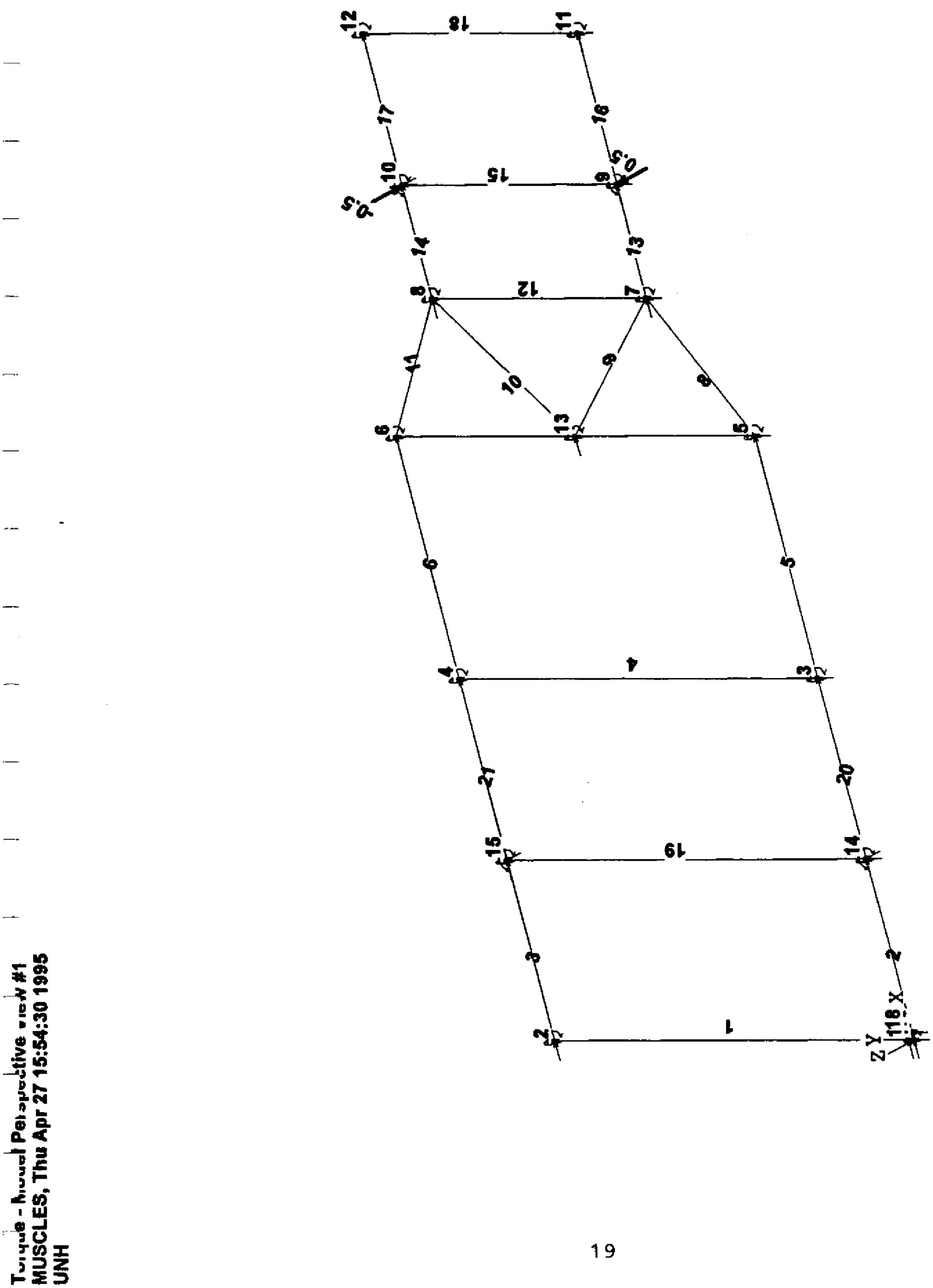
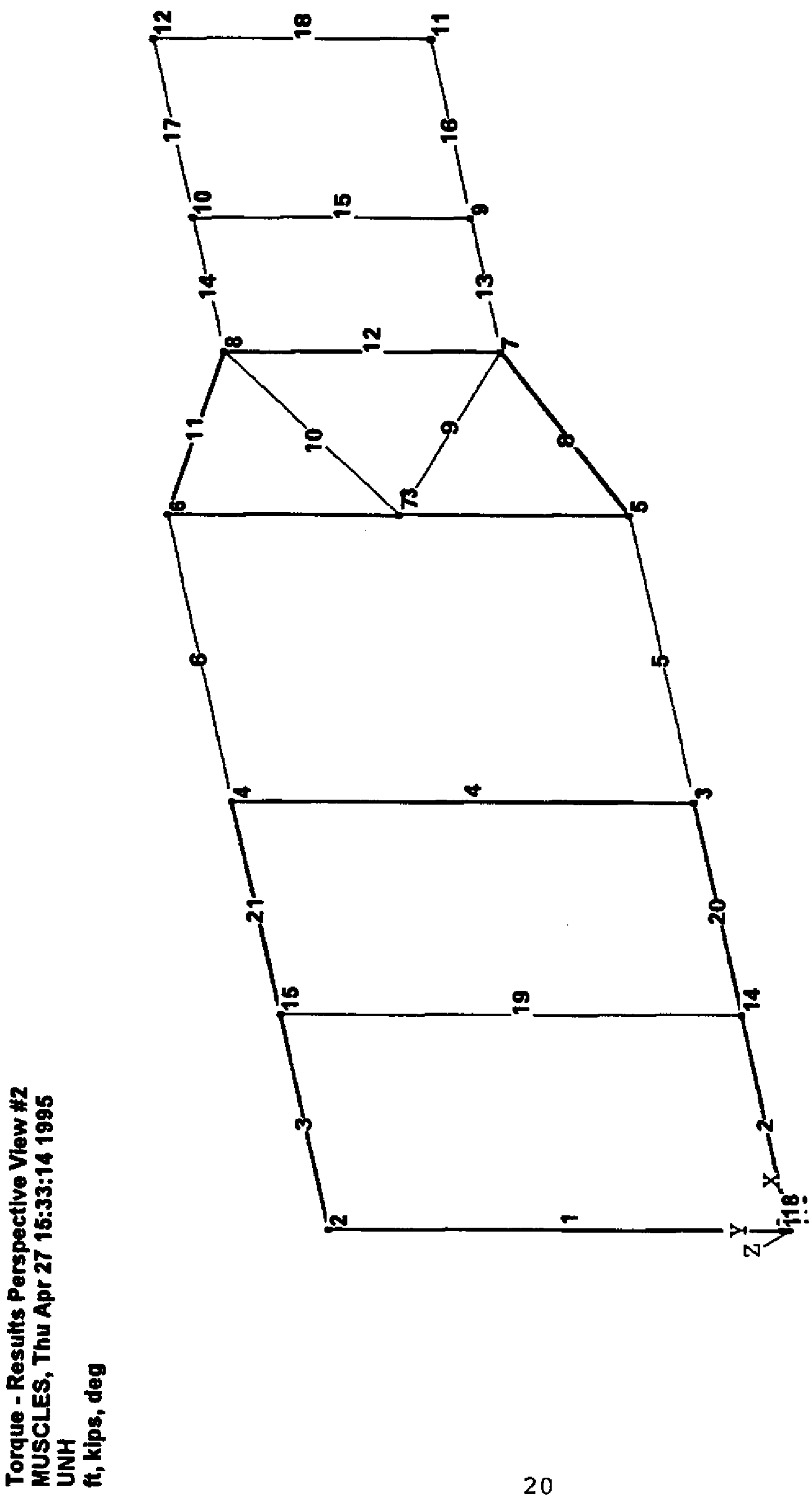


Figure 8


Torque - Results Perspective View #2
MUSCLES, Thu Apr 27 15:33:14 1995
UNH
ft, kips, deg

STEERING AND SUSPENSION SYSTEM

FUNCTIONAL REQUIREMENTS

The steering system must provide on and off road maneuverability. To permit loading and unloading of the vehicle from a large van or truck, the system must be capable of adequate vehicle control on a paved surface. The steering system must be capable of turning in hard packed and loose sand to allow for convenient launch and recovery of the submarine. The system is also required to incorporate a suspension system to reduce torsional stresses on the vehicle chassis.

EVALUATION FACTORS

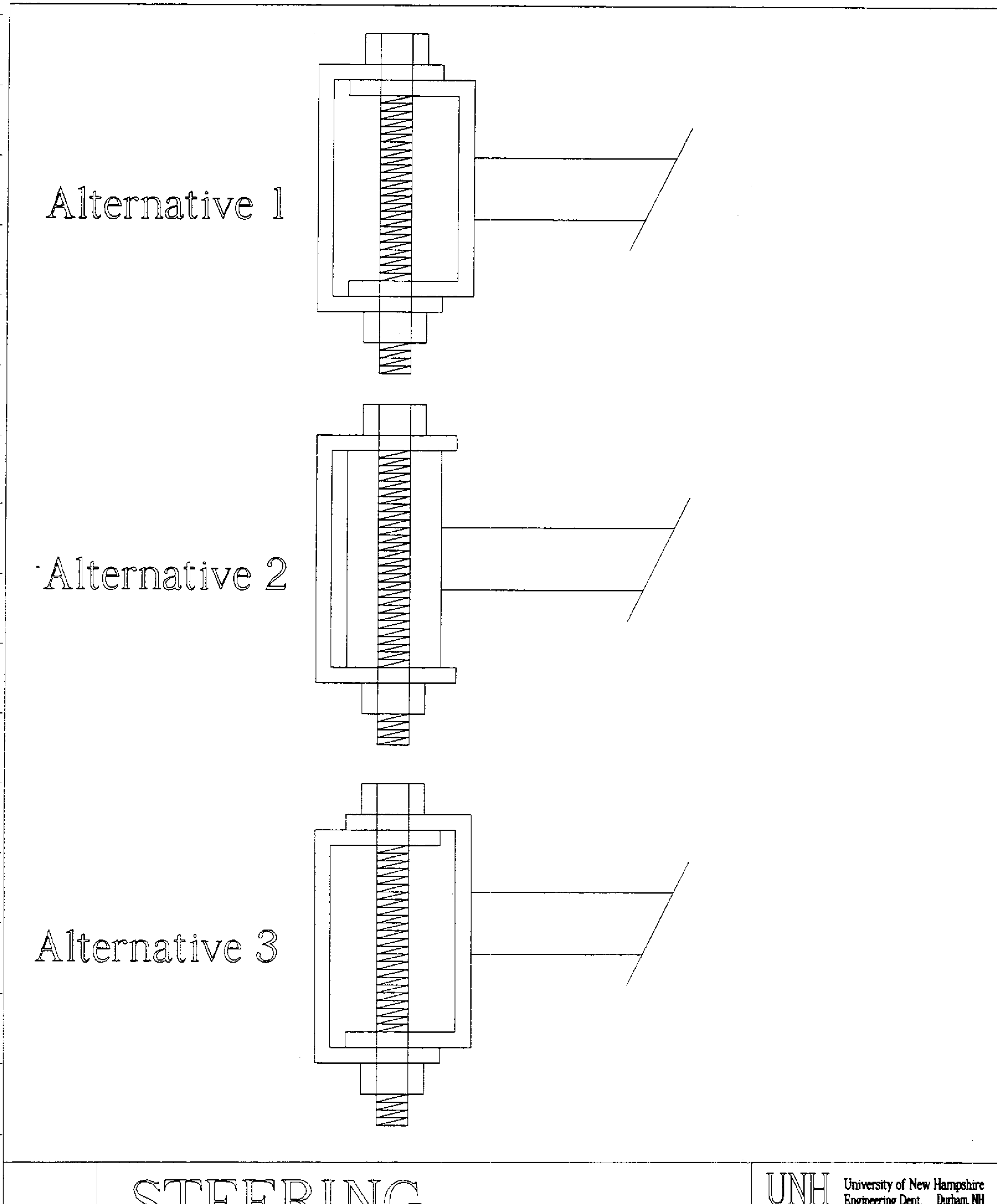
The main factors which determined the selection of the steering system were ease of implementation, durability, and the effectiveness of the suspension.

DESIGN ALTERNATIVES

Two design options involving the suspension system were considered in the selection of the steering system. One included a pin beam suspension system, the other required the adaption of an existing steering and suspension system from a small vehicle.

Three different wheel attachments for the pin beam suspension were considered (Figure 9). One alternative consisted of two C-channel members mated together, one inside the other, with a single bolt running through them. The shaft and tire assembly would be mounted to the inner C-channel. Another alternative was a single piece of C-

Figure 9


Alternative 1
Alternative 2
Alternative 3
STEERING
UNH
University of New Hampshire
Engineering Dept. Durham, NH
DESIGN M.U.S.C.L.E.S.
SCALE NTS
DRAWN Mike Spottswood
DATE 3-20-95

channel with a bolt through it. The bolt would have a steel sleeve over it with the shaft and tire attached to the sleeve. The third alternative for the pin beam suspension was composed of two C-channels. Two bolts would be used to mate an outer C-channel member above an inner C-channel member. This alternative was considered to be the optimal pin beam suspension system as the vehicle weight tends to pull the mating channels apart lowering the contact surface of the two members which ultimately decreases friction and wear. Teflon washers can also be placed between the mating surfaces to further reduce friction.

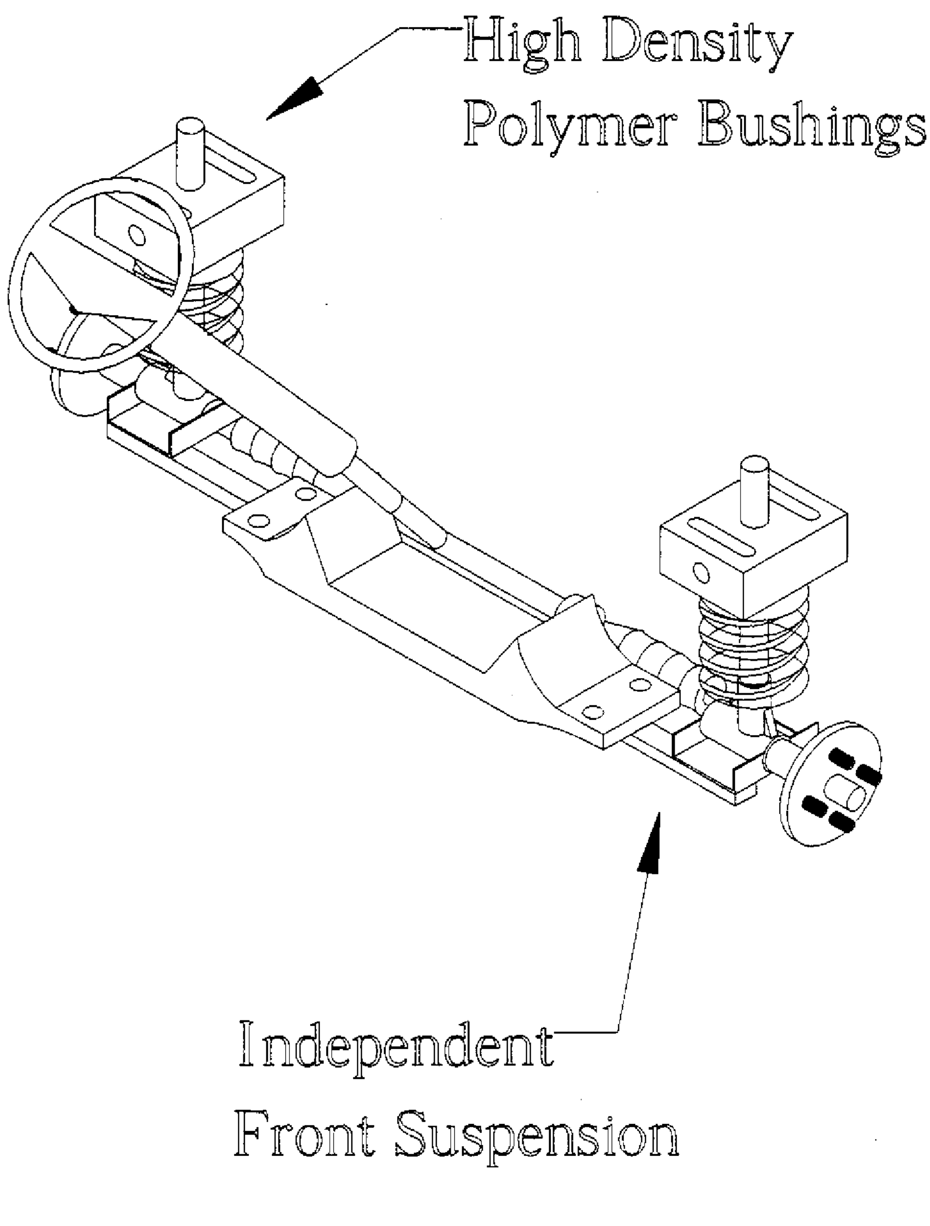
A manual steering system from an automobile was investigated and determined to be more than adequate for the MUSCLE System mission and was readily available. Adapting an existing suspension system from a small automobile proved to be the most cost effective alternative. This system required minimal machining and fabrication of parts to incorporate it with the MUSCLES System.

SELECTED DESIGN

The chosen steering system is a combination of designed and manufactured components. A lower independent suspension and steering unit from a small car was adapted to satisfy the functional requirements of the vehicle. This system is a one piece unit consisting of a manual rack and pinion steering box, lower suspension control arms, lower ball joint, and tie rods (Figure 10). Front hub mounts and struts used to guide the suspension are implemented with this unit. Boat trailer hubs and spindles are mounted on

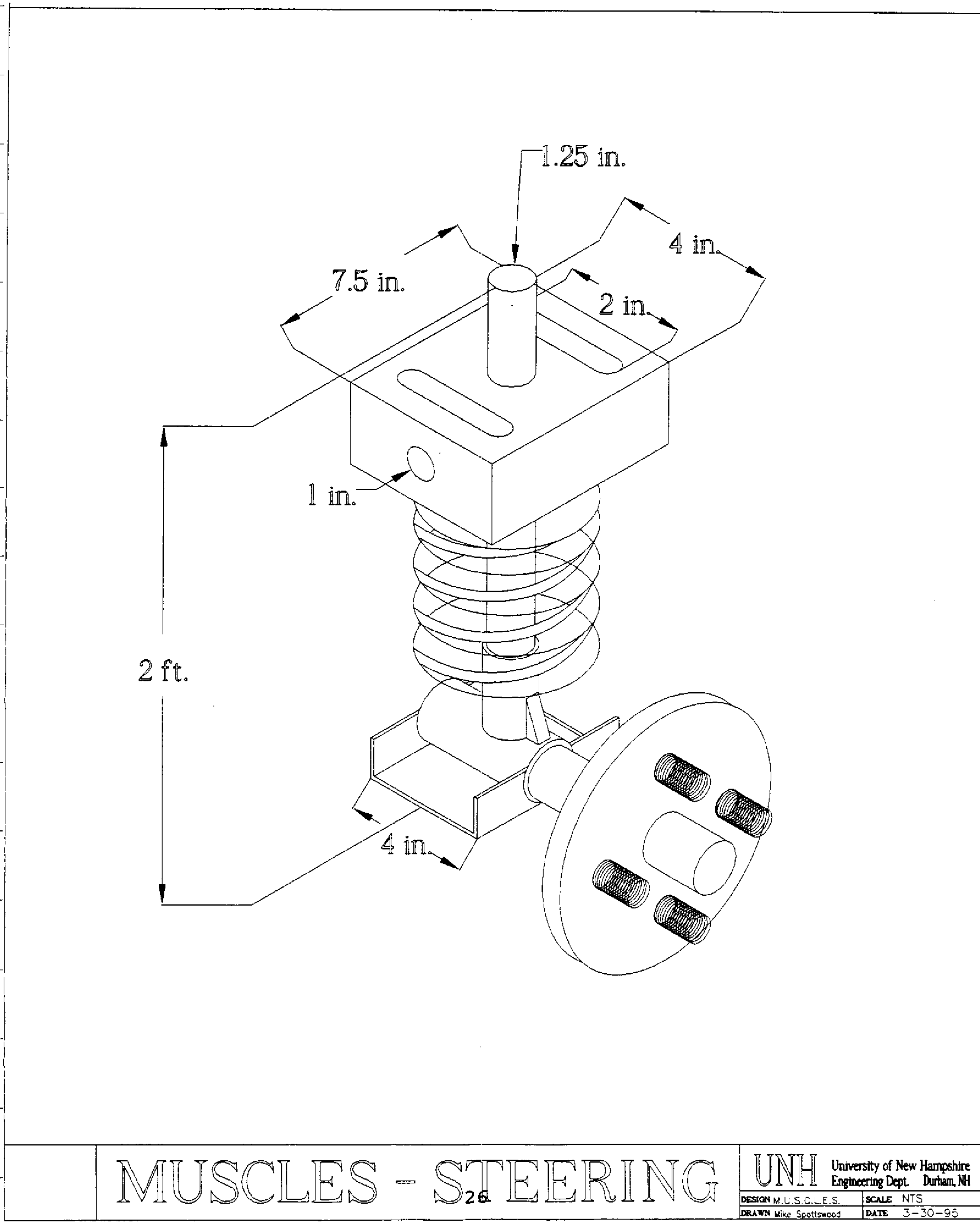
Figure 10

Rack & Pinion Steering System



the lower ball joint. The vertical movement of the strut is controlled by a polyethylene bushing located at the top of the strut. The suspension is supported by springs mounted between the hub mounts and the polyethylene bushings (Figure 11). To protect against wear on the control bushing, another polyethylene bushing with a steel plate at the spring/bushing interface is installed between the top of the spring and the bushing controlling the strut motion.

Figure 11


1.25 in.
4 in.
7.5 in.
2 in.
1 in.
2 ft.
4 in.
MUSCLES - STEERING
UNH
University of New Hampshire
Engineering Dept. Durham, NH
DESIGN M.U.S.C.L.E.S.
SCALE NTS
DRAWN Mike Spottswood
DATE 3-30-95

DESIGN ANALYSIS

The steering system selected for the vehicle was originally designed for much higher operating speeds and loads than will be encountered in the MUSCLE System mission. Thus, the steering unit was assumed to be more than adequate.

The forces acting on the suspension components were determined from a kinematic and dynamic analysis of each piece. The spring rate equation was used to determine the spring rate of the compression springs mounted on the front end of the vehicle. The spring rate equation is defined as:

$$F = Kx \tag{1}$$

where F is the weight off the front end of the vehicle, x is the desired compression displacement of the springs, and K is the required spring rate. The weight off the front end of the vehicle was estimated to be 600 lbs, and 3 inches was used as the desired compression to maintain a normal riding height.

A modified form of the spring rate equation was used to determine the maximum force applied to the polyethylene bushings:

$$F = (x+c)K \tag{2}$$

where c represents the maximum vertical displacement of the suspension system. For this analysis, 6 inches was selected as the maximum desired range of motion for the suspension system.

The maximum force calculated using equation (2) was used in the stress equation:

$$\sigma = \frac{K_t F_{perbolt}}{A} \qquad (3)$$

where $F_{perbolt}$ is the force on each bushing bolt, A is the area of the bushing, and K_t is the stress concentration factor. The stress on the polyethylene bushings was found to be 831 psi which is well below the tensile strength of the polyethylene.

The hubs and spindles were manufactured for the use on boat trailers and are rated at 1300 lbs each. This rating is safely beyond the MUSCLES System application.

DRIVE TRAIN

FUNCTIONAL REQUIREMENTS

The drive train system provides the capability to move and stop the MUSCLE System on various terrain. The power system of the vehicle must be easily controlled and reliable, provide reasonable speed, allow for both forward and reverse movement, and provide sufficient fuel for the entire mission. Figure 12 displays the power versus time requirements of a single mission.

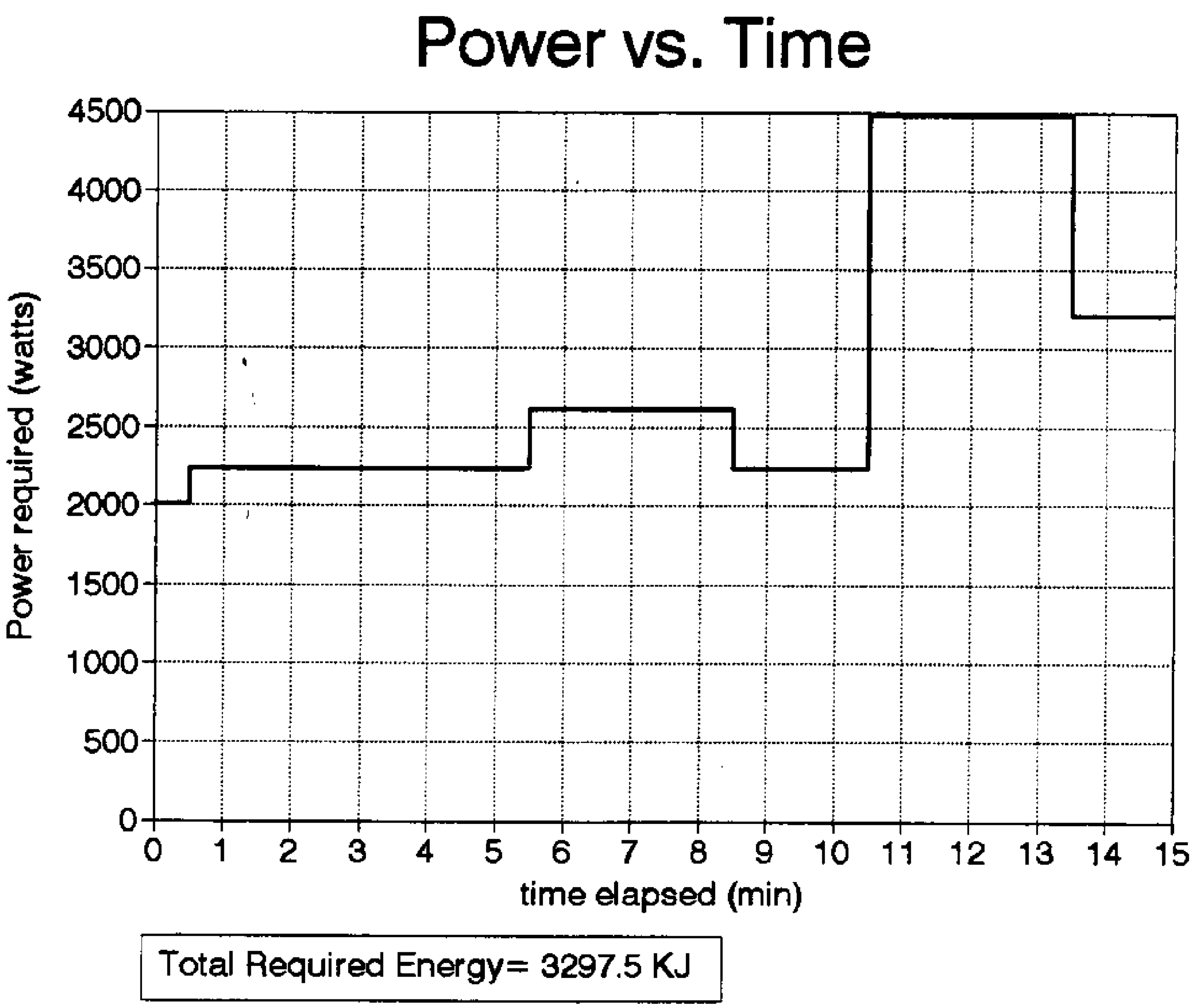
EVALUATION FACTORS

The main factors deciding the selection of the drive train were the required power, mission range, ease of implementation, size, weight, and cost.

DESIGN ALTERNATIVES

Both gas and electric power systems were considered as possible power sources for the vehicle. AC and DC motors reduce the complexity of the system controls and can be operated without producing exhaust emissions. Electric power offers easy implementation as no transmission is required to operate the vehicle in the reverse direction. Since crew members and vehicle operators will be working on and around the vehicle for extended periods of time, operation without exhaust emissions is desirable. The DC motor is the only option which guarantees operation without emissions, however, this system requires the use of several very large and heavy Lead-Acid batteries. The

Figure 12


Power vs. Time
Power required (watts)
4500
4000
3500
3000
2500
2000
1500
1000
500
0
0
1
2
3
4
5
6
7
8
9
10
11
12
13
14
15
time elapsed (min)
Total Required Energy= 3297.5 KJ

added weight of the batteries proved to be a great disadvantage and made this option ineffective. Batteries with higher specific power and specific energy were investigated in an attempt to reduce the weight, however these alternatives were too costly. Low voltage DC motors were also eliminated based on their high expense. Another disadvantage of this system was the necessity of water tight compartments to house the batteries.

Utilizing an AC motor with a generator would also be a feasible power system for the specified mission time. This proposed power system, however, would not be able to operate without producing emissions. The option was eliminated based on cost analysis.

A gas engine offers an affordable supply of power without the added weight of batteries or the danger of the batteries dying before the mission is completed. The gas engine and 5-speed trans-axle from a garden variety tractor provide reversing capability at a reasonable price and can be adapted to the vehicle drive system. Table 1 provides a comparative analysis summary of AC, DC, and gasoline engine drive systems.

DESIGN DESCRIPTION

A 16 horsepower gas engine, from a garden tractor, with a 2:1 gear reduction provides the power for the vehicle. The power is transferred from the engine using a centrifugal clutch. The main advantage of using a centrifugal clutch instead of the clutch directly off the tractor is the ease of implementation in relation to the throttle. The throttle of the engine can be feathered allowing slow operation and slow starts. The clutch also serve as a safety mechanism. If the operator accidentally falls from the

Table 1

MUSCLES SYSTEM ANALYSIS: AC vs DC vs GAS ENGINE

SYSTEM	AC MOTORS	DC MOTORS	GAS ENGINE
MOTOR/COST	2@ $650 ea.	2@ $1500 ea.	1@ $500
ENERGY SOURCE/COST	1 GENERATOR @ $1300	12-24 BATTERIES @ $40 ea	$1.34 PER GAL
CHARGING SYSTEM	NONE	SEPARATE CHARGING SYSTEM NEEDED	NONE
COOLING SYSTEM(S)	NEEDED FOR MOTORS ONLY	NEEDED FOR BATTERIES AND MOTORS	NONE
SEALED COMPARTMENT	NEEDED FOR MOTORS ONLY	NEEDED FOR BATTERIES AND MOTORS	NONE
GEARING	SAME	SAME	SAME
BRAKING	SAME/UMBIL-ICAL CONTROLLED	SAME/UMBEL-ICAL CONTROLLED	MANUAL
STEERING	MOTOR CONTROLLED	MOTOR CONTROLLED	MANUAL
CONTROLS	UMBILICAL	UMBILICAL	MANUAL
MATERIALS/ WEIGHT	NEEDED FOR MOTOR COMPARTMENT, FRAME, AND GENERATOR PLATFORM	NEEDED FOR ALL ADDITIONAL COMPARTMENT AND FRAME	NEEDED FOR FRAME (CONSIDER-ABLY LESS MATERIAL)
BUOYANCY CONSIDERAT-IONS	COMPARTMENT	COMPARTMENT	NONE
LAUNCH AND RECOVERY	DIRECT	DIRECT	CART SYSTEM

vehicle, the vehicle will stop. There is a possibility that the vehicle could be splashed during the mission, in which case the belt drive from the tractor would be useless. By utilizing a centrifugal clutch, a chain drive can be employed and this problem is entirely avoided.

The engine drives the trans-axle which is accomplished through the chain drive. The trans-axle from the tractor comes equipped with a differential. For maximum traction in the sand, a solid rear axle is desired, however, to reduce the stresses put on the axle and drivetrain components when manuvering on paved surfaces a split shaft is desireable. To incorporate the solid rear axle, the chain drive is run off each side of the trans-axle to the solid rear axle. To provide differential action on paved surfaces, the rear axle is split between the to sprockets and a locking spline is placed between them.

The tire size, available transmission ratio, and final gear reduction between the transmission and rear drive axle were considered when determining the gear ratio between the engine and transmission. The two variables which were the deciding factors in the selection of the gear ratios between the transmission and the rear axle were the required ground clearance at the rear axle and the effective braking power generated at the trans-axle. The sprocket located on the rear axle was limited by size in that it was required to clear the ground. Also, the tractor's trans-axle has an internal brake; however, the stock brakes are not sufficient for the size of the vehicle. As a result, a gear reduction was employed to increase the effective braking power. If the brakes of the selected tractor are deemed insufficient during the overall operational testing, external mechanically actuated disc brakes from ClubCadet can be utilized.

The rear axle is a split solid steel shaft with both sides supported at both ends by four pillow block bearings which are mounted on the C-channel frame. The bolt pattern of the hubs on the ends of the axle are designed to match those of the tire rims.

The tire rims are low pressure flotation tires, similar to those used on recreational vehicles. Flotation tires provide the best traction in sand and aid in the suspension of the MUSCLE System. Figures 13 & 14 are drawings of the drive system which include the gasoline engine, trans-axle, chain drive, and locking spline installed in the main shaft.

Figure 13

Drive System

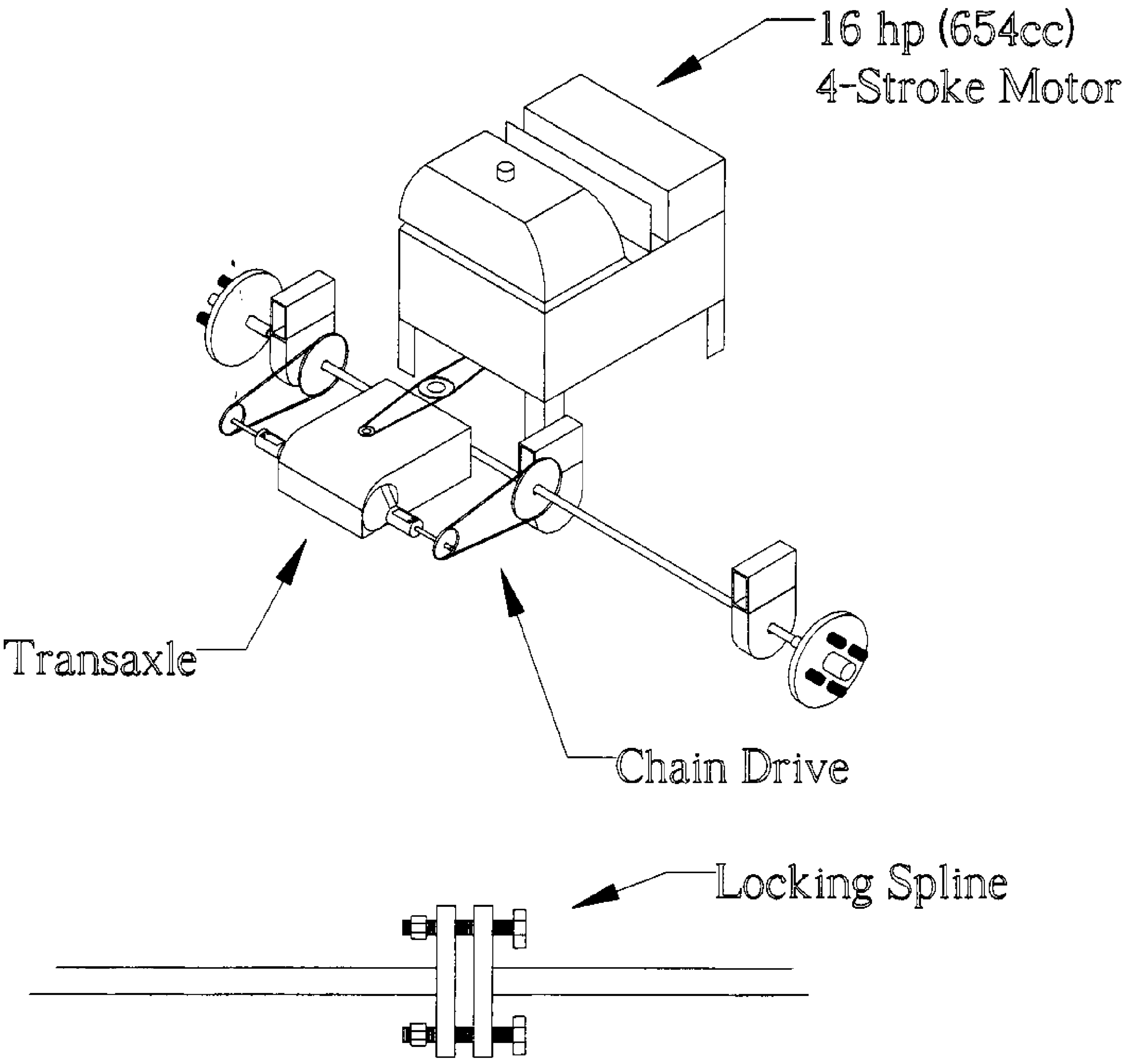
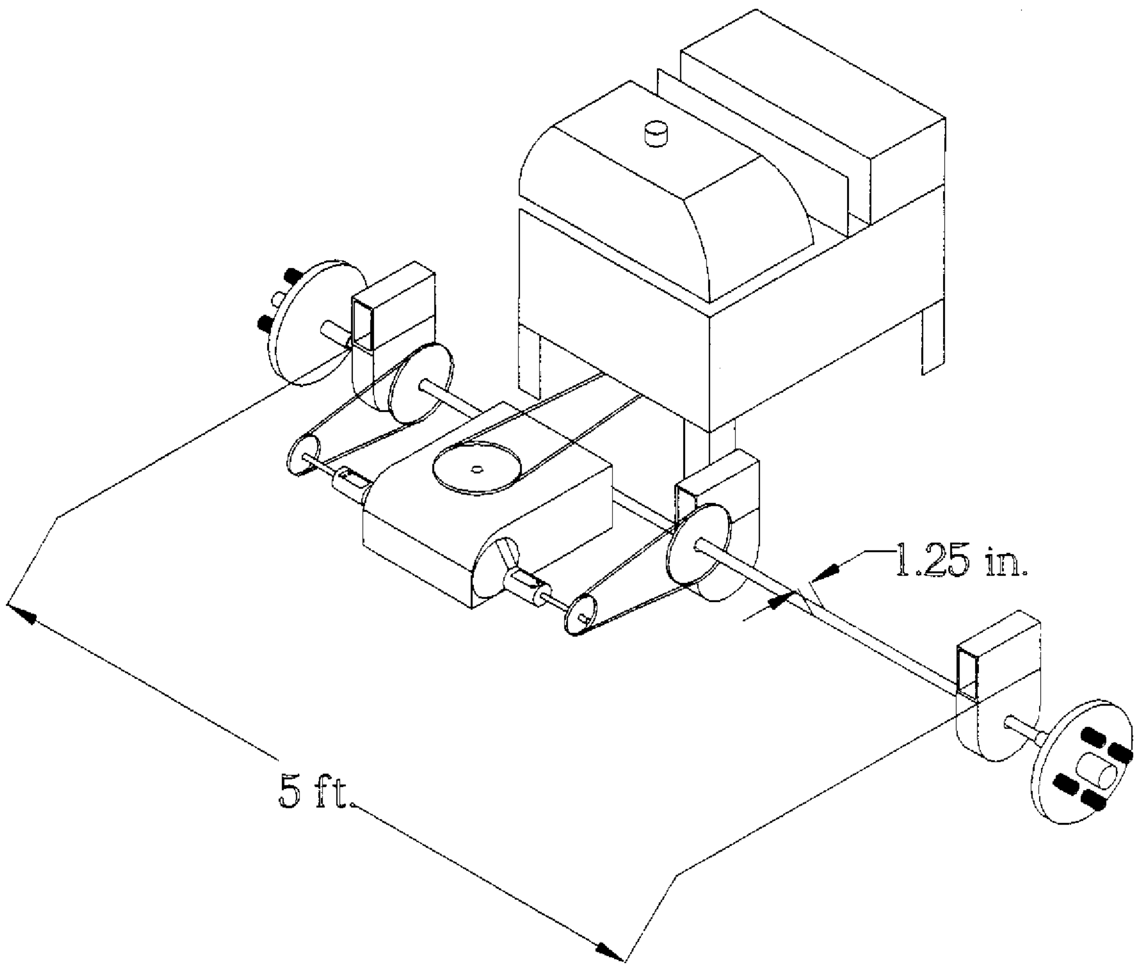


Figure 14



MUSCLES - DRIVE

UNH University of New Hampshire Engineering Dept. Durham, NH

DESIGN M.U.S.C.L.E.S.	SCALE NTS
DRAWN Mike Spottswood	DATE 3-30-95

DESIGN ANALYSIS

The required engine horsepower was determined using the power equation:

$$H = \frac{T\,\omega}{\xi} \qquad (1)$$

where H is the required horsepower, T is the torque at the rear wheel, ω is the angular velocity or tire rpm, and ξ is the efficiency of the drive train. The torque at the rear wheel was calculated using the torque equation:

$$T = FL \qquad (2)$$

where F is the force on the vehicle and L is the tire radius. The force on the vehicle was determined for the a worst case scenario in which the vehicle would be on a 20 degree incline. The required engine power was found to be 12 HP.

By manipulating equation (2) to solve for force, the maximum force supported by the chain drive was found. The value of T still remains the rear wheel torque, however, L is now the rear sprocket radius. The force of the chain under the most demanding operating conditions was 2.978 Kips.

Equation (2) was also applied to the braking system, where F was the weight of the vehicle rear end, and L was the tire radius. The force due to the weight off the back end of the vehicle was estimated using a coefficient of static friction equal to .9 (rubber on asphalt) and approximating the back end weight to be 60% of the total vehicle weight. Incorporating a factor of safety of 2, the braking torque was found to be $1.35 * 10^3$ ft*lbs.

Due to the fact that the tractor brakes were employed in the design, the braking torque cannot be rated. Operational tests were conducted to verify that the braking capabilities of the vehicle are adequate. It is estimated that the vehicle is capable of providing a braking torque equal to 500 ft*lbs.

The bending and torsional loads on the rear axle were used to determine the minimum required shaft diameter. The following equation was used to calculate the shaft diameter:

$$d_{shaft} = [(16/\pi S_y)*(M^2+T^2)^{1/2}]^{1/3} \quad (3)$$

where S_y is the allowable shear stress, M is the bending moment due to the vehicle weight, and T is the maximum torque due to the vehicle weight. Equation (3) is derived in Design of Mechanical Elements (Fifth Edition) by Shigley and Mischke.

CART SYSTEM

MISSION DESCRIPTION

The cart system is used to support the submersible and launch it from the vehicle ramp, through the surf, and to a depth of 4 to 5 feet. A manual hand winch is used in conjunction with the cart to control the launch and aid in the retrieval of the submersible after operation. The cart is also used to support and easily transport the submersible during routine maintenance, repairs, or design alterations.

FUNCTIONAL REQUIREMENTS

The cart is required to handle various sized submersibles, the largest having a 3 foot diameter and a length of 16 feet. To maximize operator safety and avoid damaging the submarine, the submarine must be safely secured to the cart. However, the launching and recovery process must be simple and quick. The size of the cart is limited by the available space on the vehicle and the design objective of handling of the cart by two persons. The cart material should be compatible with the ocean environment. A simple frame structure is required to reduce operating complications, maintenance requirements, material costs, and complexity of construction. Also, by eliminating intricate framework, the reliability of the cart is enhanced, and there are fewer obstructions while making repairs on the submersible. The cart is required to be highly maneuverable for loading purposes and when transporting the submersible to a work station. The cart is also built relatively low to the ground to ensure stability through the surf, across rough terrain, and while being used as work station. While the cart serves as a work station, divers must be able to safely climb in and out of the submarine supported on the cart without disrupting the balance of the system.

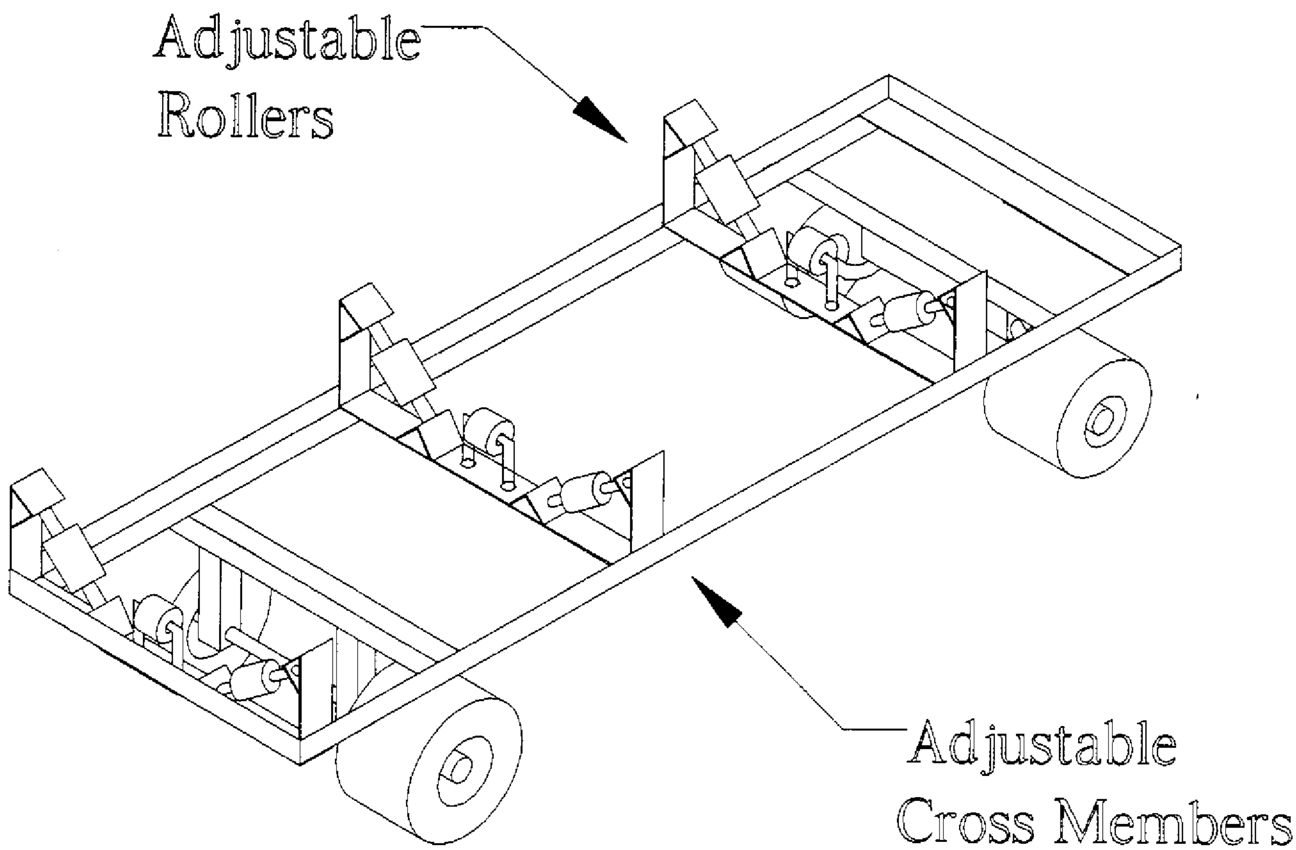
DESIGN DESCRIPTION

The frame structure was designed to support a partially flooded submersible and maintain stability in all modes of operation. Structural angle steel was used as the building material as it was the most cost effective choice and easier to work with than aluminum. By priming and painting the steel structure, the cart was made compatible with the ocean environment. Part of the construction entailed the welding of steel members for structural rigidity. However, as most of the sections of the cart must be adjustable to accommodate various shaped hulls, bolt fasteners were utilized for installing and supporting the rollers and pads which come in contact with the submarine hull.

The cart is fairly compact and spans a length of 7 feet and is 3 feet wide. Figures 15 & 16 display the cart structure and components. Rollers are located along the triangular members of the frame to maintain a non-damaging contact surface with the submarine. The rollers also help guide and control the submarine during the loading and launching process. To keep the keel of the submersible level, the triangular supports can be adjusted in the vertical direction by resetting the threaded rods running through the triangular supports and the transverse members of the main structural frame. The

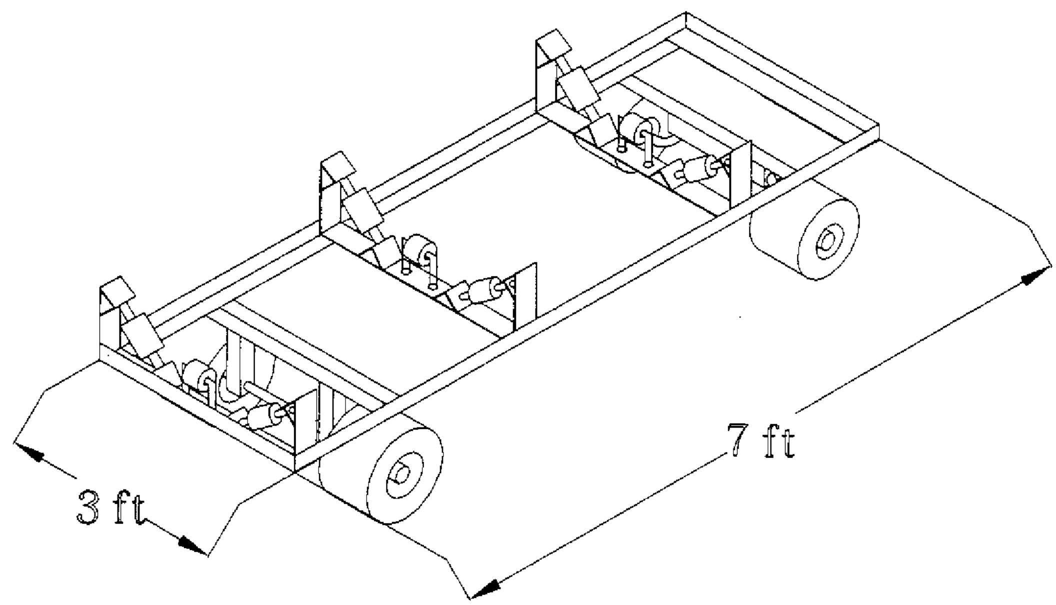
Figure 15

Cart System



Frame Sides are 2 in. Angle-Iron
Cross Members are 2 in. Angle-Iron

Figure 16



MUSCLES - CART

UNH	University of New Hampshire Engineering Dept. Durham, NH		
DESIGN	M.U.S.C.L.E.S.	SCALE	NTS
DRAWN	Mike Spottswood	DATE	3-30-95

transverse members are 3 feet in length and support the triangular members. These transverse members are also adjustable and can be repositioned in 1 foot increments along the length of the cart. The submersible is easily secured to the cart with the help of nylon straps and manually engaged buckles. Good cart maneuverability is achieved by adapting the front end of a lawn tractor which employs a rack and pinion steering system.

DESIGN ANALYSIS

The front end and rear axle were assumed to satisfy the loads supported by the cart as they were taken from the garden tractor. As in the frame analysis, Visual Analysis was also employed to analyze the maximum deflection of the cart members under a uniform loading equal to a partially flooded submarine weighing 3000 lbs. The loading of the cart can be seen in Figure 17, and the resulting cart member deformations are shown in Figure 18. The maximum deflection of the cart is .036 inches which is acceptable.

Figure 17

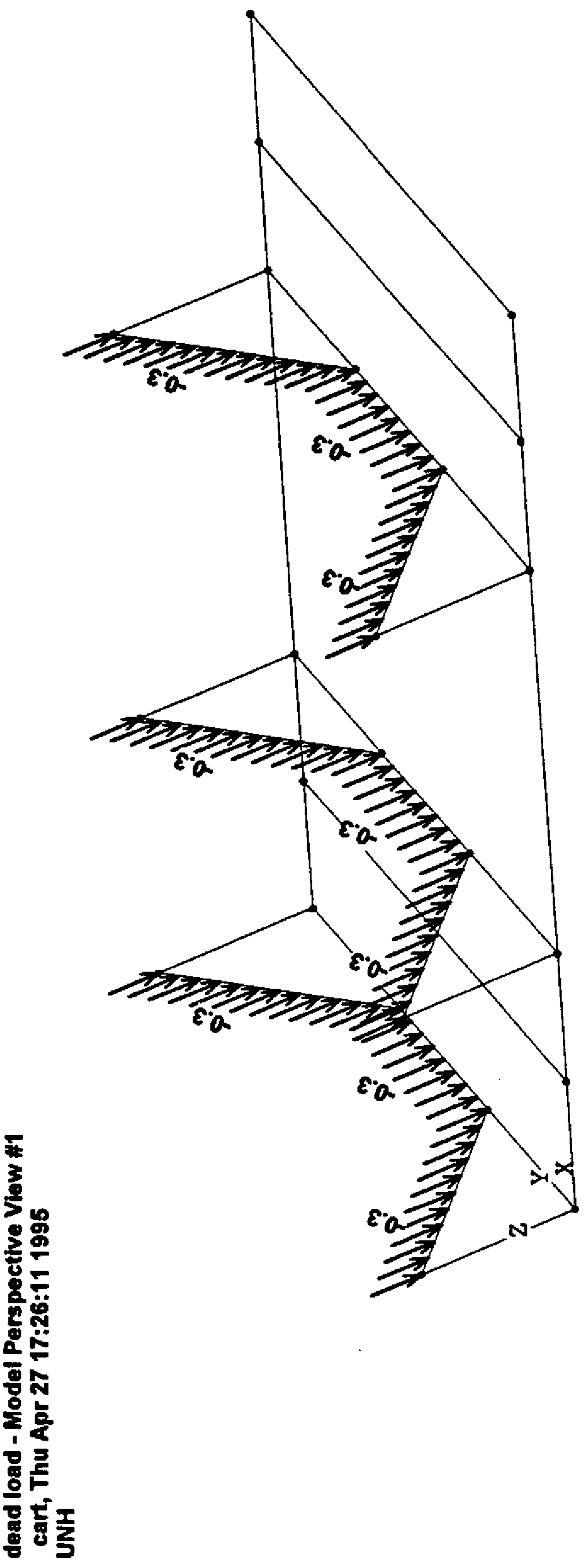
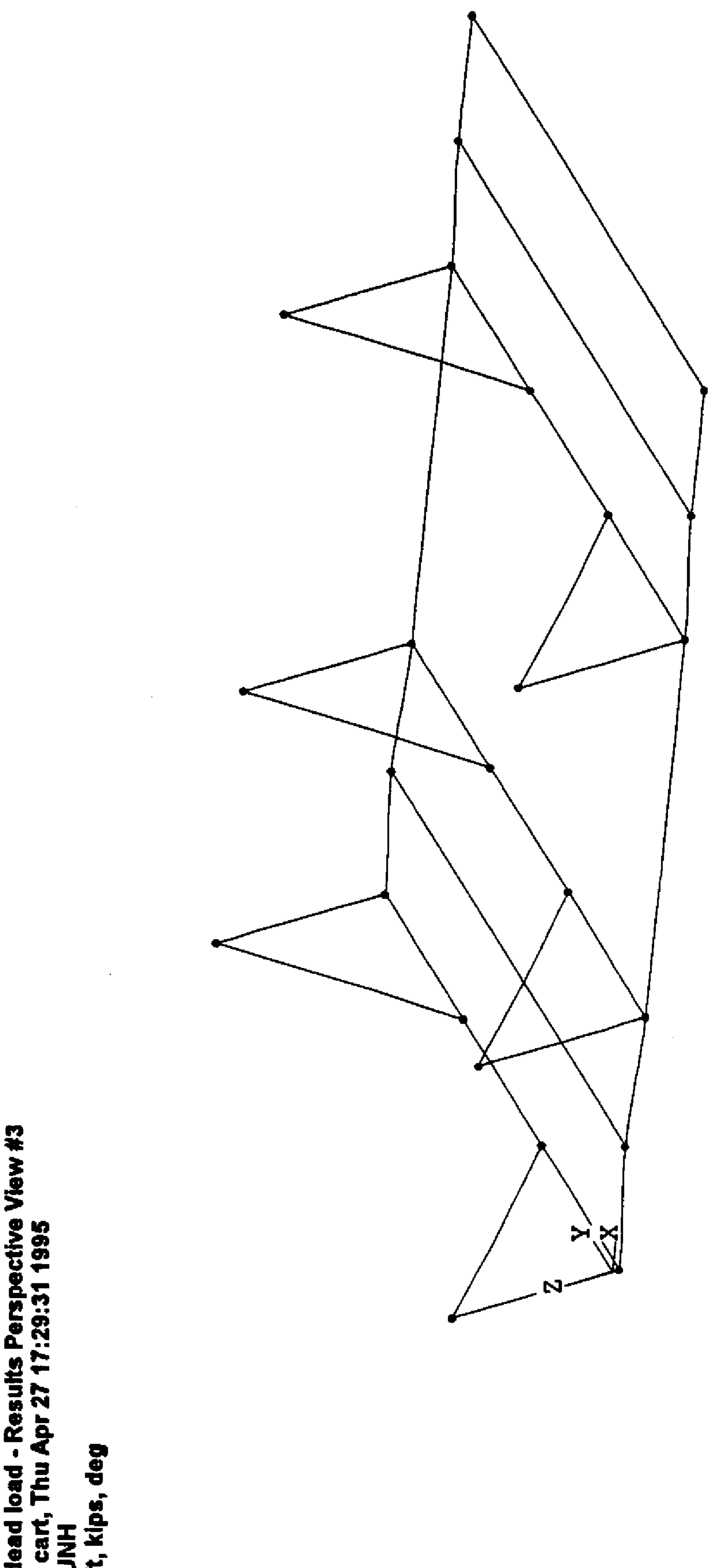

dead load - Model Perspective View #1
cart, Thu Apr 27 17:26:11 1995
UNH
-0.3
X
Y
Z

Figure 18



dead load - Results Perspective View #3
cart, Thu Apr 27 17:29:31 1995
UNH
ft, kips, deg
Y
X
Z

SYSTEM TESTING

A test program was designed to verify system and component operability. The land vehicle was operated at very low speeds to test whether the drive system and steering/suspension system are performing their required functions. The chain drive was checked to ensure an adequate interface between the chain and the sprockets, and to ensure successful transmission of power. The steering capabilities were evaluated by determining the turning radius of the vehicle. The ability to maneuver on paved ground with a solid rear axle was assessed. As a result of this test, it was determined that a solid rear axle was not sufficient and a locking spline was designed and incorporated into the rear axle to alleviate the shearing stresses. The braking capabilities of the vehicle were tested on flat and inclined grounds. The vehicle will be operated near a shore area to test the maneuvering and braking capabilities, the traction of the tires, and the performance of the solid rear axle and engine in dry and wet sand. While the vehicle was operated over various terrain, the ability of the suspension system to maintain a desired riding elevation was evaluated and determined to be satisfactory.

During construction, the welds were tested and examined for gaps to ensure the strength and rigidity of the frame. Structural supports were also built into the frame to guarantee the durability of the frame.

The submarine was loaded onto the cart to check the cart strength. With the submarine secured to the cart, the maneuverability and stability of the cart was evaluated and determined to be satisfactory.

The ramp was tested to ensure that it was capable of supporting the cart and submarine. A test run including the loading of the cart onto the vehicle via the ramp and winch demonstrated the operability of this system. The cart was placed on the transport vehicle to test for stability of the vehicle, and to test the security of the cart and submersible on the transport vehicle.

Upon completion of all necessary preparations, the MUSCLE System is scheduled for an integrated operational test in which the system is run through the entire operational scenario. This includes the transport of the MUSCLE System to a lake or beach area, driving the system to the shore's edge, down loading the cart and submarine from the vehicle using the ramp and winch, guiding the cart to an acceptable launching depth, a mock launching of the submarine, recovering the submarine, strapping the submarine back onto the cart, reloading the cart with the partially flooded submarine back onto the vehicle using the ramp and winch, driving the vehicle back up the shore, and finally into the rental truck. This operational test was not completed at the time that his report was submitted, but will be accomplished prior to completion of the project.

TEST AND ASSEMBLY PROBLEMS

During the assembly of the system, individual tests were conducted as the assembly progressed. This allowed problems to be found and corrected before they were magnified. As expected, problems arose as a result of adapting various existing systems to the MUSCLES design. The major systems adapted to the MUSCLES vehicle were the engine, trans-axle and various linkages from a Gilson garden tractor, and the lower half of the steering and suspension system from a compact car. The major problems encountered are summarized below, along with the solutions implemented to resolve the problems.

1. System: Transport Vehicle Frame

Source: 4 inch C-channel

Problem #1: Due to torsional loads, the narrow front end of the transport vehicle deflected slightly beyond an acceptable level.

Solution: Gussets were welded from the top of the C-channel at the narrow front section, to the midpoint of the bottom cross member located at the front of the wider rear section.

2. System: Front end suspension and steering

Source: The lower half of an independent front suspension and rack and pinion steering system were obtained from a private source at no cost.

Problem #1: The original design of the vehicle incorporated a pin-beam suspension to relieve stress on the transport vehicles frame, and to keep all four wheels firmly planted at all times.

Solution: The use of this existing system required specially designed mounts for the lower suspension system, hubs, spindles, and struts to allow for adequate suspension and steering.

Problem #2: After the implementation and testing of the front end suspension and steering system, it was determined that the suspension was too soft and the ride height of the vehicle was too low.

Solution: Two inch thick bushings were fabricated and installed to compress the springs. A metal plate was placed between the spring/bushing interface to prevent wear when the wheels are rotated.

3. System: Rear axle

Source: Fabricated rear wheel hubs.

Problem #1: To facilitate repairs on subsequent parts incorporated into the axle, the rear hubs were designed to be easily removed from the rear axle. This was achieved by fabricating half inch steel plate with a key way which locked them to the shaft. Locking collars on the backside were also implemented. This design was not 'hefty' enough to take loads applied parallel to the axle (those experienced during use on off-camber traverses). During the initial testing, this was quickly discovered.

Solution: The hubs were removed and one inch thick sprockets were welded to the back of the hubs. As a result, the cross sectional area of the key way and the mating surface was tripled. This design was tested again and proved more than sufficient.

Problem #2: The initial rear axle design was a solid shaft. Using a lockable spline to split the axle was considered to facilitate maneuvering on paved surfaces. The spline was not originally incorporated into the design in an attempt to reduce the complexity of the design and remain 'user friendly'. During operational testing on pavement, the vehicle was maneuverable, however the power required to turn the vehicle was enormous, along with the stress on the axle, and the stress on the side walls of the low-pressure tires.

Solution: A locking spline had to be implemented. Due to the time constraints of the project an easily operated spline was not an option. A spline that would lock, if needed, was implemented. In subsequent vehicle testing, it was found that without the spline locked the vehicle had enough traction for most situations, and if needed the spline could be engaged by reaching under the vehicle and sliding it into position.

4. System: Drive System

Source: The main components of the drive system are the engine and trans-axle from a Gilson garden tractor.

Problem #1: The chain running from the engine to the trans-axle covered a two foot horizontal span. Initially the chain was assembled so tight that during testing, the master link snapped.

Solution: An idler gear was fabricated to apply pressure outward on the inner side of the chain. This increased the tension and reduced the slack in the middle of the chain.

Problem #2: The chain drive from the trans-axle to the rear axle, applied up to ten times the force the trans-axle would experience with twenty inch tires. No supports were installed on the trans-axle shafts to counteract this force. During high load conditions, the trans-axle's axles bent outward and caused the trans-axle to skip.

Solution: To counteract these forces, instead of using the rear housing as the support, the trans-axle's axles were isolated by placing pillow block bearings on either side of the sprockets located on each axle.

PROJECT MANAGEMENT

Dan Shores, the team leader of MUSCLES, organized meetings, planned the agenda and progress of the team, corresponded with the project advisor, Dr. Gerald Sedor, and planned the initial construction schedule. Team meetings with the project advisor were scheduled once every week to review the MUSCLES System design, project progress, and agenda. In the initial stages of the design process, the design mission and requirements were discussed and design concepts were individually developed. These concepts were analyzed by the team as a whole. Once the final design was selected in January, the construction and design of specific subsystems were distributed among the team members to speed up the construction process.

Lyndaker/Rizzo/Shores	Drive System
Lyndaker/Rizzo	Steering/Suspension
Lyndaker	Winch/Ramp
Shores/Spottswood	Frame
Hollenberg/Spottswood	Cart

Construction of the MUSCLES System began in February and progressed until the end of April. The work schedule was organized between members working on the same subsystems. The design analysis for each of the subsystems were also completed by the members who were responsible for the design and construction of the subsystem.

Figure 19 shows the MUSCLES project schedule and Table 2 is a summary of the project budget.

Figure 19

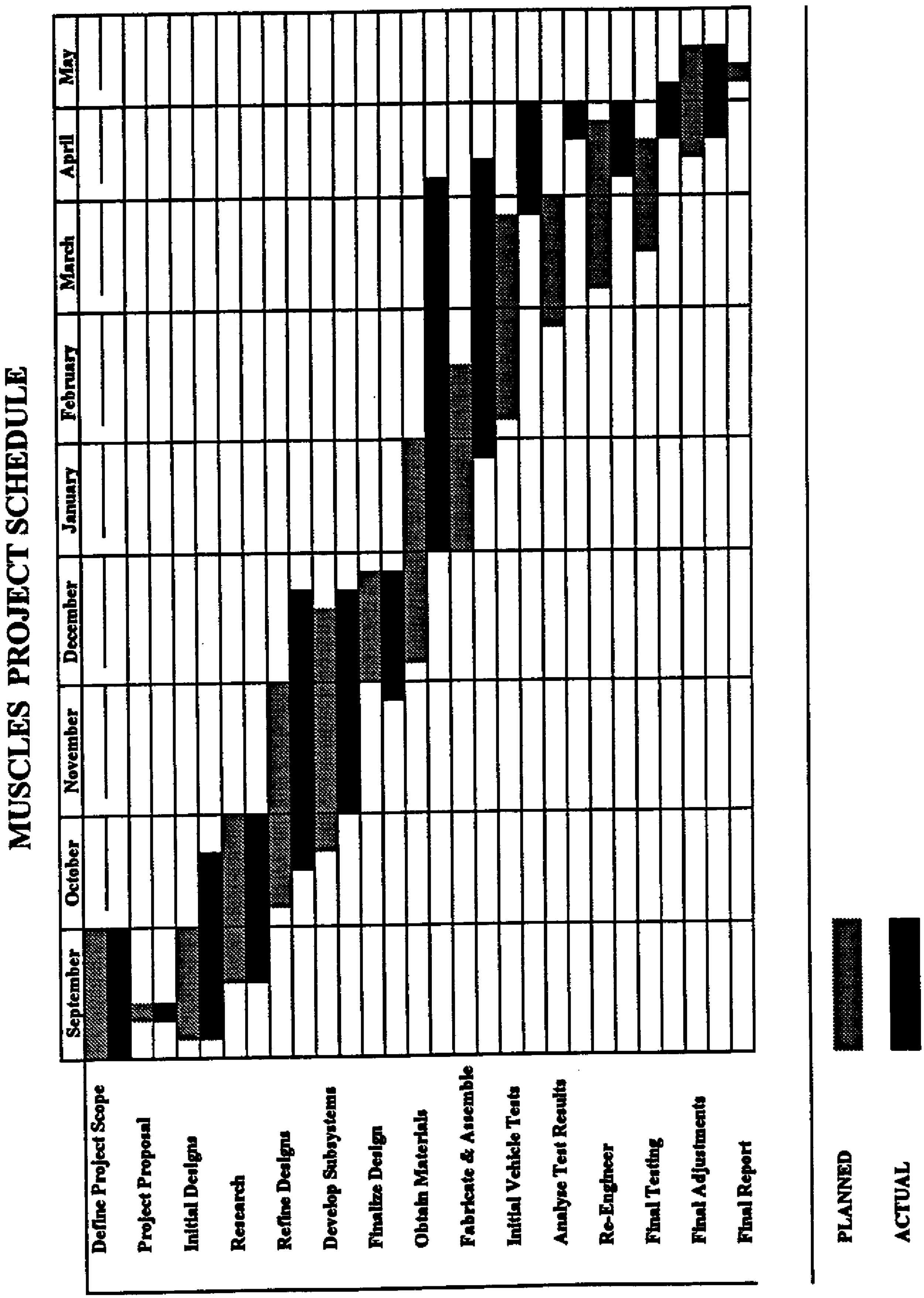

MUSCLES PROJECT SCHEDULE
September
October
November
December
January
February
March
April
May
Define Project Scope
Project Proposal
Initial Designs
Research
Refine Designs
Develop Subsystems
Finalize Design
Obtain Materials
Fabricate & Assemble
Initial Vehicle Tests
Analyze Test Results
Re-Engineer
Final Testing
Final Adjustments
Final Report
PLANNED
ACTUAL

Table 2

MUSCLES PROJECT BUDGET

	EXPECTED COST	ACTUAL COST
MATERIALS		
FRAME SYSTEM	$175.00	$100.00
DRIVE SYSTEM	$1830.00	$1830.00
MANEUVERING SYSTEM	$660.00	$360.00
CART SYSTEM	$375.00	$250.00
MISCELLANEOUS	$555.00	$555.00
LABOR		
METAL WORK	$300.00	$0.00
ADMINISTRATIVE		
TRAVEL, TELEPHONE, PRINTING, etc.	$615.00	$450.00
TOTALS	$4510.00	$3545.00 < *$3660.00* (TECH 797 BUDGET)

CONCLUSIONS

The following conclusion are made on the design, construction, testing, and operation of the MUSCLE System:

1. The MUSCLE System mission requirements established at the beginning of this project can be met by the system developed by the project team. With the exception of the requirements involving operations in an ocean environment, the mission requirements were validated by system testing. The water operations are scheduled to be completed prior to the completion of the project.

2. Adapting components from an existing system designed for a different mission can be cost-effective, but required careful analysis and thorough testing to verify that requirements were met.

3. The use of steel, properly preserved, provides for an optimum structural design which is cost-effective, relatively easy to assemble, and able to provide the required strength.

4. A power system using a gasoline engine was selected over other alternatives incorporating batteries and electric motors, and is considered optimum for this application.

5. An amphibious vehicle with both land and water propulsion capabilities would provide additional operational abilities and flexibility beyond those available with the existing MUSCLE System, but the additional cost and complexity associated with such a system dictated against the selection of such a system.

6. Detailed scheduling and frequent reviews of status and problems are vital to the

successful conclusion of a project of this type.

7. Budgetary limitations can be overcome through aggressive pursuit of alternative sources of required materials and components.

8. The project oriented experience gained by the MUSCLES team was invaluable and should be beneficial in future career growth.

REFERENCES

Bearings Incorporated, consulting information, 1995.

Beer, Ferdinand P. and Russell, Jonston Jr. Mechanics of Materials. 2nd Edition New York: McGraw-Hill Inc., 1992.

Sedor, Gerald. Naval Architecture in Ocean Engineering OE/ME - 751/851. University of New Hamphsire, 1994.

Shigley, Joseph Edward and Mishke, Charles R. Mechanical Engineering Design. 5th Edition New York: McGraw-Hill Inc., 1989.

Visual Analysis, computer software, Integrated Engineering Software, 1994.

John Wilson, consulting information, 1995.

Roger Winslow, consulting information, 1995.

ACKNOWLEDGMENTS

Dr. Gerald Sedor
Without the unconditional devotion of this true educator, we never would have learned the lessons needed to succeed in anything we may encounter in life.

Mr. Roger Winslow
We were very fortunate to cross paths with this man. Mr. Winslow dedicated his working facilities, tools, and most importantly, his great engineering mind to our project. We won't forget his words "God gave us many gifts, but the most beautiful gift is to create from our minds with our hands in order to produce something nothing less than perfect".

Mr. Paul Lavoie
Thanks for your advice and, most importantly, your patience.

Mr. Jon Scott
Thank you for all the times which you put everything aside in order to help our project progress.

Mr. Bob Champlin
Thank you for your advice, skills, and patience.

Professor M. Robinson Swift
Thank you for organizing the course which made this project possible.

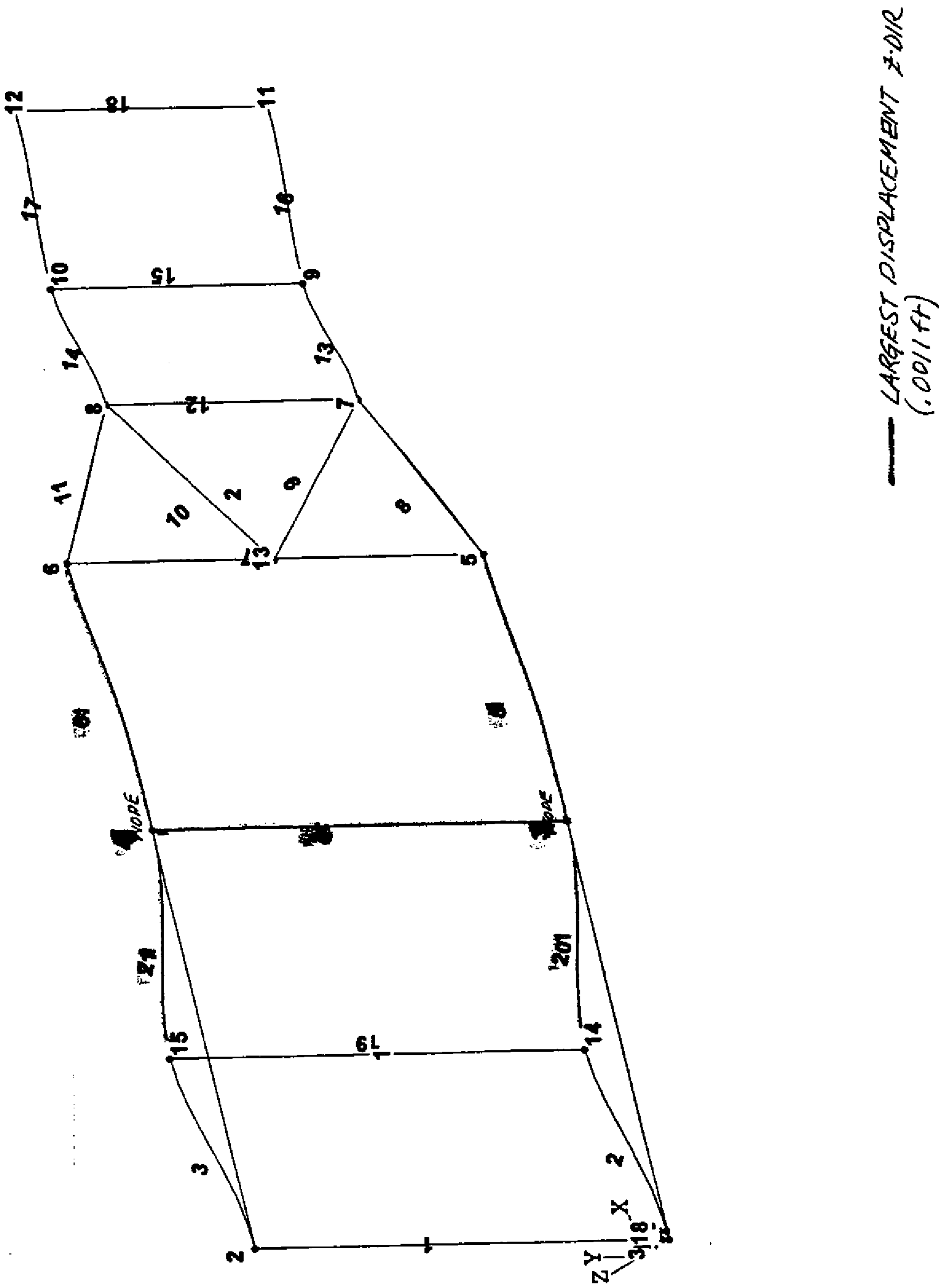
Seacoast Lawn and Tractor
Jim, thanks for helping us out last minute with the trans-axle.

TECH 797
"This work is the result of research sponsored in part, by the National Sea Grant College Program, NOAA, Department of Commerce, under grant #NA36RG0110 through the University of New Hampshire/University of Maine Sea Grant College Program."

APPENDIX

APPENDIX A

FRAME STRUCTURAL ANALYSIS RESULTS & DATA


LARGEST DISPLACEMENT Z-DIR
(.0011 ft)

UNH, UNH April 26, 1995

Durham, N.H., For Educational Use Only

Project Name: MUSCLES

Billing Info:

Default Units: ft, kips, deg, °F/ft

Complete Analysis Results Load Case: Dead-Load

NODAL DISPLACEMENTS

Node	DX	DY	DZ	RX	RY	RZ
1	0.0000	0.0000	0.0008	0.0000	0.0000	0.0000
2	0.0000	0.0000	0.0008	0.0000	0.0000	0.0000
[illegible]	0.0000	0.0000	0.0017	0.0000	0.0000	0.0000
[illegible]	0.0000	0.0000	[illegible]	0.0000	0.0000	0.0000
5	0.0000	0.0000	0.0007	0.0000	0.0000	0.0000
6	0.0000	0.0000	0.0007	0.0000	0.0000	0.0000
7	0.0000	0.0000	0.0006	0.0000	0.0000	0.0000
8	0.0000	0.0000	0.0006	0.0000	0.0000	0.0000
9	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000
10	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000
11	0.0000	0.0000	0.0003	0.0000	0.0000	0.0000
12	0.0000	0.0000	0.0003	0.0000	0.0000	0.0000
13	0.0000	0.0000	0.0006	0.0000	0.0000	0.0000
14	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000

15	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000
16	0.0000	0.0000	0.0008	0.0000	0.0000	0.0000
17	0.0000	0.0000	0.0008	0.0000	0.0000	0.0000
18	0.0000	0.0000	0.0008	0.0000	0.0000	0.0000

NODAL REACTIONS

Node	FX	FY	FZ	MX	MY	MZ
1	0.0000	0.0000	0.0000	-0.0198	-0.0241	0.0000
2	0.0000	0.0000	0.0000	0.0198	-0.0241	0.0000
3	0.0000	0.0000	0.0000	0.0198	0.1899	0.0000
4	0.0000	0.0000	0.0000	-0.0198	0.1899	0.0000
5	0.0000	0.0000	0.0000	0.1527	-0.2126	0.0000
6	0.0000	0.0000	0.0000	-0.1527	-0.2126	0.0000
7	0.0000	0.0000	0.0000	0.0003	-0.3595	0.0000
8	0.0000	0.0000	0.0000	-0.0003	-0.3595	0.0000
9	0.0000	0.0000	-0.5068	0.0000	-0.1380	0.0000
10	0.0000	0.0000	-0.5068	0.0000	-0.1380	0.0000
11	0.0000	0.0000	0.0000	0.0000	0.0372	0.0000
12	0.0000	0.0000	0.0000	0.0000	0.0372	0.0000
13	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000
14	0.0000	0.0000	-0.4067	0.0000	0.0281	0.0000
15	0.0000	0.0000	-0.4067	0.0000	0.0281	0.0000
16	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000

17	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000
18	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000

MEMBER LOCAL DISPLACEMENTS

Member	Offset	Dx	Dy	Dz
1				
	0.0000	0.0000	0.0000	0.0008
	1.2500	0.0000	0.0000	0.0008
	2.5000	0.0000	0.0000	0.0008
	3.7500	0.0000	0.0000	0.0008
	5.0000	0.0000	0.0000	0.0008
2				
	0.0000	0.0000	0.0000	0.0008
	0.5000	0.0000	0.0000	0.0007
	1.0000	0.0000	0.0000	0.0005
	1.5000	0.0000	0.0000	0.0002
	2.0000	0.0000	0.0000	0.0000
3				
	0.0000	0.0000	0.0000	0.0008
	0.5000	0.0000	0.0000	0.0007
	1.0000	0.0000	0.0000	0.0005
	1.5000	0.0000	0.0000	0.0002
	2.0000	0.0000	0.0000	0.0000

[redacted]

0.0000	0.0000	0.0000	[redacted]
1.2500	0.0000	0.0000	[redacted]
2.5000	0.0000	0.0000	[redacted]
3.7500	0.0000	0.0000	[redacted]
5.0000	0.0000	0.0000	[redacted]

[redacted]

0.0000	0.0000	0.0000	[redacted]
0.6675	0.0000	0.0000	[redacted]
1.3350	0.0000	0.0000	0.0010
2.0025	0.0000	0.0000	0.0008
2.6700	0.0000	0.0000	0.0007

[redacted]

0.0000	0.0000	0.0000	[redacted]
0.6675	0.0000	0.0000	[redacted]
1.3350	0.0000	0.0000	0.0010
2.0025	0.0000	0.0000	0.0008
2.6700	0.0000	0.0000	0.0007

7

0.0000	0.0000	0.0000	0.0007
1.2500	0.0000	0.0000	0.0007
2.5000	0.0000	0.0000	0.0007
3.7500	0.0000	0.0000	0.0007

	5.0000	0.0000	0.0000	0.0007
8				
	0.0000	0.0000	0.0000	0.0007
	0.4570	0.0000	0.0000	0.0007
	0.9139	0.0000	0.0000	0.0007
	1.3709	0.0000	0.0000	0.0006
	1.8278	0.0000	0.0000	0.0006
9				
	0.0000	0.0000	0.0000	-0.0006
	0.5357	0.0000	0.0000	-0.0006
	1.0713	0.0000	0.0000	-0.0006
	1.6070	0.0000	0.0000	-0.0006
	2.1426	0.0000	0.0000	-0.0006
10				
	0.0000	0.0000	0.0000	0.0006
	0.5357	0.0000	0.0000	0.0006
	1.0713	0.0000	0.0000	0.0006
	1.6070	0.0000	0.0000	0.0006
	2.1426	0.0000	0.0000	0.0006
11				
	0.0000	0.0000	0.0000	0.0007
	0.4570	0.0000	0.0000	0.0007
	0.9139	0.0000	0.0000	0.0007

	1.3709	0.0000	0.0000	0.0006
	1.8278	0.0000	0.0000	0.0006
12				
	0.0000	0.0000	0.0000	0.0006
	0.7500	0.0000	0.0000	0.0006
	1.5000	0.0000	0.0000	0.0006
	2.2500	0.0000	0.0000	0.0006
	3.0000	0.0000	0.0000	0.0006
13				
	0.0000	0.0000	0.0000	0.0006
	0.3125	0.0000	0.0000	0.0005
	0.6250	0.0000	0.0000	0.0003
	0.9375	0.0000	0.0000	0.0001
	1.2500	0.0000	0.0000	0.0000
14				
	0.0000	0.0000	0.0000	0.0006
	0.3125	0.0000	0.0000	0.0005
	0.6250	0.0000	0.0000	0.0003
	0.9375	0.0000	0.0000	0.0001
	1.2500	0.0000	0.0000	0.0000
15				
	0.0000	0.0000	0.0000	0.0000
	0.7500	0.0000	0.0000	0.0000

	1.5000	0.0000	0.0000	0.0000
	2.2500	0.0000	0.0000	0.0000
	3.0000	0.0000	0.0000	0.0000
16				
	0.0000	0.0000	0.0000	0.0000
	0.4175	0.0000	0.0000	0.0001
	0.8350	0.0000	0.0000	0.0002
	1.2525	0.0000	0.0000	0.0003
	1.6700	0.0000	0.0000	0.0003
17				
	0.0000	0.0000	0.0000	0.0000
	0.4175	0.0000	0.0000	0.0001
	0.8350	0.0000	0.0000	0.0002
	1.2525	0.0000	0.0000	0.0003
	1.6700	0.0000	0.0000	0.0003
18				
	0.0000	0.0000	0.0000	0.0003
	0.7500	0.0000	0.0000	0.0003
	1.5000	0.0000	0.0000	0.0003
	2.2500	0.0000	0.0000	0.0003
	3.0000	0.0000	0.0000	0.0003
19				
	0.0000	0.0000	0.0000	0.0000

1.2500	0.0000	0.0000	0.0000
2.5000	0.0000	0.0000	0.0000
3.7500	0.0000	0.0000	0.0000
5.0000	0.0000	0.0000	0.0000

20

0.0000	0.0000	0.0000	0.0000
0.5000	0.0000	0.0000	0.0002
1.0000	0.0000	0.0000	0.0006
1.5000	0.0000	0.0000	0.0009
2.0000	0.0000	0.0000	0.0011

21

0.0000	0.0000	0.0000	0.0000
0.5000	0.0000	0.0000	0.0002
1.0000	0.0000	0.0000	0.0006
1.5000	0.0000	0.0000	0.0009
2.0000	0.0000	0.0000	0.0011

MEMBER INTERNAL FORCES

Member	Offset	Fx	Fy	Fz	Mx	My	Mz
1							
	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000
	1.2500	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000
	2.5000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000

APPENDIX B

STEERING & SUSPENSION ANALYSIS RESULTS

Steering and Suspension System Analysis:

The steering unit is designed for use at much higher speeds and with much higher loads than will occur during the mission. Thus, it was assumed to be more than adequate for use.

The analysis will be concerned with the design of the hub mounts, bushings, and the spring size.

To find the forces and reactions of the different suspension components, a kinematic and dynamic analysis was started. Because of the orientation of the suspension system components, this analysis simplified to a few simple force balance equations.

Assumptions:
- The mass of the components is negligible compared to the mass of the vehicle.
- The weight of the vehicle is parallel to the strut and stays very close to this through out the motion of the suspension.

Each piece can be analyzed seperate from the rest of the system:

Determination of spring rate:

$Weightoffront := 600$ lbs

$Weightpertire := \frac{Weightoffront}{2}$

$x := 3$ inches Desired compression of springs to obtain normal ride height

$SpringRate := \frac{Weightpertire}{x}$

$SpringRate = 100$ lbs/inch

Maximum force applied to polyethelyne bushings

$c := 6$ inches Maximum travel of the suspension system

$Force := (x + c) \cdot SpringRate$

$Force = 900$ lbs $F := \frac{Force}{2}$ $F = 450$ lbs on each bushing bolt

$width\_of\_bushing := 4$ inches

$diameter\_of\_hole := 0.75$ inches

$thickness := 1$ inch

$A := (width\_of\_bushing - diameter\_of\_hole) \cdot thickness$

$A = 3.25$ in^2 Shear area of bushing

$Kt := 6$ stress concentration factor

$\sigma := \frac{Kt \cdot F}{A}$ $\sigma = 830.769$ psi Maximum stress on the bushing

Bushing material: high density polyethelyne

$Tensile\_strength := 3000$ psi

Analysis of hub/spindle mounts:

The hubs and spindles are manufactured for use on a boat trailer and are rated at 1300 pounds each.

APPENDIX C

DRIVE TRAIN ANALYSIS RESULTS

Drivetrain Analysis:

Determination of required engine power:

$Tireradius := 12.5$ inches

$Vehiclespeed := 3$ Miles per hour

$Vehicleweight := 1200$ Pounds

$Resistancefactor := .35$ percent of total vehicle weight needed to roll vehicle

$Drivetrainefficiency := .8$ efficiency power transferal

$Maximumincline := 20$ Maximum expected incline in degrees

$$Forceuphill := \sin\left(\frac{Maximumincline}{180}\cdot 2\cdot\pi\right)\cdot Vehicleweight$$

$Forceuphill = 771.345$

$Force := Resistancefactor\cdot Vehicleweight + Forceuphill$

$Force = 1.191\cdot 10^3$ Pounds

$$Rearwheeltorque := \frac{Force\cdot Tireradius}{12}$$

$Rearwheeltorque = 1.241\cdot 10^3$ Foot*Pounds — This is more torque than could possibly be applied by the rear tires

$$ReartireRPM := Vehiclespeed\cdot\frac{88}{60}\cdot\frac{12}{2\cdot\pi\cdot Tireradius}\cdot 2\cdot\pi$$

$ReartireRPM = 4.224$ radians/second

$$Enginepower := \frac{Rearwheeltorque\cdot\frac{ReartireRPM}{550}}{.8}$$

$Enginepower = 11.913$ Horsepower

The transaxle being used has 5 speeds therefore a lower gear can be selected if needed

Maximum force needed to be supported by the chain:

$Rearsprocketradius := 2.5$

$Maximumtorque := Rearwheeltorque$

$$Forceonchain := \frac{\frac{Maximumtorque}{\left(\frac{Rearsprocketradius}{12}\right)}}{2}$$

$Forceonchain = 2.978\cdot 10^3$ Pounds

Required braking torque:

$staticfrictioncoef := .9$

$Maxforce := .6 \cdot Vehicleweight \cdot staticfrictioncoef$

$Maxforce = 648$ **Pounds**

$FS := 2$

$$Neededtorque := Maxforce \cdot FS \cdot \frac{Tireradius}{12}$$

$Neededtorque = 1.35 \cdot 10^3$ **Foot*pounds**

Needed diameter of the rear axle:

$YSsteel := 60000$ $offset := 10$

$$Maxtorque := staticfrictioncoef \cdot \frac{Vehicleweight}{2}$$

$$Bendingmoment := \frac{Vehicleweight}{4} \cdot (offset)$$

$Bendingmoment = 3 \cdot 10^3$

$$Allowableshearstress := \frac{YSsteel}{2 \cdot FS}$$ $Allowableshearstress = 1.5 \cdot 10^4$

$$shaftdiameter := \left[\frac{16}{\pi \cdot Allowableshearstress} \cdot \left(Bendingmoment^2 + Maxtorque^2\right)^{\left(\frac{1}{2}\right)}\right]^{\left(\frac{1}{3}\right)}$$

$shaftdiameter = 1.012$

Stress due to bending

$Rearwheeloffset := 8$ $r := \frac{1.25}{2}$

$MaxForce := 600$

$Moment := Maxforce \cdot Rearwheeloffset$

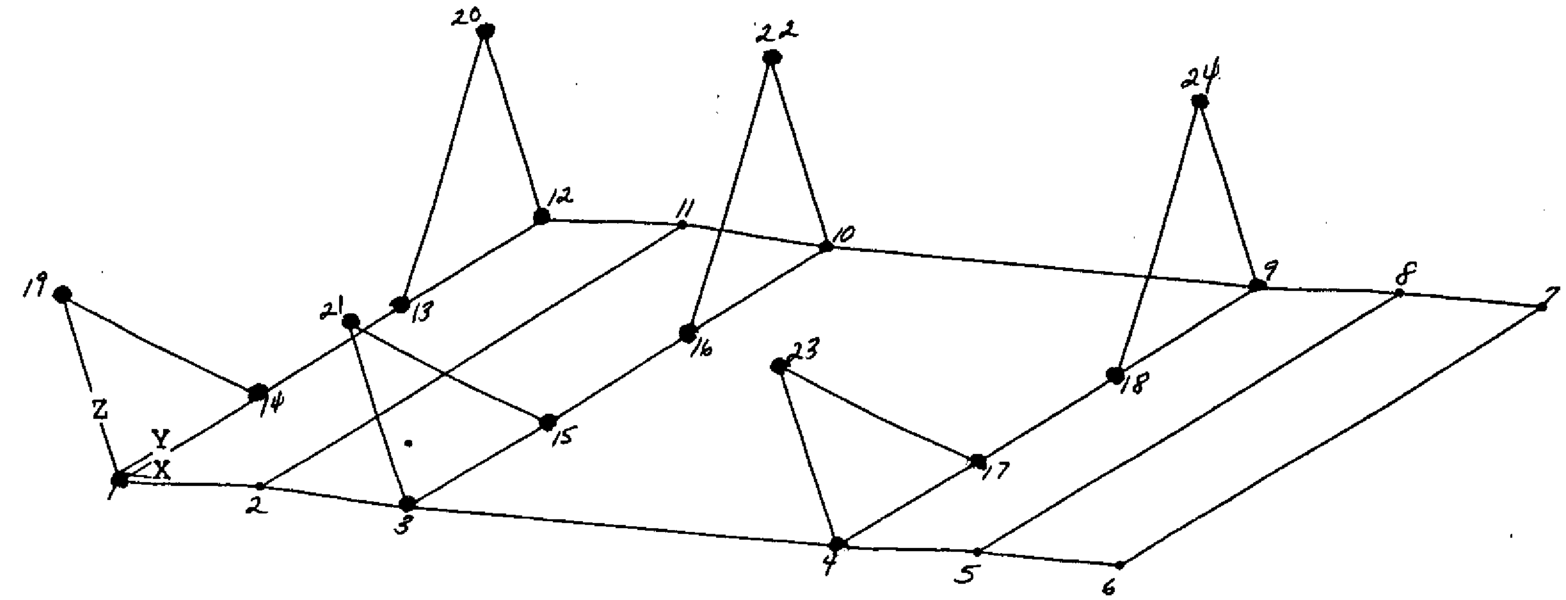
$Moment = 5.184 \cdot 10^3$

$I := .25 \cdot \pi \cdot r^4$

$stress := Moment \cdot \frac{r}{I}$ $stress = 2.704 \cdot 10^4$ **psi**

APPENDIX D

CART STRUCTURAL ANALYSIS RESULTS & DATA



NODE LOCATIONS

● NODAL DISPLACEMENT = -.003 ft z-dir

UNH, UNH April 27, 1995

Durham, N.H., For Educational Use Only

Project Name: cart

Billing Info:

Default Units: ft, kips, deg, °F/ft

Complete Analysis Results Load Case: dead load

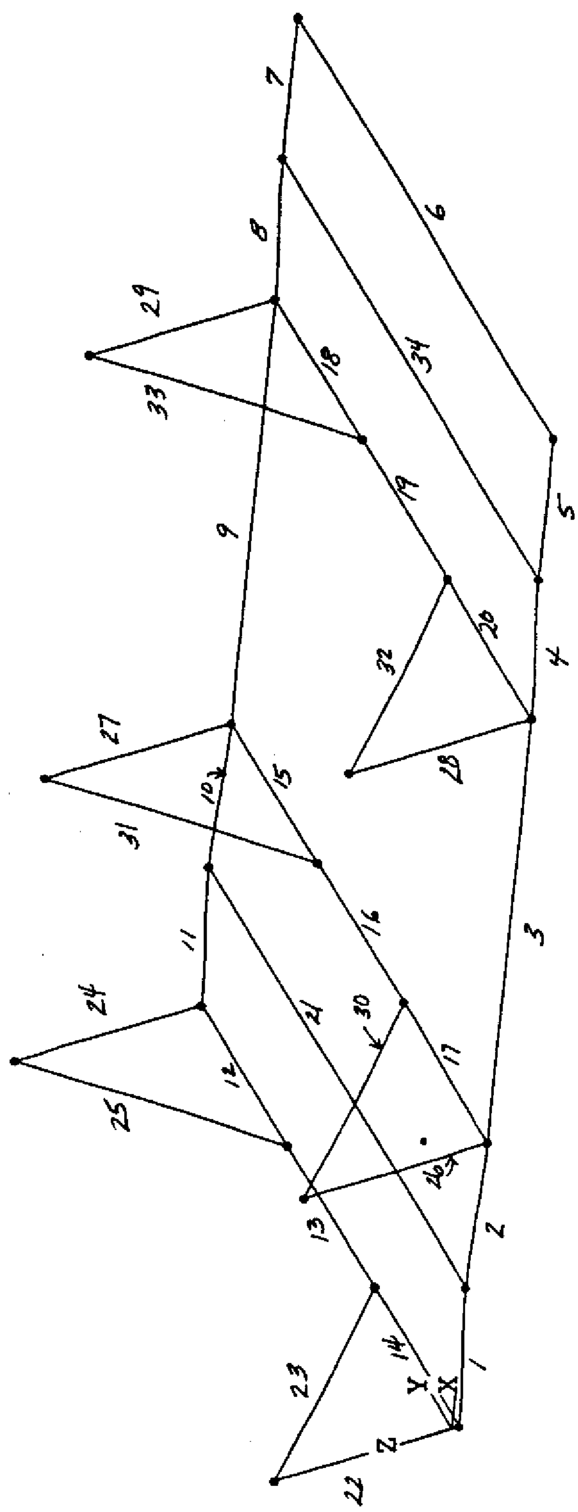
NODAL DISPLACEMENTS

Node	DX	DY	DZ	RX	RY	RZ
1	0.000	0.000	-0.003	0.000	0.000	0.000
2	0.000	0.000	0.000	0.000	0.000	0.000
3	0.000	0.000	-0.003	0.000	0.000	0.000
4	0.000	0.000	-0.003	0.000	0.000	0.000
5	0.000	0.000	0.000	0.000	0.000	0.000
6	0.000	0.000	0.000	0.000	0.000	0.000
7	0.000	0.000	0.000	0.000	0.000	0.000
8	0.000	0.000	0.000	0.000	0.000	0.000
9	0.000	0.000	-0.003	0.000	0.000	0.000
10	0.000	0.000	-0.003	0.000	0.000	0.000
11	0.000	0.000	0.000	0.000	0.000	0.000
12	0.000	0.000	-0.003	0.000	0.000	0.000
13	0.000	0.000	-0.003	0.000	0.000	0.000
14	0.000	0.000	-0.003	0.000	0.000	0.000

15	0.000	0.000	-0.003	0.000	0.000	0.000
16	0.000	0.000	-0.003	0.000	0.000	0.000
17	0.000	0.000	-0.003	0.000	0.000	0.000
18	0.000	0.000	-0.003	0.000	0.000	0.000
19	0.000	0.000	-0.003	0.000	0.000	0.000
20	0.000	0.000	-0.003	0.000	0.000	0.000
21	0.000	0.000	-0.003	0.000	0.000	0.000
22	0.000	0.000	-0.003	0.000	0.000	0.000
23	0.000	0.000	-0.003	0.000	0.000	0.000
24	0.000	0.000	-0.003	0.000	0.000	0.000

NODAL REACTIONS

Node	FX	FY	FZ	MX	MY	MZ
1	0.000	0.000	0.000	0.008	0.288	0.000
2	0.000	0.000	1.151	0.000	0.001	0.000
3	0.000	0.000	0.000	0.011	-0.287	0.000
4	0.000	0.000	0.000	0.011	0.287	0.000
5	0.000	0.000	0.574	0.000	0.287	0.000
6	0.000	0.000	0.000	0.000	0.000	0.000
7	0.000	0.000	0.000	0.000	0.000	0.000
8	0.000	0.000	0.574	0.000	0.287	0.000
9	0.000	0.000	0.000	-0.011	0.287	0.000
10	0.000	0.000	0.000	-0.011	-0.287	0.000



MEMBER LOCATIONS

11	0.000	0.000	1.147	0.000	-0.001	0.000
12	0.000	0.000	0.000	-0.011	0.286	0.000
13	0.000	-0.338	0.000	0.000	0.000	0.000
14	0.000	0.348	0.000	-0.001	0.000	0.000
15	0.000	0.340	0.000	0.001	0.000	0.000
16	0.000	-0.340	0.000	-0.001	0.000	0.000
17	0.000	0.340	0.000	0.001	0.000	0.000
18	0.000	-0.340	0.000	-0.001	0.000	0.000
19	0.000	-0.348	0.000	0.036	0.000	0.000
20	0.000	0.338	0.000	-0.036	0.000	0.000
21	0.000	-0.340	0.000	0.036	0.000	0.000
22	0.000	0.340	0.000	-0.036	0.000	0.000
23	0.000	-0.340	0.000	0.036	0.000	0.000
24	0.000	0.340	0.000	-0.036	0.000	0.000

MEMBER LOCAL DISPLACEMENTS

Member	Offset	Dx	Dy	Dz
1				
	0.000	0.000	0.000	-0.003
	0.250	0.000	0.000	-0.003
	0.500	0.000	0.000	-0.002
	0.750	0.000	0.000	0.000
	1.000	0.000	0.000	0.000

2

0.000	0.000	0.000	0.000
0.250	0.000	0.000	0.000
0.500	0.000	0.000	-0.002
0.750	0.000	0.000	-0.003
1.000	0.000	0.000	-0.003

3

0.000	0.000	0.000	-0.003
0.750	0.000	0.000	-0.003
1.500	0.000	0.000	-0.003
2.250	0.000	0.000	-0.003
3.000	0.000	0.000	-0.003

4

0.000	0.000	0.000	-0.003
0.250	0.000	0.000	-0.003
0.500	0.000	0.000	-0.002
0.750	0.000	0.000	0.000
1.000	0.000	0.000	0.000

5

0.000	0.000	0.000	0.000
0.250	0.000	0.000	0.000
0.500	0.000	0.000	0.000
0.750	0.000	0.000	0.000

	1.000	0.000	0.000	0.000
6				
	0.000	0.000	0.000	0.000
	0.750	0.000	0.000	0.000
	1.500	0.000	0.000	0.000
	2.250	0.000	0.000	0.000
	3.000	0.000	0.000	0.000
7				
	0.000	0.000	0.000	0.000
	0.250	0.000	0.000	0.000
	0.500	0.000	0.000	0.000
	0.750	0.000	0.000	0.000
	1.000	0.000	0.000	0.000
8				
	0.000	0.000	0.000	0.000
	0.250	0.000	0.000	0.000
	0.500	0.000	0.000	0.002
	0.750	0.000	0.000	0.003
	1.000	0.000	0.000	0.003
9				
	0.000	0.000	0.000	0.003
	0.750	0.000	0.000	0.003
	1.500	0.000	0.000	0.003

	2.250	0.000	0.000	0.003
	3.000	0.000	0.000	0.003
10				
	0.000	0.000	0.000	0.003
	0.250	0.000	0.000	0.003
	0.500	0.000	0.000	0.002
	0.750	0.000	0.000	0.000
	1.000	0.000	0.000	0.000
11				
	0.000	0.000	0.000	0.000
	0.250	0.000	0.000	0.000
	0.500	0.000	0.000	0.002
	0.750	0.000	0.000	0.003
	1.000	0.000	0.000	0.003
12				
	0.000	0.000	0.000	-0.003
	0.250	0.000	0.000	-0.003
	0.500	0.000	0.000	-0.003
	0.750	0.000	0.000	-0.003
	1.000	0.000	0.000	-0.003
13				
	0.000	0.000	0.000	-0.003
	0.250	0.000	0.000	-0.003

	0.500	0.000	0.000	-0.003
	0.750	0.000	0.000	-0.003
	1.000	0.000	0.000	-0.003
14				
	0.000	0.000	0.000	-0.003
	0.250	0.000	0.000	-0.003
	0.500	0.000	0.000	-0.003
	0.750	0.000	0.000	-0.003
	1.000	0.000	0.000	-0.003
15				
	0.000	0.000	0.000	-0.003
	0.250	0.000	0.000	-0.003
	0.500	0.000	0.000	-0.003
	0.750	0.000	0.000	-0.003
	1.000	0.000	0.000	-0.003
16				
	0.000	0.000	0.000	-0.003
	0.250	0.000	0.000	-0.003
	0.500	0.000	0.000	-0.003
	0.750	0.000	0.000	-0.003
	1.000	0.000	0.000	-0.003
17				
	0.000	0.000	0.000	-0.003

	0.250	0.000	0.000	-0.003
	0.500	0.000	0.000	-0.003
	0.750	0.000	0.000	-0.003
	1.000	0.000	0.000	-0.003
18				
	0.000	0.000	0.000	-0.003
	0.250	0.000	0.000	-0.003
	0.500	0.000	0.000	-0.003
	0.750	0.000	0.000	-0.003
	1.000	0.000	0.000	-0.003
19				
	0.000	0.000	0.000	-0.003
	0.250	0.000	0.000	-0.003
	0.500	0.000	0.000	-0.003
	0.750	0.000	0.000	-0.003
	1.000	0.000	0.000	-0.003
20				
	0.000	0.000	0.000	-0.003
	0.250	0.000	0.000	-0.003
	0.500	0.000	0.000	-0.003
	0.750	0.000	0.000	-0.003
	1.000	0.000	0.000	-0.003
21				

	0.000	0.000	0.000	0.000
	0.750	0.000	0.000	0.000
	1.500	0.000	0.000	0.000
	2.250	0.000	0.000	0.000
	3.000	0.000	0.000	0.000
22				
	0.000	-0.003	0.000	0.000
	0.250	-0.003	0.000	0.000
	0.500	-0.003	0.000	0.000
	0.750	-0.003	0.000	0.000
	1.000	-0.003	0.000	0.000
23				
	0.000	0.002	-0.002	0.000
	0.354	0.002	-0.003	0.000
	0.707	0.002	-0.003	0.000
	1.061	0.002	-0.003	0.000
	1.414	0.002	-0.002	0.000
24				
	0.000	-0.003	0.000	0.000
	0.250	-0.003	0.000	0.000
	0.500	-0.003	0.000	0.000
	0.750	-0.003	0.000	0.000
	1.000	-0.003	0.000	0.000

25				
	0.000	0.002	0.002	0.000
	0.354	0.002	0.003	0.000
	0.707	0.002	0.003	0.000
	1.061	0.002	0.003	0.000
	1.414	0.002	0.002	0.000
26				
	0.000	-0.003	0.000	0.000
	0.250	-0.003	0.000	0.000
	0.500	-0.003	0.000	0.000
	0.750	-0.003	0.000	0.000
	1.000	-0.003	0.000	0.000
27				
	0.000	-0.003	0.000	0.000
	0.250	-0.003	0.000	0.000
	0.500	-0.003	0.000	0.000
	0.750	-0.003	0.000	0.000
	1.000	-0.003	0.000	0.000
28				
	0.000	-0.003	0.000	0.000
	0.250	-0.003	0.000	0.000
	0.500	-0.003	0.000	0.000
	0.750	-0.003	0.000	0.000

	1.000	-0.003	0.000	0.000
29				
	0.000	-0.003	0.000	0.000
	0.250	-0.003	0.000	0.000
	0.500	-0.003	0.000	0.000
	0.750	-0.003	0.000	0.000
	1.000	-0.003	0.000	0.000
30				
	0.000	0.002	-0.002	0.000
	0.354	0.002	-0.003	0.000
	0.707	0.002	-0.003	0.000
	1.061	0.002	-0.003	0.000
	1.414	0.002	-0.002	0.000
31				
	0.000	0.002	0.002	0.000
	0.354	0.002	0.003	0.000
	0.707	0.002	0.003	0.000
	1.061	0.002	0.003	0.000
	1.414	0.002	0.002	0.000
32				
	0.000	0.002	-0.002	0.000
	0.354	0.002	-0.003	0.000
	0.707	0.002	-0.003	0.000

1.061	0.002	-0.003	0.000
1.414	0.002	-0.002	0.000

33

0.000	0.002	0.002	0.000
0.354	0.002	0.003	0.000
0.707	0.002	0.003	0.000
1.061	0.002	0.003	0.000
1.414	0.002	0.002	0.000

34

0.000	0.000	0.000	0.000
0.750	0.000	0.000	0.000
1.500	0.000	0.000	0.000
2.250	0.000	0.000	0.000
3.000	0.000	0.000	0.000

MEMBER INTERNAL FORCES

Member	Offset	Fx	Fy	Fz	Mx	My	Mz
1							
	0.000	0.000	0.000	0.576	0.000	-0.288	0.000
	0.250	0.000	0.000	0.576	0.000	-0.144	0.000
	0.500	0.000	0.000	0.576	0.000	0.000	0.000
	0.750	0.000	0.000	0.576	0.000	0.144	0.000
	1.000	0.000	0.000	0.576	0.000	0.288	0.000

APPENDIX E

CART WEIGHT & DISPLACEMENT ESTIMATES

CART WEIGHT AND DISPLACEMENT ESTIMATES (EXCLUDING SUBMERSIBLE)

Weight

number of lateral members = 6 — $6 \cdot 3 = 18$ ft

number of longtitudinal members = 2 — $2 \cdot 7 = 14$ ft

number of triangular pieces = 6 — $6 \cdot 2.67 = 16.02$ ft

number of rods (for rollers) = 6 — $6 \cdot 1 = 6$ ft

material properties of steel angle

$w := 2.34$ lb/ft $A := .69$ in^2 $\rho steel := 490.752$ lb/ft^3

$Wangle := 48.02 \cdot w$ $Wangle = $ lb2.367 lb

roller rods

$r := \frac{.3125}{12}$ ft $V := 6 \cdot \pi \cdot r^2$ $V = 0.013$ ft^3

$Wrods := V \cdot \rho steel$ $Wrods = 6.273$ lb

$Wcomponents := 100$ lb

$Wtotal := Wangle + Wrods + Wcomponents$ $Wtotal = 218.64$ lb

Displacement

tire dimensions

$r := 6$ in $h := 6$ in

$Vtires := 4 \cdot \pi \cdot r^2 \cdot h$ $Vtires = 2.714 \cdot 10^3$ in^3

$Vstructure := A \cdot 48.02 \cdot 12$ $Vstructure = 397.60$in^3

$Vtotal := Vtires + Vstructure$ $Vtotal = 3.112 \cdot 10^3$ in^3

$displacement := \frac{Vtotal}{12^3} \cdot \frac{2240}{35}$ $displacement = 115.257$ lb (Buoyancy force)

weight force >buoyancy force

negatively buoyant cart