Conceptual Design and Analysis of Roads and Road Construction Machinery for Initial Lunar Base Operations

Submitted to:

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MECHANICAL ENGINEERING DESIGN PROJECTS PROGRAM
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9 April 1990

Mr. James Aliberti
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Dear Mr. Aliberti:

Attached is our final report for the design of a road construction system for initial lunar base operations. Our design team selected a compacted lunar soil road as the most feasible type of road for initial lunar base operations and concluded that only a grader and compactor were necessary to build this road. The road construction system we developed consists of a main drive unit for propulsion and control, a detachable grader assembly, and a towed compactor.

This report contains a complete description of the alternate designs we investigated, our decision matrices, and the final design solution we developed, including machinery configuration, mass, and power requirements, and operating features of each of the three machines. We also present a thorough evaluation of the design and recommendations for further work.

It has been a pleasure working on this project for NASA/USRA, and we look forward to seeing you at our design presentation. The presentation will take place on Tuesday, April 24, 1990, at 10:00 am in the Engineering Teaching Center II, Room 4.110, on the campus of The University of Texas at Austin. A catered luncheon will follow the presentation.

Sincerely,

Joel Banks

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ACKNOWLEDGEMENTS

The design team members thank the National Aeronautics and Space Administration and the Universities Space Research Association (NASA/USRA) for sponsoring this project and Mr. James Aliberti for his continued support of our project.

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A special thanks goes to Dr. Kristin Wood (faculty advisor) for his unending support and valuable technical expertise on many different areas of our project. His enthusiasm and optimism gave us a positive attitude for our project at all times and made us strive to produce the best design we could.

We also thank Mr. Brian Muirhead and Mr. Don Bickler (JPL engineers) for their assistance in the areas of material selection and lunar soil mechanics. Mr. Frank Wood (of the New Mexico State Highway and Transportation Department) provided much needed information on terrestrial road construction and machinery.

We also express our appreciation to Professor Torfason for helping us maintain our design notebooks, to Mr. Wendell Deen for critiquing our design drawings, and to Mr. Bert Herigstad for providing administrative services, material, and advice.

Finally, we would like to thank Dr. Steven P. Nichols for overseeing the Mechanical Engineering Design Projects Program and providing informative and practical lectures on the many different aspects of engineering.

ABSTRACT

Conceptual Design and Analysis of Roads and Road Construction Machinery for Initial Lunar Base Operations

Recent developments have made it possible for scientists and engineers to consider returning to the Moon to build a manned lunar base. The base can be used to conduct scientific research, develop new space technology, and utilize the natural resources of the Moon. Areas of the base will be separated, connected by a system of roads that reduce the power requirements of vehicles traveling on them. For a senior design project sponsored by the University of Texas Mechanical Engineering Design Project Program, NASA/USRA asked the design team to analyze feasible road types for the lunar surface and design a road construction system for initial lunar base operations. The design team also constructed a model to show the system configuration and key operating features.

The alternate designs for the lunar road construction system were developed in four stages: analyze and select a road type, determine operations and machinery needed to produce the road, develop machinery configurations, and develop alternates for several machine components.

The design team selected a compacted lunar soil road for initial lunar base operations. The only machinery required to produce this road were a grader and a compactor. The road construction system the design team developed consists of a main drive unit which is used for propulsion, a detachable grader assembly, and a towed compactor.

KEY WORDS: MOTOR GRADER, COMPACTOR, ROAD CONSTRUCTION, LUNAR BASE

Joel Banks

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INTRODUCTION

In 1958, the United States Congress established the National Aeronautics and Space Administration (NASA), an independent government agency whose purposes include coordinating and conducting space research and exploration. NASA has many research centers across the United States. One of the main centers, the Lyndon B. Johnson Space Center (JSC), is located near Houston, Texas. In 1969, the National Academy of Sciences created the Universities Space Research Association (USRA), also based in Houston, to encourage joint projects between universities across the United States and commercial research companies. Together, NASA and USRA have established the Advanced Design Program, which brings NASA engineers and university students and faculty together on projects. The main purposes of the Advanced Design Program are to enhance the students' design experience and provide NASA engineers with new design ideas for their research.

In conjunction with the University of Texas at Austin's Mechanical Engineering Design Project Program, NASA/USRA has asked he design team to analyze feasible road types for the lunar surface and design a road construction system for initial lunar base operations.

Background

In July 1969, Neil Armstrong and Edwin Aldrin made the first moon landing. Recent developments have made it possible for JSC scientists and engineers to consider returning to the Moon and building a manned lunar base soon after the year 2000. The base can be used to conduct scientific research, develop new space technology, and utilize the natural resources of the Moon. The Solar System Exploration Division at JSC has recommended a site on the Mare Orientale plains as a possible location for the lunar outpost [1].* This region has a relatively high concentration of minerals which can be mined and processed to extract lunar oxygen (LOX). LOX will be an important element in life support systems, water production, and fuel for the lunar landers and other vehicles.

One lunar base design is comprised of a habitat facility, a lunar oxygen pilot plant, a power plant, and a landing site. (See Figure 1.) The landing site must be separated from the habitation area in order to minimize the damage from material expelled by rocket exhaust, chemical contamination from the rocket plumes, and potential damage in case of a lander crash or explosion. The power plant will be separated from the other areas of the base for safety reasons. One way to connect these areas together and provide easy travel between them is by a system of roads. These roads must be smoother than the natural surface to reduce the power requirements of vehicles traveling on them. In addition, a finished surface will reduce problems caused by dust raised from the lunar surface by the vehicles.

* References are listed on pages 77-79 of this report.

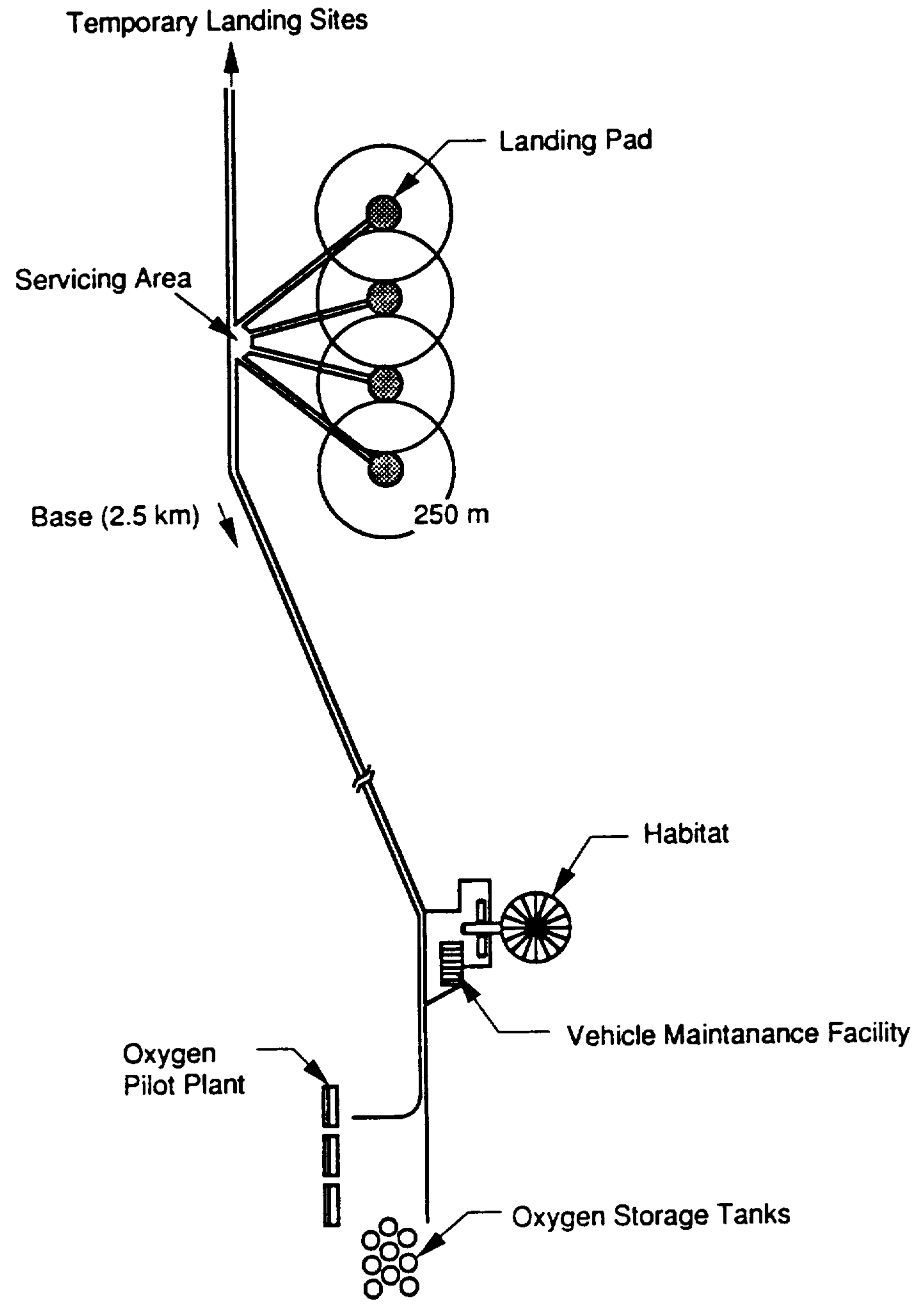


Figure 1: ROADS CONNECTING THE DIFFERENT AREAS OF THE PROPOSED LUNAR BASE (FROM LUNAR OUTPOST, ADVANCED PROGRAM OFFICE, JOHNSON SPACE CENTER. USED WITH PERMISSION.)

Project Requirements

The design team was asked to accomplish the following two tasks:

1. Select a feasible road type for the lunar surface and design a road construction system for initial lunar base operations.
2. Construct a demonstration model to show the system configuration and key operating features.

Design Criteria

NASA/USRA has set forth the following design goals for all lunar base equipment:

1. Minimize mass and size. The mass of the system must be minimized to reduce the cost of transportation to the lunar surface, and the size must be minimized to meet the space requirements on the lift vehicle.
2. Minimize power required for operation. Power will be a valuable commodity on the Moon and there will be intense competition for the limited amount of power that will be available. For this reason, the power consumption of the system should be minimized.
3. Use materials and components which are resistant to the lunar environment. All material and components must be able to withstand the harsh lunar environment, including a lack of atmosphere (vacuum), extreme temperature variations, high solar and galactic radiation, and abrasive lunar dust.
4. Minimize manpower. The first phases of construction of the lunar base will be dependent on supplies brought from Earth. In order to conserve resources such as food, water, and oxygen, the machinery should require minimal human operation. The machinery should be automated or require little human attention so that the astronauts will be able to concentrate on other tasks.

The design team set additional design criteria for the road construction system:

5. Maximize operator safety. Safety is the highest priority in all space operations. The road construction system must incorporate several safety features to minimize the risk of harm to the operator and base facilities.

6. Maximize the use of indigenous lunar materials for road construction. Due to the high cost of transportation, the design team must avoid construction processes which required materials from Earth.

7. Produce a smooth road. A smooth surface (compared to the natural lunar surface layer) is needed to reduce the power requirements of vehicles traveling on the road.

8. Multiple-purpose components. The road construction system must perform all operations necessary to produce a finished road. When road construction is not in progress, the components of the system should be adaptable to other tasks such as mining and transportation.

9. Easily maintained roads and machinery. Due to limited manpower on the lunar base, the roads and machinery must require little maintenance to remain functional. Maintenance procedures should be simple and take into account the limited mobility of EVA suits and remote manipulators. The procedures should also be designed to take little time.

Solution Methodology

The first phase of the design project was a thorough review of pertinent background information. This review included information about the physical characteristics of the lunar surface layer, the effects of the lunar environment on the materials used in the design of the construction equipment, and the methods and machinery used to construct roads on Earth. The design team also conducted a patent search of motor graders and grader equipment.

The design team examined several alternates for feasible road types for the lunar surface. The best alternate was the one that required the least amount of energy to construct, minimized the use of non-lunar materials, was easy to maintain, and provided a smooth surface to minimize the power requirements of the vehicles traveling on the roads.

The design team selected the type of road, defined the construction processes needed to produce the road, and developed several alternate designs for the road construction system. The best overall configuration was chosen for full development based on the design criteria given above and on criteria that were developed during the formulation and synthesis of the alternates.

The design team also built a scale model of the road construction system. This model shows the overall configuration and demonstrates key operating features of the machinery.

ALTERNATE DESIGNS

The alternate designs for the lunar road construction system were developed in four stages: analysis and selection of a road type, determination of the operations and machinery needed to produce the road, development of machinery configurations, and development of alternates for several machine components.

The design team examined the types of roads that are constructed on Earth to determine which road type was suitable for initial lunar base development and what operations and machinery were required to construct the road. A simple dirt road was selected as the most feasible road type, with only a grader and a compactor needed for construction.

To determine the overall machinery configuration, the design team considered both the modification of terrestrial graders and compactors and the development of entirely new machines. The alternates were designed considering the high energy costs and limited amount of space available for transport to the Moon, the effects of the lunar environment on the machinery, and the ability to adapt the machinery for uses other than their primary tasks.

The alternates for individual components for the grader and compactor were developed independently from the general configuration of the machinery. The components the design team concentrated on were the blade and drive mechanism for the grader and the roller for the compactor.

SELECTION OF A FEASIBLE ROAD TYPE

The first design issue was the selection of the type of road to build on the lunar surface. This choice specified the materials needed and the operations required to produce the road, which formed the basis for the design of the construction machinery.

There were several criteria for the selection of a feasible road type for the lunar surface. The road surface must be smooth and firm to reduce vehicle power requirements, must withstand the lunar environment and vehicle traffic, and must be easy to maintain. The construction process must minimize energy use and the need for non-lunar materials. In addition, road building should begin during the early stages of construction of the lunar base, and not depend on other activities such as mining or oxygen extraction to provide materials or equipment.

The types of roads the design team studied were those with paved surfaces (concrete, asphalt, paving tiles, and fused soil) and those using only compacted lunar soil (gravel and dirt).

Characteristics of Road Surfaces

Concrete

Concrete is a strong, durable material which can be formed with a smooth surface finish. It can withstand extreme temperature variations with little loss of strength, is resistant to the radiation and vacuum, and is resistant to abrasive wear from vehicle traffic.

However, concrete does have limitations. It requires reinforcement to provide tensile strength, and point impacts (such as from micrometeorites) can cause fragmentation of the surface. Thermal shock from rapidly changing temperature can also cause cracking.

Most lunar soils and rocks can be processed to produce cement. This is done by melting the rock in a vacuum and boiling off the more volatile elements. If the process is carried out at 1800 degrees Celsius (°C), the composition of the molten residue becomes that of alumina cement. However, this process has a low yield, producing only 5% cement by weight from the original material. The process is also energy-intensive and requires complicated equipment [2].

Concrete also requires water, which must be brought from the Earth or produced on the Moon from oxygen and hydrogen gases. Oxygen can be extracted from lunar soils (which is one of the primary resource development objectives of the lunar base), but virtually all of the hydrogen must be brought from Earth. The production of water also requires a large amount of energy. As a result, water will be too precious a commodity to be produced and then locked away in concrete road surfaces [3].

Concrete must also be cast and cured under controlled temperature and humidity for up to 28 days. This time is needed for the concrete to reach full strength and to remove all excess water to prevent cracking from freezing or rapid drying [4]. Since the lunar environment cannot be regulated, the concrete cannot be cast in place to produce the road surface. Facilities with a controlled environment will have to be built to cast and cure slabs of concrete, and these slabs will then have to be moved out to the road site.

Although the physical properties of concrete are well suited to road paving, the great cost in energy and materials makes it unsatisfactory for use on the lunar base. In addition, the manufacturing facilities will not be available during the initial stages of the construction of a lunar base.

Asphalt

Another choice for paving materials was to use asphalt (a heavy, viscous petroleum product) as a binder for lunar gravel or soil. Asphalt produces a surface which is more resilient than concrete, is fairly resistant to vehicle wear, and is easily maintained and repaired. However, there are severe problems with this material. All asphalt must be brought from Earth since it is unlikely that suitable compounds will be found on the Moon.

The asphalt content for a paving mix is 5-10% by weight, so large quantities of asphalt will be required for even a modest system of roads [5].

The physical properties of asphalt also make it unsuitable for lunar use. Asphalt has a glass transition temperature of approximately -20°C, below which it becomes brittle and is likely to fracture on impact. It has poor strength at high temperatures due to a decrease in viscosity. This reduces the life of the road, and may cause the asphalt to drain away from the aggregate, resulting in a loss of surface cohesion. Asphalt is usually applied at temperatures below 105°C [6]. Because this is less than the lunar daytime temperature of 110-130°C, the asphalt will be too fluid to support vehicle loads. The high cost of transportation to the Moon and its poor physical properties made asphalt unsuitable for lunar paving.

Paving Tiles

The road surface can also be made of preformed blocks or tiles placed on a prepared road base. Tiles share most of the drawbacks of concrete: high use of energy and materials, difficult production, and the need for extensive road base preparation. It also takes a good deal of time and effort to place the tiles, and they are difficult to stabilize so that the surface remains level. The roads will also require frequent maintenance. For these reasons, this road surface was not suitable.

Fused Soil

Focused solar heat can be used to fuse the lunar soil, also called *regolith*, to produce a hard, durable, dust-free surface [7]. There are difficulties with this process, however. It will take a great deal of energy to melt the regolith to a useful depth for supporting vehicles. The fused surface must have adequate support and will require extensive road base preparation. The fused material will be glassy and brittle and have poor resistance to impact loading from vehicles and meteorites. This road type was also not suitable for the lunar base.

Gravel

A type of road that provides a smooth, level surface without paving is a gravel road. The gravel is spread over a prepared base, leveled, and compacted to produce the finished surface. Gravel roads have several advantages. They require only lunar materials, provide a relatively smooth surface with a fairly simple construction process, can support large vehicle loads, are very easy to repair, and are fairly insensitive to the thermal effects of the lunar day/night cycle. In addition, the base required for a paved surface is essentially a gravel road, so that the surface may be improved later with little additional work.

The disadvantages are that the road will not be as smooth and durable as a paved surface, and large amounts of gravel of appropriate size and physical properties are required. This gravel can be produced by sifting the regolith, by crushing larger rocks, or as a by-product of mining and oxygen extraction processes.

Dirt (Compacted Regolith)

A dirt road, the simplest type of improved surface, requires the least amount of time or energy for construction. The only operations required are leveling and compaction and the only material needed is regolith, so little excavation or material processing will be required. The surface will be fairly smooth and firm, and will be very easy to repair. However, the surface will have relatively low durability and may be easily damaged by heavy vehicles and meteorites.

Road Type Selected

The paved surfaces have severe drawbacks because of high energy requirements for production, the need for materials to be brought from Earth, and physical properties which are not suited for the lunar environment. As a result, the choice of road type was limited to gravel and dirt roads, which use only lunar materials and require less time and energy to construct. They are not as durable as paved surfaces, but their ease of repair overcomes this disadvantage.

One problem with gravel roads is in finding or producing sufficient amounts of gravel of an appropriate size. This is especially significant in the early phases of establishing the lunar base, when roads are desired, but mining and oxygen extraction activities have not yet begun. As a result, any gravel needed for road construction must be extracted from the regolith. The soil found at a typical base site on the lunar plains is relatively fine, with only 1-5% of the grains being larger than 4 millimeters (mm) [8]. This will require the sifting of extremely large amounts of regolith to extract enough gravel for road construction (roughly 4000 metric tons of gravel per kilometer of road).

Because of this limitation, the design team decided that a compacted soil road was the most appropriate alternative for the early stages of lunar base operation. A dirt road will be an improvement over the natural lunar surface; it will be smooth and firm, reducing vehicle energy requirements and problems from lunar dust. It is not as durable as gravel, but it is much more practical for the early stages of base operation. The dirt roads will also provide a base for later gravel or paved roads. Once mining and oxygen extraction begins, waste gravel and sintered soil from these operations can be spread over the dirt roads, providing a better surface.

CONSTRUCTION PROCESSES AND MACHINERY SELECTION

There are three basic steps needed to construct a dirt road: removal of the surface dust layer, grading and leveling, and compaction. The construction machinery must be able to grade and level the lunar surface, perform minor excavation and fill work, smooth and spread the soil, and compact both loose and undisturbed soil to a desired density.

There are three machines used for road building on Earth that will be suitable for these operations: a bulldozer, a grader, and a compactor. The design team examined the functions of each of these machines and determined that only the grader and compactor were needed to construct a basic dirt road on the Moon.

The lunar base will most likely be located on a relatively flat plain of the Moon [9]. Although the surveying for the lunar base roads was beyond the scope of this report, the design team assumed that the roadways were designed to avoid obstacles such as large craters and rock outcrops. For these reasons, the excavation and movement of large amounts of regolith was not required, so the power of a bulldozer was not needed. A grader can perform light-duty soil work as well as level and finish the road surface, and is thus more versatile than a bulldozer. Most fill work will only require regolith to be moved short distances, usually from areas at the side of the roadway, so there was no need for trucks or other specialized soil transport equipment.

The purpose of a compactor is to increase the density of soil so that it can support a greater load than an uncompacted surface. Compaction can be accomplished with both loose soil that has been spread during fill work and with undisturbed soil. The compaction is done by rearranging the soil grains and forcing them into a more tightly packed arrangement.

There are four main methods of providing the force for compaction: static weight, vibration, kneading, and impact. A concentrated static weight can overcome the frictional forces between the grains and cause them to slide past each other. Dynamic methods such as vibration, kneading and impact can also cause movement between grains, increasing the effective weight of the compactor. The most commonly used methods to produce vibration are with a mass mounted eccentrically on a rotating shaft or with a piezoelectric transducer, a device which converts electric current into vibration.

Because the lunar soil density increases exponentially with depth [10], the grader can be used to remove the soil until a certain percentage of the maximum density is reached. The compactor can then be used to further increase the density to a level necessary to support vehicle loads and to provide a smooth, firm surface. Measurements of soil density must be made during the road surveying and layout process, and an appropriate depth of excavation determined to minimize overall energy use.

MACHINERY CONFIGURATIONS

The design team followed two approaches to develop the road construction machinery. The first was to adapt terrestrial machinery for use in the lunar environment, and the other was to develop entirely new machinery based on the functional requirements and the environmental constraints.

Some features were common to all configurations, and were thus not considered in the development of alternates. Electrical power provided by batteries, fuel cells, or radioisotope generators will be used by all machines. The motors will be mounted as close as possible to the components being driven in order to simplify power transmission and minimize mechanical losses, and control systems for the machinery will allow both remote and local operation.

Graders

Grader Alternate #1

A typical grader has two main characteristics: (1) a long wheelbase to ensure little variation in cutting depth and to smooth out surface irregularities, and (2) a blade to remove layers of soil. The design team used these characteristics to develop the first alternate for a lunar grader. (See Figure 2.) This alternate has a retractable frame to give the grader a shorter turning radius when it is used as a drive unit for other machinery and to decrease its size during transport to the Moon. Each wheel of the grader is powered by an independent motor, eliminating the problem of transferring power to the wheels through the telescoping frame. A scarifier may be mounted in front of the blade to loosen the regolith prior to grading and to remove rocks which could damage the grader blade. The blade and scarifier can be securely stowed or completely removed when the grader is used as the drive unit for other machines. (See Figure 3.)

Although this design very versatile, it requires a complicated telescoping mechanism for the retractable frame. This mechanism has problems with thermal expansion due to the extreme temperature variations on the Moon. The design also produces stress concentrations at the points of contact between the sections of the frame. In addition, there will be high loads on the frame because most of the weight is concentrated in the center of the vehicle.

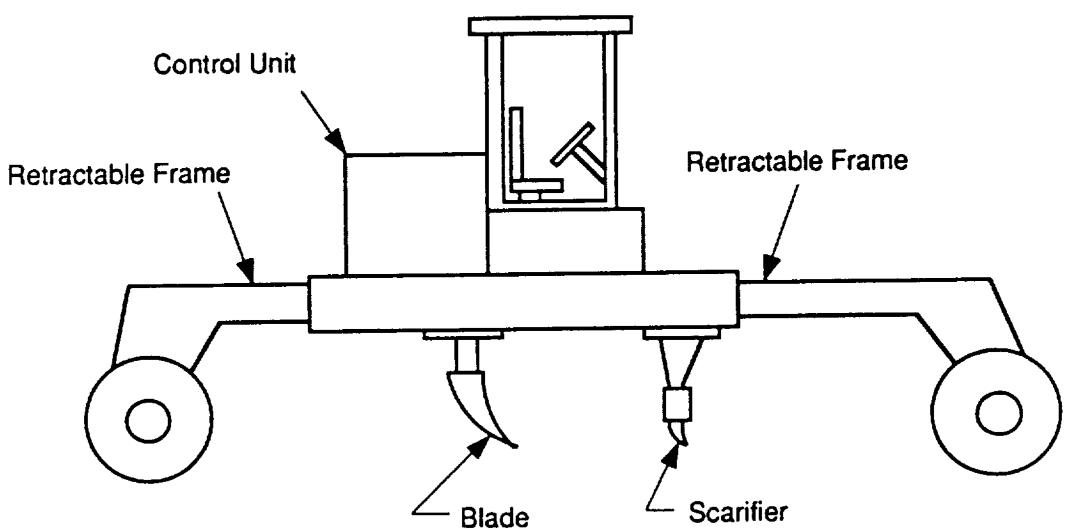


Figure 2: GRADER WITH RETRACTABLE FRAME AND STOWABLE BLADE AND SCARIFIER

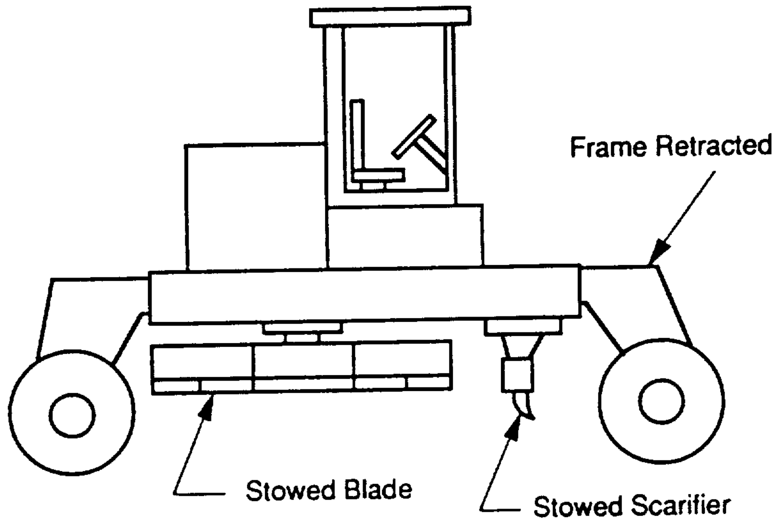


Figure 3: MAIN DRIVE UNIT CONFIGURATION WITH FRAME RETRACTED AND BLADE AND SCARIFIER STOWED

Grader Alternate #2

Another alternate was a modification of a grader used on Earth for road leveling. (See Figure 4.) This design also incorporates independent motors for each wheel and a telescoping frame to reduce the size of the machine when it is transported to the Moon. The grader can have a scarifier mounted in front of the main blade to loosen the rocks and the lunar soil. A hopper filled with regolith mounted above the front wheels provides additional weight to increase the wheels' traction in the low-gravity environment of the Moon. However, the addition of the hopper requires a device to load regolith.

This design also has difficulties with thermal expansion of the telescoping frame and will not function well as a drive unit for other machines due to the limited maneuverability caused by the large turning radius.

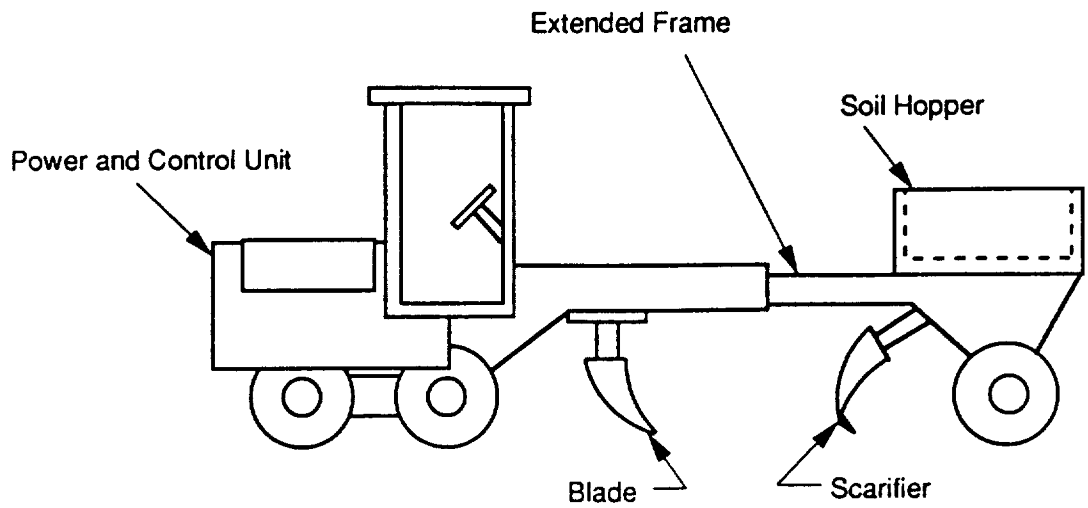


Figure 4: GRADER WITH EXTENDED FRAME, BLADE, SCARIFIER, AND SOIL HOPPER

Grader Alternate #3

The terrestrial grader can also be modified to have a removable center section. (See Figure 5.) During transport to the Moon, the center frame disconnects from the main body and the front wheel assembly of the grader in order to save space. The front wheel assembly attaches directly to the main body to use the grader as a drive unit for other machinery. This design has no telescoping parts, but will still be subject to thermal stresses at the joints between the main body, center section, and front wheel assembly. This alternate requires assembly on the Moon before use and takes more time and effort to change configurations than the telescoping designs.

The joints between the sections can be fixed or articulated. (See Figure 6.) An articulated frame allows greater flexibility in blade positioning, but requires a complex mechanism to keep the frame positioned properly. The loads will be concentrated at the articulated joints, requiring high-strength bearing assemblies at the joints.

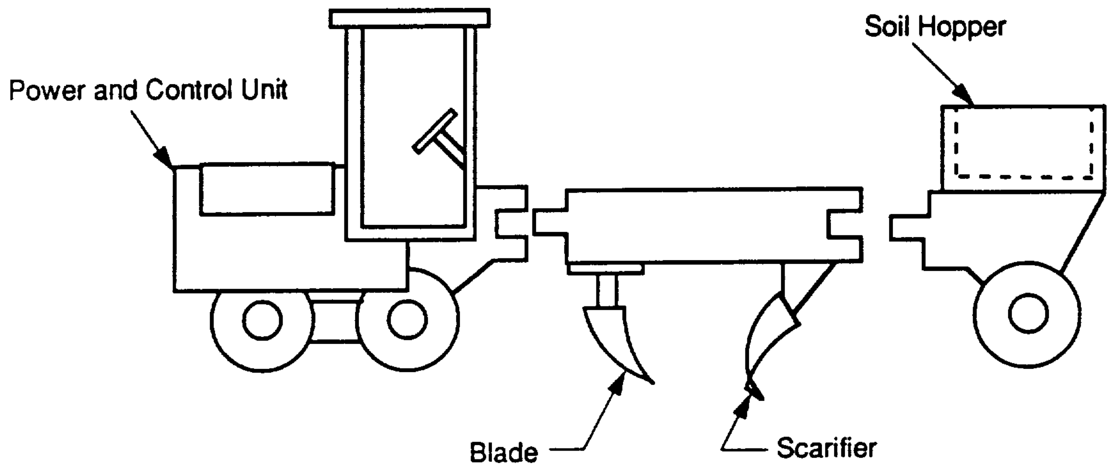


Figure 5: MODIFIED TERRESTRIAL GRADER WITH REMOVABLE CENTER SECTION

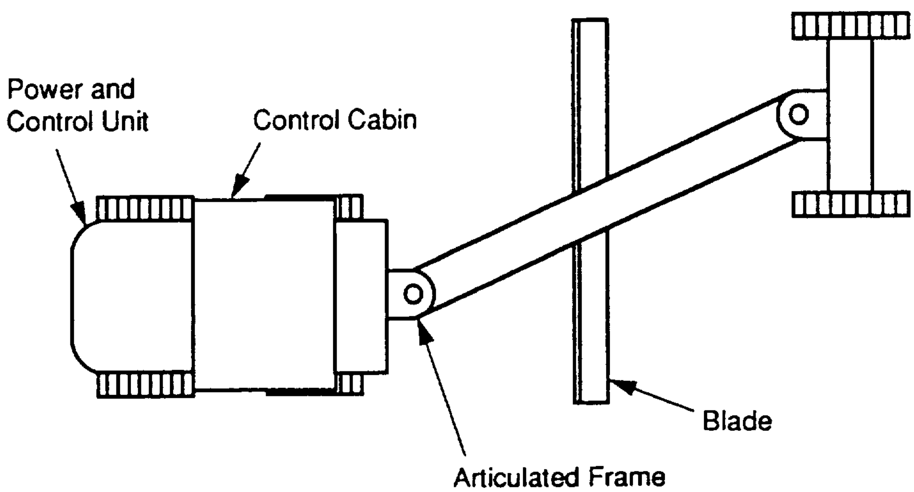


Figure 6: MODIFIED TERRESTRIAL GRADER WITH REMOVABLE, ARTICULATED CENTER SECTION

Compactors

Compactor Alternate #1

One of the simplest forms of a compactor is a roller. (See Figure 7.) The roller is hollow to minimize mass for transport and can be filled with regolith to increase its mass for use. The roller uses a combination of the static weight of the roller and a dynamic force produced by a vibrating mass to compact the soil. The roller is towed behind a drive unit, and several rollers can be joined to form a train for additional compacting. The roller is small compared to the other types of compactors and can be easily used for minor road repairs.

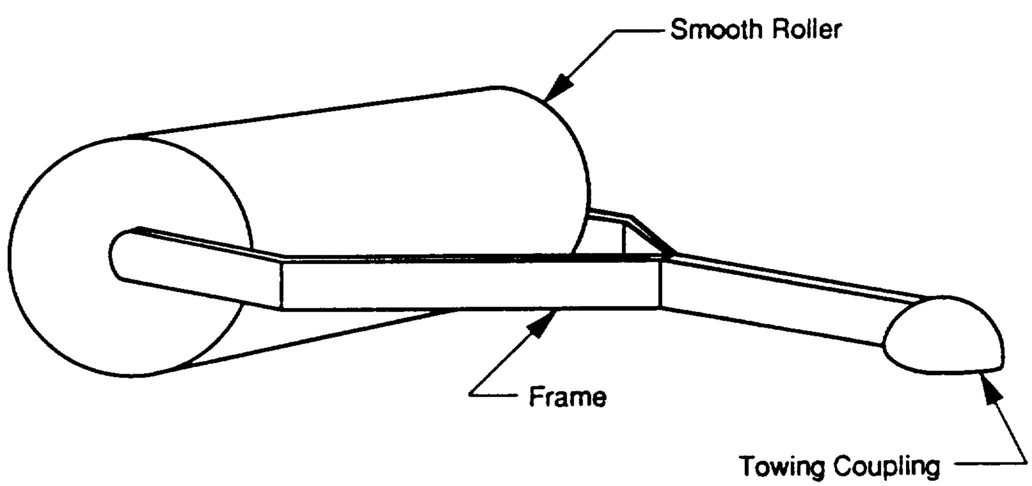


Figure 7: ROLLER COMPACTOR

Compactor Alternate #2

Compaction can also be achieved by the static weight of a vehicle alone. An alternate using this concept was the static compactor, which uses a large hopper for regolith to provide the compacting force for the wheels. (See Figure 8.) This compactor is a simple mechanical design, has low mass for transport, and can be used as a soil or cargo carrier when it is not used for compaction. The static compactor requires a large amount of regolith to produce the necessary compacting forces and requires additional machinery to load the regolith into the hopper.

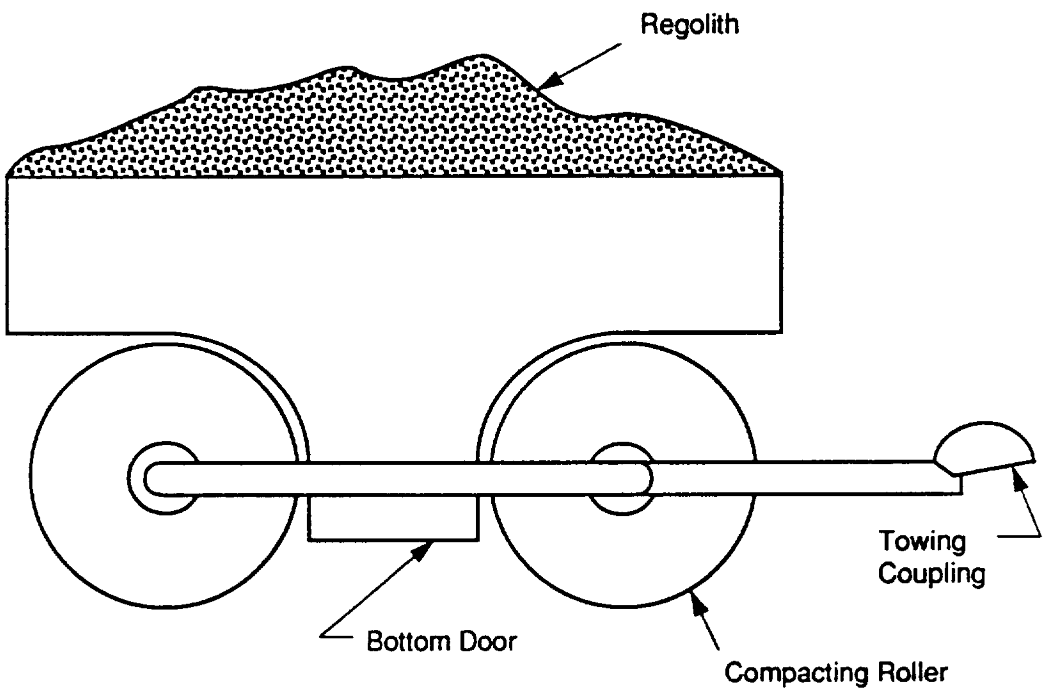


Figure 8: STATIC COMPACTOR

Compactor Alternate #3

A small, lightweight "thumper" compactor uses impact forces to dynamically compact the lunar soil. (See Figure 9.) The pads of the compactor pound the soil at a high rate, producing movement between the soil grains and packing them to the desired density. A high-power piezoelectric transducer can be used to supply the vibratory force of the thumper.

The impact force of the thumper compactor is limited by its total weight. In order to obtain sufficient pressure for compaction, the contact pad must be relatively small, requiring many pads to obtain a sufficient rate of operation. This results in a complicated mechanism with many moving parts. There will also be difficulties in producing even compaction and a smooth, level surface.

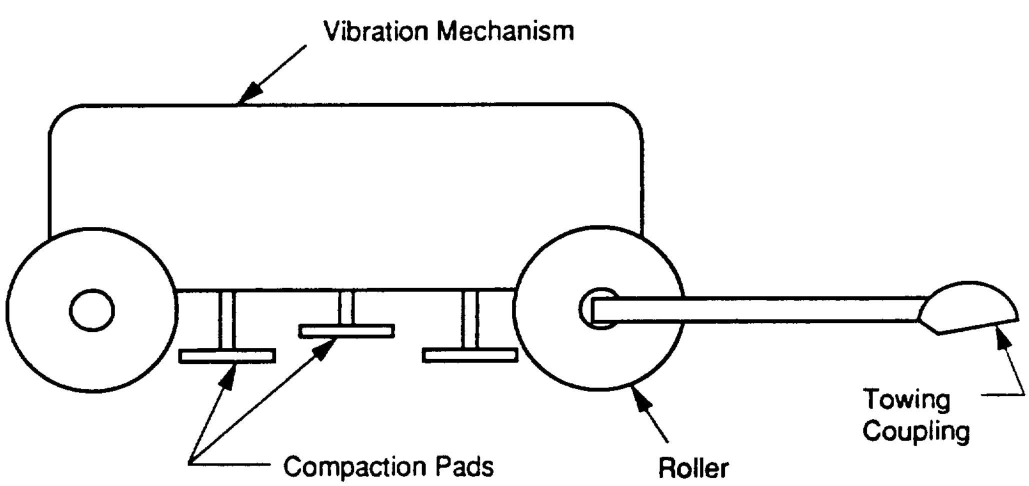


Figure 9: "THUMPER"-TYPE IMPACT COMPACTOR

Compactor Alternate #4

The compactor can also be a self-driven, independent machine. (See Figure 10.) Because it can operate independently, compaction can be accomplished at the same time as grading, with each machine able to operate at an optimum rate. This parallel operation speeds the road construction process and simplifies roadway maintenance and repair procedures. Fitted with a lightweight blade, the compactor can move regolith to fill in meteorite craters and wheel ruts, and then compact the regolith to restore the road surface.

Because this machine has its own power and control system, it is more massive than the towed designs and requires more energy for transport to the Moon. Simultaneous operation of both machines requires more complex remote control systems and more attention from the operators.

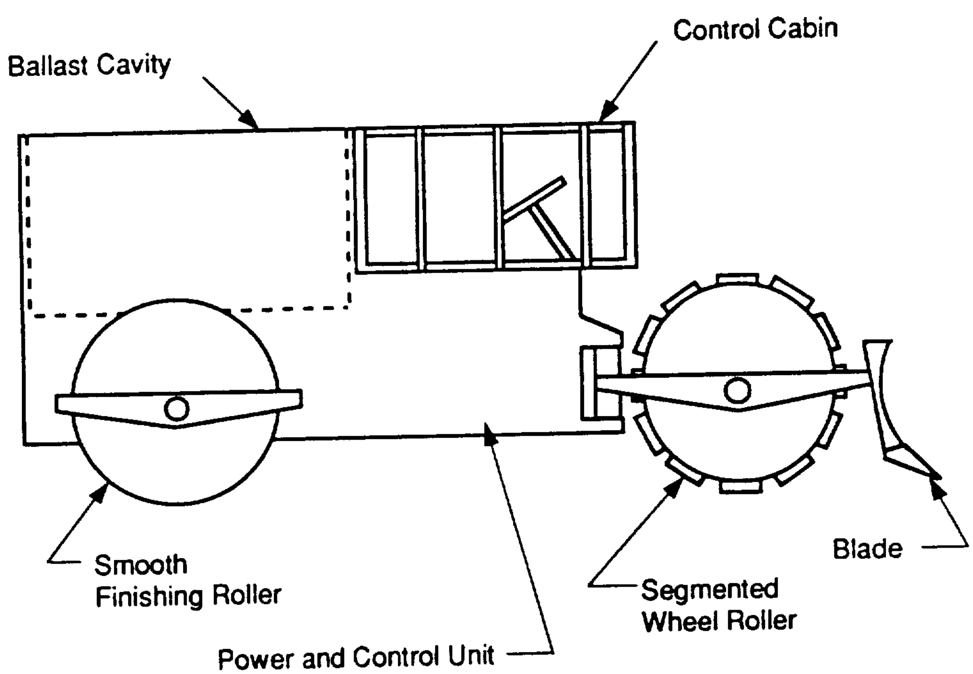


Figure 10: SELF-DRIVEN COMPACTOR

Component Selection for the Grader and Compactor

There were several alternates for the components of the grader and compactor. The design team had to select a feasible blade and wheel design for the grader and an effective roller for the compactor.

Blade Types

There were three alternates for the grader blade: a standard blade used on terrestrial machines, an articulated blade, and a combination of the standard and articulated types. Another option was to use an auger to excavate the regolith instead of a blade. In addition, the design team considered the use of a scarifier in conjunction with the blade to loosen soil and remove rocks.

Standard Blade

The standard grader blade is a curved plate mounted below the center frame of the grader. (See Figure 11.) This blade is rigid and strong due to the curvature, is mechanically simple, and has good soil cutting characteristics. The curvature of the blade creates a rolling action in the loosened soil that reduces the force required to move it [11]. During normal grading operations, the loose soil fills in low spots, with the excess soil spilling off the end of the blade. This spillage limits the amount of soil carried if the grader is used for bulldozing operations.

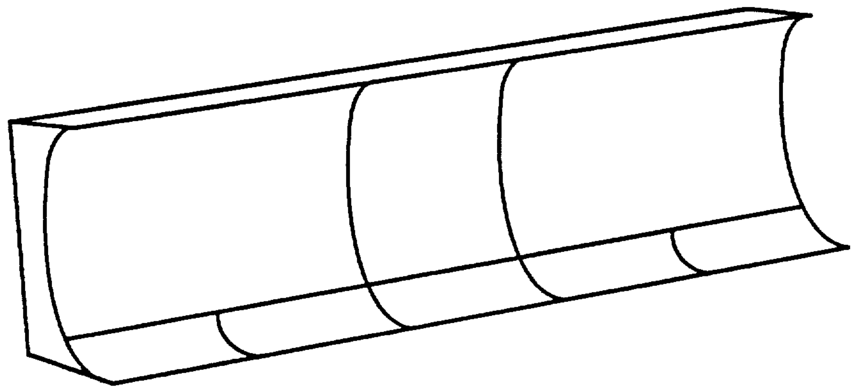
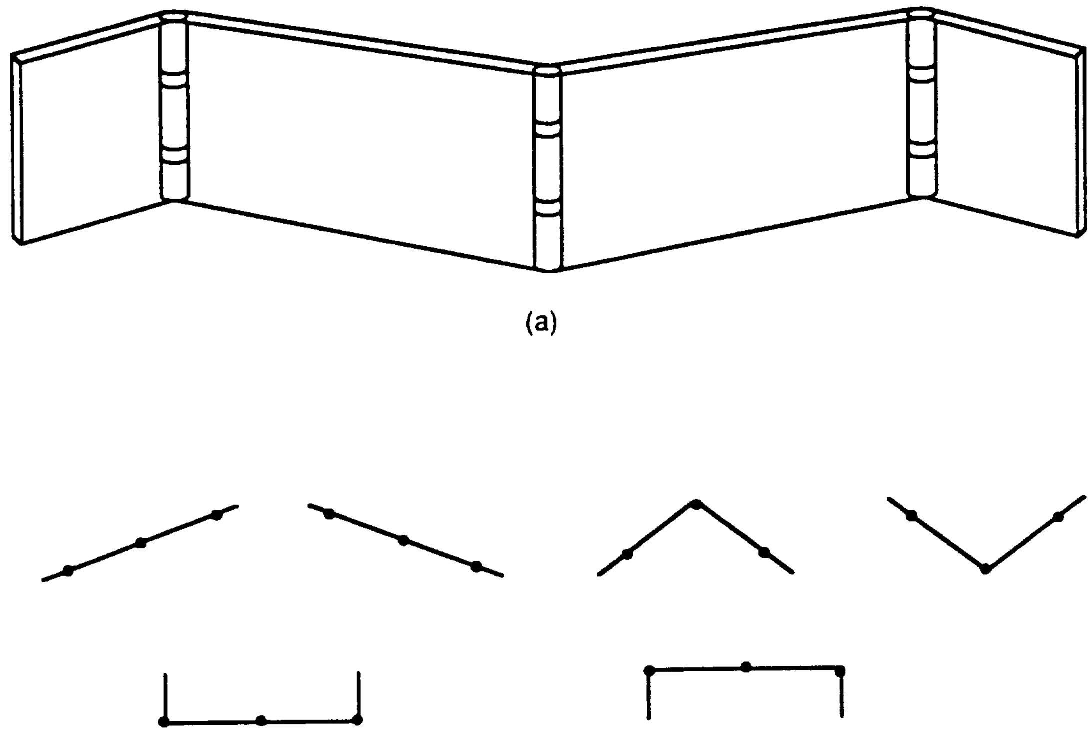


Figure 11: STANDARD GRADER BLADE

Articulated Blade

An articulated blade can be used in many configurations. (See Figure 12.) One problem with a standard blade is that when the blade is at an angle to the direction of travel, a side force is produced on the grader. An articulated blade set in a wedge-shaped configuration allows for symmetrical grading, eliminating this side force. Also, a U-shaped configuration allows the blade to push a larger volume of soil than the standard blade if the grader is used as a bulldozer. However, the articulated blade must have a flat profile to allow the sections to pivot forwards and backwards and the entire blade must be tilted to obtain the best cutting action. The mechanism for positioning the blade will be complicated and the pivot mechanisms will be subject to abrasion and can be jammed by the lunar dust.



(a)

(b)

Figure 12: ARTICULATED BLADE (a) OVERALL CONFIGURATION
(b) POSSIBLE POSITIONS FOR BLADE

Combination Blade

A standard blade can be modified with flaps on each end to aid in collecting the soil for bulldozer operations. (See Figure 13.) The flaps are strong and easily positioned, but are more mechanically complex than a standard blade. The pivots for the end plates require lubrication and dust seals.

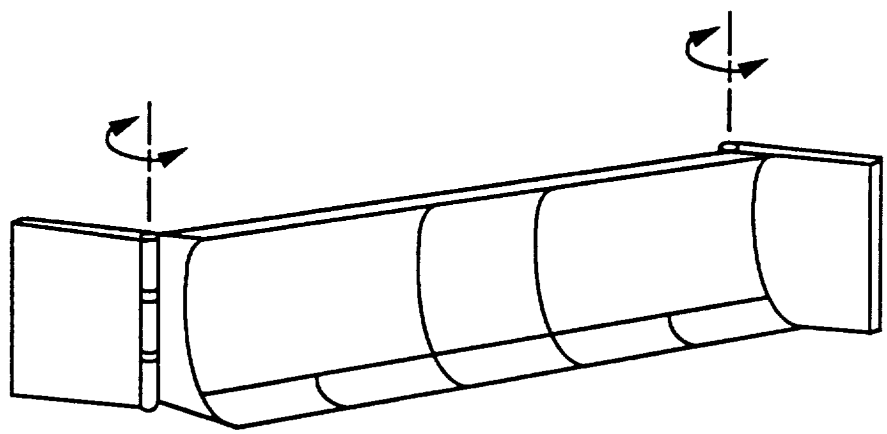


Figure 13: COMBINATION GRADER BLADE

<u>Auger</u>

An auger is a power screw mechanism that cuts into the soil and moves it to the side. (See Figure 14.) Using an auger instead of a blade reduces the magnitude of tractive forces required from the drive unit because the cutting is not done by pushing the blade through the soil. The auger can have two screw sections, each separately controlled for direction and speed, allowing the soil to be moved to either side, to both sides, or to the middle. Symmetrical cutting eliminates the side thrust on the grader [12]. However, the auger is more mechanically complicated than the standard blade, has a limited depth of cut, and is subject to high abrasive wear from the soil. If two screw sections are used, the drive and control systems will be more complicated.

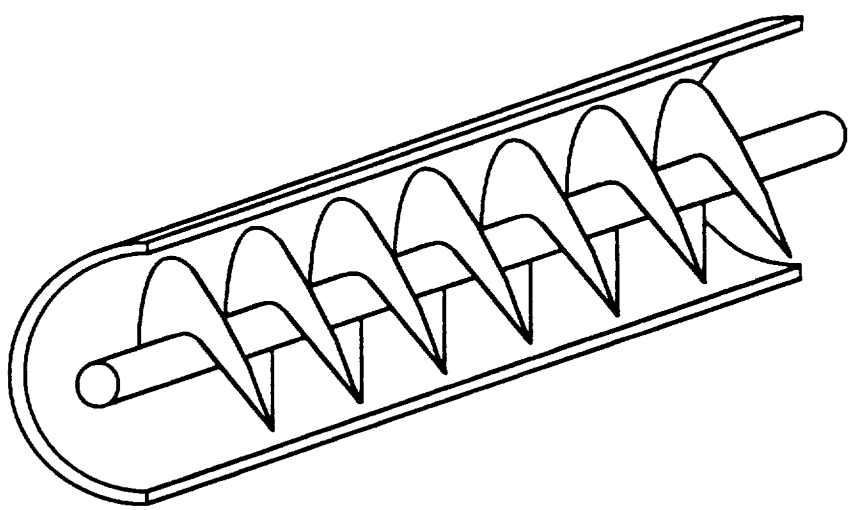


Figure 14: AUGER

Scarifier

A scarifier can be mounted in front of the blade to break up packed regolith and rake out large rocks. (See Figure 15.) This reduces the cutting force on the blade and protects the blade from rock damage [13]. The addition of the scarifier increases the overall weight of the grader and requires support and positioning mechanisms.

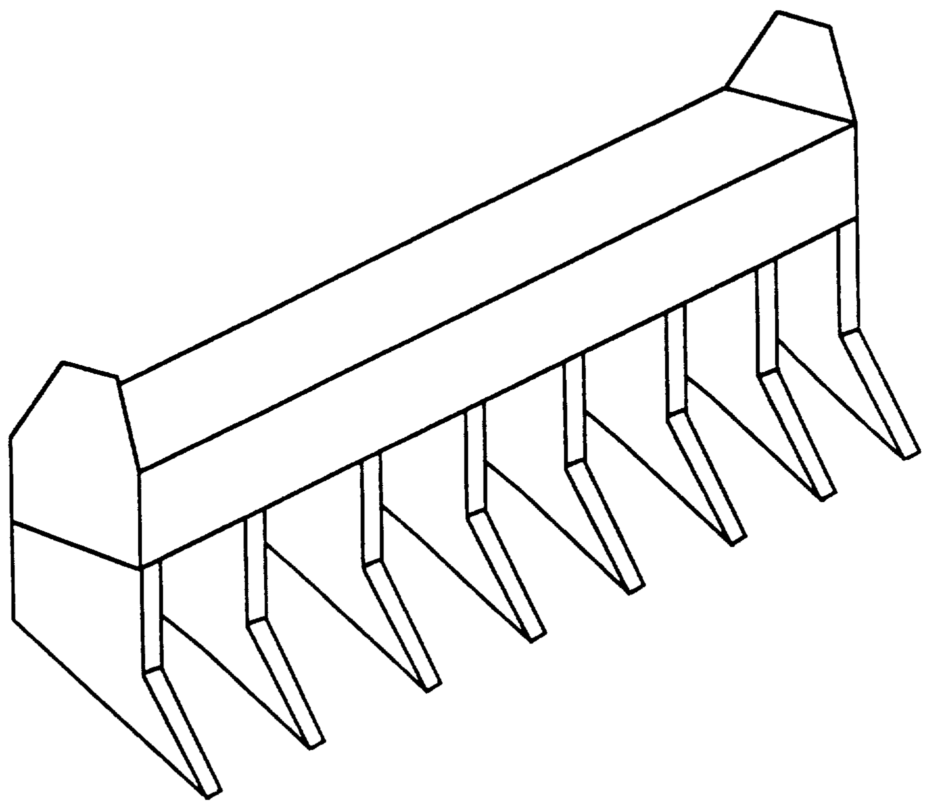


Figure 15: SCARIFIER

Drive Mechanisms

The standard terrestrial grader's drive mechanism will need to be modified to accommodate the low-gravity conditions of the Moon. The two basic options for the drive components were tracks and wheels.

Tracks

Tracks have a large bearing surface in contact with the ground, which produces more traction than wheels [14]. (See Figure 16.) Because the tracks distribute the weight over a larger area, the vehicle is less likely to sink into soft soil. However, tracks are mechanically complex, subject to fatigue, and require lubrication and seals at joints between sections. Tracks also require a complicated suspension system and tend to "dig up" road surfaces, especially when turning.

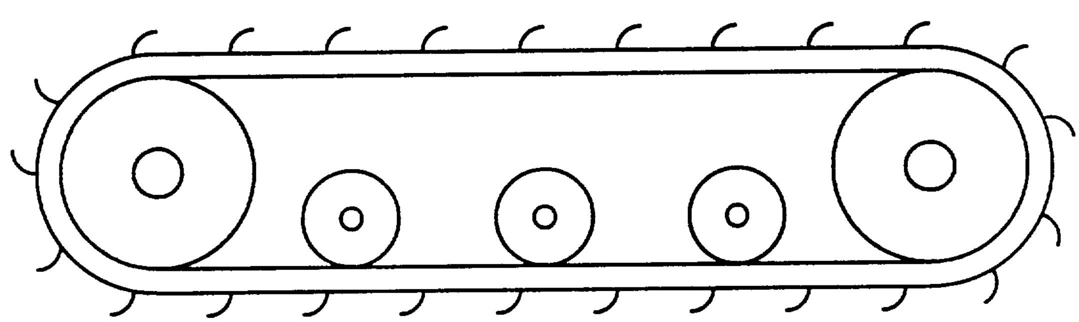


Figure 16: TRACKED DRIVE MECHANISM

Wheels

Wheels are a very simple type of drive mechanism which are easy to steer and can pivot to track along a desired path. Since wheels have a smaller bearing surface than tracks, they generate less traction before slippage occurs, and unless all the wheels on the vehicle are driven, the effective weight of the vehicle which produces traction is reduced. Wheels also tend to penetrate the surface more, especially in soft soil [15].

Wire Mesh Wheels. Wire mesh wheels were used on the original lunar rover due to their flotation characteristics, light weight, and resilient suspension. (See Figure 17.) On heavy machinery, these wheels produce very low traction unless they are fitted with separate treads [16]. Wire mesh wheels have a limited weight bearing capacity and raise a large amount of dust.

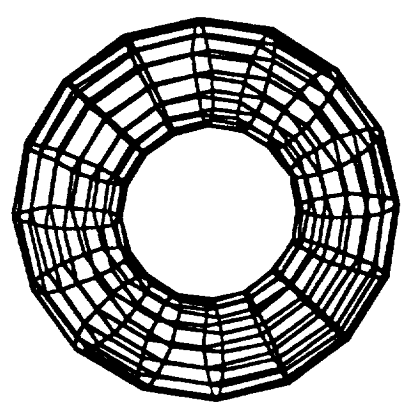


Figure 17: WIRE MESH WHEEL

<u>Metal Elastic Wheels</u>. Metal elastic wheels consist of a concentric hub and tread joined with cylindrical springs. (See Figure 18.) These wheels are lightweight, provide passive suspension, and have a moderate weight capacity [17]. However, these wheels are inefficient in torque transmission due to the deformation of the spring elements (a "wind-up action") and undergo substantial deformation when heavily loaded.

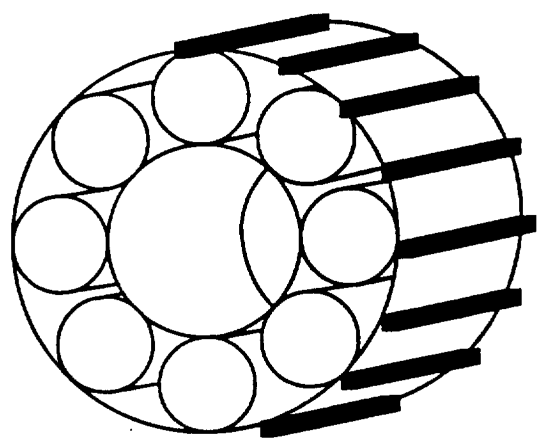


Figure 18: METAL ELASTIC WHEEL

<u>Cone Wheels</u>. Cone wheels can support heavy loads and have a simply constructed, durable tread and rim [18]. (See Figure 19.) These wheels are relatively heavy and have a high stress concentration at the points where the rim joins the body.

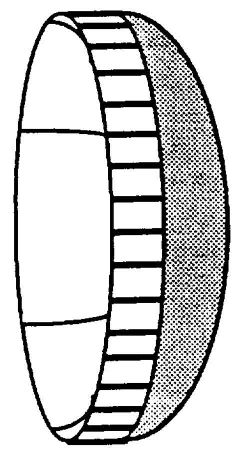


Figure 19: CONE SHAPED WHEEL

Hemispherical Dome Wheels. Hemispherical dome wheels are similar to the cone wheels except that the body is curved to eliminate the angle where the rim joins the body, thus eliminating the high stress concentration at the joint [19]. (See Figure 20.)

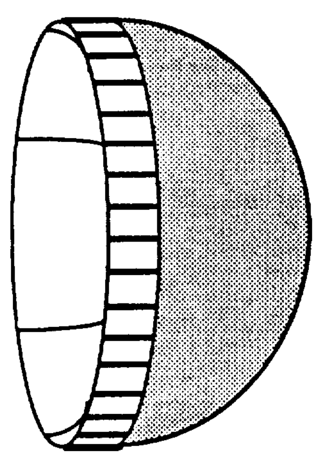


Figure 20: HEMISPHERICAL DOME WHEEL

Compactor Roller Types

Smooth Rollers

Smooth rollers produce a good surface finish, are lightweight, and are of simple construction. (See Figure 21.) Smooth rollers have low traction if driven and require ballast or a vibratory mechanism to produce good compaction [20].

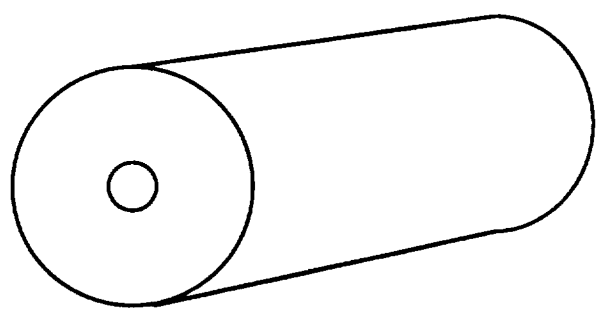


Figure 21: SMOOTH ROLLER

Tamping-Foot Roller

A tamping-foot roller is a drum with projecting feet. (See Figure 22.) This configuration provides several mechanisms for soil compaction. The reduced area of the individual feet concentrates the weight of the roller, increasing the static pressure for compaction. The feet penetrate the loose soil, creating a kneading action which eliminates voids and compacts the soil from the bottom up. In addition, as the roller turns, the point of maximum pressure moves from foot to foot. This produces a vibrating action which aids soil settlement [21]. Although the tamping-foot roller produces effective compaction, it is difficult to achieve uniform density and a smooth surface. It is usually necessary to finish the compaction with a smooth roller [22].

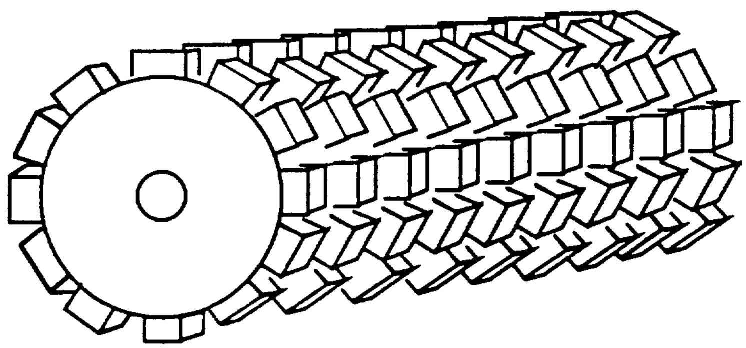


Figure 22: TAMPING-FOOT ROLLER

Design Solution

The road construction system the design team developed consists of a main drive unit for propulsion and control, a detachable grader assembly, and a towed compactor. These three components are the only machinery required to produce a compacted lunar soil road. The main drive unit (MDU) was designed as a multiple-purpose machine and can be used for mining operations and cargo transportation in addition to road construction. The grader assembly connects to the MDU and is used to excavate and contour the lunar regolith to produce a level road. The compactor does the finishing work on the road to produce a smooth, firm surface for vehicle traffic.

This section presents the final design developed for the lunar road construction equipment. The process of road construction is described to explain the use of the machinery. A discussion of the decision process the design team used to choose among the alternate designs is given, followed by considerations for materials selection and power sources. The final design is then presented, beginning with an overall description of the system and followed by a detailed discussion of each machine, including final configuration, operation and features, power requirements, and mass. Finally, the team gives its evaluation of the final design and recommendations for further work.

Evaluation of Alternate Designs

Machine Configuration

The alternate designs developed for the grader and compactor were evaluated according to the design criteria specified by NASA/USRA and the design team. A decision matrix was set up for this purpose based on the method of pairs [23]. Each criterion was compared to the others one at a time, with the more important criterion given a tally mark. The number of tally marks for each criterion was divided by the total number of tally marks to yield a weighting factor. The most important criterion was received the highest weighting factor, with the sum of all the weighting factors being unity. Each alternate design was rated on how well it satisfied each criterion and given a score between 1 and 10. The scores were multiplied by the weighting factors and then added to give an overall score for each alternate.

The decision matrices are given in Appendix A. The alternate selected for the grader was the terrestrial grader with the removable center section, and the best alternate for the compactor was the static compactor. Both alternates have been further developed and modified for the final design, which will be discussed in detail later in this section.

Machine Components

Decision matrices were not used for the individual components such as wheels, blades, and rollers. These components were judged according to their performance characteristics and suitability for the lunar environment.

Drive Mechanism

The drive mechanism selected for the MDU and the grader is the hemispherical dome wheel. These wheels have a greater weight capacity, are more durable, and are simpler to construct than the wire mesh and metal elastic wheels. The treads on the dome wheels provide good traction, and the smooth curve where the tread joins the hemispherical shell eliminates the stress concentration found in cone wheels.

Tracks are poorly suited to the lunar environment. They require a mechanically complicated suspension and drive system. Segmented tracks require lubricated joints with complicated seals to prevent abrasion from lunar dust, while elastic treads are subject to fatigue and damage from rock penetration and radiation. Although tracks have a greater contact area with the ground, this does not produce a significant increase in traction because the lunar soil is loose and noncohesive.

Grader Blade

The blade type used for the grader is the standard blade used on terrestrial machines. It is mechanically simple, strong, and lightweight. The articulated and combination blades and the auger are mechanically complicated and have problems with lubrication and lunar dust. The main drive unit is capable of producing sufficient traction to push the blade through the regolith, so the auger is not required. The steering mechanism of the grader can compensate for the side force on the blade, eliminating the need for the symmetrical cutting of the articulated blade and auger. Because the lunar soil is relatively noncohesive and fine grained, the scarifier is not needed to loosen the soil and remove rocks.

Rollers

Of the two rollers introduced in the alternate design section, the design team selected the smooth roller as the best suited to the lunar soil. Although the tamping foot roller provides compacting action to a greater depth than a smooth roller, it is most effective for compacting cohesive soils [24]. Also, the tamping foot roller produces a rough surface and must be followed by a smooth roller to finish the roadway. This roller also requires more power to overcome rolling resistance. For these reasons, the design team decided that smooth rollers are better suited to lunar road construction.

Materials Selection

The materials selected for the machinery of the lunar road construction system must be capable of withstanding the harsh lunar environment, including exposure to high radiation levels, micrometeorite bombardment, abrasive dust, extreme variations in temperature, and a vacuum. A detailed discussion of the factors involved in material selection is given in Appendix B. Based on information from several sources, including a conversation with Mr. Brian Muirhead of NASA's Jet Propulsion Laboratory [25], the design team selected an aluminum alloy and a titanium alloy as the two main structural materials for the machinery.

The aluminum alloy, Al-Li 2090, has a low density, a high strength-to-weight ratio, and is relatively inexpensive to produce and machine. The titanium alloy, Ti-6 Al-4 V, has a very high strength-to-weight ratio and is more resistant to residual stresses and abrasive wear than aluminum. Although it is considerably more expensive than aluminum, the physical properties of titanium make it more suitable for critical load-bearing members and interfacing parts. The properties of these materials are shown in Table 1 below and in Appendix B.

Table 1: PROPERTIES OF ALLOYS USED BY NASA

ALLOY	TENSILE STRENGTH (psi)	YIELD STRENGTH (psi)	DENSITY (g/cm^3)
Al 7075	70,000	57,000	2.80
Al 2024	70,000	64,000	2.80
Al-Li 2090	78,000	72,000	2.56
Ti-6 Al-4 V	160,000	145,000	4.42

Power Sources

The road construction machinery will use electrical power. Although the source of the electricity was not specifically addressed in the design, there are several established technologies which can be used. These include batteries, fuel cells, radioisotope generators, and solar panels. The power source can be contained within the main drive unit or placed on an independent vehicle that follows the machinery it supplies. The power source must have relatively low mass and provide enough power to operate the machinery.

Batteries tend to be relatively massive and have limited power storage capacity. Solar panels require a large surface area to produce sufficient power, and are not practical for mobile machinery. Radioisotope generators present safety and mass problems due to the need for radiation shielding.

One alternative for the power supply is a mobile fuel-cell cart proposed by Eagle Engineering [26]. This cart carries tanks of oxygen and hydrogen which are combined in a fuel cell to produce electricity and water. The cart can be self-propelled or towed behind the construction machinery, with electrical power distributed by a connecting cable. The system can be recharged by separating the water produced into its component gases by electrolysis, using electricity from the main base power supply. The disadvantage of having a mobile fuel cart is that the cart may interfere with the operation and maneuverability of the construction equipment. Also, a complicated control system will be required to synchronize the movements and speed of the fuel cart and the construction machinery.

Another alternative is to have this same type of power system mounted on the main drive unit. The advantage of this alternative is that there will be no need to have a complicated control system and the construction process will not be interfered with by the fuel cart. However, the addition of the power supply will increase the size and mass of the drive unit.

Since the power source was beyond the scope of this project, the design team did not consider the location of the supply in the final design solution.

ROAD CONSTRUCTION PROCESSES

Before actual construction can begin, the lunar surface must be surveyed and the proposed roadways laid out. The land surveying and the road planning was beyond the scope of this project, so the design team assumed that these tasks were already completed and that navigation markers and remote control systems were in place.

The first task of road construction is to scrape off the dust layer and grade the lunar soil to the desired depth. The grader assembly is attached to the main drive unit and the grader blade is placed in the desired position. The grader proceeds along the path laid out, excavating and filling as necessary to produce a level road. This process is primarily designed for remote control from the lunar base or from Earth, but in tight maneuvering situations, or if the remote control system fails, the grader can be manually controlled by an astronaut in an EVA suit.

The second task is to compact the graded surface to achieve an optimum soil density and give the road a smooth finish. This is accomplished by loading the compactor hopper with regolith, removing the grader assembly from the MDU, and attaching the compactor. The MDU/compactor assembly is then ready for operation. This process is also designed for remote control, but can be accomplished by manual operation.

Upon completion of compaction, the dirt road will be ready for vehicle traffic. If the road is damaged by micrometeorite impact or normal traffic wear, the grader and compactor can be used to perform repairs using the same techniques used to construct the road.

ROAD CONSTRUCTION MACHINERY

The final design of the road construction machinery has three main components: a main drive unit, a grader attachment, and a compactor. (See Figure 23.) The main drive unit provides propulsion for the grading and compacting processes, and can be used for other tasks when it is not used for road construction. The grader attachment mounts to the front of the drive unit and consists of a support and positioning mechanism for the grader blade and a steerable front wheel assembly. The compactor is towed by the drive unit and has two smooth, wide rollers which compact the road surface. The force needed for compaction is provided by regolith ballast carried in a hopper and by a vibratory mechanism. With the rollers on the compactor replaced with wheels, the hopper can be used to carry soil or other materials for mining operations.

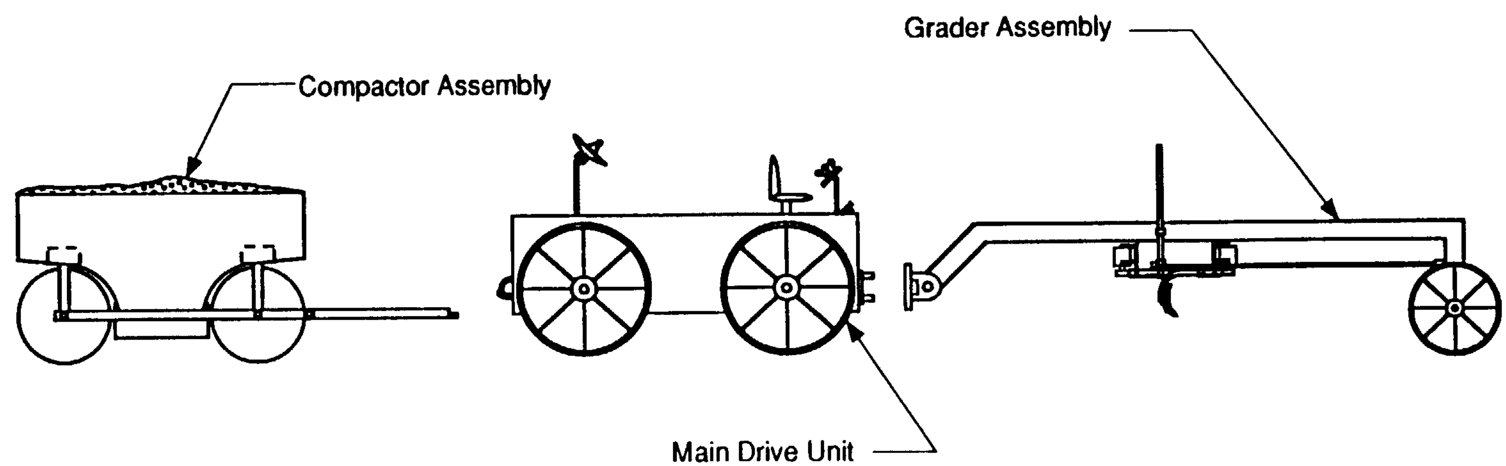


Figure 23: ROAD CONSTRUCTION MACHINERY

Main Drive Unit

The main drive unit (MDU) is a tractor-like machine which provides propulsion and control for the grader and compactor. It has been designed as a multiple-purpose machine and can be used for mining operations and transport of cargo in addition to road construction. This section presents the configuration, mass, power requirements, and operating features of the MDU.

Configuration

The main drive unit was based on the lunar truck proposed by the Advanced Programs Office of the Johnson Space Center [27]. The drive unit is a compact vehicle with four independently driven wheels mounted on a rigid frame. (See Figure 24.) Directional control is provided by differential wheel speed, and the relatively short and wide frame allows for a short turning radius. The MDU has a central hopper for regolith ballast, which increases the working mass of the drive unit to provide greater traction. A bench seat and control panel is provided for direct operation of the MDU by the base crew. The grader section connects to a mounting plate at the front of the MDU, and a hitch at the rear is used for towing the compactor or other equipment. (See Figure 25.)

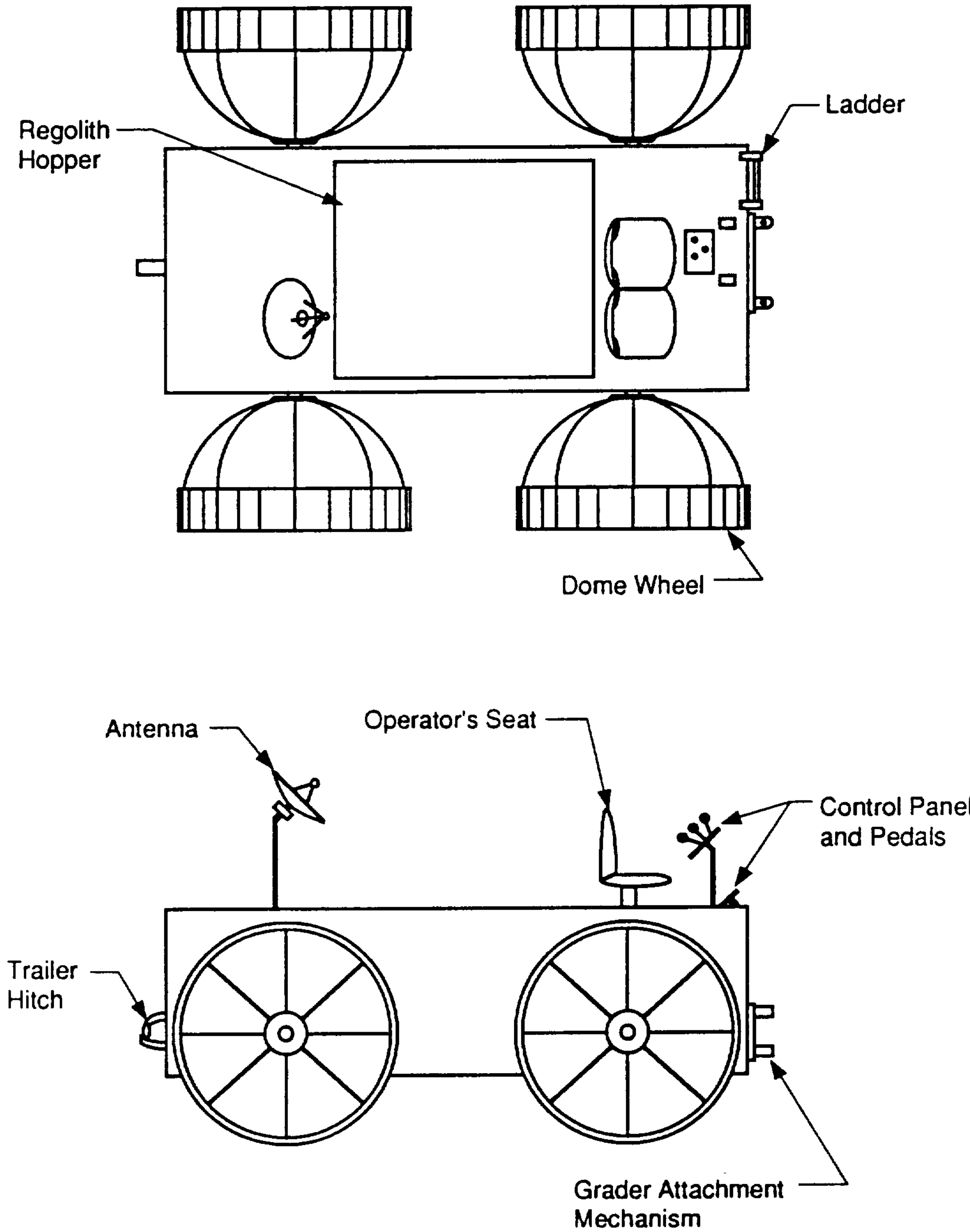


Figure 24: MAIN DRIVE UNIT

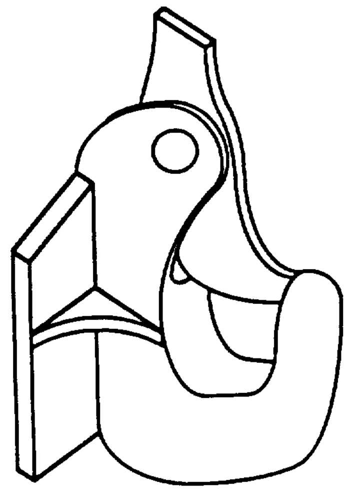


Figure 25: MAIN DRIVE UNIT TOWING HITCH

The wheels of the main drive unit are hemispherical dome wheels, which are well suited to supporting heavy loads at the relatively low speeds (2-5 kilometers per hour [km/hr]) of the MDU. Flexing of the hemispherical shell under load provides some passive suspension, but due to the low operating speed, dynamic loads on the vehicle will not be large and a complicated suspension system will not be necessary. To provide the required traction, the hemispherical dome wheels must be 1.6 meters (m) in diameter, with the shell made of 5 mm thick aluminum plate. The treads of the wheels are titanium, 30 centimeters (cm) wide and 2 cm thick. The calculations used to determine the traction and derive these dimensions are shown in Appendix C.

Mass

Because time did not permit a detailed analysis of the mass of the MDU, the design team assumed that the mass will be 1400-1600 kilograms (kg), similar to that of the lunar truck. Regolith ballast is added to increase the working mass of the main drive unit to provide adequate traction for grading and compaction. The total working mass will be 5000-7000 kg for typical road construction operations.

Power Requirements

The traction force requirements of the MDU were based on an analysis of the cutting forces on the grader blade when making a 3 cm deep cut. As shown in the description of the grader and in Appendix D, this force was calculated to be 6 kilonewtons (kN). The drive unit must be able to provide this force in addition to the force required to overcome the rolling resistance of the MDU wheels. A complete discussion of traction, rolling resistance, and the related soil mechanics is given in Appendix C. The resulting gross traction force developed by the MDU is 8.5 kN. At an operating speed of 2 km/hr, the power transmitted to the wheels is approximately 4.75 kilowatts (kW). Assuming 50% electrical and mechanical losses in the motors and transmission, and allowing 500 W for control systems and sensors, the input power required by the MDU for grading or compacting operations is 10 kW. The 50% efficiency is a conservative figure and includes an allowance for additional power required when climbing typical lunar slopes of up to 5 degrees.

Operating Features

Automatic and Remote Operation. Although the MDU can be operated directly by a base crew member, it has the capacity for automatic or remote control. Automatic operation frees crew members for other tasks and help to conserve resources such as food, water, and oxygen. Automatic or remote operation also minimizes the amount of time that crew members must work on the lunar surface, exposed to the risks of radiation, vacuum, and accidents.

The design team did not specifically address the details of remote or automatic operation, but several approaches are possible. The machinery may be designed to follow guide markers placed during the survey and layout process. During grading, sensing equipment determines when the roadway has been excavated to the proper soil density and surface contour. Similar sensors control the speed of the machinery during compacting so that the finished road surface has the proper firmness.

Remote operation, from the lunar base or from Earth, is probably more practical than automatic operation. Because road construction is a rather complex operation requiring constant monitoring of progress, an automatic control system needs to be highly sophisticated and have feedback and artificial intelligence capabilities.

Control signals to and from the machinery can be carried by a high-frequency digital radio beam. The operator receives stereoscopic images from cameras on the MDU, as well as information on vehicle forces and power consumption. Because radio signals travel in a straight line in a vacuum, it will be necessary to maintain a line of sight between the control station and the MDU. This can be done with a series of relay antennas, or a satellite can be placed in synchronous orbit above the lunar operations area.

If the machinery is controlled from Earth, the crew at the lunar base will be free to concentrate on other tasks. However, the 3 second time delay involved in signal transmission from the Earth to the Moon may make this approach impractical.

For safety purposes, any automatic or remote control system will be designed to shut the machinery down if the control signal is lost or if the machinery malfunctions. The system will also send an alarm signal to the lunar base indicating the need for attention.

Rapid Travel. The transmission of the drive unit is equipped with a selectable high gear to permit faster travel when grading or compaction is not being performed. This allows more efficient use of the operator's time, which is especially important if a crew member is directly controlling the MDU. In an emergency, the operator can travel to a place of shelter quickly.

Alternate Uses. Because the MDU has been designed as an all-purpose tractor, it can be used for operations other than road construction, such as cargo transport or mining. The trailer hitch at the rear allows the drive unit to pull cargo wagons, mobile fuel tanks, or other equipment needed for lunar base operations. It can similarly be used to pull regolith carriers for mining operations. Excavators or other equipment can be mounted to the front of the MDU in place of the grader assembly. The amount of ballast can be increased to provide greater traction or reduced to lessen losses from rolling resistance. The short wheelbase makes the MDU highly maneuverable, and the wide stance and low center of gravity provide for good stability on slopes and under heavy load.

Grader Assembly

The grader assembly is connected to the main drive unit to excavate and contour the lunar regolith to produce a level road. This section will discuss the grader assembly in detail, including a description of the structure and configuration, mass, power requirements, connection to the main drive unit, and operating features.

Configuration

The grader assembly consists of a grader blade, a blade positioning mechanism, a main support beam, a front end assembly, two front wheels, and a grader attachment mechanism. (See Figure 26.) The dimensions of the grader assembly are similar to those of terrestrial graders, and were not determined by detailed structural analysis.

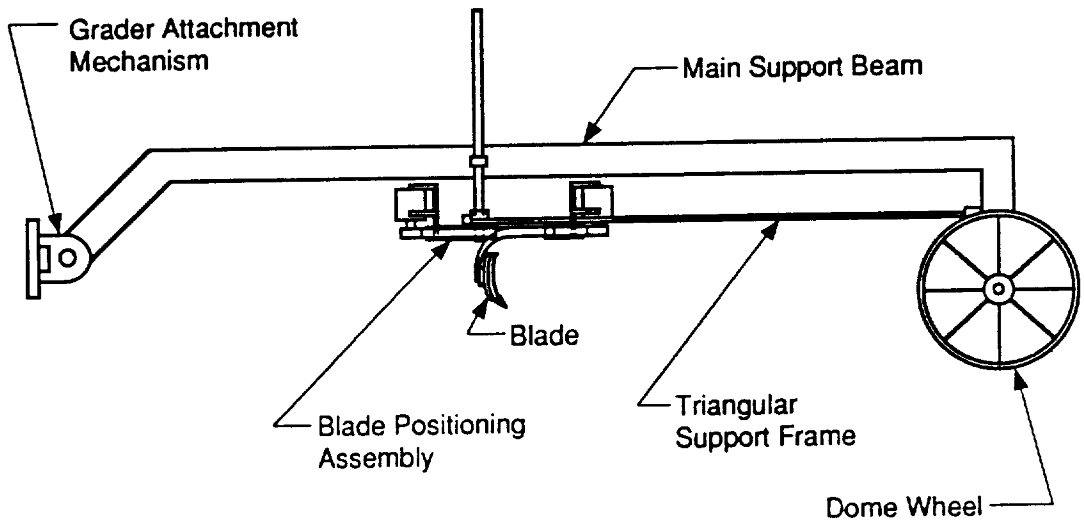


Figure 26: COMPONENTS OF THE GRADER ASSEMBLY

Grader Blade. The grader blade is a standard blade design commonly used on terrestrial graders [28]. The body of the blade is made of aluminum and is 3 m long, 50 cm high, and 2.5 cm thick. The radius of curvature of the blade is 40 cm. The cutting edge is made of titanium and is in 3 sections, each 1 m long, 10 cm high, and 5 cm thick. The edge sections are bolted to the body of the blade, and are easily replaced in case of wear or damage. (See Figure 27.)

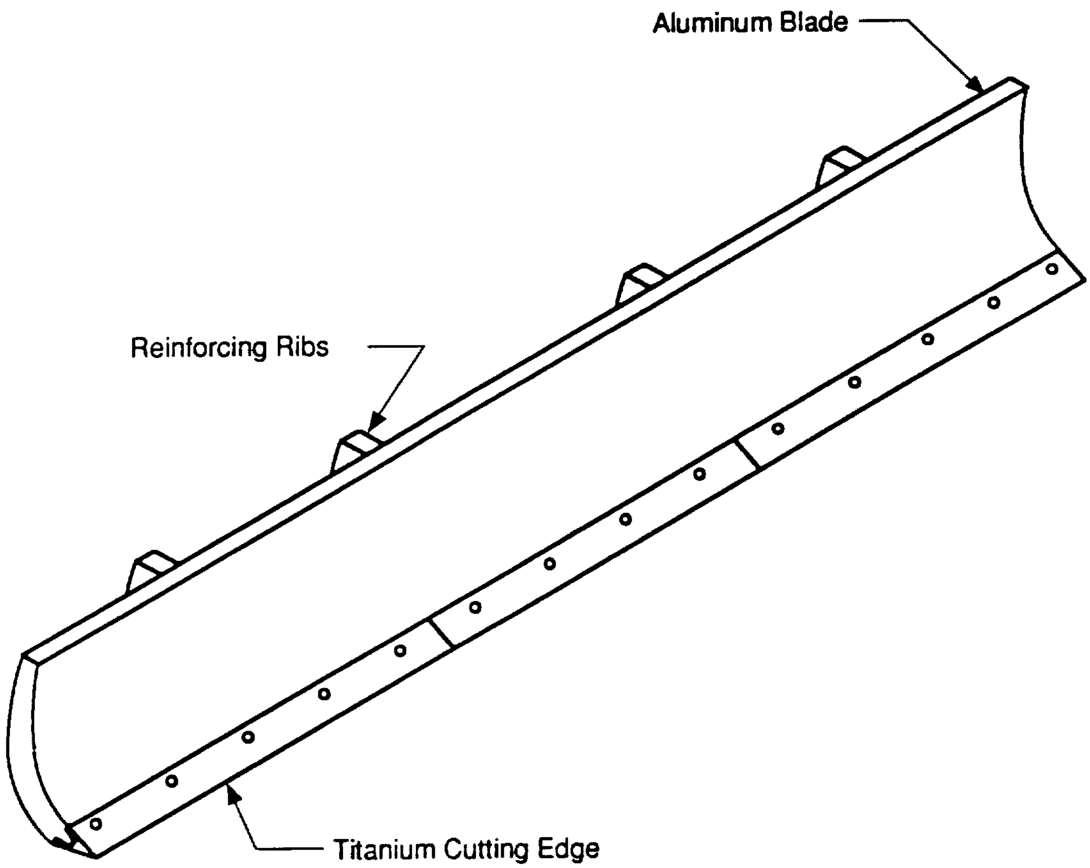


Figure 27: GRADER BLADE

Blade Positioning Assembly. The blade positioning assembly consists of a rotating positioning ring, two support arms which connect the blade to the ring, a triangular frame which supports the ring, motors which rotate the ring assembly, and power screws which change the height and horizontal angle of the blade. (See Figure 28.)

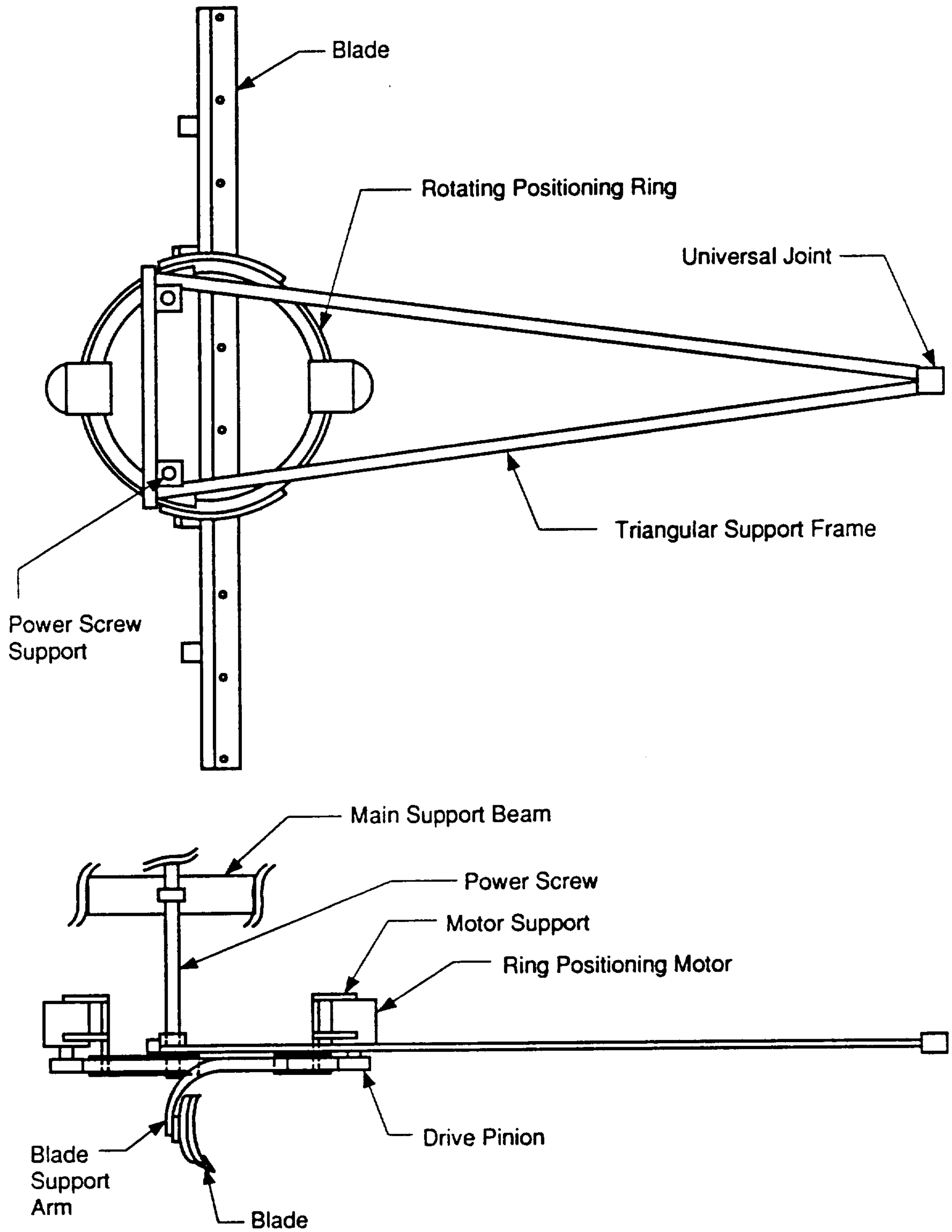


Figure 28: BLADE POSITIONING ASSEMBLY

The rotating positioning ring is used to change the blade's angle of bite, which is the angle the blade makes to the direction of motion. The ring is made of titanium and has an outside diameter of 1.2 m, an inside diameter of 1.0 m, and a thickness of 5 cm. The outside of the ring has gear teeth which mesh with pinions on the shafts of the ring positioning motors. The two aluminum blade support arms are attached to the ring 180 degrees apart and connect to the back of the blade.

The triangular ring support is connected to the grader frame with a universal joint at the front and two power screws at the rear. The universal joint allows vertical and tilting movement of the blade. Two 1.5 m long, 5 cm diameter power screws connected between the main support beam and the triangular frame raise and lower the blade assembly to adjust the depth of cut and horizontal tilt of the blade. When the grader is not connected to the drive unit, the blade is lowered to support the free end of the assembly. The triangular frame is made of aluminum and the positioning gears and power screws are made of titanium.

Main Support Beam. The main support beam is a 6 m long box beam made of aluminum. One end of the beam connects to the front wheel assembly and the other end has a mounting plate which connects the grader assembly to the main drive unit. A preliminary stress calculation was performed assuming that a maximum cutting force of 6kN acted on the center of the blade. The stress calculation resulted in the section modulus required for the box beam and the beam's dimensions. A 13 cm x 13 cm wide, 6 mm thick box beam will adequately support the load expected on the main support beam. The full analysis of cutting forces is given in Appendix D, and the stress calculations for the support structure are given in Appendix E.

Front End Assembly. The front end assembly consists of a front end support containing the steering mechanism, a universal joint for the ring support frame, and a rocking axle with an axle support frame. (See Figure 29.) The axle shafts and wheel treads are titanium; all other components are aluminum.

The front end support is a box frame which attaches to the main support beam at a 90 degree angle. The support houses the steering mechanism and provides a mount for the universal joint that attaches to the triangular ring support. The exact configurations and dimensions of the steering mechanism and universal joint were not determined.

The rocking axle allows the front wheels to follow the contour of the lunar surface independently from the main drive unit. The axle frame rotates about the vertical axis for steering and joins the rocking axle to the front end support. The axle shafts are separate to allow the wheels to rotate independently. (See Figure 30.)

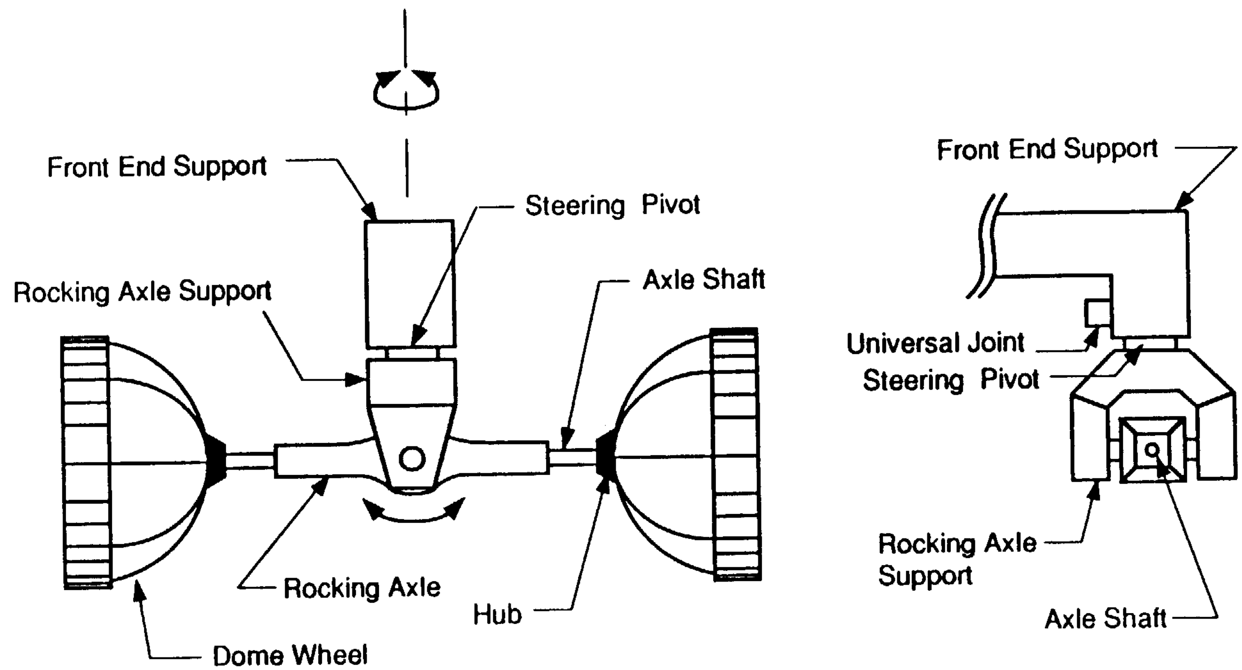


Figure 29: FRONT END ASSEMBLY

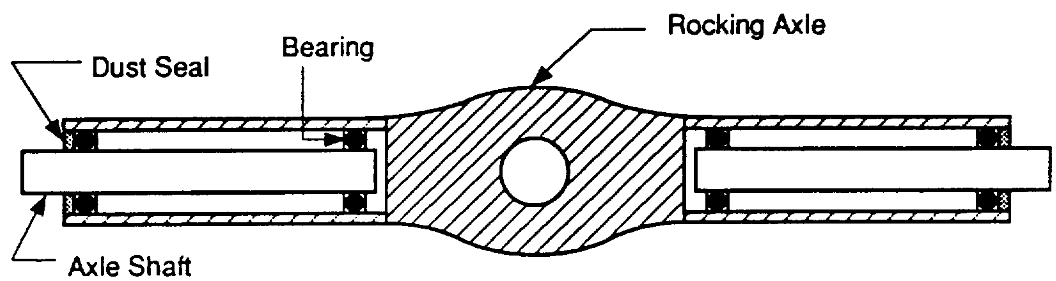


Figure 30: SECTION THROUGH ROCKING AXLE, FRONT VIEW

Front Wheels. The front wheels are hemispherical dome wheels like those on the MDU, although much smaller and not driven. The hemispherical shells are aluminum, with a diameter of 1.0 m and a thickness of 5 mm. A titanium tread 15 cm wide and 2 cm thick is connected to the outer edge of the shell.

Grader Attachment Mechanism. The grader attachment mechanism is used to connect the grader assembly to the main drive unit. It consists of one plate mounted to the main drive unit and a mating plate mounted on the end of the grader main support beam. (See Figure 31.) Sleeves on the MDU plate fit through slots in the grader plate. The sleeves are tapered to help align the plates when the grader assembly is joined to the MDU, and the openings of the sleeves on the grader plate are slightly tapered to accommodate any final misalignment when the plates are joined. Two locking pins are inserted through the sleeves to securely connect the assemblies. (See Figure 32.)

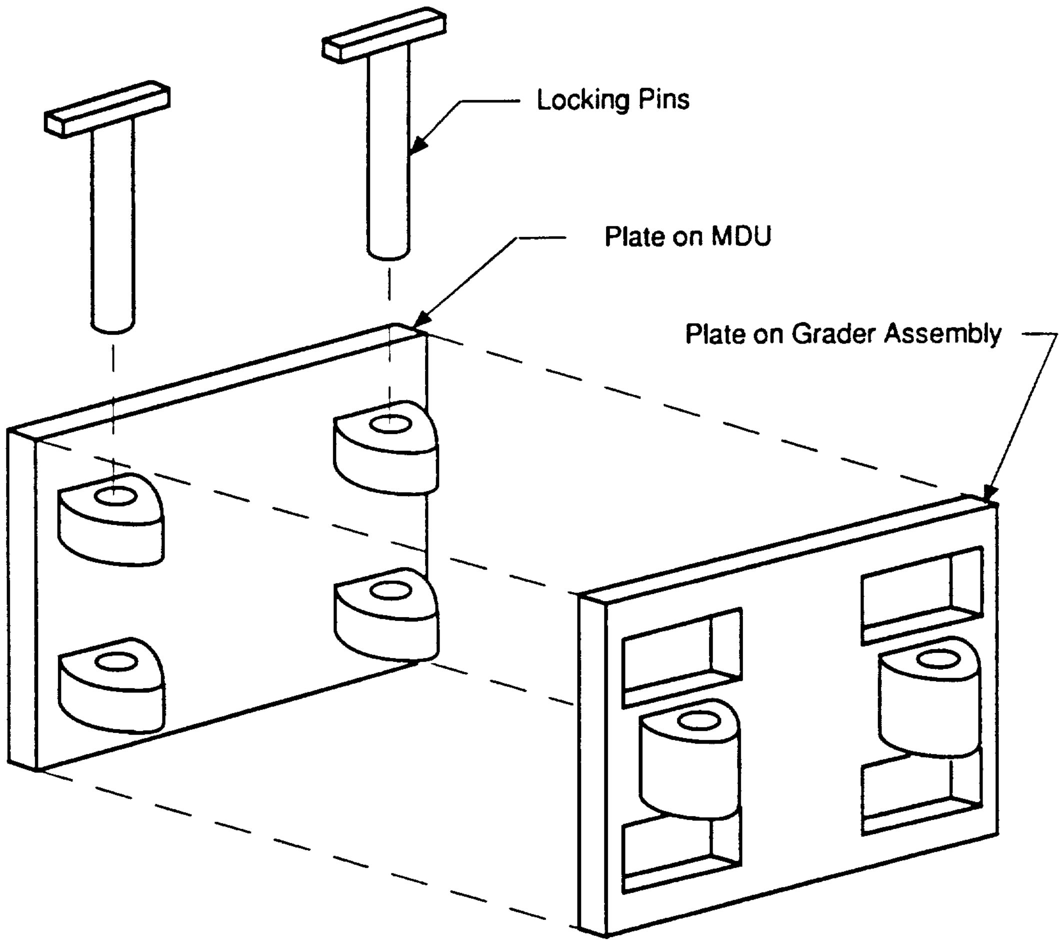


Figure 31: GRADER ATTACHMENT MECHANISM

The sleeves and locking pins provide the mechanical coupling between the drive unit and grader, and must therefore be strong and fit together tightly. They will be subject to cyclical shear loads from the grader cutting and steering forces. Due to the need for strength, the pins, sleeves, and mounting plates are made of titanium.

The design team selected this pinned connection instead of a bolted connection for two reasons. The pins can be inserted more quickly and easily than bolts; the only tool required may be a hammer. In addition, bolts are subject to binding due to dust accumulation on the threads.

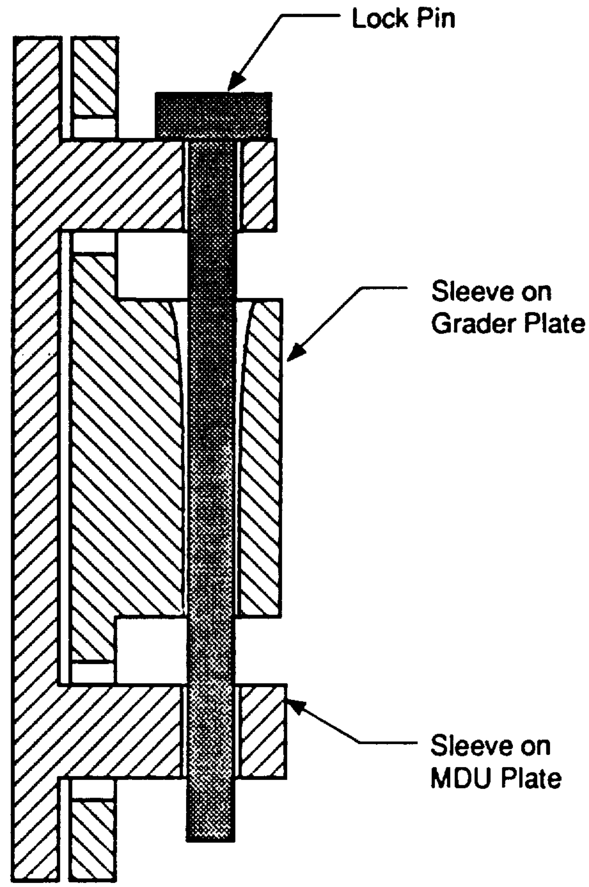


Figure 32: SECTION THROUGH GRADER ATTACHMENT MECHANISM

The connection between the grader attachment plate and the main support beam is a pivot which allows vertical movement of the front wheels of the grader relative to the drive unit. This arrangement minimizes the stresses placed on the support beam due to variations in terrain. It also allows the weight distribution on the MDU wheels to remain constant for best traction characteristics. The pivot can be locked to keep the mechanism properly aligned for coupling to the drive unit.

Alignment of the drive unit and the grader is achieved by means of a "gunsight" device and the use of the grader blade positioning mechanism. The gunsight device consists of a circular sight mounted on the front of the MDU and a diamond-shaped sight and a vertical post mounted on top of the main support beam of the grader assembly. (See Figure 33.) As the MDU approaches the grader, the driver keeps all three sighting posts centered so that the drive unit and grader are properly aligned for joining. Before connecting the two machines, the operator attaches the power cable to the grader section and uses the blade positioning mechanism to make any necessary height adjustments. The operator then makes the final connection, and inserts the locking pins to hold the MDU and grader together.

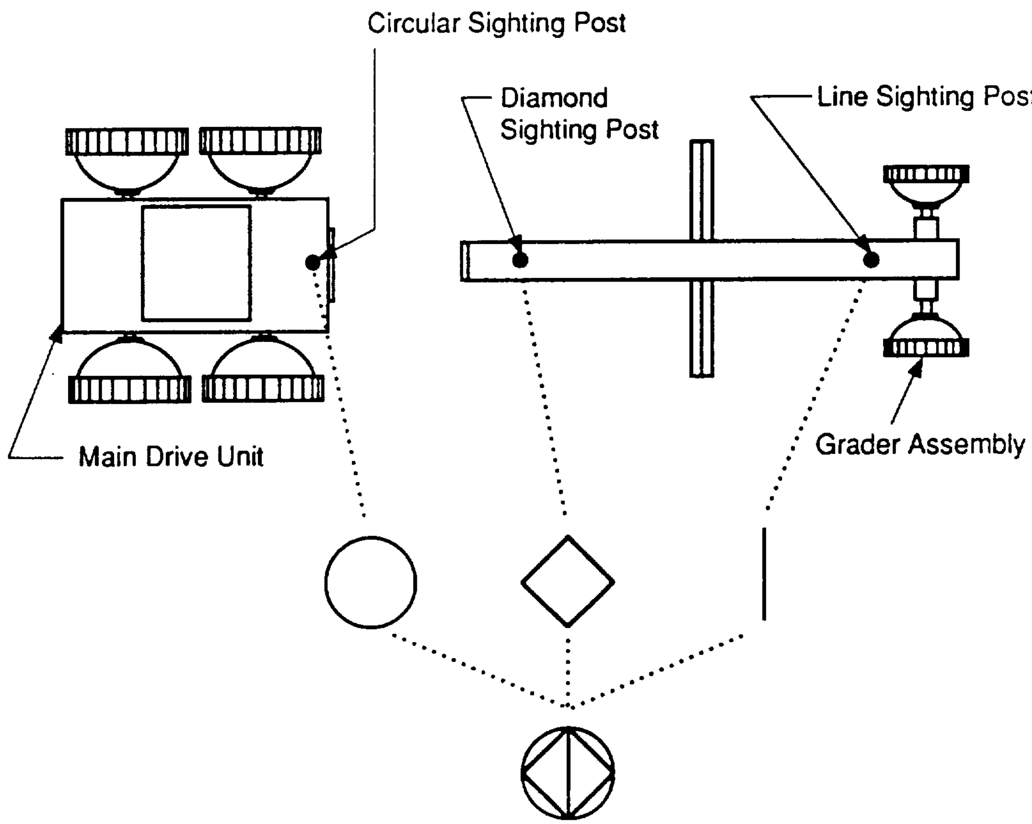


Figure 33: ALIGNMENT MECHANISM FOR JOINING DRIVE UNIT AND GRADER

Mass

The grader assembly mass was calculated using estimates for the shape and size of each component. The dimensions of the components were based on terrestrial graders since time did not permit a detailed structural analysis. The design team also assumed that much of the technology for terrestrial road construction machinery is valid for lunar applications, so a complete redesign of the grader would not be necessary. Other than a preliminary stress analysis used to determine the cross-section of the main support beam, no structural analyses were performed.

The calculations for the grader assembly mass are shown in Appendix F and are summarized in Table 2 below. The total mass of the grader assembly is approximately 400kg. Because the mass calculations only include the major components of the grader, an adjustment factor of 25% is added to the mass. This brings the total mass to approximately 500 kg.

Table 2: MASS OF THE GRADER ASSEMBLY

COMPONENT	MASS (kg)
Grader Blade	142
Blade Positioning Ring	76
Blade Connectors (2)	20
Triangular Ring Support	16
Power Screws (2)	26
Main Support Beam	45
Front End Support	8
Front Wheels (2)	40
Titanium Wheel Rims	18
TOTAL MASS	391

Power Requirements

In order to determine the power required for the grader attachment, four separate equations were used to calculate the force to cut the lunar soil. These equations are shown in Appendix D, along with a description and value for each variable in the equations. Although the results of the equations varied, they substantially agreed for shallow depths of cut. A cutting force of 6 kN was calculated for a depth of cut of 3 cm. This cutting force is small compared to those experienced on Earth, but is reasonable considering the low cohesion of lunar soil, low lunar gravity, and shallow depth of cut. Time did not allow optimization of the depth of cut, and since the value of the cutting force was within the traction capabilities of the main drive unit, the design team selected the 3 cm cut for all subsequent calculations.

The total grading depth was determined by two factors. The first factor is that the regolith must be contoured to produce a level road by removing high spots and filling in low areas. It will take several passes with the grader to achieve this. Secondly, the optimum soil density for a firm road is 95% of maximum density [29], and since the density of the regolith increases logarithmically with depth, this optimum density is reached within 15 cm of the surface [30]. The road construction process involves using the grader to excavate to a certain depth (with a corresponding density) and then compacting the remaining soil to the density required for the road. The process may be optimized to find the best grading depth and density to minimize power use by both the grader and the compactor. However, time did not permit this optimization analysis, so the design team selected 10 cm (80% of maximum density) as an estimate of the optimum depth.

Using a value of 6 kN for the cutting force and assuming a grading rate of 2 km/hr, 3.3 kW will be required to push the blade through the regolith and move the excavated soil aside. A higher grading rate can be used if desired, but the design team selected the 2km/hr speed in order to keep the power requirements low.

The design team also calculated the energy required to grade a 1 km long, 8 m wide road. Assuming a depth of cut of 3 cm, an angle of bite of 35 degrees, and a blade length of 3 m, it will take 16 passes to grade to a depth of 10 cm. The total energy required per kilometer of road is 96 megajoules (MJ). The time required to fully grade 1 km of road will be 8 hours. This estimate does not include the time required to turn the grader around after each pass. This also assumes a grading rate of 2 km/hr, which is slow compared to the grading speeds on Earth. (See calculations in Appendix D.)

Operating Features

The grader blade has the capability of rotating from an angle of bite of -40° to an angle of +40°. One source suggests that an angle of -35° or +35° is the most desirable angle for economic reasons [31]. The blade also has an adjustable cutting angle with a range from 20° to 45° [32]. The depth of cut and the horizontal angle of the blade can be adjusted by means of the power screws of the blade positioning assembly.

Unlike the main drive unit and the compactor, the grader is specialized for road construction. It can be used for excavation in mining operations and other similar tasks, but alternate uses are limited.

Compactor

The compactor does the finishing work on the road. Towed behind the main drive unit, it compacts the soil to the density needed to support vehicle traffic and produces a smooth surface. This section will discuss the configuration, mass, power requirements, and operating features of the compactor.

Configuration

The final configuration selected for the compactor was a modification of the static compactor design, with a vibratory mechanism added to increase the compaction force. (See Figure 34.) The machine consists of two compacting rollers, a soil hopper for regolith ballast, a vibratory mechanism, and a frame which supports the components and provides connection to the drive unit.

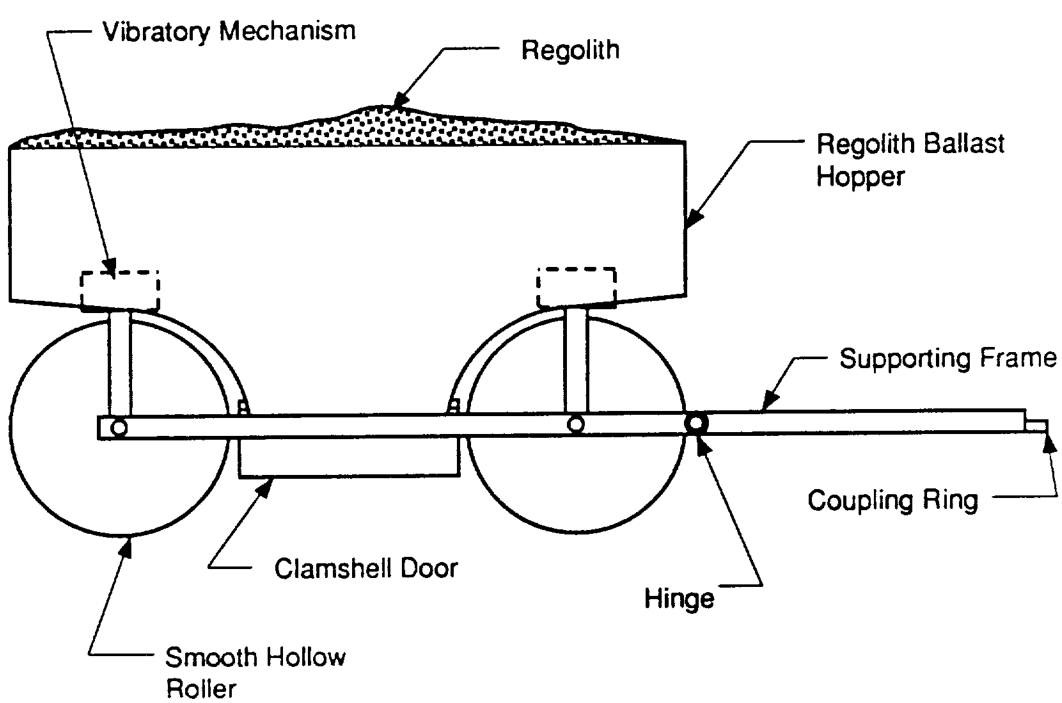


Figure 34: COMPACTOR

Rollers. The design team selected the smooth roller as best suited to the lunar soil. The rollers used on the compactor are hollow cylinders made of titanium, 1 m in diameter, 1.5 m wide, and with a wall thickness of 1 cm. The rollers are narrower than the grader blade and drive unit track to provide adequate pressure under the rollers without making the compactor excessively massive. Increasing the roller mass causes a greater rolling resistance and requires more pulling power from the drive unit. A full discussion of the compaction process and the analysis to determine the roller dimensions and forces needed to compact the soil is given in Appendix G.

Regolith Ballast Hopper. The hopper carries regolith to increase the static weight of the compactor, producing a greater compaction force. Using lunar regolith for ballast reduces the mass which has to be transported from Earth. The hopper is made from thin aluminum sheet with reinforcement trusses to add rigidity and strength. (See Figure 35.) Although time did not permit a complete development of the structure, the hopper will be designed to collapse for transportation from the Earth to the Moon to minimize the space used in the lift vehicle. The storage capacity of the hopper must be approximately 5 cubic meters (m^3) to hold the regolith required for the ballast.

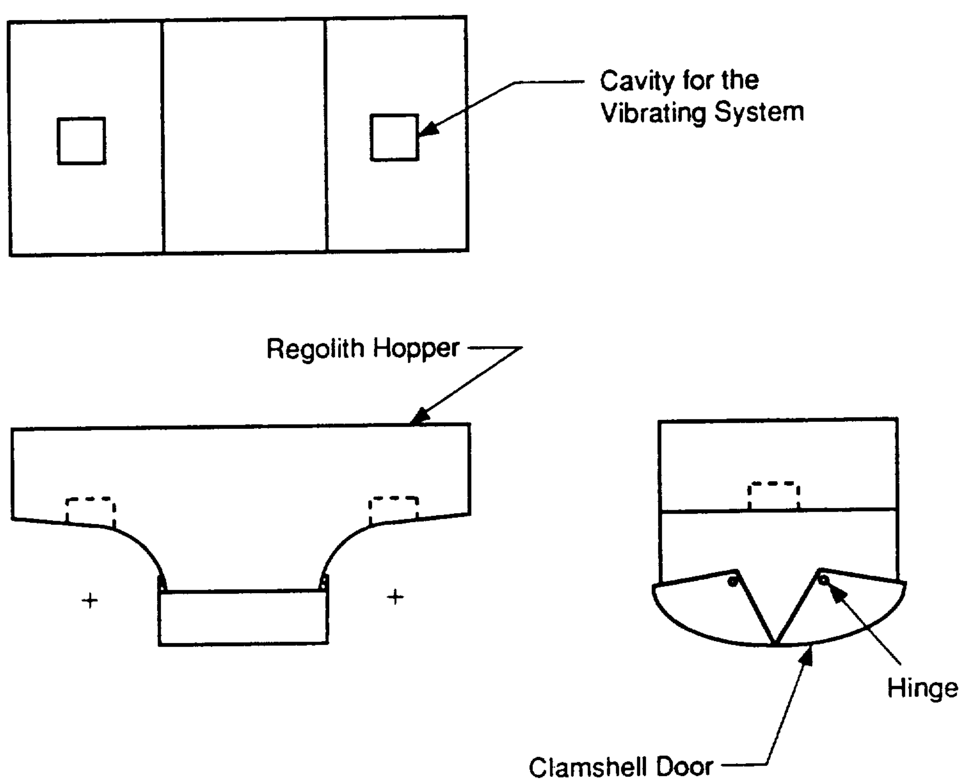


Figure 35: REGOLITH BALLAST HOPPER

Vibratory Mechanism. The calculations presented in Appendix G show that the mass of the compactor, even when the hopper is loaded with regolith, is not enough to produce the desired compaction force. The additional force required is provided by a vibratory mechanism which adds dynamic loading to the static weight of the rollers and ballast. Vibration also helps to overcome cohesive and frictional forces, inducing relative motion between the soil particles. The combined effects of the static weight and vibration compact the soil to a greater depth than any other method evaluated.

The vibratory force is produced by rapidly rotating, eccentrically-mounted weights. As shown in Appendix G, for the roller described above, the best compaction will be produced with an effective force of 8120 N. The weight of the compactor and the ballast per roller is approximately 7340 N. The remaining 780 N is obtained from the vibratory mechanism. To obtain this force, an eccentric mass of 15 kg is required to rotate at a 20 cm radius and at a speed of 154 revolutions per minute (rpm). Increasing the power required by 10% to account for expected losses in the motor, a 2.75 kW motor will be needed for each roller. The calculations for the required power are given in Appendix G.

Frame. The frame supports the rollers, hopper, and vibratory mechanism, and connects the compactor to the grader. The frame has a simple design and is made of aluminum. (See Figure 36.) The towing coupling is a ring at the end of the frame which connects to a hook mounted on the back of the drive unit. This arrangement is commonly used for terrestrial equipment, and provides for secure attachment and free movement between the machines. The dimensions of the frame are estimates; the exact dimensions must be determined by a detailed structural analysis.

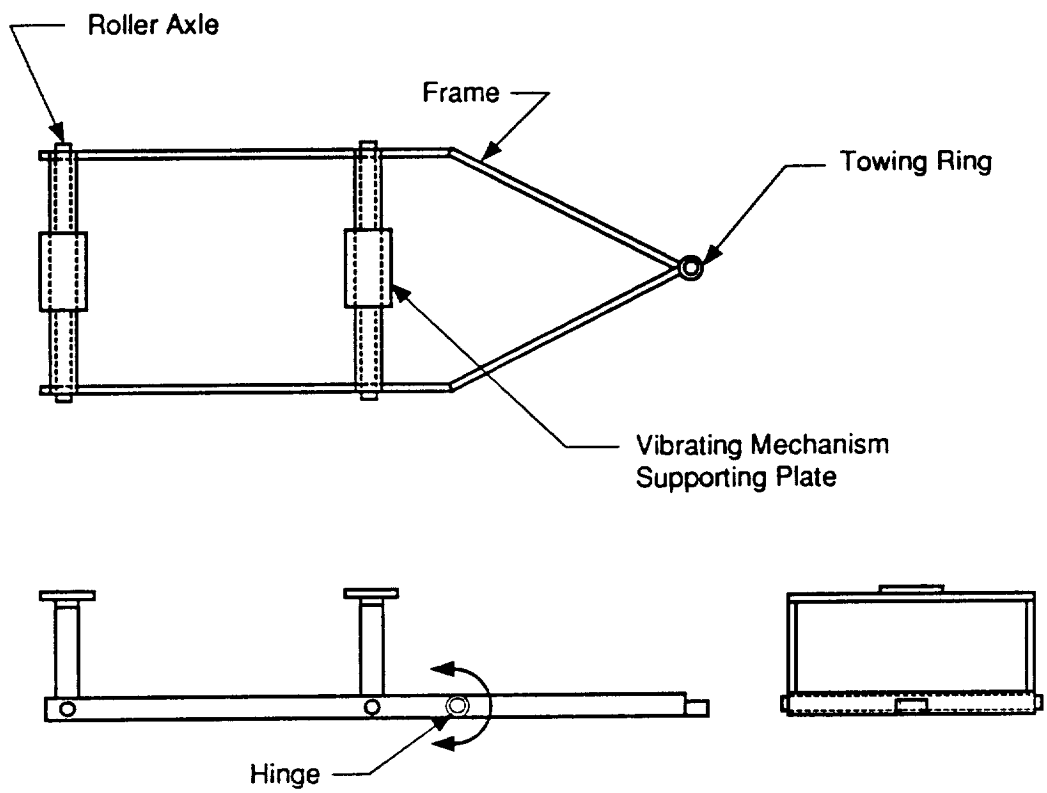


Figure 36: COMPACTOR FRAME AND TOWING RING

Mass

The design team estimated the mass of the compactor from the approximate dimensions and materials of the components. The calculations are shown in Appendix H and the results are summarized in Table 3. Because a more detailed mass calculation was performed for the compactor, only an additional 10% was added for oversimplifying the calculations (compared to 25% for the grader mass). The unloaded mass of the compactor is approximately 920 kg. The total mass of the compactor will be 9050 kg when the hopper is filled with regolith.

Table 3: MASS OF THE COMPACTOR ASSEMBLY

COMPONENT	MASS (kg)
Rollers (2)	551
Frame	67
Axles (2)	67
Hopper	86
Vibrating Mechanisms (2)	50
Towing Coupling	9
Hopper Truss	10
TOTAL MASS	840

Power Requirements

There are two factors involved in compactor power consumption. The main drive unit must provide enough tractive power to overcome the rolling resistance of the compactor. In addition, electrical power must be supplied to the compactor to operate the vibratory mechanism.

The rolling resistance of the fully loaded compactor is approximately 7.6 kN. The main drive unit requires approximately 12 kW to tow this load at a speed of 2 km/hr. The weight of the ballast on the drive unit must also be increased to produce a greater traction than is required for the grader. As stated earlier, the motors of the vibratory mechanism for each roller require 2.75 kW, so the electrical power required is 5.5 kW. The total power required for compacting operations is approximately 17.5 kW. Experience shows that for noncohesive soils on Earth, 4-6 passes are required to obtain the optimum 95% density [33]. However, the number of passes required for compaction of the lunar soil will have to be determined empirically.

Operating Features

Ballast Loading. One aspect of the compactor design that was not addressed was the loading of regolith into the hopper. A loading device can be developed at a later time, or other lunar base equipment can be used if available. Most lunar base plans suggest covering habitats with regolith to provide radiation shielding, and the equipment used for this purpose can also be used to load regolith into the ballast hopper.

Alternate Uses. The compactor can also be used as a soil carrier for mining operations by replacing the rollers with large-diameter dome wheels. The design team did not have time to work out the details of this conversion.

Evaluation of the Design Solution

The design team feels that the selection of a compacted dirt road and the design of the lunar road construction system satisfy the design criteria set forth by NASA/USRA and the design team. Six of the more important aspects of the lunar road and road construction machinery are listed below, with an evaluation of how well these features satisfy the design criteria.

Road Type

A compacted dirt road is the most feasible type of surface for initial lunar base operations given the limited material resources available. The dirt road has a relatively smooth surface and requires no material which must be brought from Earth or transported over long distances on the Moon. The main disadvantage of the dirt road is that it is easily damaged by micrometeorite impacts and vehicle traffic. However, a dirt road is also easily repaired and can be upgraded to a gravel or paved surface at later stages of base operation.

System Mass

The mass of the entire system was kept low, which was an essential design criterion. The total mass of the road construction machinery is approximately 3,020 kg, as shown in Table 4 below. This figure compares favorably to the 2,650 kg estimate by the JSC Advanced Program Office for a simpler road construction system which does not include a machine for compaction [34]. In addition, the design team did not attempt to optimize the machinery for low mass. A detailed structural analysis of the design and a more varied selection of materials will likely result in a lower overall mass.

Table 4: MASS OF THE LUNAR ROAD CONSTRUCTION SYSTEM

COMPONENT	MASS (kg)
Main Drive Unit	1600
Grader Assembly	500
Compactor	920
TOTAL MASS	3020

Power Requirements

It is difficult to evaluate the accuracy of the power requirements for the machinery. The forces for compacting and grading were based on soil properties for which the data available are incomplete and conflicting. Values used for the efficiencies of motors and drive components were only estimates based on textbook examples which assume worst-case conditions. No attempt was made to optimize the road construction processes to minimize energy use. The design team expects that a thorough analysis will produce more accurate and reliable results.

The power levels calculated for the machinery were higher than those specified for similar machines developed by others. The excavator and angle dozer proposed by the JSC Advanced Program Office has an estimated power consumption of 5 kW [35]. However, due to a lack of information on the methods used by JSC to calculate the power requirements and the uncertainties stated above, it is difficult to ascertain whether this difference in power levels is significant.

Material Selection

The materials used for the structure of the road construction machinery were selected for to their ability to withstand the harsh lunar environment. The aluminum alloy is resistant to the extreme temperature variations, the vacuum, and the high solar and galactic radiation level. The titanium alloy also possesses these properties, and is resistant to residual stresses and abrasive wear. Both alloys have a high strength-to-weight ratio. Since the purpose of this project was to produce a conceptual design for the road construction machinery, only the materials for the structure and interfacing components were selected. A more detailed material selection analysis for each part of the machinery must be performed.

Automatic and Remote Operation

The final design solution minimizes the amount of manpower required to operate the road construction machinery by providing the capability for automated operation. Some processes, such as connecting or disconnecting the grader and compactor or critical grading and road maintenance operations, require direct manipulation by base crew members. However, other processes such as compacting can be fully automated. In most cases, the machinery can be remotely operated either from Earth or the lunar base, minimizing the need for crew members to be exposed to the lunar surface environment.

Machine Versatility

The main drive unit and compactor were designed to be used for tasks other than grading and compacting. The drive unit is a multiple-purpose tractor, and can be used for a wide range of operations. The compactor can be used for soil or cargo transportation when it is not used for road construction. The grader assembly, however, will be limited in the number of other tasks that it will be able to perform. These tasks are primarily limited to excavation, mining, and light bulldozing. Modifications of the grader assembly may make it more versatile.

RECOMMENDATIONS FOR FURTHER WORK

The aim of this project was to conceptually design a road construction system for the lunar surface. The design team accomplished this task, but much detailed work must be completed in order to produce a more thorough design. Three major areas recommended for further study are optimization of the designs for minimum mass, optimization of the road construction process for minimum energy use, and experimentation with the lunar soil to obtain accurate values for soil properties. Other areas for further work include remote control and guidance systems, power systems, and roadway survey and layout. An annotated bibliography of material related to lunar engineering would also be useful for future design projects.

Machine and Process Optimization

Much analysis must be performed to optimize the road construction machines and processes in order to minimize the system mass and energy consumption. Loads must be analyzed to design structures which are strong yet lightweight. The operating characteristics of the mechanical components such as the wheels and the grader blade must be examined in order to maximize performance. For example, an ellipsoidal wheel shell will reduce wheel width, and may also give better performance than a hemispherical wheel.

The road construction process must also be carefully studied. The grading and compaction process can be optimized to find the best grading depth to minimize the power used by both machines. Although road construction uses a large amount of energy, the power required can be reduced by slowing down the machinery. A review of the mission schedule will help to determine the optimum rate for the road construction process.

Lunar Soil Mechanics

One of the most important areas requiring further study is the soil mechanics of the lunar surface. Soil mechanics form the basis for traction, grading, and compaction analysis. The design team had a difficult time finding information on lunar soil mechanics, and much of this information was contradictory. More accurate and extensive information on the cohesive and frictional moduli of soil deformation, which are essential for calculation of traction and rolling resistance, would be particularly useful.

Control Systems

The remote and automatic control systems for the machinery need to be developed. Remote control systems using digital radio control are currently available, and can be adapted for the road construction machinery. Sensing equipment which allows the operator to monitor machine operation must be developed and refined.

The option for remote control from Earth needs to be studied. This approach has several advantages. The first is that the crew of the lunar base will be freed for other tasks, conserving resources such as oxygen, food, and energy. Another advantage of Earth-based operation is that experienced road-construction personnel can operate the equipment, eliminating the need to train base crew. One obstacle to Earth-based control is the time delay involved in signal transmission. However, the machinery is designed to move slowly, so this should not present major difficulties.

The options for automatic control vary widely. The machinery may be designed to follow a series of guide markers using radio or laser beams. This will be suitable for simple processes such as compacting the fully graded road. More complicated systems involving feedback and artificial intelligence may allow the entire road construction process to be fully automated. Various approaches must be studied and evaluated.

Power Systems

A practical power system for the road construction machinery must be developed. It must be portable, have relatively low mass, and provide the power for extended operation without the need for recharging. The system must be able to respond to dynamic loads from the machinery.

As discussed before, one of the most likely approaches is the use of fuel cells. However, high-efficiency batteries and dynamic radioisotope generator systems also offer possibilities.

Road Survey and Layout

Before construction of the road can start, the path for the road must be selected and marked. Survey work can be done from lunar orbit by satellites equipped with optical or radar sensors. Automated or remotely-controlled rovers may be used to explore the lunar surface, measuring soil properties and placing guidance markers.

It is likely that some work will have to be done on the lunar surface by the base crew. Although the path for the road will be chosen to avoid obstacles such as large rocks and craters, it may be necessary for base personnel to use explosives to remove some rocks.

The best method for road survey work must be selected, and any necessary equipment and procedures developed.

Annotated Bibliography

One of the most frustrating problems the design team faced was finding information on the lunar environment and work performed by other NASA design teams that was relevant to this project. Some sources could be found in the reference sections of other reports, or in collections of technical papers such as Engineering, Construction, and Operations in Space. However, it was often necessary to go through long lists of titles, and it was difficult to tell what material would be relevant. This problem can be alleviated by an annotated bibliography that will provide references for future design teams.

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APPENDICES

APPENDIX A

Decision Matrices

Table A.1: DECISION MATRIX FOR GRADER

Decision Criteria	Weighting Factor	Alternate Design							
		# 1		# 2		# 3		# 4	
Use as Main Drive Unit (MDU)	0.1875	5	0.94	1	0.19	8	1.50	8	1.50
Ease of Conversion to MDU	0.1875	8	1.50	8	1.50	6	1.13	2	0.38
Dust Insensitivity	0.3125	2	0.63	3	0.94	9	2.81	5	1.56
Temp. Variation Insensitivity	0.25	3	0.75	2	0.50	9	2.25	5	1.25
Mechanical Simplicity	0.0625	4	0.25	5	0.31	8	0.50	3	0.19
Sum	**1.00**		**4.06**		**3.44**		**8.19**		**4.88**

Alternate Design #1: Grader with Front and Rear Retractable Frames
Alternate Design #2: Modified Terrestrial Grader with Retractable Frame
Alternate Design #3: Modified Terrestrial Grader with Removable Center Section (Fixed Joints)
Alternate Design #4: Modified Terrestrial Grader with Removable Center Section (Articulated Joints)

Table A.2: DECISION MATRIX FOR COMPACTOR

Decision Criteria	Weighting Factor	Alternate Design							
		#	1	#	2	#	3	#	4
Mass	0.1212	9	1.09	8	0.97	4	0.48	1	0.12
Size	0.0152	8	0.12	7	0.11	5	0.08	2	0.03
Efficiency (# of Passes)	0.1061	3	0.32	8	0.85	8	0.85	9	0.95
Methods of Compaction	0.0606	6	0.36	6	0.36	2	0.12	9	0.55
Road Maintenance	0.1061	5	0.53	5	0.53	2	0.21	8	0.85
Ability to Do Other Tasks	0.1212	1	0.12	9	1.09	1	0.12	2	0.24
Mechanical Simplicity	0.0303	9	0.27	8	0.24	1	0.03	1	0.03
Independence	0.0758	0	0.00	0	0.00	0	0.00	10	0.76
Rate of Compaction	0.0606	4	0.24	6	0.36	3	0.18	7	0.42
Production of Smooth Surface	0.1515	9	1.36	10	1.52	3	0.45	9	1.36
Production of Level Surface	0.1515	9	1.36	10	1.52	3	0.45	9	1.36
Sum	**1.00**		**5.79**		**7.55**		**2.98**		**6.68**

Alternate Design #1: Roller Compactor
Alternate Design #2: Static Compactor with Vibratory Mechanism
Alternate Design #3: "Thumper"-Type Impact Compactor
Alternate Design #4: Self-Driven Compactor

APPENDIX B

Material Selection

APPENDIX B
Material Selection

The materials selected for the machinery of the lunar road construction system must be capable of withstanding the harsh lunar environment, including exposure to high radiation levels, micrometeorite bombardment, abrasive dust, extreme variations in temperature, and a vacuum.

Radiation can cause severe changes in the mechanical properties of metals. Neutron and gamma radiation may embrittle structural materials as well as cause thermal stresses by creating severe temperature gradients [1]. The main type of damage mechanism that metals undergo is an atomic displacement which results in defects in the crystal lattice of materials. Neutron and gamma interactions produce vacancies and interstitial point defects, which cause swelling (due to void formation) and radiation-enhanced creep in stainless steels [2]. Clusters of these defects, called displacement spikes, cause an increase in the yield strength and a reduction of ductility of the metal, which may lead to brittle fracture.

The high levels of radiation on the moon are a result of two sources: the galactic cosmic radiation (GCR) and solar particle events (SPE). GCR is a continuous, intense, omni-directional flux of protons, alpha particles, and heavier nuclei emitted from interstellar sources such as supernovas [3]. SPE is a flux of radioactive particles leaving the sun during a solar sunspot cycle, which occurs on the average once in every 11.1 years, but ranges from 7 to 17 years [4]. Up to one hundred SPEs can occur during a sunspot cycle.

The material selected for the road construction must also resist the constant bombardment from micrometeorites and the abrasive lunar dust produced by the collision of these micrometeorites with the lunar surface. The lack of an atmosphere on the moon produces a very hard vacuum, which allows metals to experience a loss of material due to direct evaporation from the surface. Some metals may loose as much as 0.004 inches in the course of a year [5].

The surface temperature on the moon can vary from -250°F (-150°C) during the lunar night to 250°F (120°C) during the lunar day [6]. This wide temperature range may cause thermal stresses on the structure of a vehicle when metals of widely different coefficients of thermal expansion are joined or when the structure is confined by another member.

Factors for Material Selection. The main factors in selecting a metal for the lunar road construction machinery were the cost, density, strength and ductility, coefficient of thermal expansion, susceptibility to evaporative losses in a vacuum, resistance to the abrasive lunar dust and micrometeorite bombardment, and resistance to damage from radiation exposure.

Materials Considered. The metals the design team considered for the lunar road construction machinery were alloys of steel, aluminum, titanium, magnesium, beryllium, cobalt, molybdenum, tantalum and tungsten. A matrix metal, a new material still under consideration by NASA, was also considered.

Selection of Materials. Several alloys can be eliminated because of their cost, poor material properties, and susceptibility to the lunar environment. Alloys of cobalt, molybdenum, tantalum, and tungsten are extremely expensive compared to the other metal alloys. Alloys of molybdenum, beryllium, and tungsten have very low ductility and may undergo brittle fracture if impacted by a micrometeorite or a sudden heavy load. Magnesium has a very low sublimation pressure and may lose as much as 0.004 inches of material from its surface in the course of a year due to evaporation. Steel is a very strong and inexpensive metal, but it has a very low strength to weight ratio.

The matrix metals now being considered by NASA are extremely strong metals formed by placing metal fibers of titanium, silicon, carbon, or graphite in a metal matrix, usually aluminum. These metals have a high strength to weight ratio, but are extremely expensive, hard to manufacture, and are not easily welded. Their use will primarily be restricted for components requiring very stiff members [7].

Aluminum is a very lightweight, strong, and inexpensive metal. It has a high strength to weight ratio and is has very little evaporative losses in a vacuum. Titanium is expensive, but it has a high strength to weight ratio.

The most common materials now used by NASA in the space program are listed in Table B.1, along with some of their most important properties. At the advice of Mr. Brian Muirhead of NASA's Jet Propulsion Laboratory, the design team selected an aluminum alloy (Al-Li 2090) for the structures of the road construction machinery and a titanium alloy (Ti-6 Al-4 V) for load-bearing and moving parts.

Table B.1: PROPERTIES OF ALLOYS USED BY NASA

ALLOY	TENSILE STRENGTH (psi)	YIELD STRENGTH (psi)	DENSITY (g/cm^3)
Al 7075	70,000	57,000	2.80
Al 2024	70,000	64,000	2.80
Al-Li 2090	78,000	72,000	2.56
Ti-6 Al-4 V	160,000	145,000	4.42

Appendix B: References

1. Principles of Radiation Shielding, Arthur B. Chilton, J. Kenneth Shultis, and Richard E. Faw, (Prentice-Hall, Inc., Englewood Cliffs, NJ, 1984), p. 366.

2. Ibid, p. 379.

3 Lunar Split Mission: A Robot Constructed Lunar Base Scenario, George W. Davis, etc., NASA/USRA Advanced Space Design Program, August 1988, p. 37.

4. Ibid., p. 37-38.

5. Lunar Construction Utility Vehicle, Old Dominion University, NASA/USRA Summer Report, July 1989.

6. Lunar Bases and Space Activities of the 21st Century, W.W. Mendell, editor, Lunar Planetary Institute, Houston TX, 1985, p. 383.

7. Phone conversation with Mr. Brian Muirhead of the Jet Propulsion Lab, 20 February 1990.

APPENDIX C

Traction Analysis

APPENDIX C

Traction Analysis

Basic Principles

Traction forces are produced by the reaction of soil to imposed loads, governed by the shear strength of the soil and its resistance to deformation. The two main mechanisms are cohesion between the soil grains and frictional forces between grains due to the imposed load [1].

The developed traction is determined by the following relationship:

$$H = Ac + Mg \tan \phi. \qquad (C1)\ [2]$$

H: developed traction

A: area of wheel or track in contact with soil

c: soil coefficient of cohesion

M: mass supported by contact surface

g: acceleration of gravity

ϕ: angle of internal friction of soil
($\tan \phi$ = coefficient of friction of soil)

For plastic soils with low internal friction (such as mud or clay) traction is determined primarily by the shear forces acting on the tread contact area. For frictional soils with low cohesion (such as sand or lunar regolith) the traction force is determined primarily by vehicle weight, regardless of contact area.

The contact pressure under the tread is a function of wheel sinkage into the soil.

$$p = \left(k_c / b + k_\phi\right) z^n \qquad \text{(C2) [3]}$$

p: contact pressure

k_c: cohesive modulus of soil deformation

b: length of minor axis of contact area

k_ϕ: frictional modulus of soil deformation

z: depth of sinkage

n: dimensionless exponent of soil deformation

Rearranging equation (C2) gives sinkage as a function of contact pressure:

$$z = \left[\frac{p}{k_c / b + k_\phi}\right]^{1/n} . \qquad \text{(C3) [4]}$$

This indicates that the wider the contact area (b), the greater the sinkage, even for the same pressure. A long, narrow contact area will give less sinkage, reducing power losses from pushing a wheel through loose soil.

A narrow wheel also produces less slip. This has been shown in field tests comparing two wheels with the same contact area and ground pressure; one a large diameter wheel with a narrow footprint, the other small diameter with a wide footprint. The large, narrow wheel will produce greater traction at up to 40% slip. This effect is greatest at low-slip conditions [5].

The other factor in the net traction produced by a vehicle is the rolling resistance of the wheels or tracks. Rolling resistance is due to three effects: compaction of the soil under the wheel, "bulldozing" or pushing loose soil ahead of the wheel, and dragging of the soil by suspension and other vehicle components.

The major component of rolling resistance on the Moon is compaction. Bulldozing depends on the amount of wheel sinkage and soil characteristics as well as wheel geometry. With lunar soil and narrow wheels, bulldozing will be a minor effect. The road construction machinery is designed with clearance above the soil surface, so dragging is not a factor.

Compaction resistance is given by:

$$R_c = \frac{1}{(3-n)^{\frac{2n+2}{2n+1}}\,(n+1)\,(k_c + bk_\phi)^{\frac{1}{2n+1}}}\left[\frac{3Mg}{\sqrt{2R}}\right]^{\frac{2n+2}{2n+1}} \qquad \text{(C4) [6]}$$

R_c: compaction resistance, force opposing developed traction

R: radius of wheel

This shows that rolling resistance will increase with vehicle mass, and decrease with increasing wheel radius and width, higher soil cohesion and friction, and a larger soil deformation exponent.

In general, the best performance will be obtained with a large-diameter wheel with a fairly narrow tread. This will not allow the wheel to remain on top of the soil surface without sinkage, but it does provide several advantages. Once the surface is penetrated, a narrow wheel will give less sinkage for a given contact pressure. In low-slip conditions, the narrow wheel generates higher traction forces. A large-diameter wheel has lower rolling resistance from soil compaction. There is also less variation in penetration depth over the contact area, resulting in a more even pressure and stress distribution on the wheel due to soil compression.

Soil Properties and Vehicle Characteristics

Data on lunar soil are available from several sources. Although the exact figures vary, there are some general trends. Lunar soil is fine-grained with relatively low cohesion, with values ranging from 0.1 to 1.42 kN/m^2. The angle of internal friction is 35° to 45° [7,8]. The cohesive modulus of soil deformation (k_c) is 3.5 kN/m^2, the frictional modulus (k_ϕ) is 8.1 kN/m^3, and the dimensionless exponent of soil deformation is 1 [9].

A lunar truck proposed by the Advanced Program Office of the Johnson Space Center has a mass of 1400-1900 kg, and has four 2-meter diameter cone wheels [10]. This vehicle is similar to the drive unit developed in the team's design. Traction analysis will be performed using this vehicle mass with 1.6 m diameter, 30 cm wide hemispherical dome wheels.

Traction Analysis

The first step in the traction analysis is to find the contact area between the wheel and the soil. The wheel contacts the soil along a sector of the tread with length L. (See Figure C1.) For the analysis, the average contact pressure is assumed to act at a depth equal to the position of the centroid of the contact arc, given by:

$$\bar{r} = \frac{R \sin \theta}{\theta} \qquad \text{(C5)}$$

$\bar{r}$: distance from center of wheel to centroid of contact arc

R: radius of wheel

θ: half-angle of contact arc

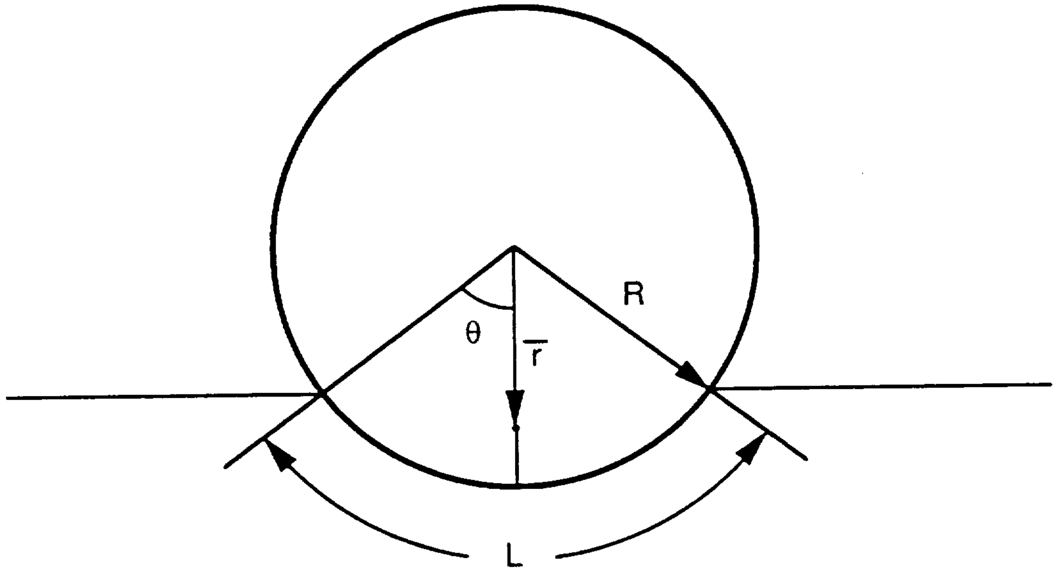

Figure C1: GEOMETRY OF WHEEL-SOIL CONTACT

From equation C2, the average contact pressure, $\bar{p}$, is:

$$\bar{p} = \frac{Mg}{A} = (k_c + bk_\phi) z^n \qquad (C6)$$

$\bar{p}$: average contact pressure

A: contact area

where

$$A = bL = bR(2\theta) \qquad (C7)$$

and

$$z = \bar{r} - R\cos\theta$$

$$= R\left(\frac{\sin\theta}{\theta} - \cos\theta\right). \qquad (C8)$$

With n = 1, combining (C6) through (C8) and rearranging yields:

$$\theta\left(\frac{\sin\theta}{\theta} - \cos\theta\right) = \frac{Mg}{2bR^2\left(k_c / b + k_\phi\right)} \qquad (C9)$$

which must be solved for θ numerically.

The equation for rolling resistance, (C4), simplifies to

$$R_c = \frac{\left[\frac{3Mg}{\sqrt{2R}}\right]^{4/3}}{2^{7/3}\left(k_c + bk_\phi\right)^{1/3}}. \qquad (C10)$$

The total net traction force developed by the vehicle is then:

$$H_N = A_T c + M_T g - R_{CT}. \qquad (C\,11)$$

H_N: total net traction force

A_T: total contact area for all wheels

M_T: total vehicle mass

R_{CT}: total rolling resistance for all wheels

Analysis Program

A computer program using these equation was used to calculate the net traction force produced by the drive unit, varying the wheel diameter from 1.0 - 2.0 m and the mass from 2000 - 10000 kg. These calculations were performed for both soft and firm soil.

The soil deformation moduli, k_c and k_ϕ,were only given for soft soil, and values for firm soil were needed for the analysis. The method given by Bekker for the determination of the modulus values indicates that k_c varies linearly with cohesion and that k_ϕ varies logarithmically with the friction angle. This gives values for firm soil of 20.588 kN/m^2 for k_c and 8.515 kN/m^3 for k_ϕ.

The gravitational acceleration of the Moon is 1.624 m/s^2 [11].

The computer program used for traction analysis appears on the following pages, along with the results for the 1.6 m diameter wheels used on the drive unit. Graphs of the results are shown in Figures C2 and C3.

Another program was used to calculate the sensitivity of the developed traction and rolling resistance to variations in wheel diameter, tread width, vehicle mass, soil cohesion, and soil friction in order to determine which parameters had the greatest effect. Each factor was varied over a reasonable range with the other factors held at an intermediate value. The results were normalized over the range to show relative effect. This program is also presented, along with the calculation results. Graphs of the traction and rolling resistance are shown in Figures C4 and C5.

Computer Program for Traction Analysis

```
      PROGRAM TRACTION
C
      REAL MASS, MRED, KC, KPHI
C
      OPEN( UNIT=6, FILE='TRACTION.DAT', STATUS='NEW' )
C
      GRAV = 1.624
      WID = 0.30
      KC = 3500.0
      KPHI = 8100.0
      WHLS = 4.0
      PHI = 0.61087
      CC = 170.0
C
      DO 10 DIA = 1.0, 2.0, 0.2
         WRITE( 6, 101 )
 101     FORMAT( ///, 1X, 'DIAM', T10, 'MASS', T20, 'THETA', T30,
     &           'AREA', T40, 'SHEAR', T50, 'FRICTION', T60, 'RESIST',
     &           T70, 'TRACTION', / )
         RAD = DIA / 2.0
C
         DO 20 MASS = 2000.0, 10000.0, 200.0
C
            MRED = MASS / WHLS
            CALL BISECT( 0.0, 1.57, 1.0E-6, THETA, 1000, MRED, GRAV,
     &                   WID, RAD, KC, KPHI, WHLS )
C
            AREA = 2.0 * WID * RAD * THETA * WHLS
            SHEAR = AREA * CC / 1000.0
            FRIC = MASS * GRAV * TAN( PHI ) / 1000.0
            RESIST = WHLS * ( ( 3.0 * MRED * GRAV / SQRT( DIA ) ) **
     &               ( 4.0 / 3.0 ) / ( 2.0 ** ( 7.0 / 3.0 ) *
     &               ( KC + WID * KPHI ) ** ( 1.0 / 3.0 ) ) ) / 1000.0
            TRACTION = SHEAR + FRIC - RESIST
C
            WRITE( 6, 102 ) DIA, MASS, THETA, AREA, SHEAR, FRIC,
     &                      RESIST, TRACTION
 102        FORMAT( 1X, F4.1, T10, F7.0, T20, F6.4, T30, F6.4, T40,
     &              F7.3, T50, F7.3, T60, F7.3, T70, F7.3 )
C
 20      CONTINUE
 10   CONTINUE
C
      STOP
      END
C
```

```
C
      SUBROUTINE BISECT( XL, XU, ES, XR, MAXIT, MRED, GRAV, WID, RAD,
     &                   KC, KPHI, WHLS )
C
      REAL MRED, KC, KPHI
C
      ITER = 0
      EA = 1.1 * ES
 10   IF( ( EA .GT. ES ) .AND. ( ITER .LT. MAXIT ) ) THEN
         XR = ( XL + XU ) / 2.0
         ITER = ITER + 1
C
         IF( XL + XU .NE. 0.0 ) THEN
            EA = ABS( ( XU - XL ) / ( XU + XL ) ) * 100.0
         ENDIF
C
         TEST = ANGLE( XL, MRED, GRAV, WID, RAD, KC, KPHI, WHLS )
     &        * ANGLE( XR, MRED, GRAV, WID, RAD, KC, KPHI, WHLS )
C
         IF( TEST = 0.0 ) THEN
            EA = 0.0
         ELSE
            IF( TEST .LT. 0.0 ) THEN
               XU = XR
            ELSE
               XL = XR
            ENDIF
         ENDIF
C
      GO TO 10
C
      ENDIF
C
      RETURN
      END
C
C
      FUNCTION ANGLE( X, MRED, GRAV, WID, RAD, KC, KPHI, WHLS )
C
      REAL MRED, KC, KPHI
C
      ANGLE = X * ( SIN( X ) / X - COS( X ) ) - ( MRED * GRAV )
     &        / ( 2.0 * WID * RAD ** 2.0 * ( KC / WID + KPHI ) )
C
      END
```

Results for soft soil

DIAM	MASS	THETA	AREA	SHEAR	FRICTION	RESIST	TRACTION
1.6	2000.	.6959	1.3361	.227	2.274	1.051	1.451
1.6	2200.	.7191	1.3807	.235	2.502	1.193	1.543
1.6	2400.	.7411	1.4229	.242	2.729	1.340	1.631
1.6	2600.	.7620	1.4629	.249	2.957	1.491	1.715
1.6	2800.	.7818	1.5011	.255	3.184	1.646	1.794
1.6	3000.	.8008	1.5376	.261	3.411	1.804	1.869
1.6	3200.	.8191	1.5726	.267	3.639	1.966	1.940
1.6	3400.	.8366	1.6063	.273	3.866	2.132	2.008
1.6	3600.	.8535	1.6388	.279	4.094	2.301	2.072
1.6	3800.	.8699	1.6702	.284	4.321	2.472	2.133
1.6	4000.	.8857	1.7006	.289	4.549	2.647	2.190
1.6	4200.	.9011	1.7301	.294	4.776	2.825	2.245
1.6	4400.	.9160	1.7588	.299	5.003	3.006	2.296
1.6	4600.	.9306	1.7867	.304	5.231	3.190	2.345
1.6	4800.	.9447	1.8138	.308	5.458	3.376	2.391
1.6	5000.	.9585	1.8403	.313	5.686	3.565	2.434
1.6	5200.	.9720	1.8662	.317	5.913	3.756	2.474
1.6	5400.	.9852	1.8915	.322	6.141	3.950	2.512
1.6	5600.	.9981	1.9163	.326	6.368	4.146	2.547
1.6	5800.	1.0107	1.9405	.330	6.595	4.345	2.580
1.6	6000.	1.0230	1.9643	.334	6.823	4.546	2.611
1.6	6200.	1.0352	1.9875	.338	7.050	4.749	2.639
1.6	6400.	1.0471	2.0104	.342	7.278	4.954	2.665
1.6	6600.	1.0588	2.0329	.346	7.505	5.162	2.689
1.6	6800.	1.0703	2.0549	.349	7.733	5.372	2.710
1.6	7000.	1.0816	2.0766	.353	7.960	5.583	2.730
1.6	7200.	1.0927	2.0980	.357	8.187	5.797	2.747
1.6	7400.	1.1036	2.1190	.360	8.415	6.013	2.763
1.6	7600.	1.1144	2.1396	.364	8.642	6.230	2.776
1.6	7800.	1.1250	2.1600	.367	8.870	6.450	2.787
1.6	8000.	1.1355	2.1801	.371	9.097	6.671	2.797
1.6	8200.	1.1458	2.1999	.374	9.325	6.895	2.804
1.6	8400.	1.1560	2.2195	.377	9.552	7.120	2.810
1.6	8600.	1.1660	2.2388	.381	9.779	7.347	2.813
1.6	8800.	1.1759	2.2578	.384	10.007	7.575	2.815
1.6	9000.	1.1857	2.2766	.387	10.234	7.806	2.816
1.6	9200.	1.1954	2.2952	.390	10.462	8.038	2.814
1.6	9400.	1.2050	2.3136	.393	10.689	8.272	2.811
1.6	9600.	1.2144	2.3317	.396	10.917	8.507	2.806
1.6	9800.	1.2238	2.3497	.399	11.144	8.744	2.799
1.6	10000.	1.2330	2.3674	.402	11.371	8.983	2.791

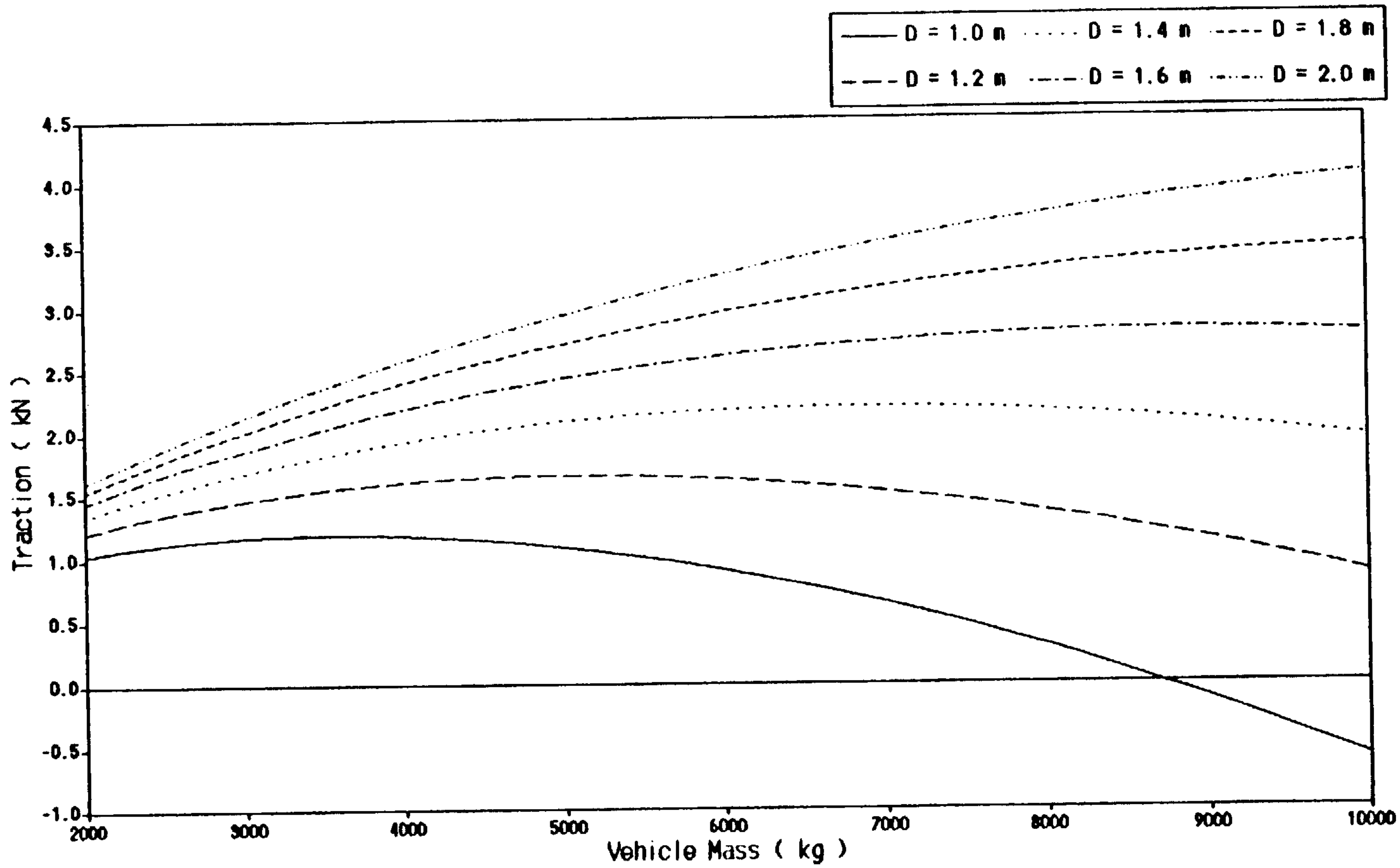

Figure C2: NET TRACTION FORCE OF DRIVE UNIT AS A FUNCTION OF VEHICLE MASS AND WHEEL DIAMETER - SOFT SOIL

Results for firm soil

DIAM	MASS	THETA	AREA	SHEAR	FRICTION	RESIST	TRACTION
1.6	2000.	.4377	.8403	.840	2.925	.667	3.097
1.6	2200.	.4520	.8678	.868	3.217	.758	3.327
1.6	2400.	.4655	.8937	.894	3.509	.851	3.552
1.6	2600.	.4783	.9182	.918	3.802	.947	3.773
1.6	2800.	.4904	.9416	.942	4.094	1.045	3.991
1.6	3000.	.5020	.9639	.964	4.387	1.146	4.205
1.6	3200.	.5131	.9852	.985	4.679	1.249	4.416
1.6	3400.	.5238	1.0057	1.006	4.972	1.354	4.623
1.6	3600.	.5341	1.0254	1.025	5.264	1.461	4.828
1.6	3800.	.5440	1.0444	1.044	5.557	1.570	5.031
1.6	4000.	.5535	1.0628	1.063	5.849	1.682	5.230
1.6	4200.	.5628	1.0806	1.081	6.141	1.795	5.427
1.6	4400.	.5718	1.0979	1.098	6.434	1.909	5.622
1.6	4600.	.5805	1.1146	1.115	6.726	2.026	5.815
1.6	4800.	.5890	1.1309	1.131	7.019	2.144	6.005
1.6	5000.	.5973	1.1468	1.147	7.311	2.264	6.194
1.6	5200.	.6054	1.1623	1.162	7.604	2.386	6.380
1.6	5400.	.6132	1.1774	1.177	7.896	2.509	6.565
1.6	5600.	.6209	1.1921	1.192	8.189	2.634	6.747
1.6	5800.	.6284	1.2065	1.207	8.481	2.760	6.928
1.6	6000.	.6357	1.2206	1.221	8.774	2.887	7.107
1.6	6200.	.6429	1.2344	1.234	9.066	3.016	7.284
1.6	6400.	.6500	1.2479	1.248	9.358	3.147	7.460
1.6	6600.	.6569	1.2612	1.261	9.651	3.279	7.633
1.6	6800.	.6636	1.2742	1.274	9.943	3.412	7.806
1.6	7000.	.6703	1.2869	1.287	10.236	3.546	7.977
1.6	7200.	.6768	1.2995	1.299	10.528	3.682	8.146
1.6	7400.	.6832	1.3118	1.312	10.821	3.819	8.314
1.6	7600.	.6895	1.3239	1.324	11.113	3.957	8.480
1.6	7800.	.6957	1.3358	1.336	11.406	4.097	8.645
1.6	8000.	.7018	1.3475	1.347	11.698	4.237	8.808
1.6	8200.	.7078	1.3590	1.359	11.991	4.379	8.970
1.6	8400.	.7137	1.3704	1.370	12.283	4.522	9.131
1.6	8600.	.7195	1.3815	1.382	12.575	4.666	9.291
1.6	8800.	.7253	1.3926	1.393	12.868	4.811	9.449
1.6	9000.	.7309	1.4034	1.403	13.160	4.958	9.606
1.6	9200.	.7365	1.4141	1.414	13.453	5.105	9.762
1.6	9400.	.7420	1.4247	1.425	13.745	5.254	9.916
1.6	9600.	.7475	1.4351	1.435	14.038	5.403	10.069
1.6	9800.	.7528	1.4454	1.445	14.330	5.554	10.222
1.6	10000.	.7581	1.4556	1.456	14.623	5.706	10.373

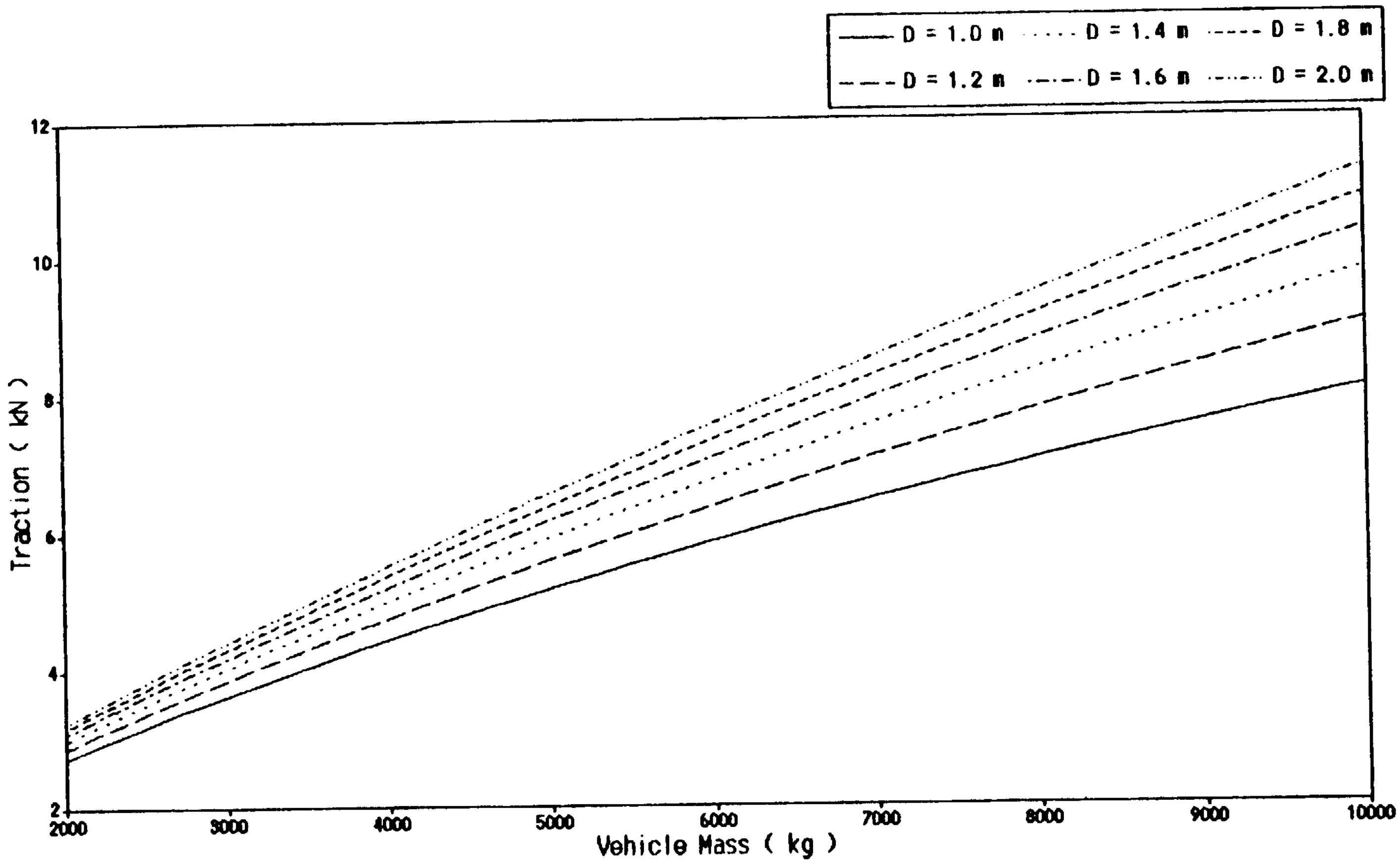

Figure C3: NET TRACTION FORCE OF DRIVE UNIT AS A FUNCTION OF VEHICLE MASS AND WHEEL DIAMETER - FIRM SOIL

<u>Computer Program for Parametric Sensitivity Analysis</u>

```
      PROGRAM PARMVARY
C
      REAL MASS, MRED, KC, KPHI
      PARAMETER( GRAV = 1.624, WHLS = 4.0 )
C
      OPEN( UNIT=6, FILE='PARMVARY.GRF', STATUS='NEW' )
C
      MASS = 5000.0
      MRED = MASS / WHLS
      WID = 0.30
      CC = 600.0
      KC = 12353.0
      PHI = 0.68068
      KPHI = 8346.0
C
      WRITE( 6, 101 ) MASS
 101  FORMAT( ///, 1X, 'MASS = ', F7.0, ' KG' )
      WRITE( 6, 102 ) WID
 102  FORMAT( 1X, 'WIDTH = ', F4.2, ' M' )
      WRITE( 6, 103 ) CC
 103  FORMAT( 1X, 'COHESION = ', F6.1, ' N/M^2' )
      WRITE( 6, 104 ) KC
 104  FORMAT( 1X, 'KC = ', F7.1, ' N/M^2' )
      WRITE( 6, 105 ) PHI
 105  FORMAT( 1X, 'PHI = ', F7.5, ' RADIANS' )
      WRITE( 6, 106 ) KPHI
 106  FORMAT( 1X, 'KPHI = ', F7.1, ' N/M^3', / )
      WRITE( 6, 107 )
 107  FORMAT( 1X, 'DIAM', T10, 'NORM', T20, 'THETA', T30, 'AREA',
     &        T40, 'SHEAR', T50, 'FRICTION', T60, 'RESIST', T70,
     &        'TRACTION', / )
C
      DO 10 DIA = 1.0, 2.05, 0.05
         RAD = DIA / 2.0
         SCL = SCALE( 1.0, 2.0, DIA )
C
         CALL FIGURE( MASS, MRED, WID, DIA, RAD, CC, KC, PHI, KPHI,
     &                THETA, AREA, SHEAR, FRIC, RESIST, TRACTION )
C
         WRITE( 6, 108 ) DIA, SCL, THETA, AREA, SHEAR, FRIC,
     &                   RESIST, TRACTION
 108     FORMAT( 1X, F4.2, T10, F6.4, T20, F6.4, T30, F6.4, T40,
     &           F7.3, T50, F7.3, T60, F7.3, T70, F7.3 )
C
 10   CONTINUE
```

```
      DIA = 1.5
      RAD = 0.75
C
C
      WRITE( 6, 101 ) MASS
      WRITE( 6, 109 ) DIA
 109  FORMAT( 1X, 'DIAMETER = ', F4.2, ' M' )
      WRITE( 6, 103 ) CC
      WRITE( 6, 104 ) KC
      WRITE( 6, 105 ) PHI
      WRITE( 6, 106 ) KPHI
      WRITE( 6, 111 )
 111  FORMAT( 1X, 'WID', T10, 'NORM', T20, 'THETA', T30, 'AREA',
     &       T40, 'SHEAR', T50, 'FRICTION', T60, 'RESIST', T70,
     &       'TRACTION', / )
C
      DO 20 WID = 0.10, 0.525, 0.025
         SCL = SCALE( 0.10, 0.50, WID )
C
         CALL FIGURE( MASS, MRED, WID, DIA, RAD, CC, KC, PHI, KPHI,
     &                THETA, AREA, SHEAR, FRIC, RESIST, TRACTION )
C
         WRITE( 6, 112 ) WID, SCL, THETA, AREA, SHEAR, FRIC,
     &                   RESIST, TRACTION
 112     FORMAT( 1X, F5.3, T10, F6.4, T20, F6.4, T30, F6.4, T40,
     &              F7.3, T50, F7.3, T60, F7.3, T70, F7.3 )
C
 20   CONTINUE
      WID = 0.30
C
C
      WRITE( 6, 113 ) DIA
 113  FORMAT( ///, 1X, 'DIAMETER = ', F4.2, ' M' )
      WRITE( 6, 102 ) WID
      WRITE( 6, 103 ) CC
      WRITE( 6, 104 ) KC
      WRITE( 6, 105 ) PHI
      WRITE( 6, 106 ) KPHI
      WRITE( 6, 114 )
 114  FORMAT( 1X, 'MASS', T10, 'NORM', T20, 'THETA', T30, 'AREA',
     &       T40, 'SHEAR', T50, 'FRICTION', T60, 'RESIST', T70,
     &       'TRACTION', / )
C
      DO 30 MASS = 2000.0, 8200.0, 200.0
         MRED = MASS / WHLS
         SCL = SCALE( 2000.0, 8000.0, MASS )
C
         CALL FIGURE( MASS, MRED, WID, DIA, RAD, CC, KC, PHI, KPHI,
     &                THETA, AREA, SHEAR, FRIC, RESIST, TRACTION )
C
```

```
      WRITE( 6, 115 ) MASS, SCL, THETA, AREA, SHEAR, FRIC,
   &                  RESIST, TRACTION
 115      FORMAT( 1X, F6.1, T10, F6.4, T20, F6.4, T30, F6.4, T40,
   &              F7.3, T50, F7.3, T60, F7.3, T70, F7.3 )
C
 30   CONTINUE
      MASS = 5000.0
      MRED = MASS / WHLS
C
C
      WRITE( 6, 101 ) MASS
      WRITE( 6, 109 ) DIA
      WRITE( 6, 102 ) WID
      WRITE( 6, 105 ) PHI
      WRITE( 6, 106 ) KPHI
      WRITE( 6, 116 )
 116  FORMAT( 1X, 'CC', T10, 'NORM', T20, 'THETA', T30, 'AREA',
   &        T40, 'SHEAR', T50, 'FRICTION', T60, 'RESIST', T70,
   &        'TRACTION', / )
C
      DO 40 CC = 200.0, 1050.0, 50.0
         KC = CC / 170.0 * 3500.0
         SCL = SCALE( 200.0, 1000.0, CC )
C
         CALL FIGURE( MASS, MRED, WID, DIA, RAD, CC, KC, PHI, KPHI,
   &                  THETA, AREA, SHEAR, FRIC, RESIST, TRACTION )
C
         WRITE( 6, 117 ) CC, SCL, THETA, AREA, SHEAR, FRIC,
   &                  RESIST, TRACTION
 117      FORMAT( 1X, F6.1, T10, F6.4, T20, F6.4, T30, F6.4, T40,
   &              F7.3, T50, F7.3, T60, F7.3, T70, F7.3 )
C
 40   CONTINUE
      CC = 600.0
      KC = 12353.0
C
C
      WRITE( 6, 101 ) MASS
      WRITE( 6, 109 ) DIA
      WRITE( 6, 102 ) WID
      WRITE( 6, 103 ) CC
      WRITE( 6, 104 ) KC
      WRITE( 6, 118 )
 118  FORMAT( /, 1X, 'PHI', T10, 'NORM', T20, 'THETA', T30, 'AREA',
   &        T40, 'SHEAR', T50, 'FRICTION', T60, 'RESIST', T70,
   &        'TRACTION', / )
C
```

```
      DO 50 PHI = 0.610865, 0.7417649, 0.0087266
         PHID = PHI * 180.0 / 3.14159
         KPHI = LOG( PHID ) / LOG( 35.0 ) * 8100.0
         SCL = SCALE( 0.61087, 0.733038, PHI )
C
         CALL FIGURE( MASS, MRED, WID, DIA, RAD, CC, KC, PHI, KPHI,
     &                THETA, AREA, SHEAR, FRIC, RESIST, TRACTION )
C
         WRITE( 6, 119 ) PHI, SCL, THETA, AREA, SHEAR, FRIC,
     &                   RESIST, TRACTION
 119     FORMAT( 1X, F6.4, T10, F6.4, T20, F6.4, T30, F6.4, T40,
     &             F7.3, T50, F7.3, T60, F7.3, T70, F7.3 )
C
 50   CONTINUE
C
C
      STOP
      END
C
C
C
      SUBROUTINE FIGURE( MASS, MRED, WID, DIA, RAD, CC, KC, PHI, KPHI,
     &                   THETA, AREA, SHEAR, FRIC, RESIST, TRACTION )
C
      REAL MASS, MRED, KC, KPHI
      PARAMETER( GRAV = 1.624, WHLS = 4.0 )
C
      CALL BISECT( THETA, MRED, WID, RAD, KC, KPHI )
C
      AREA = 2.0 * WID * RAD * THETA * WHLS
      SHEAR = AREA * CC / 1000.0
      FRIC = MASS * GRAV * TAN( PHI ) / 1000.0
      RESIST = WHLS * ( ( 3.0 * MRED * GRAV / SQRT( DIA ) ) **
     &         ( 4.0 / 3.0 ) / ( 2.0 ** ( 7.0 / 3.0 ) *
     &         ( KC + WID * KPHI ) ** ( 1.0 / 3.0 ) ) ) / 1000.0
      TRACTION = SHEAR + FRIC - RESIST
C
      RETURN
      END
C
C
C
      SUBROUTINE BISECT( XR, MRED, WID, RAD, KC, KPHI )
C
      REAL MRED, KC, KPHI
C
      XL = 0.0
      XU = 1.57
      ES = 1.0E-6
```

```
      MAXIT = 1000
      ITER = 0
      EA = 1.1 * ES
 10   IF( ( EA .GT. ES ) .AND. ( ITER .LT. MAXIT ) ) THEN
         XR = ( XL + XU ) / 2.0
         ITER = ITER + 1
C
         IF( XL + XU .NE. 0.0 ) THEN
            EA = DABS( ( XU - XL ) / ( XU + XL ) ) * 100.0
         ENDIF
C
         TEST = ANGLE( XL, MRED, WID, RAD, KC, KPHI )
     &        * ANGLE( XR, MRED, WID, RAD, KC, KPHI )
C
         IF( TEST = 0.0 ) THEN
            EA = 0.0
         ELSE
            IF( TEST .LT. 0.0 ) THEN
               XU = XR
            ELSE
               XL = XR
            ENDIF
         ENDIF
C
      GO TO 10
C
      ENDIF
C
      RETURN
      END
C
C
C
      FUNCTION ANGLE( X, MRED, WID, RAD, KC, KPHI )
C
      REAL MRED, KC, KPHI
      PARAMETER( GRAV = 1.624 )
C
      ANGLE = X * ( SIN( X ) / X - COS( X ) ) - ( MRED * GRAV )
     &        / ( 2.0 * WID * RAD ** 2.0 * ( KC / WID + KPHI ) )
C
      END
C
C
C
      FUNCTION SCALE( XLO, XHI, X )
C
      SCALE = ( X - XLO ) / ( XHI - XLO )
C
      END
```

Results of Parametric Analysis

MASS = 5000. KG
WIDTH = .30 M
COHESION = 600.0 N/M^2
KC = 12353.0 N/M^2
PHI = .68068 RADIANS
KPHI = 8346.0 N/M^3

DIAM	NORM	THETA	AREA	SHEAR	FRICTION	RESIST	TRACTION
1.00	.0000	.9659	1.1591	.695	6.575	3.591	3.680
1.05	.0500	.9330	1.1756	.705	6.575	3.476	3.805
1.10	.1000	.9028	1.1917	.715	6.575	3.370	3.921
1.15	.1500	.8749	1.2074	.724	6.575	3.271	4.029
1.20	.2000	.8492	1.2228	.734	6.575	3.180	4.129
1.25	.2500	.8253	1.2379	.743	6.575	3.094	4.224
1.30	.3000	.8030	1.2526	.752	6.575	3.014	4.313
1.35	.3500	.7821	1.2671	.760	6.575	2.940	4.396
1.40	.4000	.7626	1.2812	.769	6.575	2.869	4.475
1.45	.4500	.7443	1.2951	.777	6.575	2.803	4.550
1.50	.5000	.7270	1.3087	.785	6.575	2.740	4.620
1.55	.5500	.7108	1.3220	.793	6.575	2.681	4.688
1.60	.6000	.6954	1.3351	.801	6.575	2.625	4.752
1.65	.6500	.6808	1.3479	.809	6.575	2.571	4.813
1.70	.7000	.6669	1.3606	.816	6.575	2.521	4.871
1.75	.7500	.6538	1.3730	.824	6.575	2.473	4.927
1.80	.8000	.6413	1.3852	.831	6.575	2.427	4.980
1.85	.8500	.6294	1.3972	.838	6.575	2.383	5.031
1.90	.9000	.6180	1.4090	.845	6.575	2.341	5.080
1.95	.9500	.6071	1.4206	.852	6.575	2.300	5.127
2.00	1.0000	.5967	1.4320	.859	6.575	2.262	5.173

MASS = 5000. KG
DIAMETER = 1.50 M
COHESION = 600.0 N/M^2
KC = 12353.0 N/M^2
PHI = .68068 RADIANS
KPHI = 8346.0 N/M^3

WID	NORM	THETA	AREA	SHEAR	FRICTION	RESIST	TRACTION
.100	.0000	.7577	.4546	.273	6.575	2.851	3.997
.125	.0625	.7536	.5652	.339	6.575	2.836	4.078
.150	.1250	.7495	.6746	.405	6.575	2.822	4.158
.175	.1875	.7456	.7829	.470	6.575	2.807	4.238
.200	.2500	.7417	.8901	.534	6.575	2.793	4.316
.225	.3125	.7379	.9962	.598	6.575	2.780	4.393
.250	.3750	.7342	1.1014	.661	6.575	2.766	4.470
.275	.4375	.7306	1.2055	.723	6.575	2.753	4.546
.300	.5000	.7270	1.3087	.785	6.575	2.740	4.620
.325	.5625	.7235	1.4109	.847	6.575	2.727	4.695
.350	.6250	.7201	1.5122	.907	6.575	2.715	4.768
.375	.6875	.7167	1.6127	.968	6.575	2.703	4.840
.400	.7500	.7134	1.7123	1.027	6.575	2.691	4.912
.425	.8125	.7102	1.8110	1.087	6.575	2.679	4.983
.450	.8750	.7070	1.9089	1.145	6.575	2.667	5.054
.475	.9375	.7039	2.0060	1.204	6.575	2.656	5.123
.500	1.0000	.7008	2.1024	1.261	6.575	2.645	5.192

DIAMETER = 1.50 M
WIDTH = .30 M
COHESION = 600.0 N/M^2
KC = 12353.0 N/M^2
PHI = .68068 RADIANS
KPHI = 8346.0 N/M^3

MASS	NORM	THETA	AREA	SHEAR	FRICTION	RESIST	TRACTION
2000.0	.0000	.5313	.9563	.574	2.630	.808	2.396
2200.0	.0333	.5488	.9878	.593	2.893	.917	2.569
2400.0	.0667	.5652	1.0174	.610	3.156	1.030	2.737
2600.0	.1000	.5809	1.0456	.627	3.419	1.146	2.901
2800.0	.1333	.5958	1.0724	.643	3.682	1.265	3.061
3000.0	.1667	.6100	1.0980	.659	3.945	1.387	3.217
3200.0	.2000	.6236	1.1225	.673	4.208	1.511	3.370
3400.0	.2333	.6367	1.1460	.688	4.471	1.639	3.520
3600.0	.2667	.6493	1.1687	.701	4.734	1.768	3.667
3800.0	.3000	.6614	1.1906	.714	4.997	1.900	3.811
4000.0	.3333	.6732	1.2118	.727	5.260	2.035	3.952

4200.0	.3667	.6846	1.2323	.739	5.523	2.172	4.091
4400.0	.4000	.6957	1.2522	.751	5.786	2.311	4.227
4600.0	.4333	.7064	1.2715	.763	6.049	2.452	4.360
4800.0	.4667	.7169	1.2903	.774	6.312	2.595	4.492
5000.0	.5000	.7270	1.3087	.785	6.575	2.740	4.620
5200.0	.5333	.7370	1.3265	.796	6.838	2.887	4.747
5400.0	.5667	.7467	1.3440	.806	7.102	3.036	4.872
5600.0	.6000	.7561	1.3610	.817	7.365	3.187	4.994
5800.0	.6333	.7654	1.3777	.827	7.628	3.340	5.114
6000.0	.6667	.7745	1.3940	.836	7.891	3.494	5.233
6200.0	.7000	.7833	1.4100	.846	8.154	3.650	5.349
6400.0	.7333	.7920	1.4257	.855	8.417	3.808	5.464
6600.0	.7667	.8006	1.4410	.865	8.680	3.968	5.577
6800.0	.8000	.8090	1.4561	.874	8.943	4.129	5.687
7000.0	.8333	.8172	1.4709	.883	9.206	4.292	5.797
7200.0	.8667	.8253	1.4855	.891	9.469	4.456	5.904
7400.0	.9000	.8332	1.4998	.900	9.732	4.622	6.010
7600.0	.9333	.8410	1.5138	.908	9.995	4.789	6.114
7800.0	.9667	.8487	1.5277	.917	10.258	4.958	6.217
8000.0	1.0000	.8563	1.5413	.925	10.521	5.128	6.318

MASS = 5000. KG
DIAMETER = 1.50 M
WIDTH = .30 M
PHI = .68068 RADIANS
KPHI = 8346.0 N/M^3

CC	NORM	THETA	AREA	SHEAR	FRICTION	RESIST	TRACTION
200.0	.0000	.9649	1.7369	.347	6.575	3.587	3.336
250.0	.0625	.9167	1.6501	.413	6.575	3.419	3.569
300.0	.1250	.8768	1.5782	.473	6.575	3.278	3.771
350.0	.1875	.8430	1.5174	.531	6.575	3.158	3.949
400.0	.2500	.8138	1.4648	.586	6.575	3.053	4.108
450.0	.3125	.7882	1.4188	.638	6.575	2.962	4.252
500.0	.3750	.7656	1.3781	.689	6.575	2.880	4.385
550.0	.4375	.7453	1.3416	.738	6.575	2.807	4.507
600.0	.5000	.7270	1.3087	.785	6.575	2.740	4.620
650.0	.5625	.7104	1.2787	.831	6.575	2.680	4.727
700.0	.6250	.6952	1.2513	.876	6.575	2.624	4.827
750.0	.6875	.6812	1.2261	.920	6.575	2.573	4.922

MASS = 5000. KG
DIAMETER = 1.50 M
WIDTH = .30 M
PHI = .68068 RADIANS
KPHI = 8346.0 N/M^3

CC	NORM	THETA	AREA	SHEAR	FRICTION	RESIST	TRACTION
200.0	.0000	.9649	1.7369	.347	6.575	3.587	3.336
250.0	.0625	.9167	1.6501	.413	6.575	3.419	3.569
300.0	.1250	.8768	1.5782	.473	6.575	3.278	3.771
350.0	.1875	.8430	1.5174	.531	6.575	3.158	3.949
400.0	.2500	.8138	1.4648	.586	6.575	3.053	4.108
450.0	.3125	.7882	1.4188	.638	6.575	2.962	4.252
500.0	.3750	.7656	1.3781	.689	6.575	2.880	4.385
550.0	.4375	.7453	1.3416	.738	6.575	2.807	4.507
600.0	.5000	.7270	1.3087	.785	6.575	2.740	4.620
650.0	.5625	.7104	1.2787	.831	6.575	2.680	4.727
700.0	.6250	.6952	1.2513	.876	6.575	2.624	4.827
750.0	.6875	.6812	1.2261	.920	6.575	2.573	4.922
800.0	.7500	.6682	1.2028	.962	6.575	2.526	5.012
850.0	.8125	.6562	1.1812	1.004	6.575	2.482	5.098
900.0	.8750	.6450	1.1611	1.045	6.575	2.440	5.180
950.0	.9375	.6345	1.1422	1.085	6.575	2.402	5.259
1000.0	1.0000	.6247	1.1245	1.124	6.575	2.366	5.334

MASS = 5000. KG
DIAMETER = 1.50 M
WIDTH = .30 M
COHESION = 600.0 N/M^2
KC = 12353.0 N/M^2

PHI	NORM	THETA	AREA	SHEAR	FRICTION	RESIST	TRACTION
.6109	0.0000	.7283	1.3109	.787	5.686	2.745	3.728
.6196	.0714	.7281	1.3106	.786	5.792	2.744	3.834
.6283	.1428	.7280	1.3103	.786	5.900	2.744	3.942
.6370	.2143	.7278	1.3100	.786	6.008	2.743	4.052
.6458	.2857	.7276	1.3098	.786	6.119	2.742	4.162
.6545	.3571	.7275	1.3095	.786	6.231	2.742	4.275
.6632	.4285	.7273	1.3092	.786	6.344	2.741	4.388
.6720	.5000	.7272	1.3089	.785	6.459	2.741	4.504
.6807	.5714	.7270	1.3087	.785	6.575	2.740	4.620
.6894	.6428	.7269	1.3084	.785	6.694	2.740	4.739
.6981	.7143	.7267	1.3081	.785	6.813	2.739	4.859
.7069	.7857	.7266	1.3079	.785	6.935	2.739	4.981
.7156	.8571	.7265	1.3076	.785	7.059	2.738	5.105
.7243	.9286	.7263	1.3074	.784	7.184	2.738	5.231
.7330	1.0000	.7262	1.3071	.784	7.311	2.737	5.358

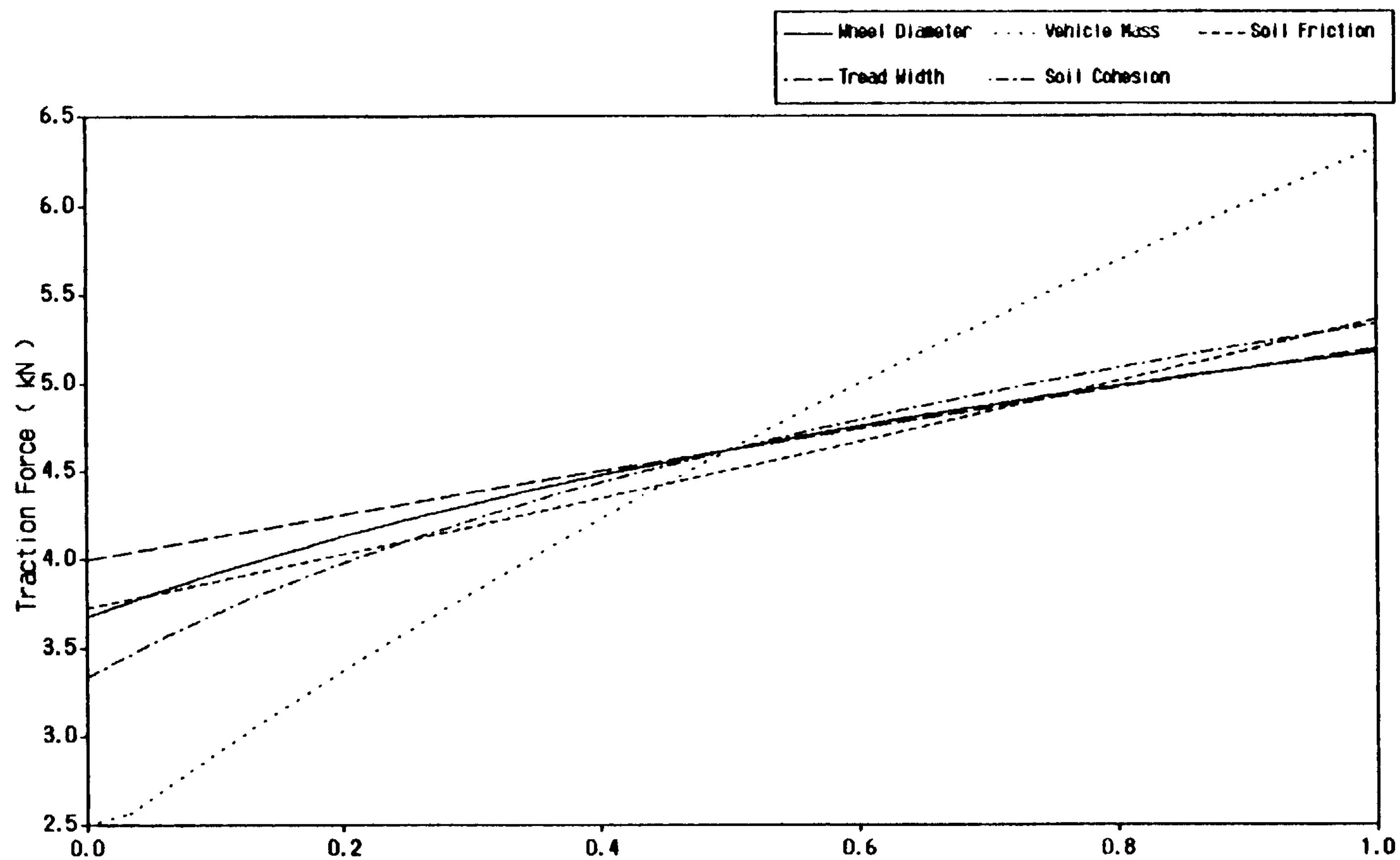

Figure C4: PARAMETRIC SENSITIVITY OF TRACTION

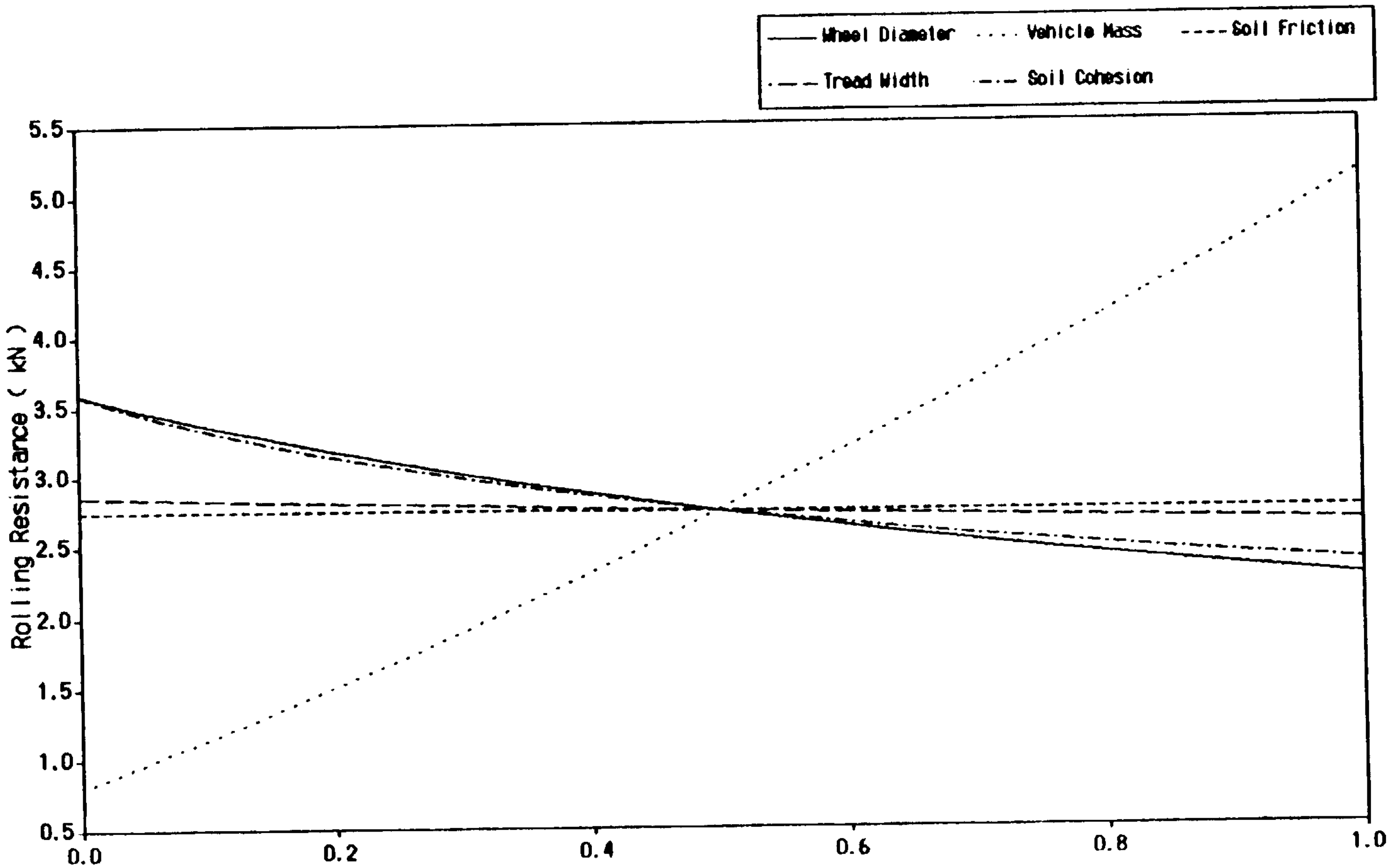

Figure C5: PARAMETRIC SENSITIVITY OF ROLLING RESISTANCE

Conclusions and Recommendations

The traction developed from the shear forces acting on the wheel contact area is a minor factor due to the low cohesion of the lunar soil. The shear contribution to developed traction is less than 4% for soft soil and less than 11% for firm soil.

The frictional traction force depends only on the total mass of the vehicle and soil friction angle. This force is not affected by the size or number of wheels.

The most important factor in the net traction force is rolling resistance. The resistance decreases with greater wheel diameter, but is affected more by increasing overall mass:

$$R_C \, \alpha \, \frac{M^{4/3}}{R^{2/3}} \qquad \text{(C12)}$$

The effects of rolling resistance are greater in soft soils because of the greater amount of wheel sinkage. As can be seen in Figure C2, as the vehicle mass increases, the net traction reaches a maximum and then decreases as rolling resistance becomes proportionally greater.

The parametric analysis shows the relative effects of vehicle and wheel characteristics, which will allow optimization of traction performance. Increasing vehicle mass results in the greatest increase in net traction. However, this also causes the greatest increase in rolling resistance. Although the increase in traction outweighs the increase in resistance, vehicle propulsion becomes less efficient as the mass increases. A higher proportion of energy is used to overcome rolling resistance, and less is used to perform useful work.

Increasing the wheel diameter increases developed traction and reduces rolling resistance. The drive unit should use the largest wheel that is practical, considering space for transport, turning radius, wheel mass, and other factors.

Increasing the tread width results in a slight increase in traction due to the increased surface area and also yields a slight decrease in rolling resistance. However, as mentioned earlier, a relatively narrow tread has several advantages which outweigh these traction and resistance benefits.

The drive unit will have the worst traction performance in loose soil. Developed traction will be relatively low, and rolling resistance will be high due to wheel sinkage. This performance is typical of the lunar dust layer. However, the cutting force needed to move the dust will also be relatively low, and the drive unit should have adequate power. As cuts are made into deeper, firmer soil, the increase in traction and decreasing rolling resistance will provide more than enough of an increase in net traction for the grader to excavate the firmer soil.

APPENDIX C: REFERENCES

1. M. G. Bekker, Off-the-Road Locomotion, (The University of Michigan Press, Ann Arbor: 1960), pp. 25-28.

2. op.cit., p. 28.

3. op. cit., p. 32.

4. op. cit., p. 53.

5. op. cit., pp. 63-65.

6. op. cit., p. 68.

7. Lunar Base Launch and Landing Facility Conceptual Design, (Eagle Engineering, Webster, TX: 1988), p. 2.

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APPENDIX D

Cutting Force, Energy, and Power Calculations

APPENDIX D

Cutting Force, Energy, and Power Calculations

Cutting Force

Four separate equations were used to calculate the force to cut the lunar soil. These equations are shown below, along with a definition and value for each variable.

Equation 1. From Machines for Moving the Earth, A.N. Zelenin, Russian Translation Series 33, 1987, p.157.

$$P_k = 10\, C\, h^{1.35} \sin^3(\alpha_b)\, (1+2.6L)(1+0.0075\alpha)(1+0.03S)V\mu + \xi FK + gq\gamma \tan(\rho) + G_i$$

Where:

P_k = cutting force (N)

C = coefficient of soil cohesion (N/cm^2)

h = depth of cut (cm)

α_b = angle blade makes to direction of motion

L = length of grader blade (cm)

α = cutting angle (radians)

S = grader blade thickness (cm)

V = geometric coefficient depending on the wedge angle

(varies from 0.81 to 1.05)

μ = geometric coefficient depending on the cutting conditions

(varies from 0.42 to 4.5)

K = specific resistance to cutting, depends on geometry of cutting tool

(varies from 0.7 to 20)

F = cross-sectional area of the soil chip = h•S (cm^2)

ξ = coefficient which accounts for the increase in K with h

(varies from 1.0 to 2.4)

g = gravitational acceleration (m/s^2)

q = volume of soil in front of blade (m^3) = $0.5\ LH^2\cos(\phi)$

where H=height of blade (m), ϕ=internal angle of friction of the soil

γ = soil density (kg/m^3)

ρ = external angle of friction of the soil (radians)

G_i = weight of the grader blade (N)

The following values were assigned to each of the above variables as an initial estimate of the grader forces: C = 0.142 N/cm^2, h = 3 cm, α_b = 35°, L = 300 cm, α = 27.5°, S = 10 cm, V = 1.05, μ = 4.5, K = 20, ξ = 2.4, g = 1.635 m/s^2, H=50 cm, ϕ = 35°, γ = 2.0 kg/m^3, ρ = 35°. The resulting grader force was 5.96 kN.

Equation 2. From New Methods for Calculating Resistance to Cutting of Soil, V.I. Balovnev, Rosvuzizdat Publishers, 1963, p. 35.

$$P = \left[\sin(\alpha_b) + \tan(\delta) \sqrt{\frac{1}{\sin^2(\alpha_c)} - \sin^2(\alpha_b)} \right] A_1 L h \left[\frac{\gamma h}{2} + c \cot(\rho) + Po \right]$$

Where:

P = horizontal component of the resistance to cutting (metric tons)

α_b = angle blade makes to the direction of motion

δ = angle of inclination of the wall

α_c = cutting angle

A_1 = coefficient determined analytically

$$A_1 = \frac{1 - \sin(\rho)\cos(2\alpha_c)}{1 - \sin(\rho)}$$

L = blade length (m)

h = depth of cut (m)

γ = specific weight of soil (metric tons/m^3)

c = coefficient of soil cohesion (metric tons/m^2)

ρ = angle of internal friction

Po = external pressure on the surface (metric tons/m^2)

The following values were assigned to each of the above variables as an initial estimate of the grader forces: $\alpha_b = 35^\circ$, $\delta = 49^\circ$, $\alpha_c = 27.5^\circ$, L = 3 m, h = 3 cm, $\gamma = 2.0$ metric tons/m^3, c = 0.145 metric tons/m^2, $\rho = 35^\circ$, Po = 0.0 metric tons/m^2. The resulting grader force was 1.0 kN.

Equation 3. From Digging of Soils by Earthmovers with Powered Parts, V.K. Rudnev, Russian Translation Series 32, 1985, pp. 40.

$$P = B\frac{\sin(\alpha)\sin(\gamma)+\tan(\varphi)\cos^2(\gamma)+\tan(\varphi)\sin^2(\gamma)\cos(\alpha)}{\sin(\alpha+\psi+\rho)+\tan(\varphi)\sin(\gamma)\cos(\alpha+\psi+\rho)}$$
$$\times\left[0.5\cos(\rho)\cos(\psi)\delta_r H^2 + \sin(\psi+\rho)(\cot(\psi)+\cot(\alpha))\delta_r hH + 0.5\cosh\left(\frac{\cos(\rho)}{\sin(\psi)}\right)\right]$$

Where:

P = cutting force (metric tons)

B = blade length (m)

α = cutting angle (radians)

γ = angle blade makes to direction of motion

φ = external angle of friction of the soil (radians)

ψ = angle of displacement ($= \pi/4 - \rho/2$)

ρ = internal angle of friction of the soil (radians)

δ_R = density of loosened soil (metric tons/m^2)

H = blade height (m)

h = depth of cut (m)

The following values were assigned to each of the above variables as an initial estimate of the grader forces: $B = 3.0$ m, $\alpha = 27.5°$, $\gamma = 35°$, $\varphi = 35°$, $\rho = 35°$, $\delta_R = 2.0$ metric tons/m^2, $H = 50$ cm, $h = 3$ cm. The resulting grader force was 49.7 kN.

Equation 4. From Digging of Soils by Earthmovers with Powered Parts, V.K. Rudnev, Russian Translation Series 32, 1985, pp. 38-39.

$$P = N[\sin(\alpha)\sin(\gamma) + \tan(\varphi)\cos^2(\gamma) + \tan(\varphi)\sin^2(\gamma)\cos(\alpha)]$$

Where:

P = cutting force (metric tons)

α = cutting angle (radians)

γ = angle blade makes to direction of motion

φ = external angle of friction of the soil (radians)

N = component of soil resistance normal to the grader blade

$$N = \frac{\frac{\sin(\alpha+\psi)}{\cos(\rho)} R_1 + \sin(\psi+\rho)\, G_{ch} + 0.5\cos(\rho)T}{\sin(\alpha+\psi+\rho) + \tan(\varphi)\sin(\gamma)\cos(\alpha+\psi+\rho)}$$

T = resistance to shear

$$T = CB\frac{h}{\sin(\psi)}$$

G_{ch} = weight of the soil chip

$$G_{ch} = Brh\,\delta_R \frac{\omega\sin(\alpha+\psi)}{\sin(\psi)}$$

R_1 = weight of the displaced prism normal to the surface of the chip

$$R_1 = \frac{\sin(\psi+\rho)\cos(\rho)}{\sin(\alpha+\psi)} G_{pr}$$

G_{pr} = weight of the displaced prism

$$G_{pr} = \delta_R B \left[r - h \frac{\sin(\alpha+\rho)}{\sin(\psi)} \right]^2 \left[\frac{2\sin(\varepsilon-\rho)\sin(\varepsilon+\psi)\sin^2(\omega/2)}{\sin(\rho+\psi)} + \frac{\omega - \sin(\omega)}{2} \right]$$

$$r = \frac{H}{2\sin(\omega/2)\sin(\varepsilon)}$$

ψ = angle of displacement (= $\pi/4 - \rho/2$)

ρ = internal angle of friction of the soil (radians)

B = blade length (m)

δ_R = density of loosened soil (metric tons/m^2)

H = blade height

h = depth of cut

r = blade radius of curvature (m)

ε = angle between horizon and cord of the grader blade

$\omega = 2\,(\varepsilon - \alpha)$

C = coefficient of soil cohesion (metric tons/m^2)

The following values were assigned to each of the above variables as an initial estimate of the grader forces: $\alpha = 27.5°$, $\gamma = 35°$, $\varphi = 35°$, $\rho = 35°$, B = 3.0 m, $\delta_R = 2.0$ metric tons/m^2, H = 50 cm, h = 3 cm, $\varepsilon = 60°$, C = 0.145 metric tons/m^2. The resulting grader force was 6.08 kN.

Although the results of the equations varied, they approximately agreed for shallow depths of cut. A cutting force of 6 kN was calculated for a depth of cut of 3 cm. This cutting force is small compared to those experienced on Earth, but is expected due to the low cohesion of lunar soil, low lunar gravity, and shallow depth of cut. The results of the force analysis above are summed up in Table D.1 below. The force from Equation 3 is much larger compared to the results of the other equations because it includes the force required to move the motor grader itself, as well as the cutting force.

Table D.1: GRADING FORCE FOR A DEPTH OF CUT OF 3 CM

EQUATION	FORCE (kN)
1	5.96
2	0.96
3	49.74
4	6.08

A parametric analysis was performed on each equation to determine how sensitive the cutting force was to each variable. Only seven major factors were varied: the depth of cut, angle of bite, cutting angle, angle of friction, soil density, soil cohesion, and blade length. (See Figures D.1-D.7). The results from Equation 3 are consistently high because this force includes the force required to move the motor grader.

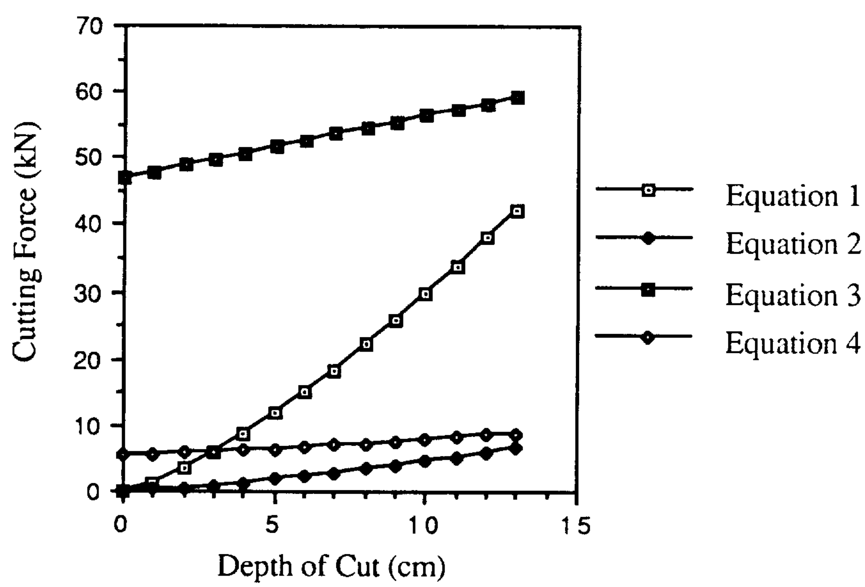

Figure D.1: CUTTING FORCE VS DEPTH OF CUT

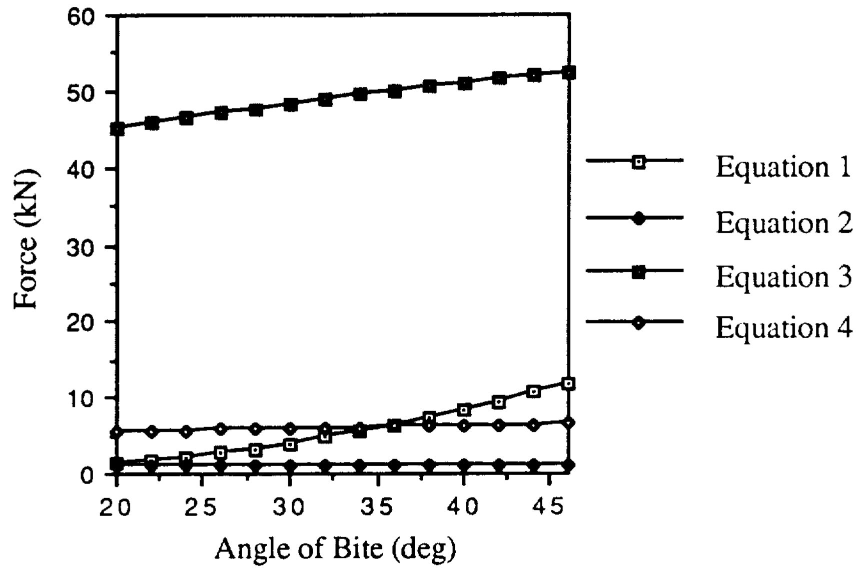

Figure D.2: CUTTING FORCE VS ANGLE OF BITE

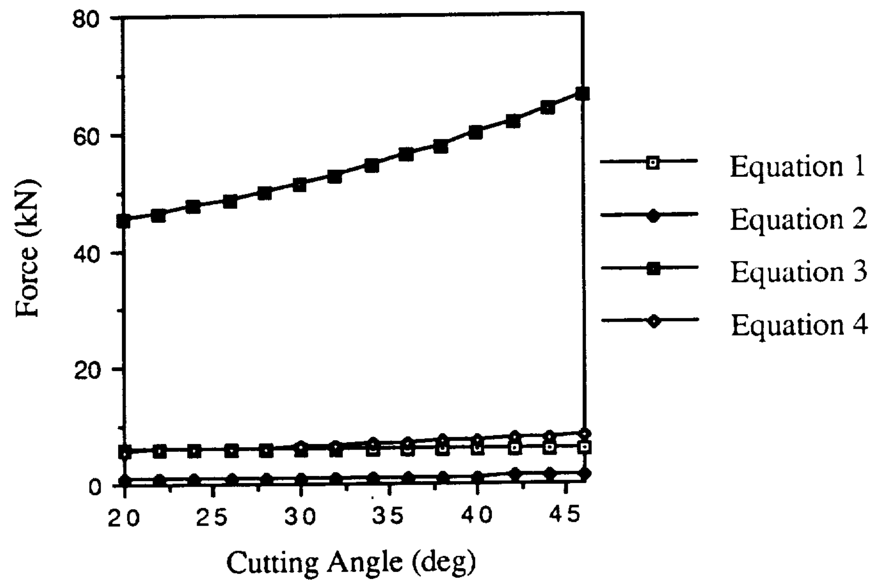

Figure D.3: CUTTING FORCE VS CUTTING ANGLE

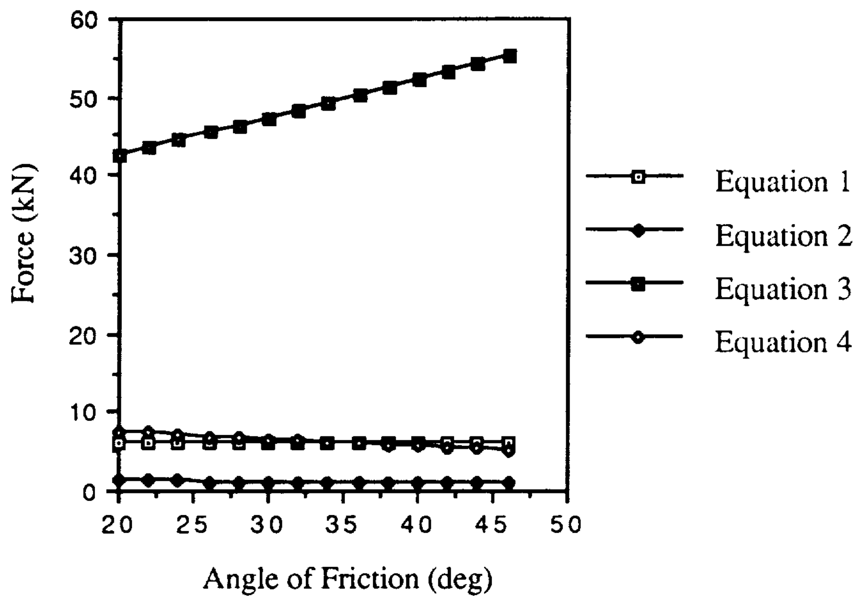

Figure D.4: CUTTING FORCE VS ANGLE OF FRICTION

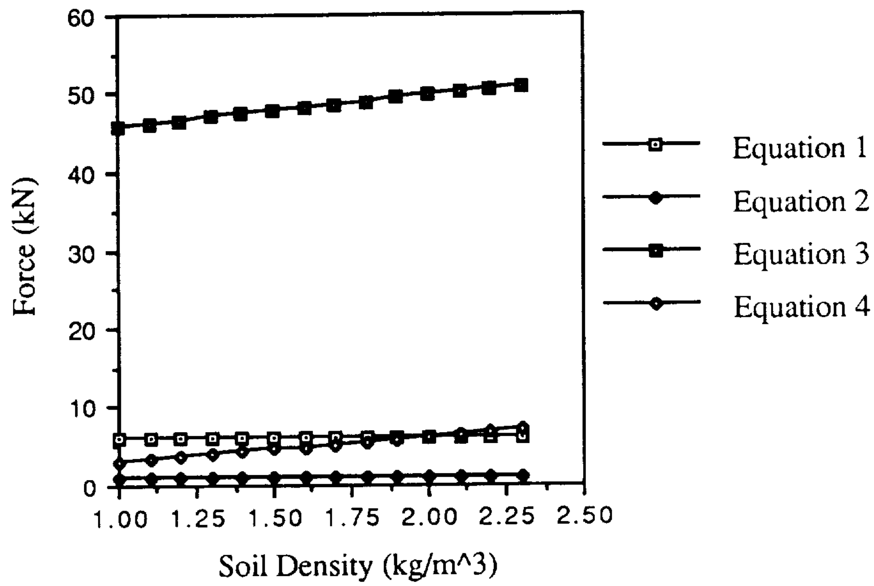

Figure D.5: CUTTING FORCE VS SOIL DENSITY

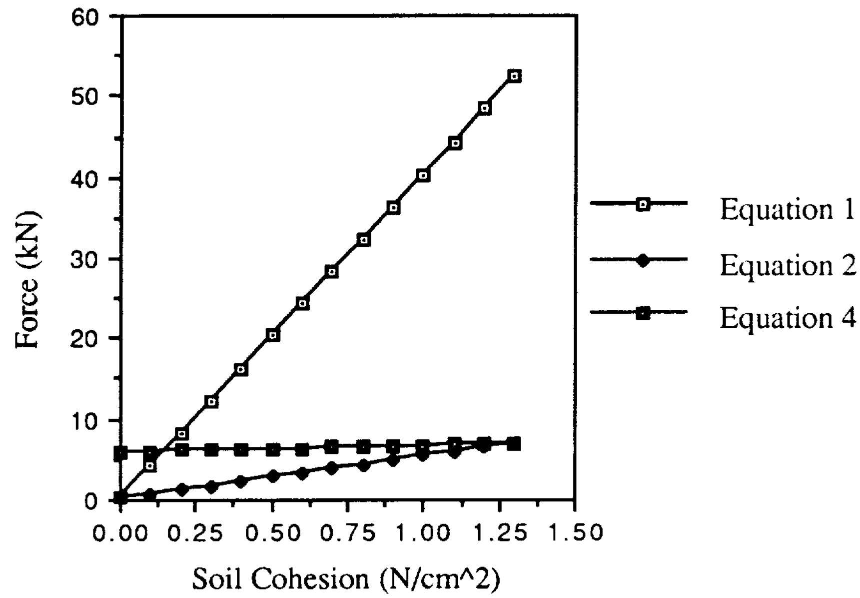

Figure D.6: CUTTING FORCE VS SOIL COHESION

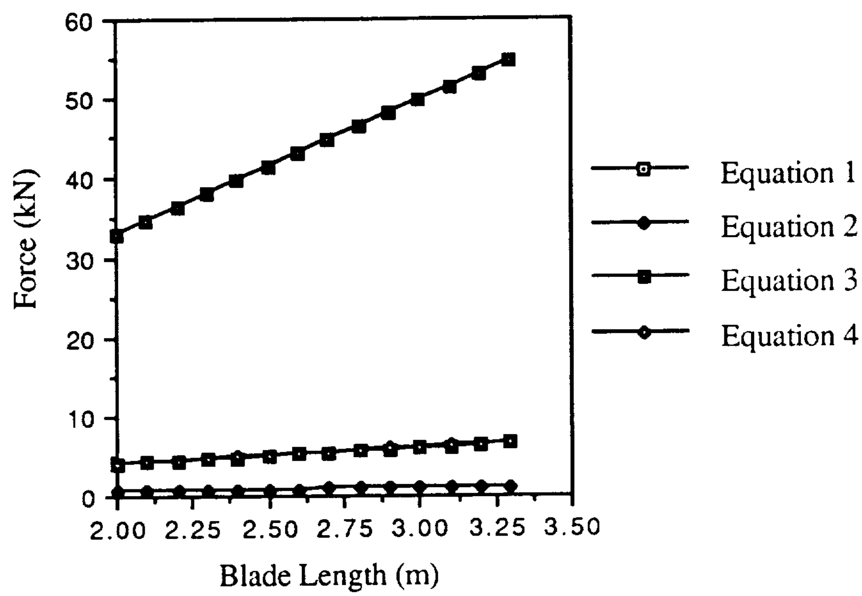

Figure D.7: CUTTING FORCE VS. BLADE LENGTH

Energy Requirments

The energy calculation required the cutting force, road width, blade length and angle of bite, depth of cut per pass, and total grading depth. The design team did not have time to perform an optimization analysis to determine how deep the grader must cut, so the team selected 10 cm (80% of maximum density) as an estimate of the optimum depth. The energy required to grade per kilometer of road was calculated based on the following assumptions:

- Road is 8 m wide
- Blade length is 3 m
- Angle of bite is 35°
- Depth of cut is 3 cm
- Total grading depth is 10 cm
- Rate of grading is 2 km/hr
- Maximum cutting force of 6 kN is constant

It was first necessary to calculate the number of passes required to grade an 8 m road a depth of 10 cm when the blade is at an angle of 35°.

$$\text{Number of Passes} = \frac{\text{Road width}}{\text{Grading width}} \times \frac{\text{Total grading depth}}{\text{Depth of Cut}}$$

$$\text{Number of Passes} = \frac{8\text{ m}}{3\sin(35)\text{ m}} \times \frac{10\text{ cm}}{3\text{ cm}} = 16\text{ passes}$$

The energy required per kilometer of road is equal to the force times the distance times the number of passes:

$$E = FdN$$

$$E = (6\text{kN})(1{,}000\ \text{m})(16) = 96{,}000\ \text{kJ}$$

Power Requirements

The power required for the grading process depends on the cutting force and grading rate. The cutting force is 6 kN and the grading rate is 2 km/hr, so the power is:

$$P = Fv$$

$$P = (6{,}000\ \text{N})(2{,}000\ \text{m/hr})(1\text{hr}/3600\ \text{sec}) = 3{,}333\ \text{W}$$

Time Requirments

The time required is equal to the energy divided by the power:

$$t = E/P$$

$$t = 96{,}000\ \text{kJ}/3{,}300\ \text{W} = 8.1\ \text{hrs}$$

APPENDIX E

Stress Calculations

APPENDIX E
Stress Calculations

In order to determine the size of the main support beam of the grader, a preliminary stress calculation was performed. The stress calculation resulted in a value for the section modulus, which was then used to determine the dimensions of a box beam. A box beam was selected because of its good torsional strength.

Assuming that the cutting force is concentrated at the center of the blade and the main support beam is approximately 2 meters off the ground, a maximum moment will be produced on the beam right above the blade. The moment is equal to the force times the moment arm.

$M = Fd$

$M = (6{,}000\text{ N})(2\text{ m}) = 12{,}000\text{ Nm}$

The allowable stress is equal to the yield stress divided by a safety factor. A safety factor of 4.0 was assumed since human life would be jeopardized if the machinery failed. The aluminum alloy, Al-Li 2090, has a yield strength of 496.4 MPa.

$\sigma_{all} = \sigma_{yield} / SF$

$\sigma_{all} = 496.4\text{ MPa} / 4.0 = 124.1\text{ MPa}$

The required section modulus for the box beam to ensure failure will not occur is equal to the maximum moment divided by the allowable stress.

$SM = M / \sigma_{all}$

$SM = 12{,}000\text{ Nm} / 124.1\text{ MPa} = 9.68 \times 10^{-5}\text{ m}^3$ (5.91 in.^3)

According to the Manual of Steel Construction published by the American Institute of Steel Construction, Inc., a 5" x 5", 1/4" thick box beam has a section modulus of 6.638 in.^{3}. This beam is sufficient to sustain the applied loading.

APPENDIX F

Grader Assembly Mass Calculations

APPENDIX F

Grader Assembly Mass Calculations

The mass of the lunar grader assembly was estimated by assuming that the dimensions of the major components were similar to the dimensions of terrestrial graders. The major components considered in the mass estimation are the grader blade, main support beam, positioning ring, ring support, blade connectors, two power screws, two wheels with titanium rims, and the front end support. Since all components were not considered and the dimensions were estimated, a 25% correction factor was added.

Blade

The body of the blade is made of Al-Li 2090, which has a density of 2.56 g/cm^3. The blade is 3 m long, 50 cm high, and 2.5 cm thick. The cutting edge of the blade is made of Ti-6 Al-4 V, which has a density of 4.42 g/cm^3. The edge is 10 cm high and 5 cm thick, with a groove for the body of the blade which is 3 cm wide and 5 cm deep.

Mass of blade body = (300 cm)(50 cm)(2.5 cm)(2.56 g/cm^3) = 96 kg

Mass of blade cutting edge = (300 cm)[(10 cm)(5 cm) - (3 cm)(5 cm)] (4.42 g/cm^3)

= 46.4 kg

Total mass of blade = 96 kg + 46 kg = 142 kg

Main Support Beam

The mass of the main support beam is equal to the area of the beam times the length times the density of the material. The beam is also made of Al-Li 2090. The area of the box beam is 29.25 cm^2. The length of the beam is 600 cm.

$$\text{Mass of beam} = (29.25\ cm^2)(600\ cm)(2.56\ g/cm^3) = 44.9\ kg$$

Blade Positioning Ring

The blade positioning ring has an outside diameter of 1.2 m, an inside diameter of 1.0 m, and a thickness of 5 cm. The ring is made of titanium.

$$\text{Mass of ring} = \pi\ [(60\ cm)^2 - (50\ cm)^2]\ (5\ cm)\ (4.42\ g/cm^3) = 76.4\ kg$$

Triangular Ring Support

The triangular ring support is made of aluminum. The base of the support is 1 m long and the lengths of the two ends are both 3 m. The supports were assumed to be 3 cm x 3 cm square.

$$\text{Mass of ring support} = (3\ cm)(3\ cm)[100\ cm + (2)(300\ cm)](2.56\ g/cm^3) = 16.3\ kg$$

Blade Connectors

The blade connectors are aluminum. They were assumed to be rectangular blocks in order to simplify the calculation. The blocks are 75 cm high, 10 cm wide, and 5 cm thick.

$$\text{Mass of connectors} = (75\text{ cm})(5\text{ cm})(10\text{ cm})(2.56\text{ g/cm}^3) = 19.2\text{ kg}$$

Power Screws

The power screws are made of titanium, are 1.5 m long, and have a diameter of 5 cm. There are two power screws.

$$\text{Mass of 2 screws} = 2(\pi)(2.5\text{ cm})^2(150\text{ cm})(4.42\text{ g/cm}^3) = 26\text{ kg}$$

Hemispherical Dome Wheels

The dome wheels have an outside radius of 50 cm and a thickness of 5 cm. The body of the wheel is made of aluminum, but the tread is made of titanium. The tread is 5 cm wide and 2.5 cm thick.

Mass of body = $(2)(1/2)(4/3)(\pi)[(50\ cm)^3 - (49.5\ cm)^3](2.56\ g/cm^3) = 39.8\ kg$

Mass of rims = $(2)(\pi/4)[(102.5\ cm)^2 - (1\ cm)^2](5\ cm)(4.42\ g/cm^3) = 17.6\ kg$

Total mass of wheels = 39.8 kg + 17.6 kg = 57.4 kg

Front End Support

The front end support is also a box beam with the same dimensions as the main support beam, except that it is only 1 m long. It is also made of aluminum.

Mass of front tne support = $(100\ cm)(29.25\ cm2)(2.56\ g/cm^3) = 7.5\ kg$

Total Grader Assembly Mass

The total mass of the grader assembly is 391 kg as shown in Table F.1. The additional 25% brings this value to 490 kg.

Table F.1: MASS OF THE GRADER ASSEMBLY

COMPONENT	MASS (kg)
Grader Blade	142
Blade Positioning Ring	76
Blade Connectors (2)	20
Triangular Ring Support	16
Power Screws (2)	26
Main Support Beam	45
Front End Support	8
Front Wheels (2)	40
Titanium Wheel Rims	18
TOTAL MASS	391

APPENDIX G

Compaction

APPENDIX G
Compaction

Compaction Theory

Soil compaction improves the mechanical properties of the soil in a number of respects, the most important of which are [1]

1. Soil compressibility is decreased
2. Soil strength increases
3. The permeability is reduced
4. Soil volume change characteristics enhance

A compactor is a self-driven or towed machine that uses one or more methods to increase the density of a medium. The methods of compaction are static weight, impact force, kneading action, and vibration [2]. Heavy smooth cylindrical rollers only apply the static weight, while tamping-foot (see Figure 21) rollers produce both static and kneading effects. An impact type compactor uses momentum to increase the soil density. Finally, a dynamic loading (vibration) is obtained by rapidly rotating, eccentrically-mounted weights. The impact type mechanisms are very similar to the vibrational type. The major difference between these two is that the impact mechanism and the ground are not in constant touch (i.e. the mechanism contacts the surface periodically) but the vibrational type never loses connection with the surface it is compacting. Also, the frequency of the impact systems is usually below 10 cycles/sec while that for the vibratory systems is as high as 80 cycles/sec or more [3].

Vibrational forces induce relative motion between the soil particles and they tend to compact the soil to a higher depth than any other method. For example, the smooth rollers can compact the soil effectively to a depth of 10-20 cm [4] while the vibratory systems have an effective compaction depth between 50-100 cm [5]. Furthermore, vibratory compactors are very effective in compacting cohesionless soils like sand [6].

Having too much weight on the soil, even though it results in a faster rate of compaction and a higher depth of effective compaction, it will tend to crack and loosen the top layer of the soil [7]. Generally, using a lightweight vibratory roller or a light to medium weight smooth roller will result in the best compacted surface [8]. To achieve the best compaction the normal contact pressure induced on the soil must be 0.9-1.0 of the elastic limit of the soil [9].

Compaction Equations

As mentioned earlier, the best compaction is resulted when the maximum normal contact pressure on the soil, σ_{max}, is within the following range

$$\sigma_{max} = (0.9 \text{ to } 1.0)\ \sigma_{p,} \qquad \text{(G1)} \quad [9]$$

where σ_p is the elastic limit of the soil. For less cohesive soils (sand, sandy loam, dusty), σ_p is 3-7 kgf/cm^2 and that for the medium cohesive soils is 7-12 kgf/cm^2 [10]. Both σ_p and σ_{max} are specified in kgf/cm^2 (kilogram force per centimeter squared).

Smooth Rollers

For smooth rollers, σ_{max} is found from Equation G2.

$$\sigma_{max} = \frac{C_1}{2R_0}\left(\sqrt[3]{\frac{3G_r R_0}{B_r C_1}}\right)^2 . \qquad \text{(G2)} \quad [11]$$

Where

C_1 = the coefficient of soil deformation, kgf/cm^3,

R_0 = the radius of the roller, cm,

G_r = the total weight on the axle of the roller, kgf, and

B_r = the width of the roller, cm.

C_1 is calculated using the following equation.

$$C_1 = E_1/D_s \qquad \text{(G3) [11]}$$

where

E_1 = the static modulus of soil deformation, kgf/cm^2 (for cohesive soil, E_1=150 to 200 kgf/cm^2, and for loose soil this is 100 to 150 kgf/cm^2), and

D_s = the diameter of the loading plate, cm (for 5-7 compactors, D_s is 15 to 20 cm).

Vibratory Rollers

A rapidly rotating, eccentric-mounted weight will produce a dynamic force which is computed from Equation G4. (See Figure G1.)

$$F = m\frac{v^2}{r} = m\frac{(r\omega)^2}{r} = mr\omega^2 \qquad \text{(G4) [12]}$$

Where

F = the dynamic force produced due to rotation, N,

m = mass of the eccentric weight, kg,

v = tangential velocity of the center of gravity of the eccentric weight, m/s,

r = the eccentricity of the rotating mass (the distance from the center of gravity), m, and

ω = the angular velocity of the eccentric weight, rad/s.

The dynamic force produced, F, is added to the static weight of the roller to give us the total dynamic loading of the vibratory roller.

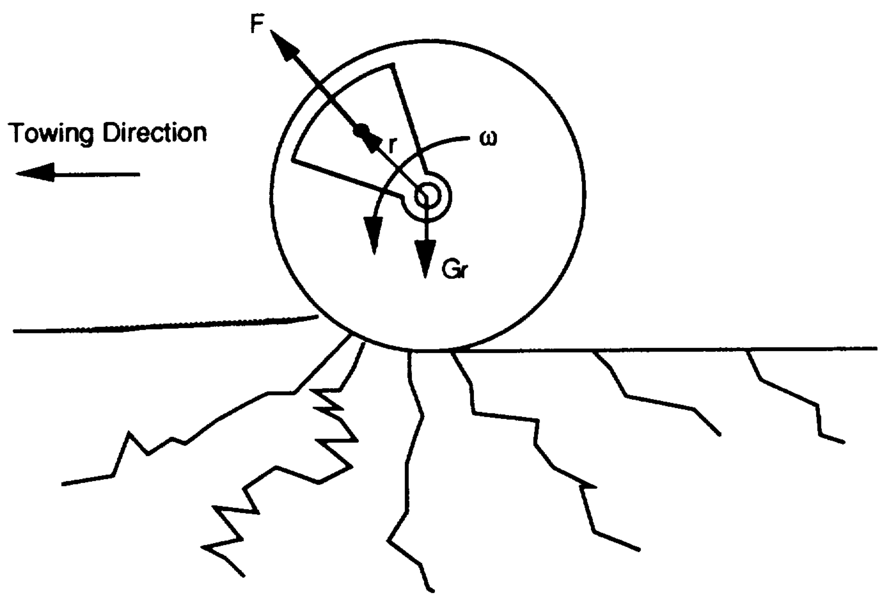

Figure G1: VIBRATORY ROLLER

The power required to rotate the eccentric-mounted weight, P, is derived in the following manner.

$$\begin{aligned} P &= \tau\omega = Fr\omega = m\frac{v^2}{r}r\,\omega \\ &= mv^2\omega = m(r\omega)^2\omega = mr^2\omega^3 . \end{aligned} \qquad \text{(G 5)}$$

where

τ = torque transmitted by the motor, N-m.

Compaction Calculations

The design team could not find any data on σ_p for regolith and, therefore, assumed a value of 6 kgf/cm^2 based on the ranges given earlier and soil characteristics of regolith (for example, its noncohesive behavior). Therefore, using Equation G1, the range of the acceptable values for σ_{max} is 5.4-6 kgf/cm^2. Furthermore, the design team assumed a value of 120 kgf/cm^2 for E_1. Finally, the design team assumed a linear relationship ($D_s = 2.5 + 0.0025\ G_r$) for D_s based on the given guidelines earlier.

Selecting a G_r and R_0 and knowing D_s and E_1, the design team calculated C_1 from Equation G3 and σ_{max} from Equation G2. In using Equation G2, the design team assumed a roller width to radius ratio of 3. The design team used the above procedure for various weights and roller radii to plot curves of σ_{max} versus R_0. (See Figure G2). This plot shows that for a 0.5 m radius roller with a width of 1.5 m and a weight of 5000 kgf (i.e., 8120 N in the lunar gravitation), σ_{max} is 5.8 kgf/cm^2 which is within the acceptable range of 5.4-6.0 kgf/cm^2.

Appendix H shows that the mass of the compactor when filled with regolith is 9040kg (14,680 N) or 4520 kg (7340 N) per roller. Since for best compaction 8120 N is required per roller, another 780 N is needed. Although the volume of the regolith ballast hopper can be increased to yield an additional 780 N, the design team decided to use a vibratory mechanism to take advantage of its beneficial features.

Knowing the vibratory force (780 N) and by varying the eccentricity and the mass of the eccentric weight, the design team used Equation G4 to plot the curves shown in Figure G3.

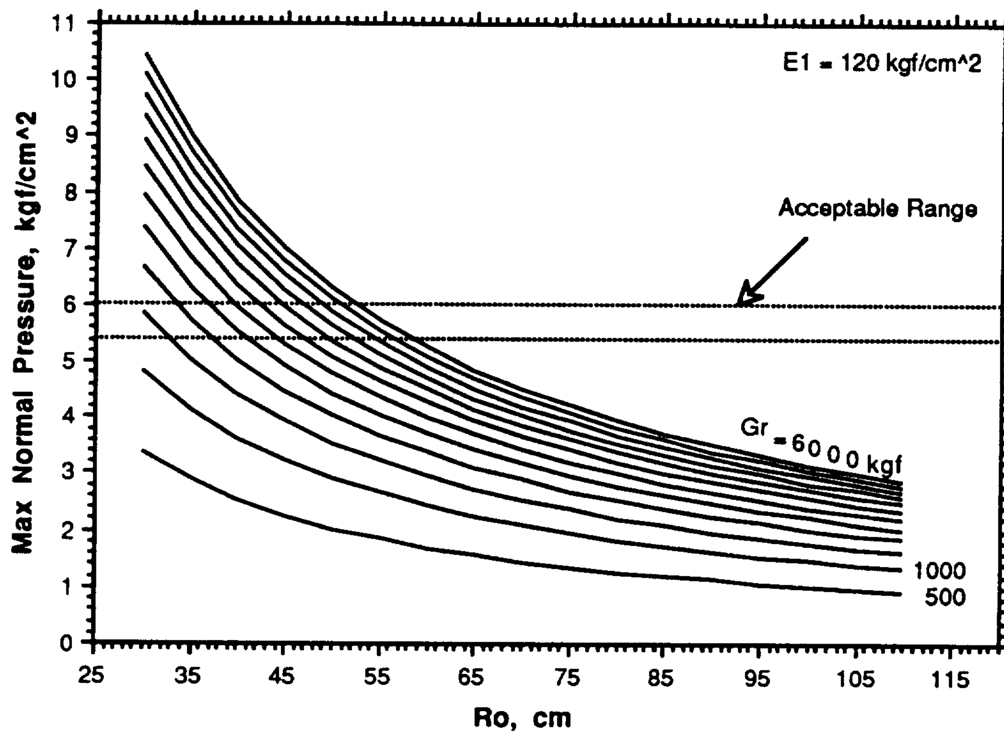

Figure G2: MAXIMUM NORMAL PRESSURE VERSUS ROLLER RADIUS

Using Equation G5, the design team calculated the power requirements of the vibrating mechanism for the same range of parameters used in Figure G3. (See Figure G4). Figures G3-G4 show that the 780 N required force can be obtained in numerous combinations of power, mass, eccentricity, and angular velocity. The design team selected a 15 kg mass, an eccentricity of 0.2 m, and an angular velocity of 16.1 rad/s (154 rpm) which will need 2.5 kW power per roller to operate. Adding a 10% power increase for inefficiencies of the motor, the total power requirements for the vibrating system of the compactor is 5.5 kW (7.4 hp).

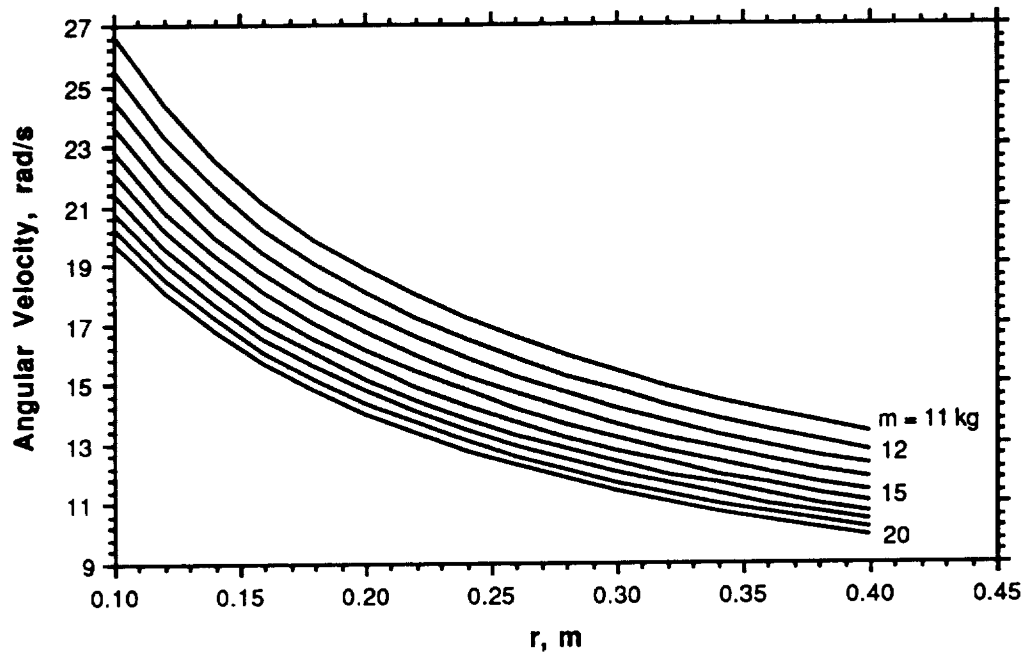

Figure G3: ANGULAR VELOCITY VERSUS ECCENTRICITY FOR F=780 N

The design team used Equation C1 to calculate the Main Drive Unit power required to push the compactor. In this equation

$M = 9041$ kg,

$R = 0.5$ m,

$g = 1.624$ m/s^2,

$k_c = 2.0588$ N/m^2,

$b = 1.5$ m, and

$k_\phi = 8515$ N/m^3.

Substituting these values in Equation C1 gives a rolling resistance of 13,200 N. Multiplying this by the speed of MDU (2 km/hr) yields the power which equals 7.3 kN.

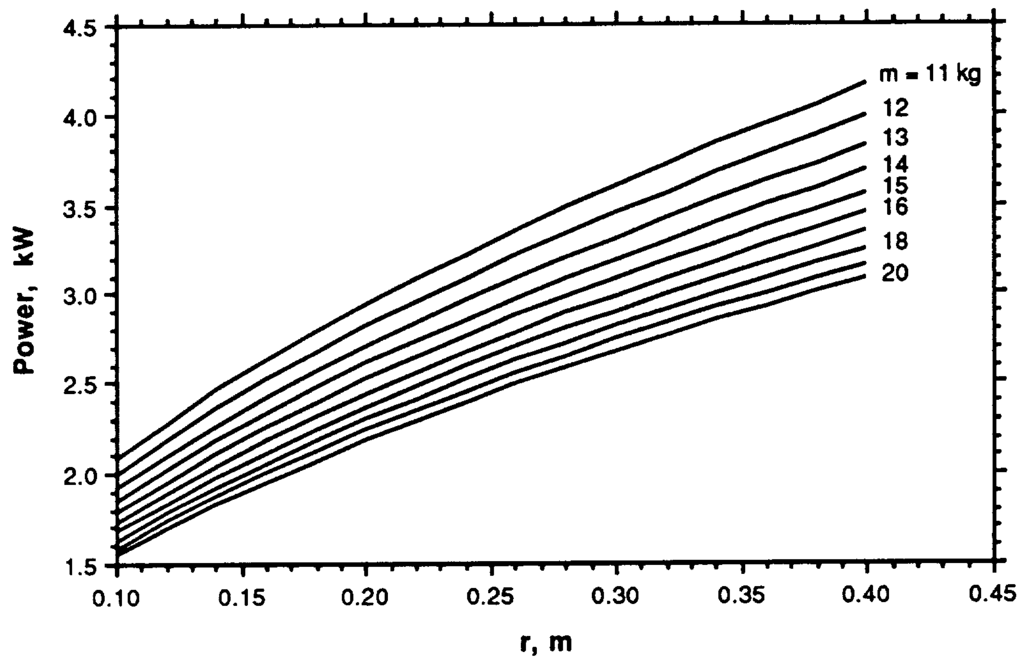

Figure G4: POWER VERSUS ECCENTRICITY FOR F=780 N

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APPENDIX H

Compactor Mass

APPENDIX H
Compactor Mass

The design team estimated the mass of the compactor assembly by estimating the dimensions of its major components and determining the materials from which these components are made. The dimensioned drawings are shown in Appendix J. The mass calculations are shown below for all of the major components.

ROLLERS

MATERIAL: $Ti_6 Al_4 V$

MASS:

Assumptions: 1) the roller is a hollow cylinder with circular plates mounted on both sides of the cylinder.

2) thickness of the roller walls = 1 cm

Abbreviations: m_R = mass of both rollers,

V_R = total volume of one roller,

R_0 = outer radius of the roller,

R_i = inner radius of the roller,

B_R = width of the roller,

t = thickness of the roller, and

ρ = density.

Analysis:

$$m_R = 2\rho V_R = 2\,\rho\,[\,\pi\,(\,R_0^2 - R_i^2\,)\,B_R + 2\,\pi\,R_0^2\,t\,)$$

$$= 2(4.42\ \text{g/cm}^3)\{\ \pi\ [(50\ \text{cm})^2\ -\ (50\ \text{cm} - 1\ \text{cm})^2](150\ \text{cm}) + 2\pi\ (50\ \text{cm})^2\ (1\text{cm})\}\ (1\text{kg}/1000\ \text{g})$$

$\therefore\ m_R = \underline{\underline{551\ \text{kg}}}.$

FRAME

MATERIAL: Al-Li 2090

MASS:

Assumptions: 1) the frame has a solid rectangular cross section

2) the cross section of the frame is 2 cm x 15cm

Abbreviations: m_F = mass of the frame,

V_F = total volume of the frame,

L = total length of the frame,

t = thickness of the frame,

w = width of the frame, and

ρ = density.

Analysis:

$$L = 2\left[(270\ \text{cm}) + \sqrt{(150\ \text{cm})^2 + (155/2\ \text{cm})^2}\right]$$

$$= 878\ \text{cm}$$

$$m_F = \rho V_F = \rho twL$$

$$= (2.56\ \text{g/cm}^3)(2\ \text{cm})(15\ \text{cm})\ (878\ \text{cm})(1\text{kg}/1000\ \text{g})$$

$\therefore\ m_F = \underline{\underline{67\ \text{kg}}}.$

ROLLER AXLES

MATERIAL: $Ti_6 Al_4 V$

MASS:

Assumptions: 1) the axles are hollow cylinders

2) the outer radius of the axle is 5 cm

3) the inner radius of the axle is 3 cm

Abbreviations: m_A = mass of both roller axles,

V_A = total volume of the roller axles,

R_0 = outer radius of the roller axles,

R_i = inner radius of the roller axles,

L = length of the roller axles, and

ρ = density.

Analysis:

$$m_A = 2\,\rho V_A = 2\,\rho\,\pi\,(R_0^2 - R_i^2)\,L$$

$$= 2(4.42\ \text{g/cm}^3)\,(\pi)\,[(5\ \text{cm})^2 - (3\ \text{cm})^2](150\ \text{cm})$$

$$\times\ (1\text{kg}/1000\ \text{g})$$

$$\therefore\ m_A = \underline{\underline{67\ \text{kg}}}.$$

REGOLITH BALLAST HOPPER

MATERIAL: Al-Li 2090

MASS:

Assumptions:
1) the hopper is made from 2 mm thick sheet metals
2) the mass of the regolith hopper is initially calculated by ignoring the housing of the vibratory mechanism; then, this calculated mass is increased by 20% to account for the mass of the housing and the fact that the clamshell door must be made from a thicker sheet metal.
3) the clamshell door is approximated by a 100 cm x 155 cm x 15 cm box
4) a reduction of 2.5% in the final mass of the regolith is appropriate to account for the volume occupied by the vibratory mechanisms.
5) density of regolith is 2000 kg/m^3

Abbreviations:

m_H = mass of the regolith ballast hopper,
V_H = volume of the sheet metal used to make the hopper,
A = total surface area of the sheet metal,
t = thickness of the sheet metal,
A_F = the surface area of the hopper's front view,
A_S = the surface area of the hopper's side view,
A_T = the remaining surface area of the hopper's top view,
V_{RH} = volume of the regolith in the hopper,
m_{RH} = mass of the regolith in the hopper,
ρ_R = density of regolith, and

ρ = density of the sheet metal.

Analysis:

$$A_F = 70\text{ cm}(300\text{ cm}) + 70\text{ cm}(100\text{ cm}) + 2\{(1/2)(5\text{ cm})(50\text{ cm}) + (50\text{ cm})(5\text{ cm}) + [(50\text{ cm})(50\text{ cm}) - (1/4)(\pi)(50\text{ cm})^2]\}$$
$$= 29{,}040\text{ cm}^2$$

$$A_S = \{15\text{ cm} + (1/4)(2\pi)(50\text{ cm}) + [(50\text{ cm})^2 + (5\text{ cm})^2]^{0.5} + 70\} \times (155\text{ cm})$$
$$= 33{,}140\text{ cm}^2$$

$$A_T = (100\text{ cm})(155\text{ cm})$$
$$= 15{,}500\text{ cm}^2$$

$$m_H = \rho V_H = \rho A t = \rho(2A_F + 2A_S + A_T)t$$
$$= (2.56\text{ g/cm}^3)[2(29{,}040\text{ cm}^2) + 2(33{,}140\text{ cm}^2) + (15{,}500\text{ cm}^2)](0.2\text{ cm})(1\text{kg}/1000\text{ g})$$
$$= 72\text{ kg}$$
$$m_H = 72\text{ kg} + 0.2(72\text{ kg})$$

$$\therefore\ m_H = \underline{\underline{86\text{ kg}}}.$$

The design team calculated the mass of regolith contained in the hopper in the following manner.

$$V_{RH} = \{3\text{ m}(0.9\text{ m}) + 2\text{ m}(0.05\text{ m}) + 0.5\text{ m}(1\text{ m}) + 0.15\text{ m}(1\text{ m}) + 2(1/2)(0.5\text{ m})(0.05\text{ m}) + 2[(0.5\text{ m})(0.5\text{ m}) - (1/4)(\pi)(0.5)^2]\}(1.55\text{ m})$$
$$= 5.55\text{ m}^3$$

$$m_{RH} = \rho_R V_{RH} = 5.55 \text{ m}^3 \,(2000 \text{ kg/m}^3) = 11{,}100 \text{ kg}$$

The design team reduced the mass of the regolith by 25% to account for the voids between the regolith particles (since, the density decreases after excavation). Then, the mass was reduced by 2.5 % for the reasons given earlier.

$$\therefore\ m_{RH} = \underline{\underline{8121 \text{ kg}}}.$$

TOTAL MASS

The total mass of the compactor is estimated as follows:

$$\begin{aligned} m_{total} &= m_R + m_F + m_A + m_H + m_{\text{hopper reinforcing ribs}} \\ &\quad + 2m_{\text{vibratory mechanism}} + m_{\text{towing coupling, bearings, fasteners}} \\ &= 551 \text{ kg} + 67 \text{ kg} + 67 \text{ kg} + 86 \text{ kg} + 9 \text{ kg} \\ &\quad + 2(25 \text{ kg}) + 10 \text{ kg} \\ &= 840 \text{ kg} \end{aligned}$$

To account for the oversimplification of the compactor, the design team decided to increase the calculated mass by 10%. Therefore, the final estimated mass of the compactor is

$$m_{compactor} = 840 \text{ kg} + 0.1 \,(840 \text{ kg}) = 924 \text{ kg}$$

$$\therefore\ m_{compactor} \approx \underline{\underline{920 \text{ kg}}}.$$

When the compactor is filled with regolith the total mass in increased to 9041 kg (14,683 N in the lunar gravitational field) or 4521 kg (7340 N) per roller.

APPENDIX I

Relevant Patents

only 11, 12

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United States Patent [19]
[11] 3,786,871
[45] Jan. 22, 1974

Long et al.

[54] **GRADER CONTROL**

[75] Inventors: George E. Long, Monroe; Dennis L. Reese, Kirkland; Floyd C. Johnson, Seattle, all of Wash.

[73] Assignee: Grad-Line Inc., Woodinville, Wash.

[27] Filed: July 26, 1971

[21] Appl. No.: 165,973

[52] U.S. Cl. 172/4.5, 172/793
[51] Int. Cl. E02f 3/76
[58] Field of Search.... 172/4.5; 94/46 AE; 250/202, 250/203; 299/1

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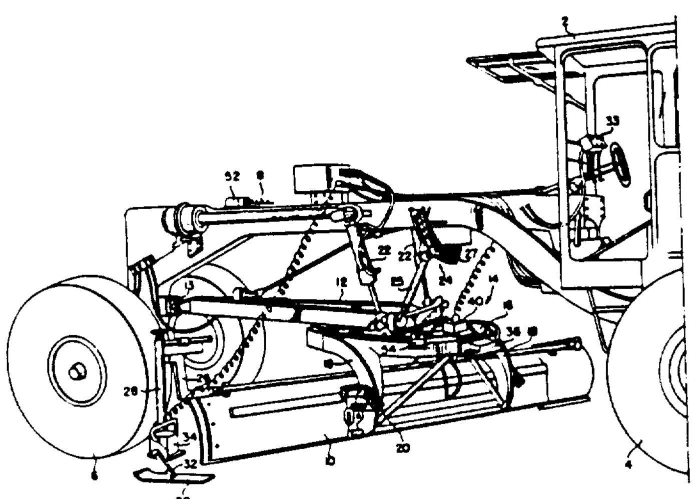
Primary Examiner—Robert E. Pulfrey
Assistant Examiner—R. T. Rader
Attorney, Agent, or Firm—Richard W. Seed et al.

[57] **ABSTRACT**

A control system for a road grader adapted to maintain the working edge of the grader blade in a preset datum plane so as to establish a uniform graded condition of the surface worked by the blade, regardless of the inclination assumed by the motor grader or the relative angular position of the blade itself. The system includes grade sensor means mounted adjacent one end of the blade to follow a pre-selected grade datum, slope sensing means mounted in fixed relation with the blade for keeping the blade at a predetermined slope with rotary position sensing means mounted on the blade supporting ring or circle to detect the relative angular position of the blade, and longitudinal grade angle sensing means with rotary position sensing means for altering the control signals to compensate for grade changes and rotary position of the blade.

8 Claims, 6 Drawing Figures



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PATENTED JAN 22 1974 3,786,871

SHEET 1 OF 3

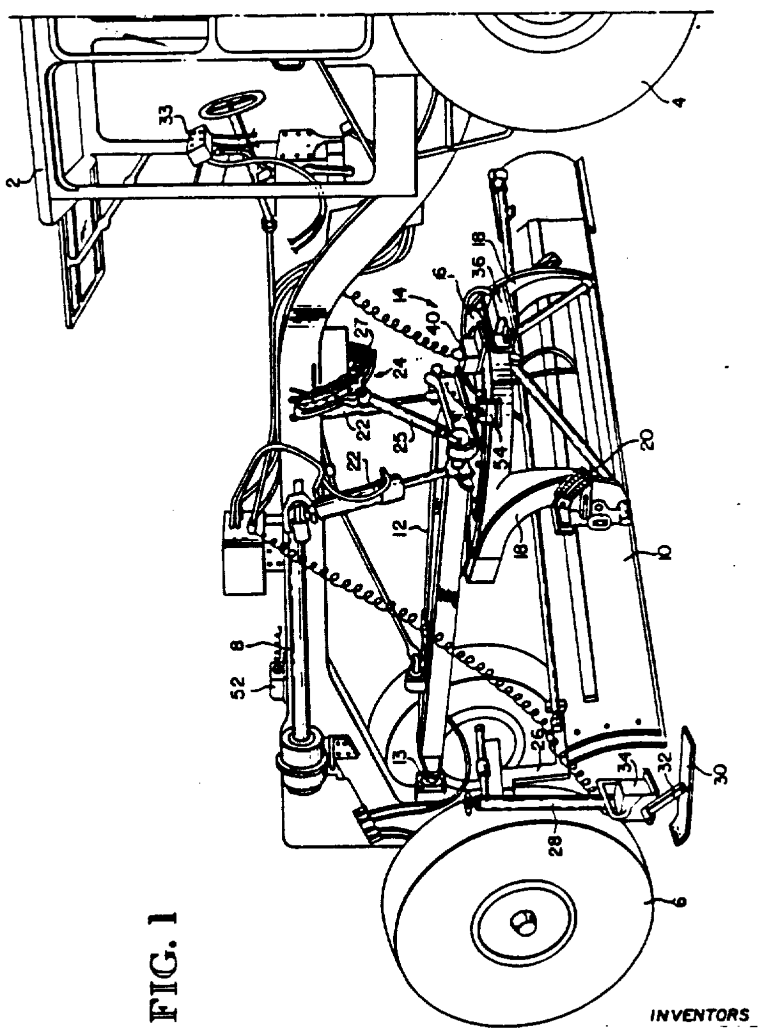


FIG. 1

INVENTORS
GEORGE E. LONG
DENNIS L. REESE
FLOYD C. JOHNSON

BY [signature]
ATTORNEYS

PATENTED JAN 22 1974

3,786,871

SHEET 2 OF 3

FIG. 2

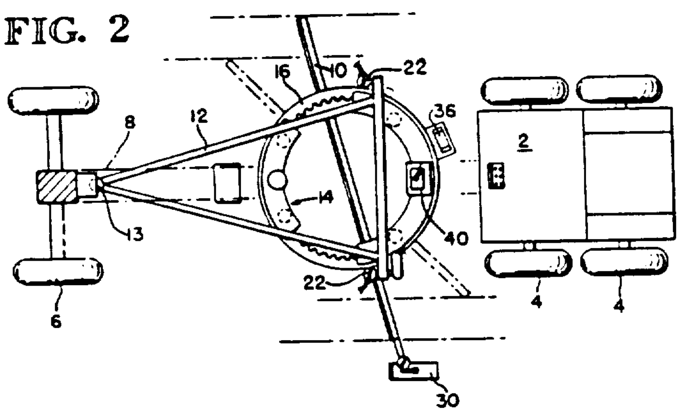


FIG. 3

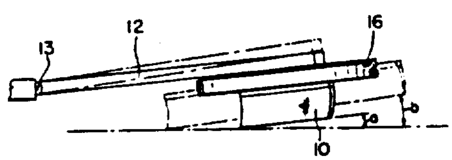
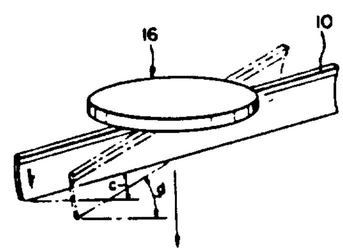


FIG. 4



GEORGE E. LONG
DENNIS L. REESE
FLOYD C. JOHNSON
INVENTORS

BY

ATTORNEYS

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United States Patent [19] | [11] 3,777,822
Stedman et al. | [45] Dec. 11, 1973

[54] **CONVEYORIZED MOTOR GRADER BLADE WITH RETRACTABLE END BITS**

[75] Inventors: Robert N. Stedman, Chillicothe, Ill.; Bobby D. Griffith, Phoenix, Ariz.; Robert J. Sullivan, Peoria, Ill.

[73] Assignee: Caterpillar Tractor Co., Peoria, Ill.

[22] Filed: Mar. 31, 1972

[21] Appl. No.: 240,167

[52] U.S. Cl. 172/33, 172/63, 172/782, 37/45

[51] Int. Cl. A01b 3/00

[58] Field of Search 172/784, 33, 63, 172/200, 782; 37/45, 82

[56] References Cited

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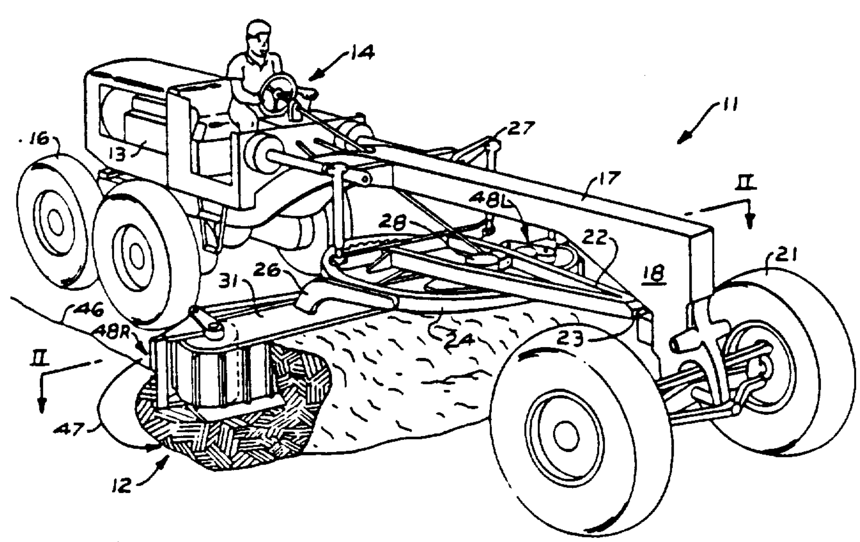
954,386	4/1910	Ganser	172/782 X

Primary Examiner—Robert E. Pulfrey
Assistant Examiner—Stephen C. Pellegrino
Attorney—Donald J. McRae et al.

[57] **ABSTRACT**

A blade assembly for a motor grader having a power driven endless belt conveyor for forcibly moving material to one side of the blade assembly is provided with end bits for facilitating operations such as bank cutting, ditching and the like in which stress may be concentrated at one end of the blade assembly. The end bits greatly reduce the wear and risk of damage which can otherwise occur at the cutting end of the conveyor. While the movement of the conveyor tends to prevent jamming of materials between the conveyor and end bit at the cutting end of the assembly, an opposite situation prevails at the discharge end. To avoid jamming at the discharge end, the end bits are retractable whereby only the bit at the cutting end need be maintained in the operative position.

5 Claims, 5 Drawing Figures



PATENTED DEC 11 1973 3,777,822

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FIG. 1.

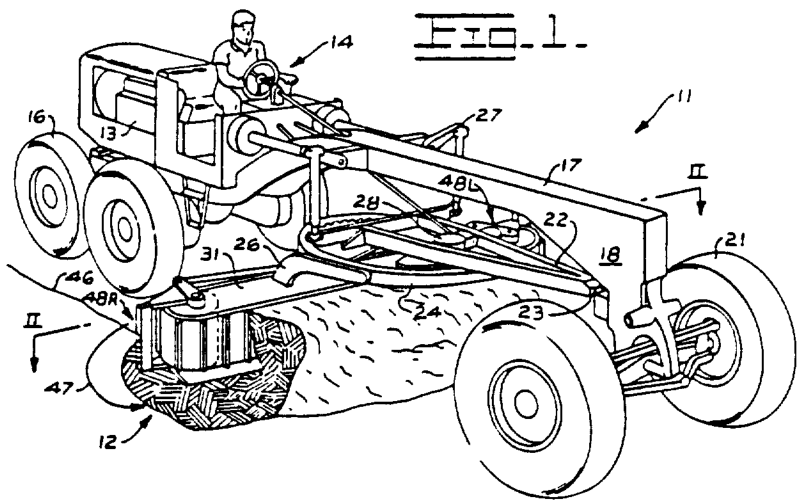

FIG. 2.

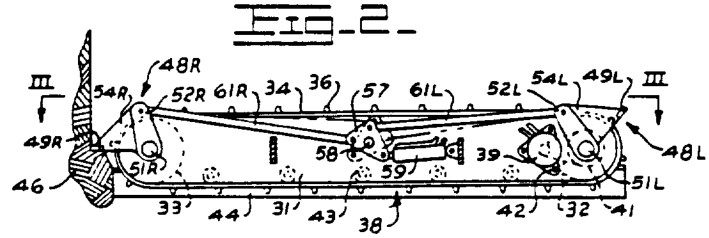

FIG. 3.

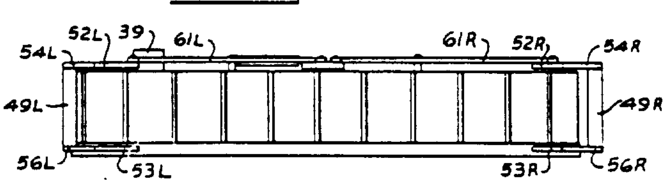

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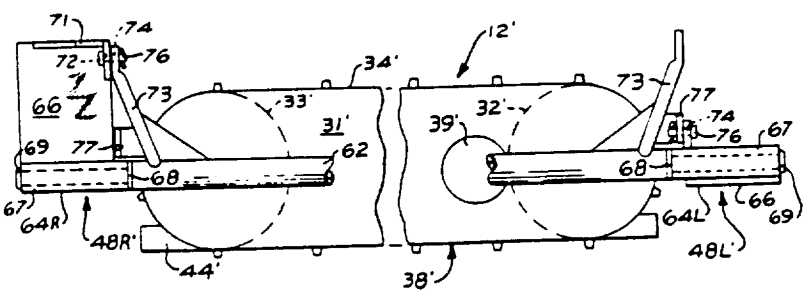
FIG. 4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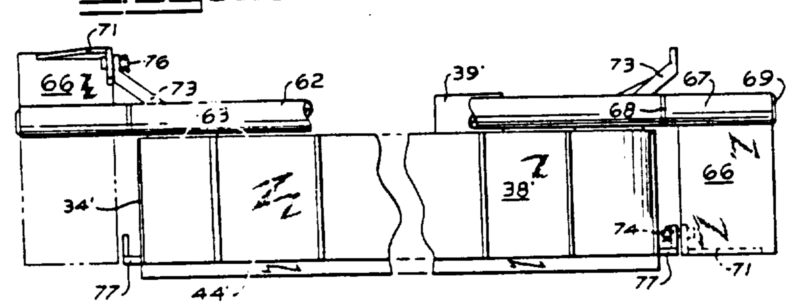
48L'

FIG. 5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United States Patent

Brown

[15] 3,693,722

[45] Sept. 26, 1972

[54] **FINE GRADING DEVICE FOR RUBBER TIRE ROAD GRADER**

[72] Inventor: **Robert L. Brown,** Chesapeake, Va.

[73] Assignee: **Brohighe, Ltd.,** Norfolk, Va.

[22] Filed: **Aug. 11, 1970**

[21] Appl. No.: **62,917**

[52] U.S. Cl. 172/4.5, 172/72, 172/785, 37/108

[51] Int. Cl. E02f 3/76, A01b 49/02, E02f 3/12

[58] Field of Search..........172/785, 779, 72, 71, 4.5, 172/784; 37/108

[56] **References Cited**

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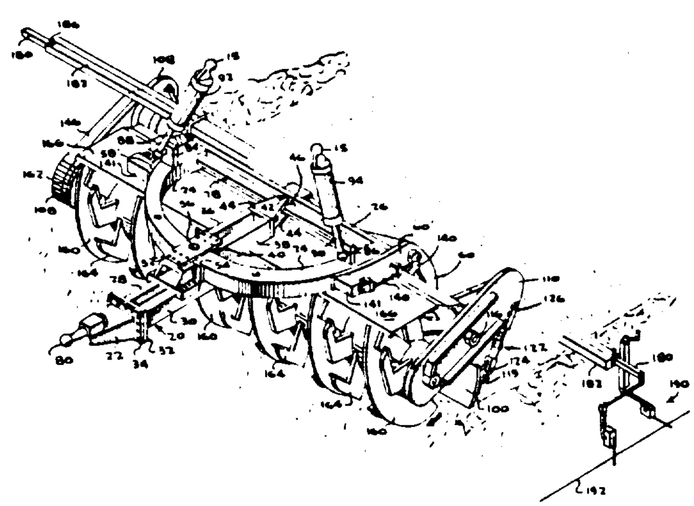
3,423,859	1/1969	Swisher et al.	37/108

Primary Examiner—Robert E. Pulfrey
Assistant Examiner—Stephen C. Pellegrino
Attorney—Mason, Fenwick & Lawrence

[57] **ABSTRACT**

A grading blade and parallel rotary auger are supported on an auxiliary frame transversely with respect to a grader vehicle from which the frame is supported with the auger being connected to the blade for unitary pivotal movement about a pivot point on the frame between the front of the blade and the rear of the auger in general vertical alignment with the cutting edge of the blade so that the height of the auger can be pivotally adjusted by hydraulic cylinder means with respect to the cutting edge of the blade for differing soil conditions by pivoting the auger and blade with there being a negligible vertical displacement of the cutting edge of the blade; an automatically operable control means actuates hydraulic cylinders supporting the auxiliary frame from the vehicle in response to signals from a guide line extending along the path of movement of the vehicle for maintaining the cutting edge of the blade at a given elevation with respect to the guide line regardless of the height of the auger with respect to the cutting edge of the blade as adjusted by the operator for differing soil conditions.

14 Claims, 7 Drawing Figures



PATENTED SEP 26 1972 3.693.722

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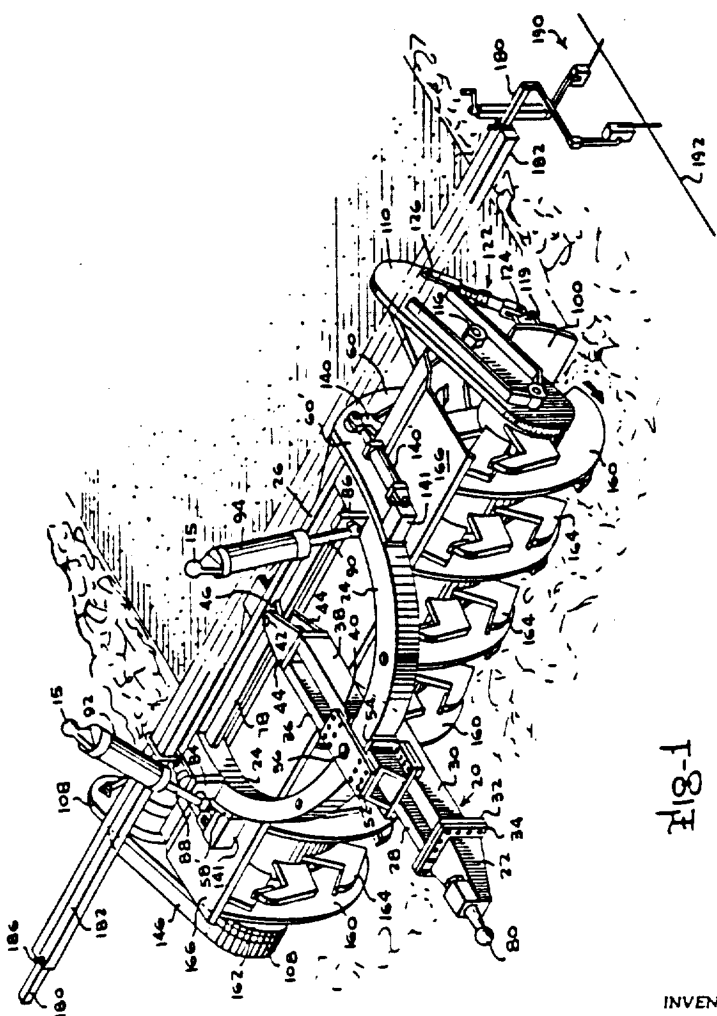

Fig-1

INVENTOR

ROBERT L. BROWN

BY

Mason, Fenwick & Lawrence
ATTORNEYS

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PATENTED SEP 26 1972 3,693,722

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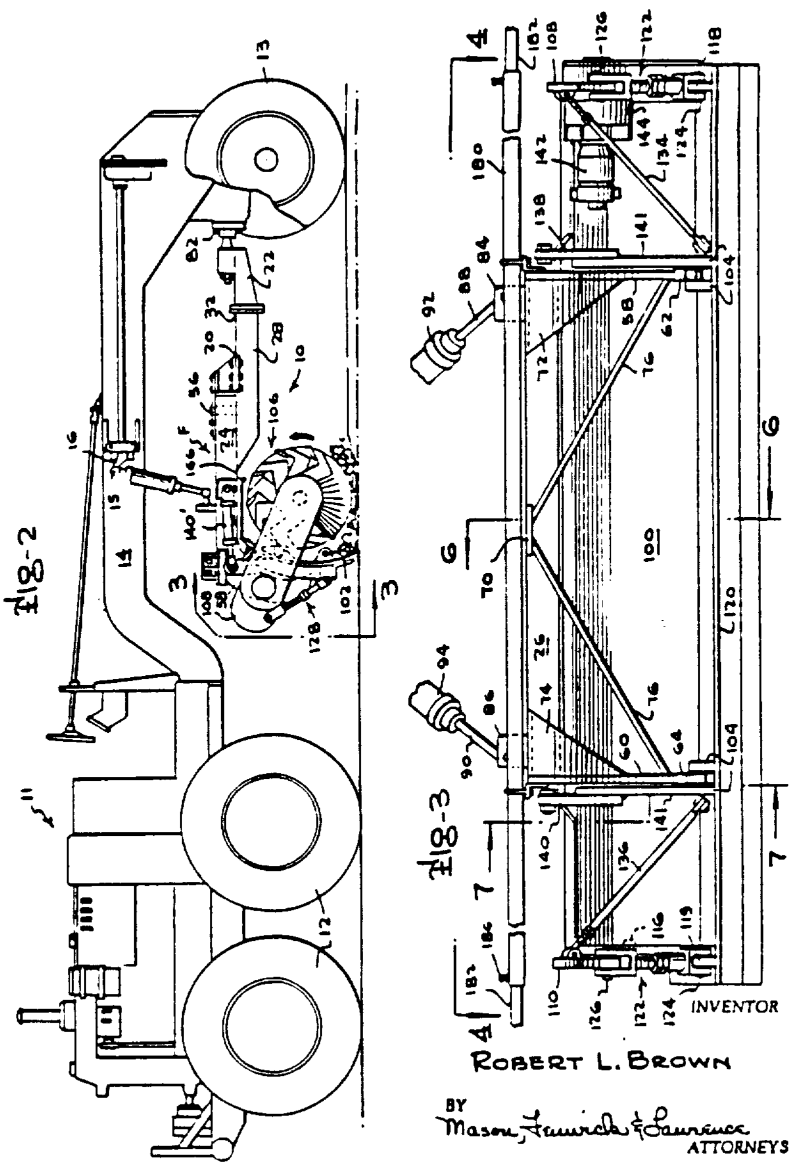

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Mason, Fenwick & Lawrence
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PATENTED SEP 26 1972 3,693,722

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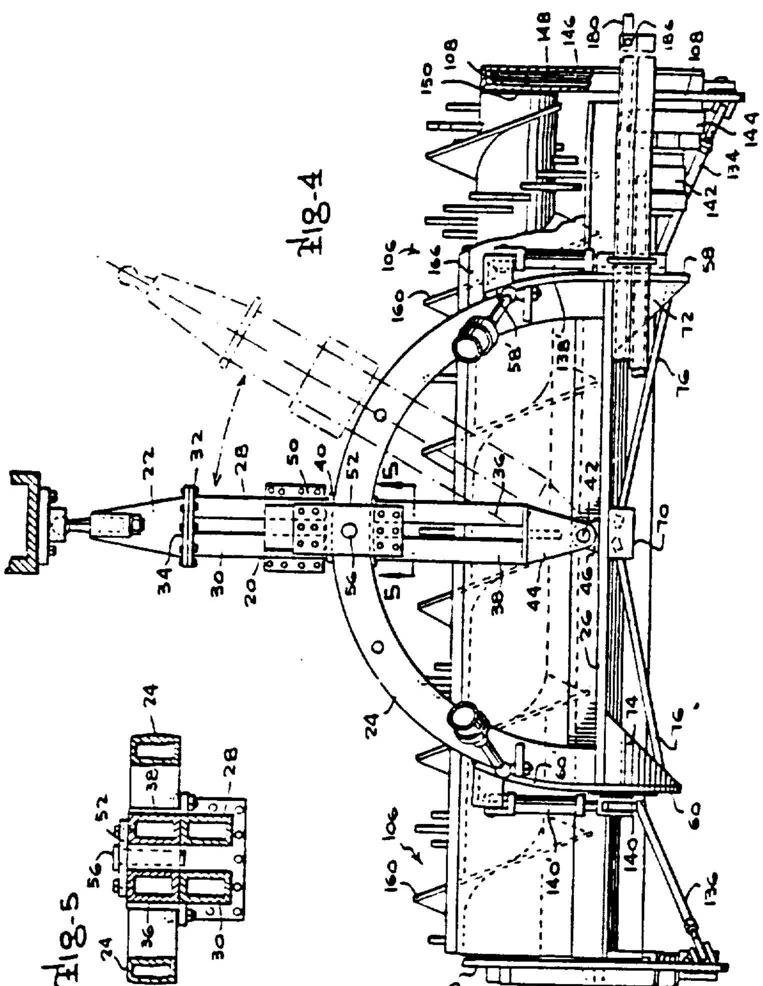

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PATENTED SEP 26 1972

3,693,722

SHEET 4 OF 4

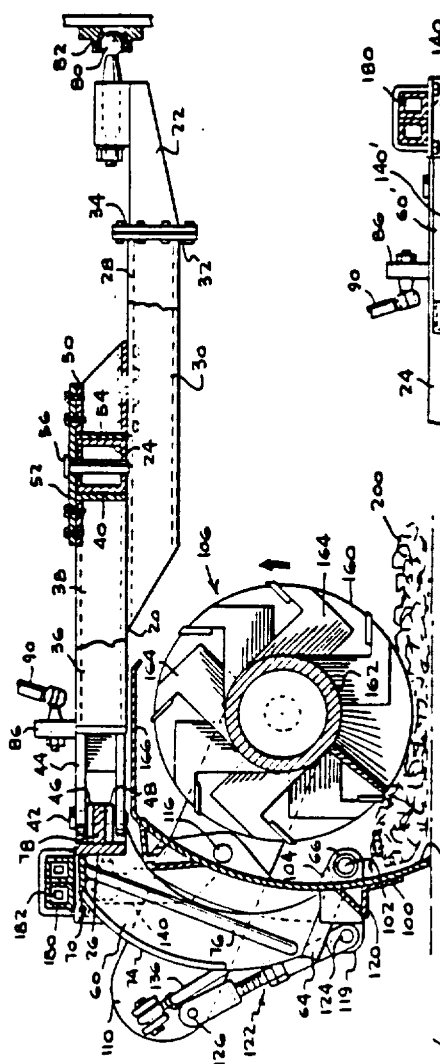

Fig-6

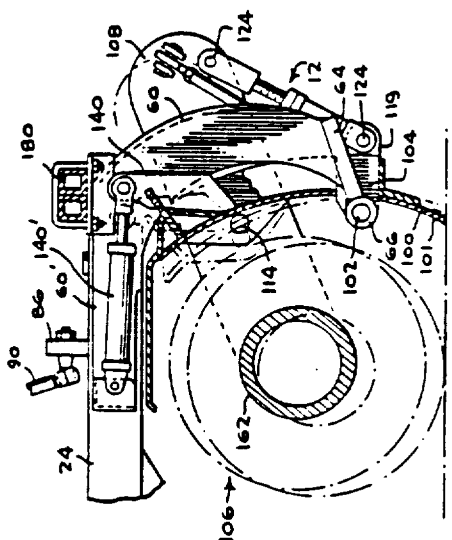

Fig-7

INVENTOR

ROBERT L. BROWN

BY

Mason, Fenwick & Lawrence
ATTORNEYS

United States Patent [19]

[11] 3,767,262

[45] Oct. 23, 1973

Pentith

[54] **ROAD CUTTING MACHINE WITH LATERALLY EXTENSIBLE DRUM AND METHOD**

[75] Inventor: Gerald R. O. Pentith, Sheffield, England

[73] Assignee: Greenside Machine Co., Limited, Durham, England

[22] Filed: Nov. 17, 1971

[21] Appl. No.: 199,677

[52] U.S. Cl. 299/10, 172/122, 299/39

[51] Int. Cl. E01c 23/09

[58] Field of Search 299/39–41, 10; 51/176; 37/117.5; 94/50; 172/122

[56] References Cited

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3,560,050 2/1971 Lockwood 299/39

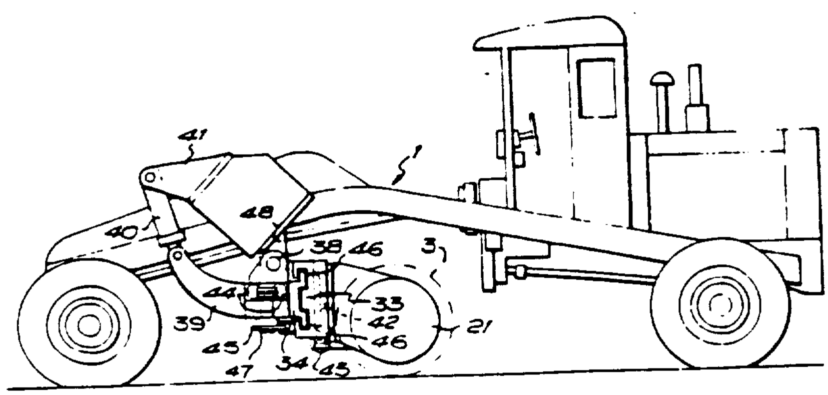
Primary Examiner—Ernest R. Purser

Attorney—C. Yardley Chittick et al.

[57] **ABSTRACT**

A machine for removing a worn road surface comprises a chassis on which is supported a rotary drum armed with cutting picks. To make possible a reduction in the inner radius of an arcuate path cut out by the rotary drum, the drum is traversable into a position lying outside the ground plan of the machine as defined by the wheels on which the machine is supported for movement.

7 Claims, 4 Drawing Figures



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PATENTED OCT 23 1973 3,767,262

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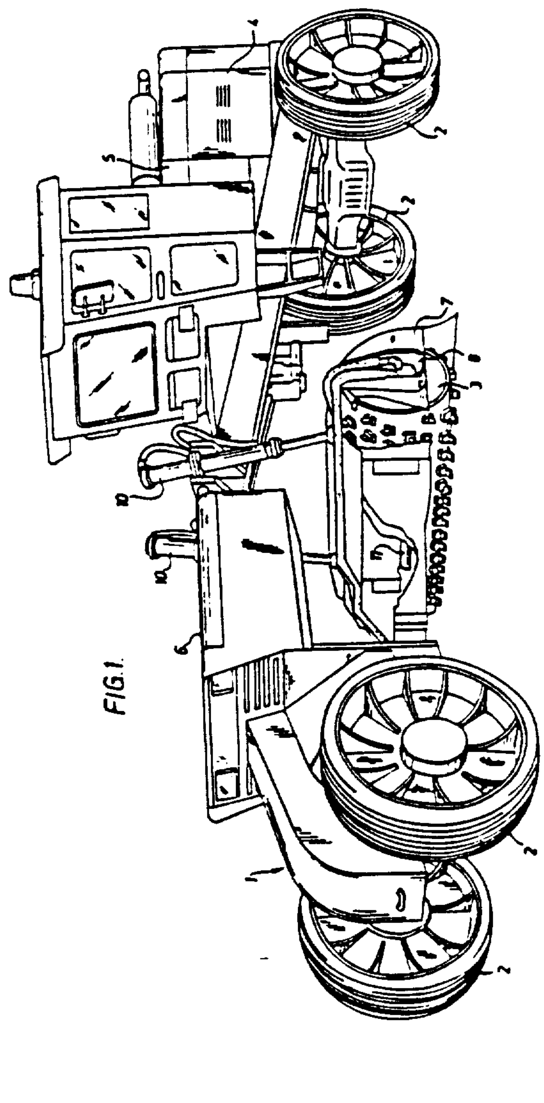
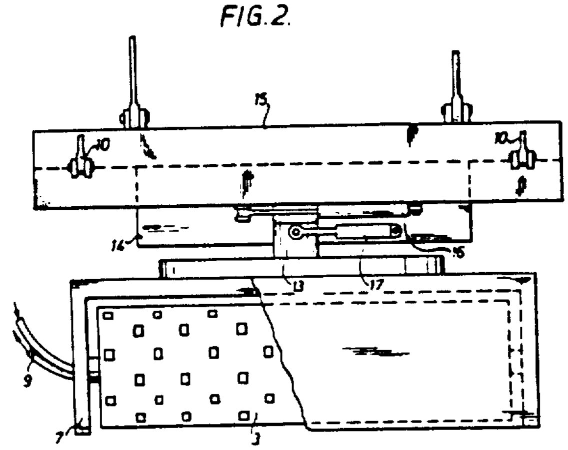
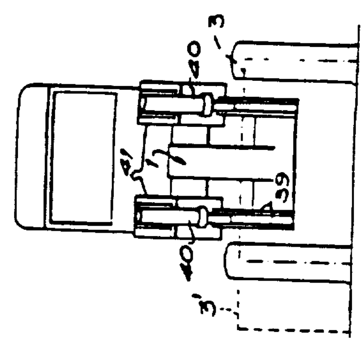
FIG.1.



PATENTED OCT 23 1973 3,767,262
SHEET 2 OF 3

FIG.2.





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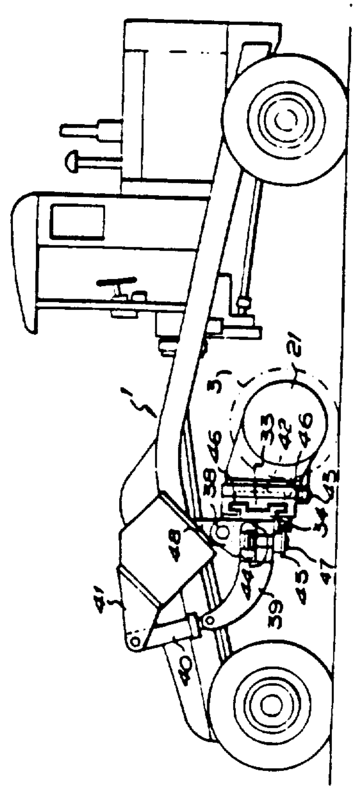

FIG 3

United States Patent [19]

Ruhter et al.

[11] **Patent Number:** 4,696,350

[45] **Date of Patent:** Sep. 29, 1987

[54] **MOTOR GRADER WITH SADDLE MOUNTED TO TRANSVERSE PIN ON MAIN FRAME**

[75] Inventors: Martin L. Ruhter; Dennis A. Brimeyer; David W. Stubben, all of Dubuque, Iowa

[73] Assignee: Deere & Company, Moline, Ill.

[21] Appl. No.: 780,048

[22] Filed: Sep. 25, 1985

[51] Int. Cl.⁴ E02F 3/80

[52] U.S. Cl. .... 172/ /93; 37/108 R; 172/753; 280/760; 403/387

[58] Field of Search .... 172/776, 781, 789, 791, 172/793, 795, 796, 797, 780, 782, 783, 784, 785, 786, 787, 788, 790, 792, 794, 798, 799, 753; 403/387, 388; 180/11, 235; 280/760; 37/108 R, 231, 234, 235, 236, DIG. 13

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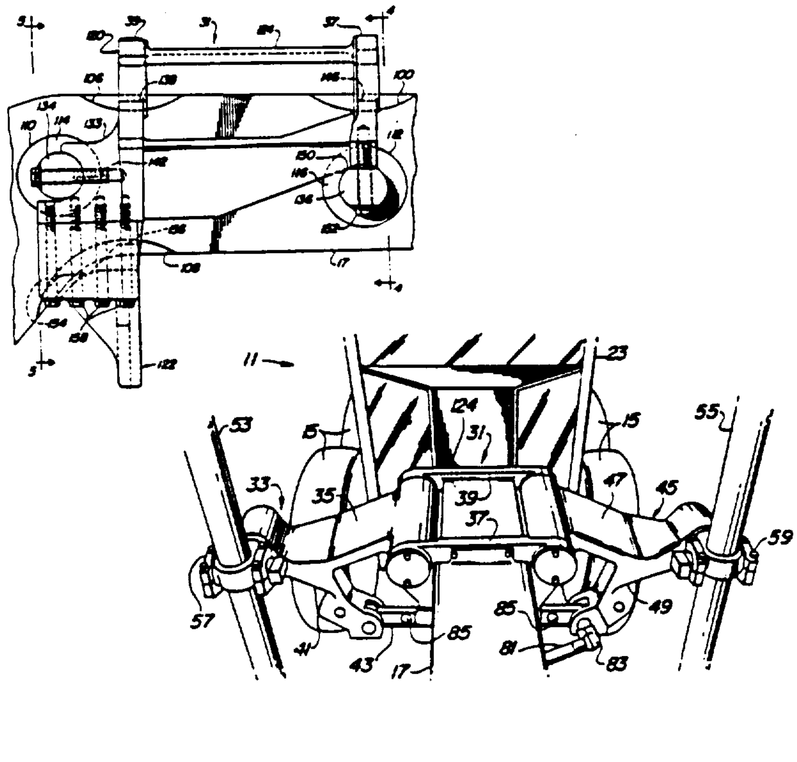
Primary Examiner—Richard T. Stouffer

[57] **ABSTRACT**

A saddle construction for use in a motor grader having a fore-and-aft extending main frame comprising an upper portion having a generally longitudinal extending section forming a forward facing and a rear facing. The facings have a generally vertical orientation. The main frame has first and second pins slidably mounted transversely in the main frame in longitudinal spaced apart relationship. The forward facing of the saddle upper portion straddles the main frame and is fixably mounted to opposite ends of the first pin. In like manner, the rear facing straddles the main frame and is fixably mounted to opposite ends of the second pin. The saddle further includes a lower portion straddling the main frame and fixably mounted the rear facing. The main frame has a plurality of recesses formed in respective side walls of the main frame in longitudinal spaced apart relationship for seatably receiving inwardly directed landings formed on the facings.

4 Claims, 5 Drawing Figures



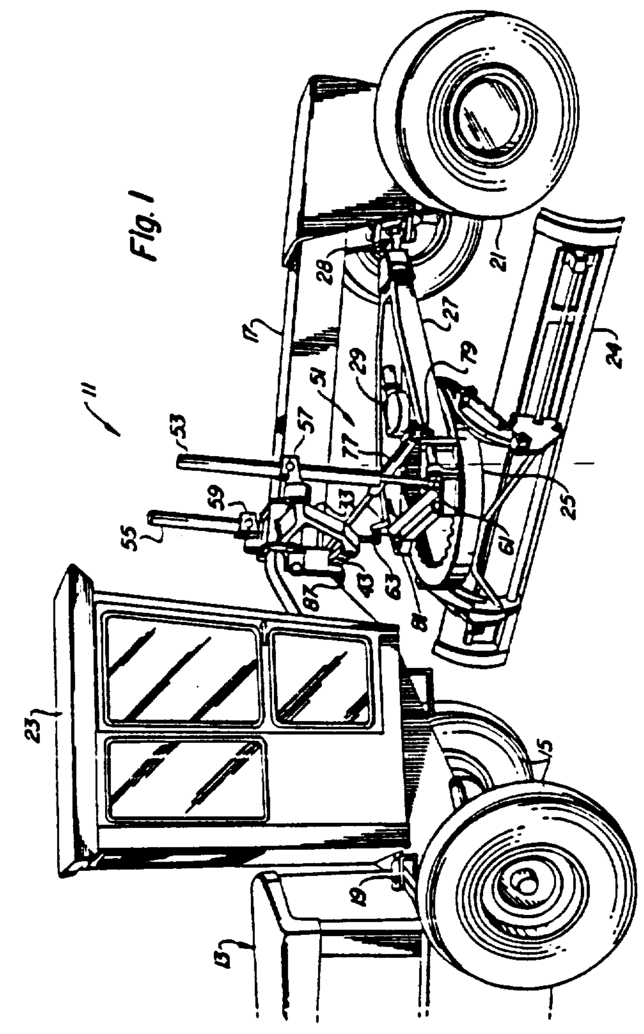


Fig. 1

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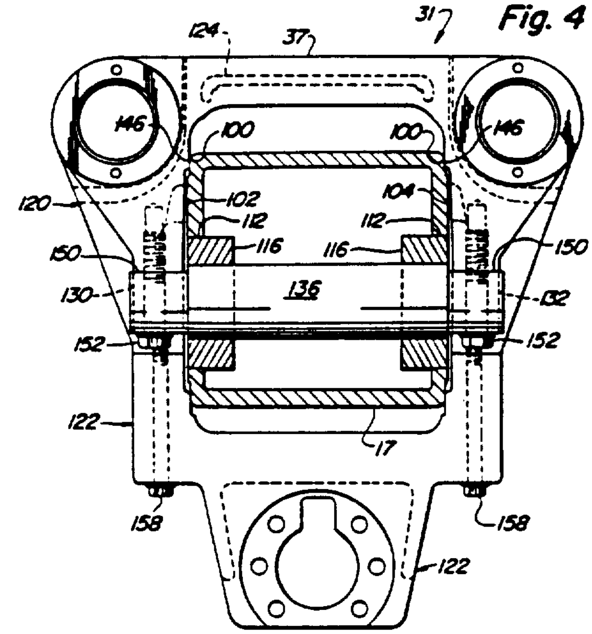
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Fig. 4

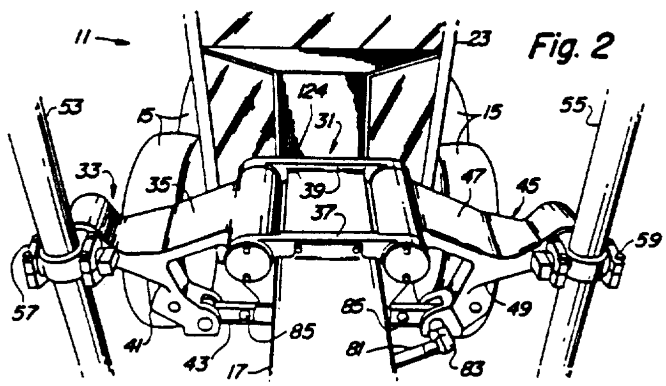
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Fig. 2

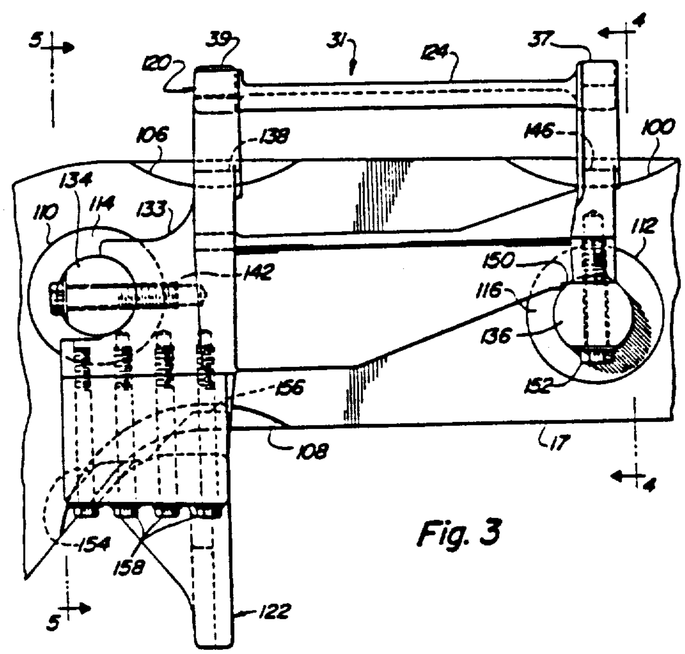
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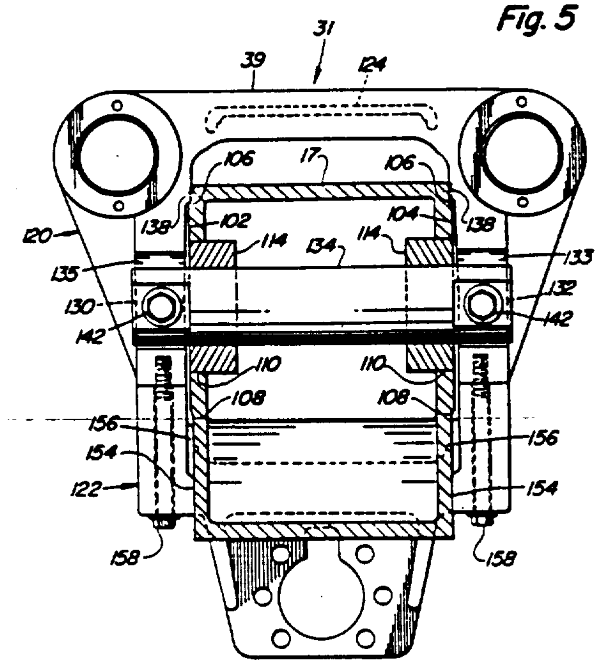
Fig. 3

Fig. 5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United States Patent [19]
Mironov et al.

[11] **3,841,006**
[45] **Oct. 15, 1974**

[54] EXCAVATING AND GRADING MACHINE WITH ADJUSTABLE ROTARY CUTTING HEAD

[76] Inventors: **Ivan Antonovich Mironov,** Institutskaya ulitsa, 16, kv. 55; **Vladislav Iosifovich Nikitin,** Institutskaya ulitsa, 10, kv. 11, both of Pushkino Moskovskoi oblasti; **Valentin Dmitrievich Kiselev,** ulitsa Tsentralnaya, 5, Moskovskaya oblast, poselok Mamontovskaya, all of U.S.S.R.

[22] Filed: **Mar. 1, 1973**

[21] Appl. No.: **337,196**

[52] **U.S. Cl.** .... **37/108 R**, 37/189
[51] **Int. Cl.** **E02f 3/18**
[58] **Field of Search** 37/189, 108; 172/807, 803, 172/804

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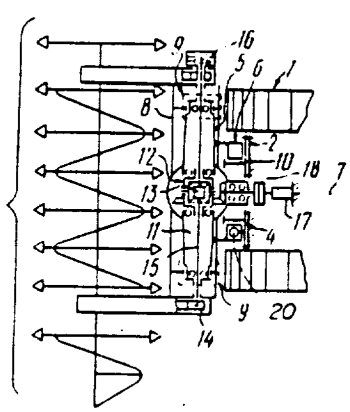
Primary Examiner—Clifford D. Crowder
Attorney, Agent, or Firm—Holman & Stern

[57] ABSTRACT

An improvement in an excavating and grading machine with a power-driven active cutting head which rotates during the process of excavation. More specifically, the improvement relates to the suspension of the rotary cutting head in which the frame of the cutting head comprises a juxtaposition of three member frames. The first member frame of the cutting head rigidly connects to the frame of a self-propelled chassis of the machine. The second member frame is connected to the first frame by an axle pivot arranged coaxially with the longitudinal axis of the frame of the chassis and is capable of turning about this pivot by means of a hydraulic cylinder. The third member frame of the cutting head is connected to the second member frame by an axial pivot arranged at right angles to the longitudinal axis of the chassis frame and is linked up with hydraulic cylinders serving to turn this member frame about the pivot which connects it to the preceeding second member frame. The axles of both pivots are tubular to pass through their bores a power transmission for imparting motion to the rotary cutting head.

1 Claim, 2 Drawing Figures



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PATENTED OCT 15 1974 3,841,006

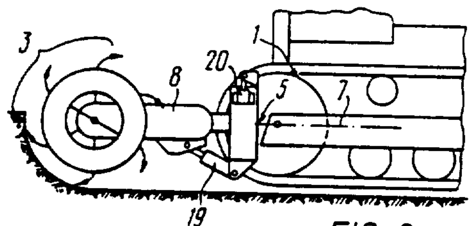

FIG. 2

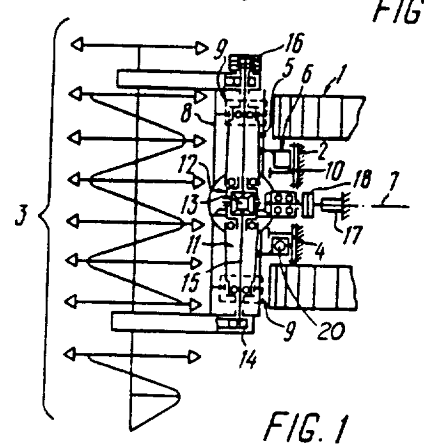

FIG. 1

United States Patent [19]

Miller et al.

[11] **3,999,314**

[45] **Dec. 28, 1976**

[54] **SMALL SUB-GRADER**

[75] Inventors: David J. Miller; Charles P. Miller, both of McHenry, Ill.

[73] Assignee: Miller Formless Co., Inc., McHenry, Ill.

[22] Filed: Nov. 5, 1975

[21] Appl. No.: 629,023

[52] U.S. Cl. .... 37/108 R; 172/4; 37/DIG. 20; 404/84; 172/26

[51] Int. Cl.² E02F 5/00

[58] Field of Search .... 37/108 R, 108 A, DIG. 4, 37/15, 20; 172/4.5, 4, 26, 779, 803, 804, 807; 404/84

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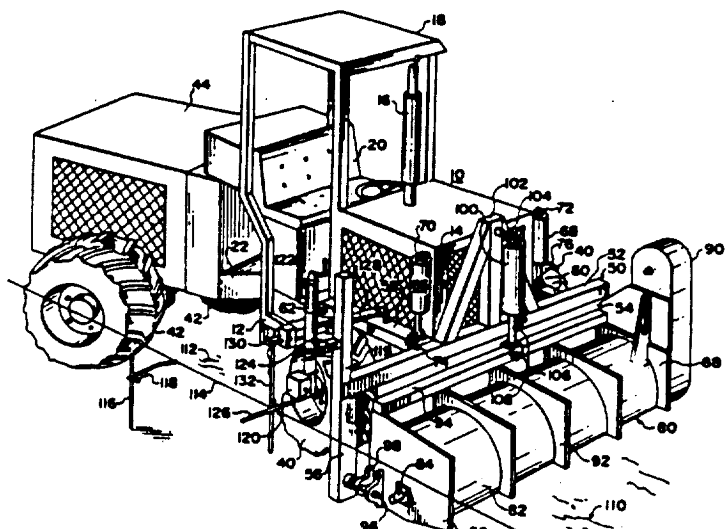
Primary Examiner—E. H. Eickholt
Attorney, Agent, or Firm—Bruce K. Thomas

[57] **ABSTRACT**

The rotating cutting tool of a sub-grader is adjustably mounted transverse the front of the frame of the machine upon a tool support in a manner which isolates changes in grade from changes in slope and which minimizes the influence of changes in slope upon the grade of the tool. The transverse pivot axis for grade control of the tool support is located at an intermediate point on the main frame, while the cutting tool, its associated drive means and housing are mounted on an independent pivot axis for slope control that extends longitudinally on one side of the main frame. Any changes made in the grade are accomplished by raising and lowering the entire tool support on the transverse axis equally at both ends which does not alter the slope of the tool. The arc of movement is relatively long so that the change in fore and aft altitude of the tool is minimal. The rotating tool is maintained perpendicular to and in the plane of its longitudinal slope control axis by the tool frame. However, any changes in slope are entirely independent of the grade deviations and the tool is maintained in a desired plane at all times.

4 Claims, 7 Drawing Figures



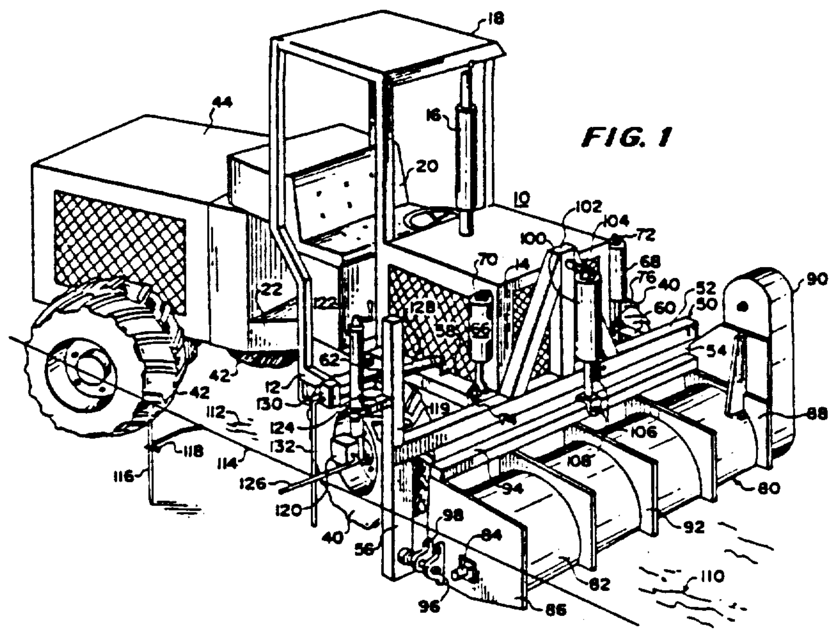


FIG. 1

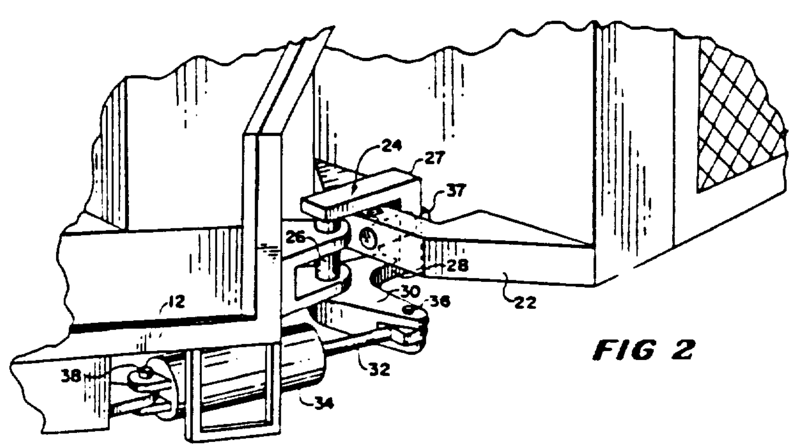


FIG 2

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FIG. 3

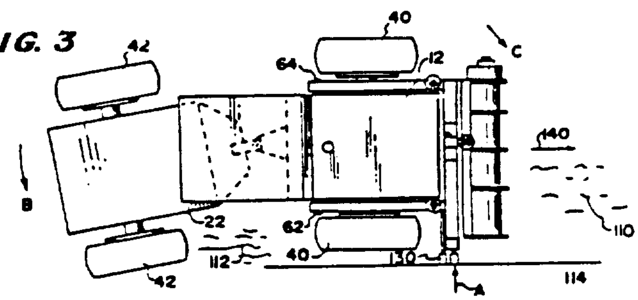


FIG. 4

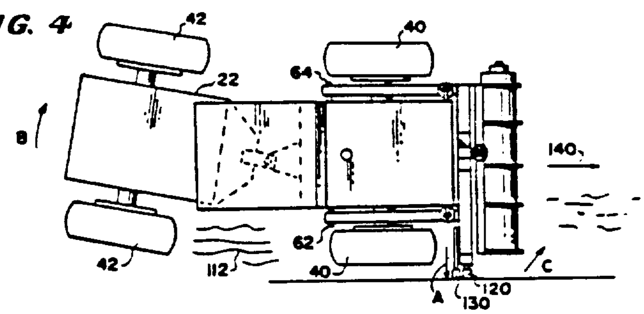


FIG. 5

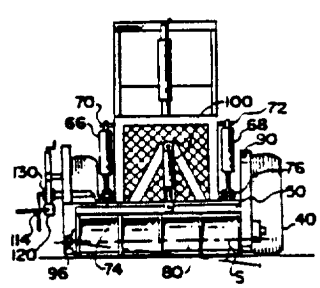
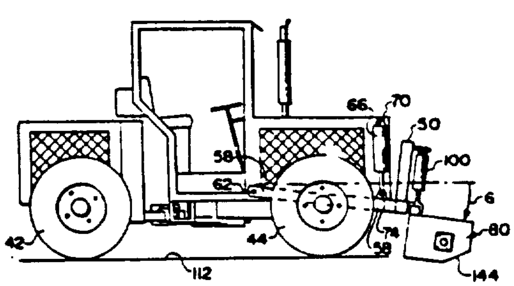


FIG. 6



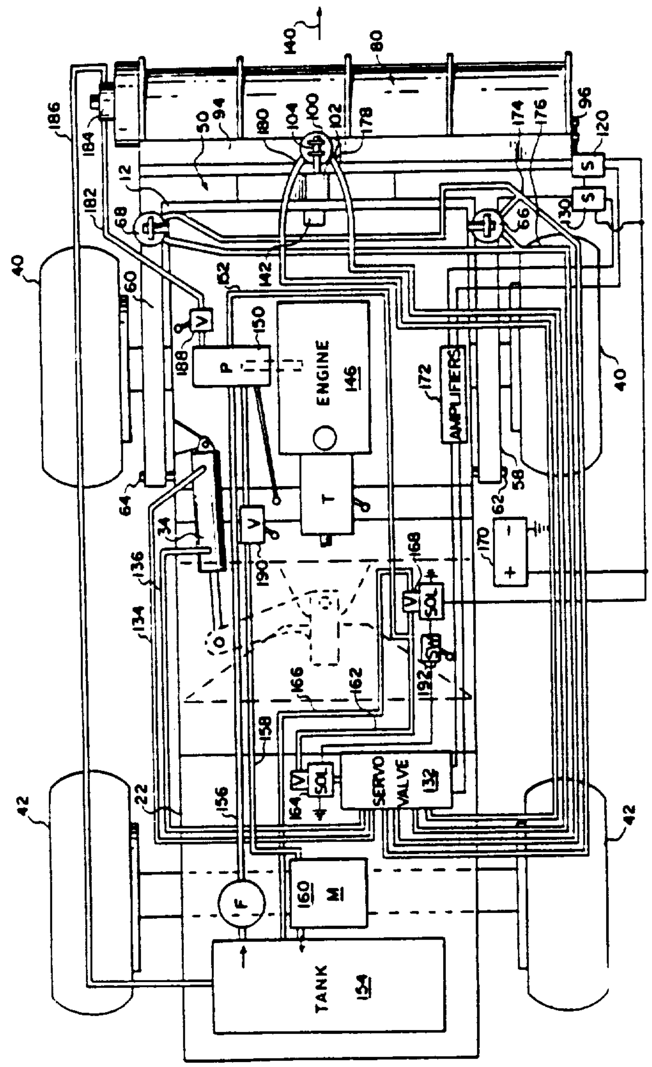
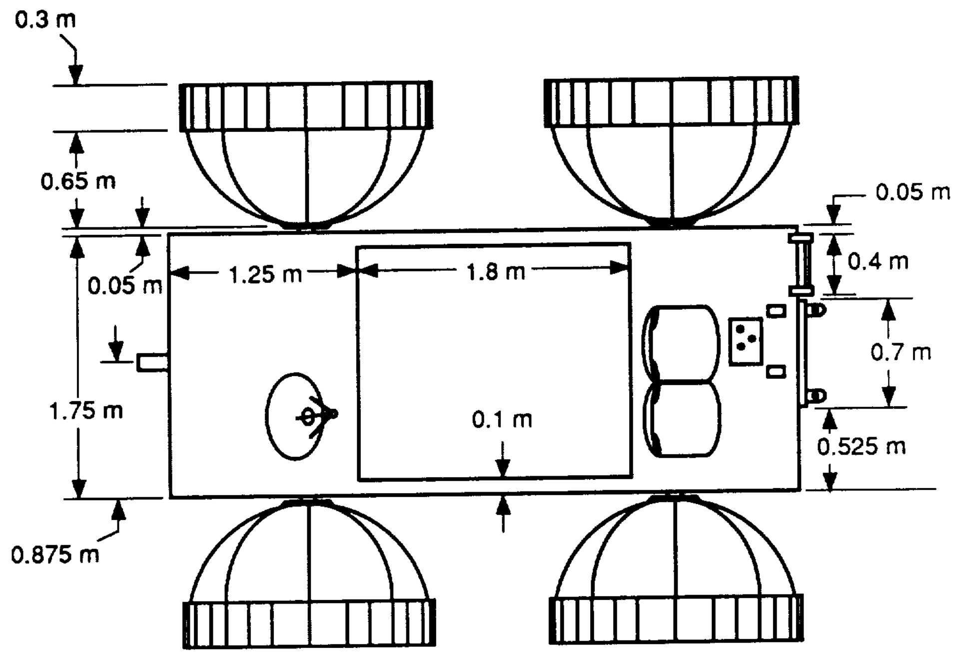


FIG. 7

APPENDIX J

Dimensioned Drawings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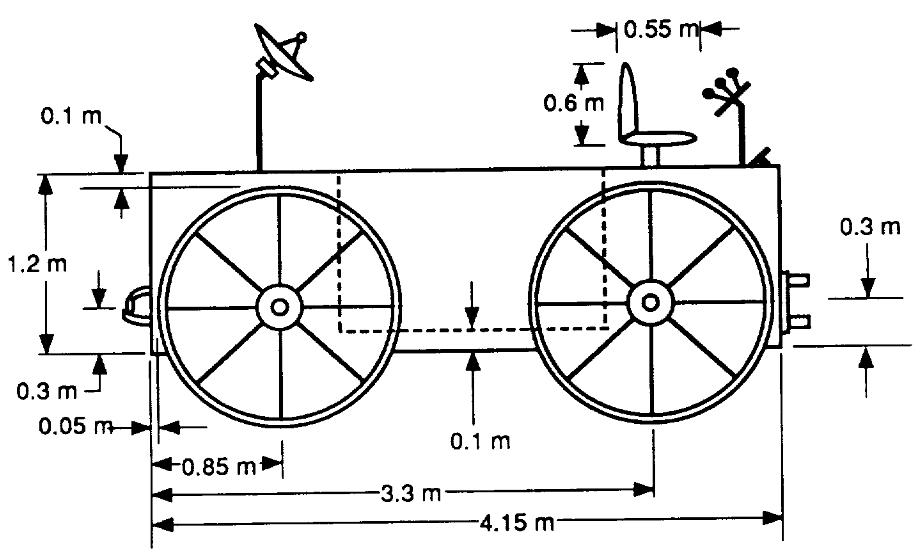
0.875 m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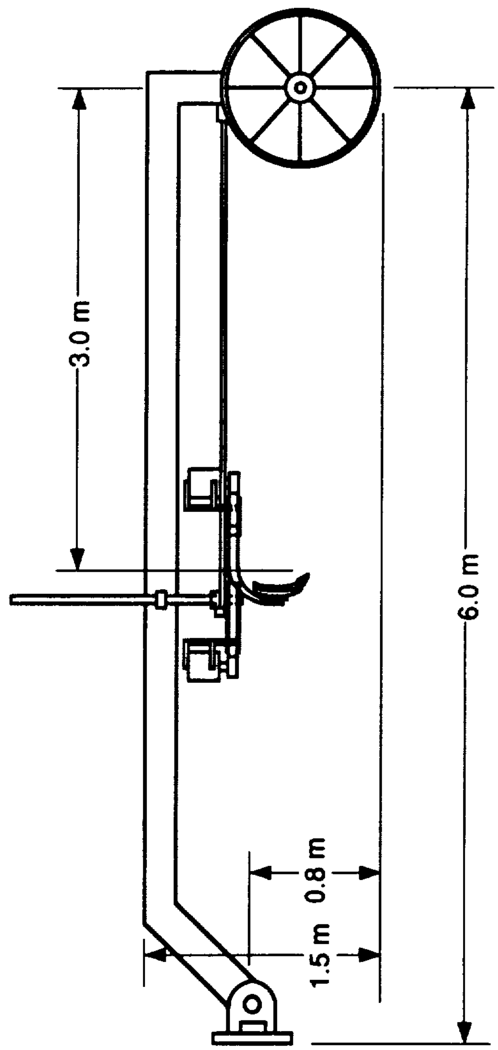
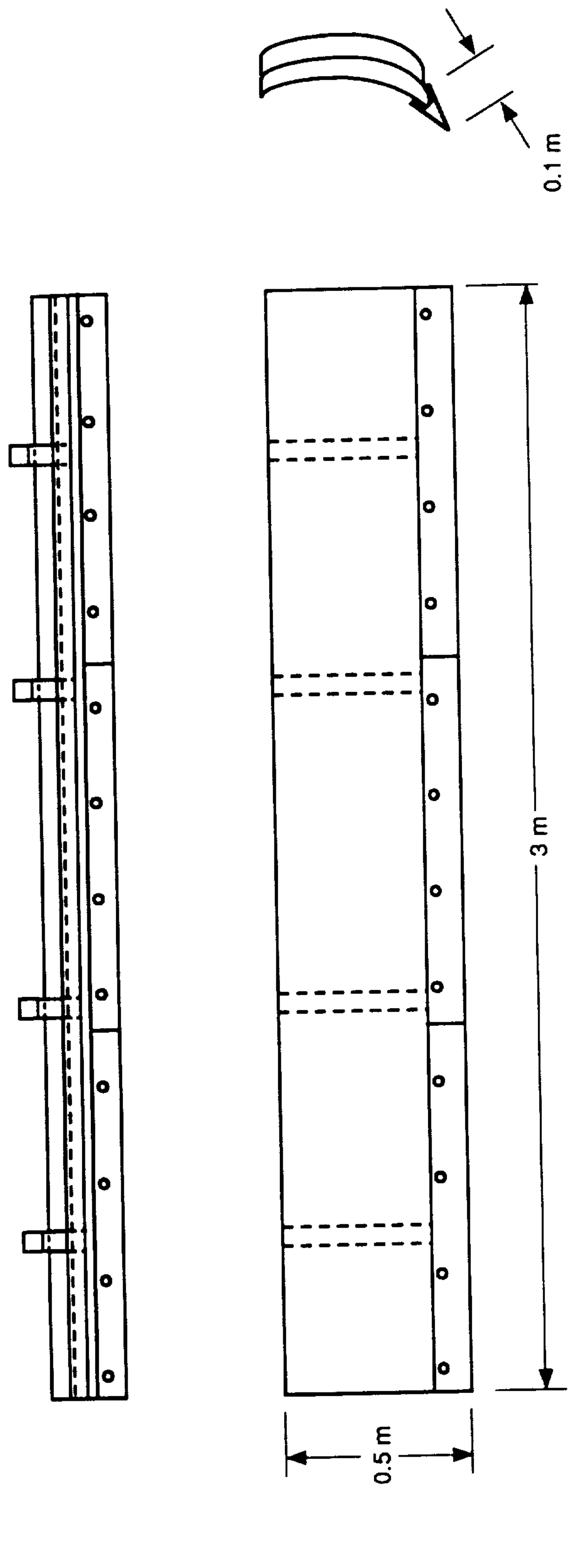
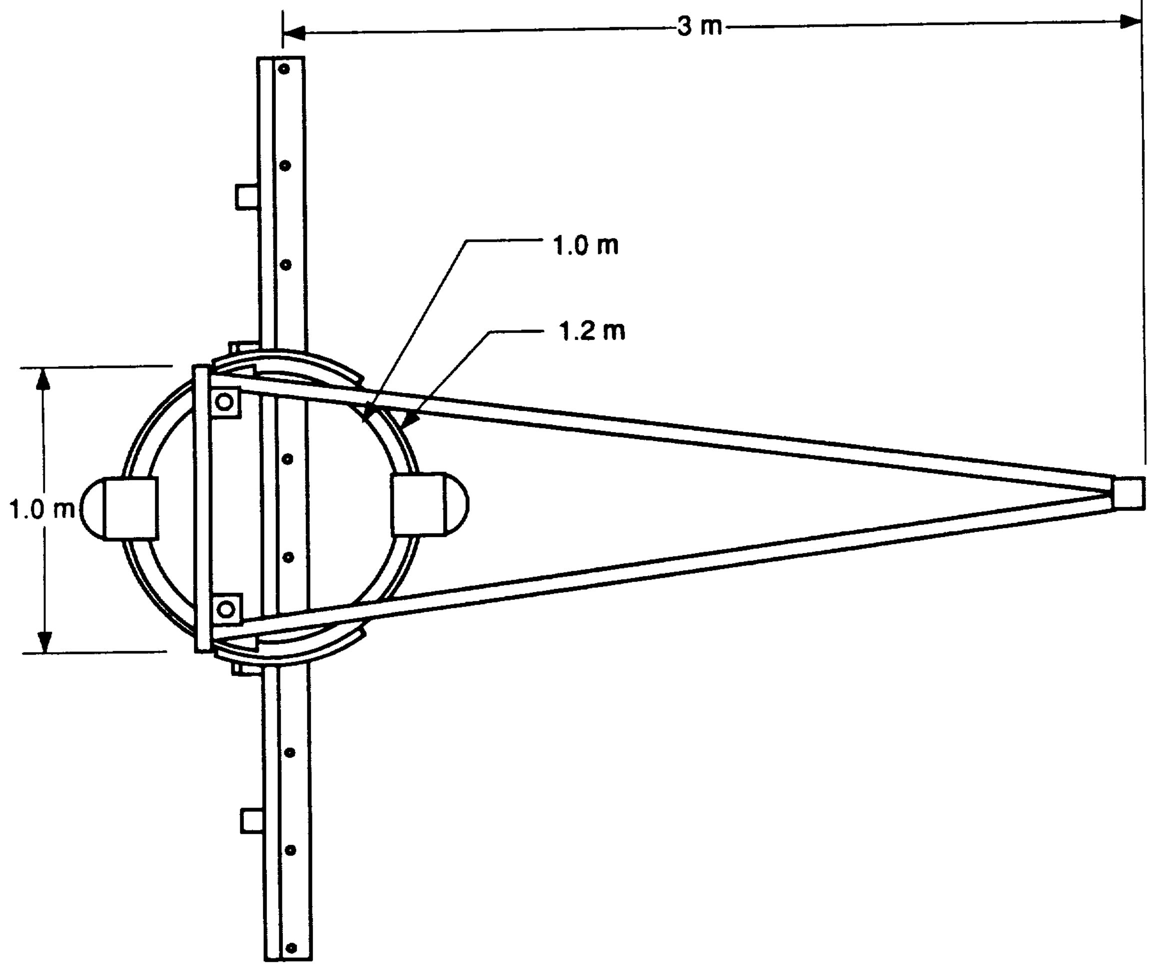
0.6 m
0.1 m
1.2 m
0.3 m
0.3 m
0.05 m
0.1 m
0.85 m
3.3 m
4.15 m

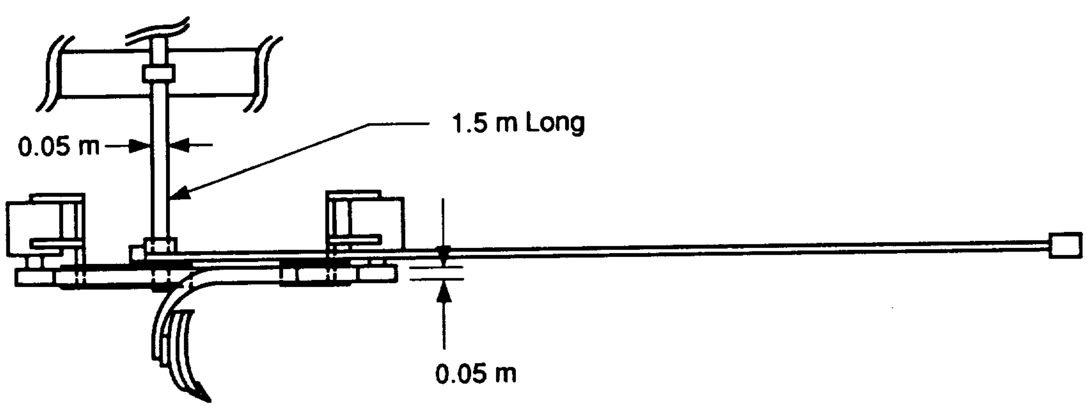

3.0 m
6.0 m
0.8 m
1.5 m

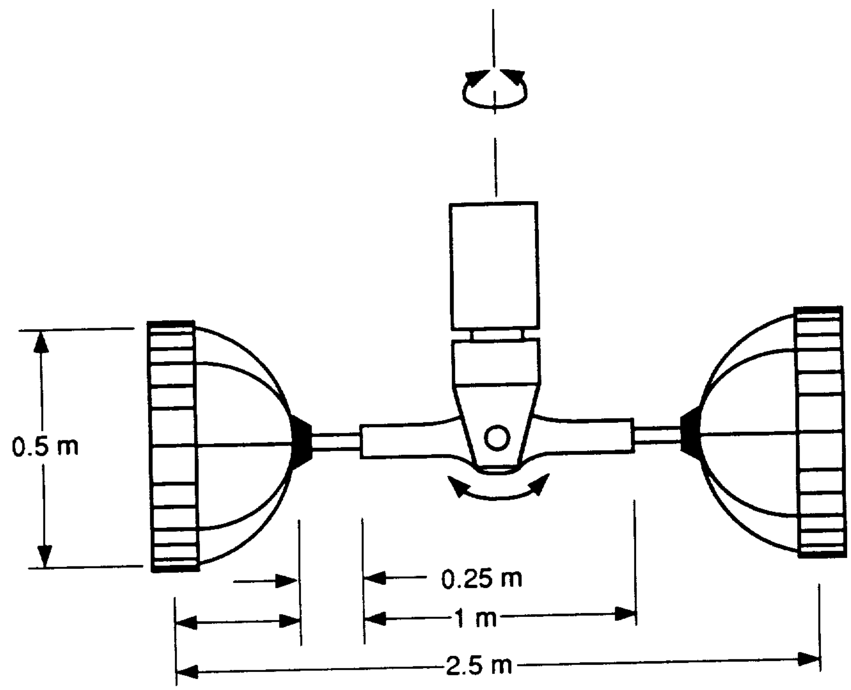
J4



J5



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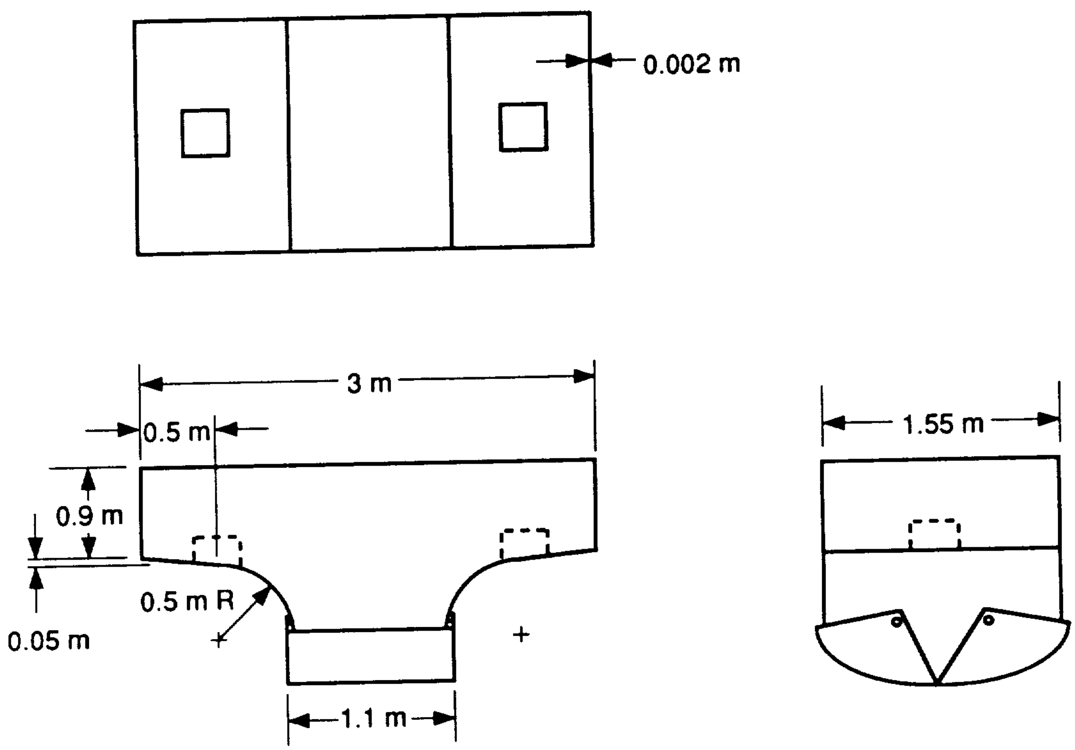
2.5 m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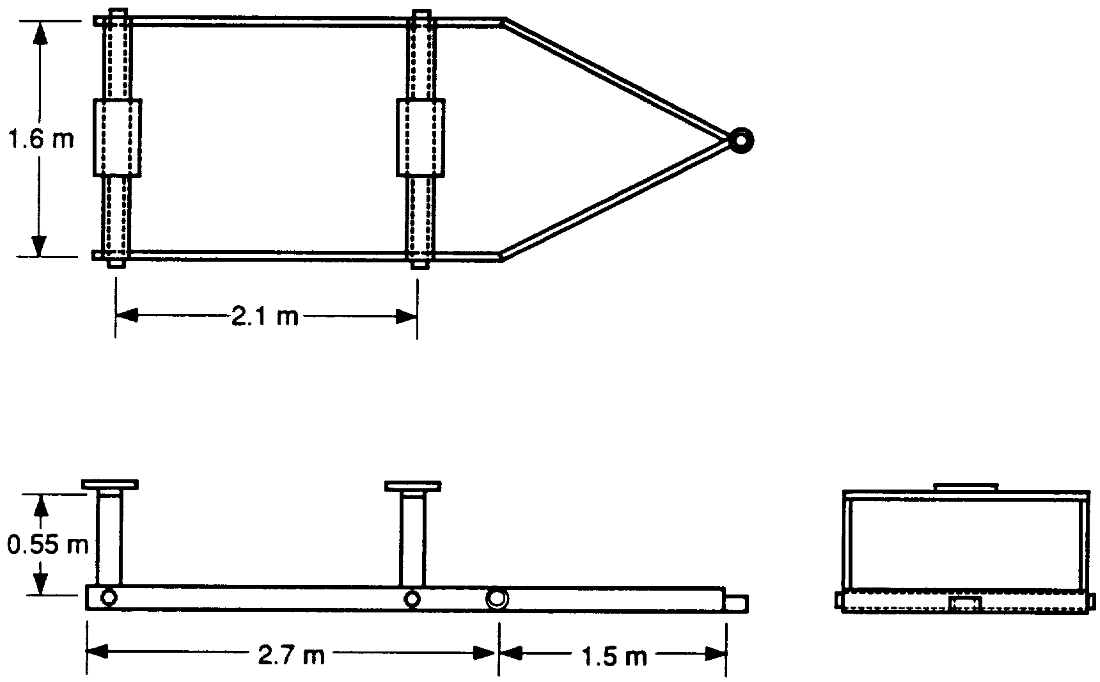
0.05 m
0.5 m R
1.1 m
1.55 m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