Georgia Tech

THE GEORGE W. WOODRUFF SCHOOL OF
MECHANICAL ENGINEERING

Georgia Institute of Technology
Atlanta, Georgia 30332-0405

ME 4182
MECHANICAL DESIGN ENGINEERING

NASA/UNIVERSITY
ADVANCED DESIGN PROGRAM

LUNAR DRILL
FOOTPLATE & CASING

MARCH 1989

Erik C. Maassen
Thomas H. Hendrix
Eddie W. Morrison
Rodrick B. Phillips
Vu Quang Le
Bruce A. Works

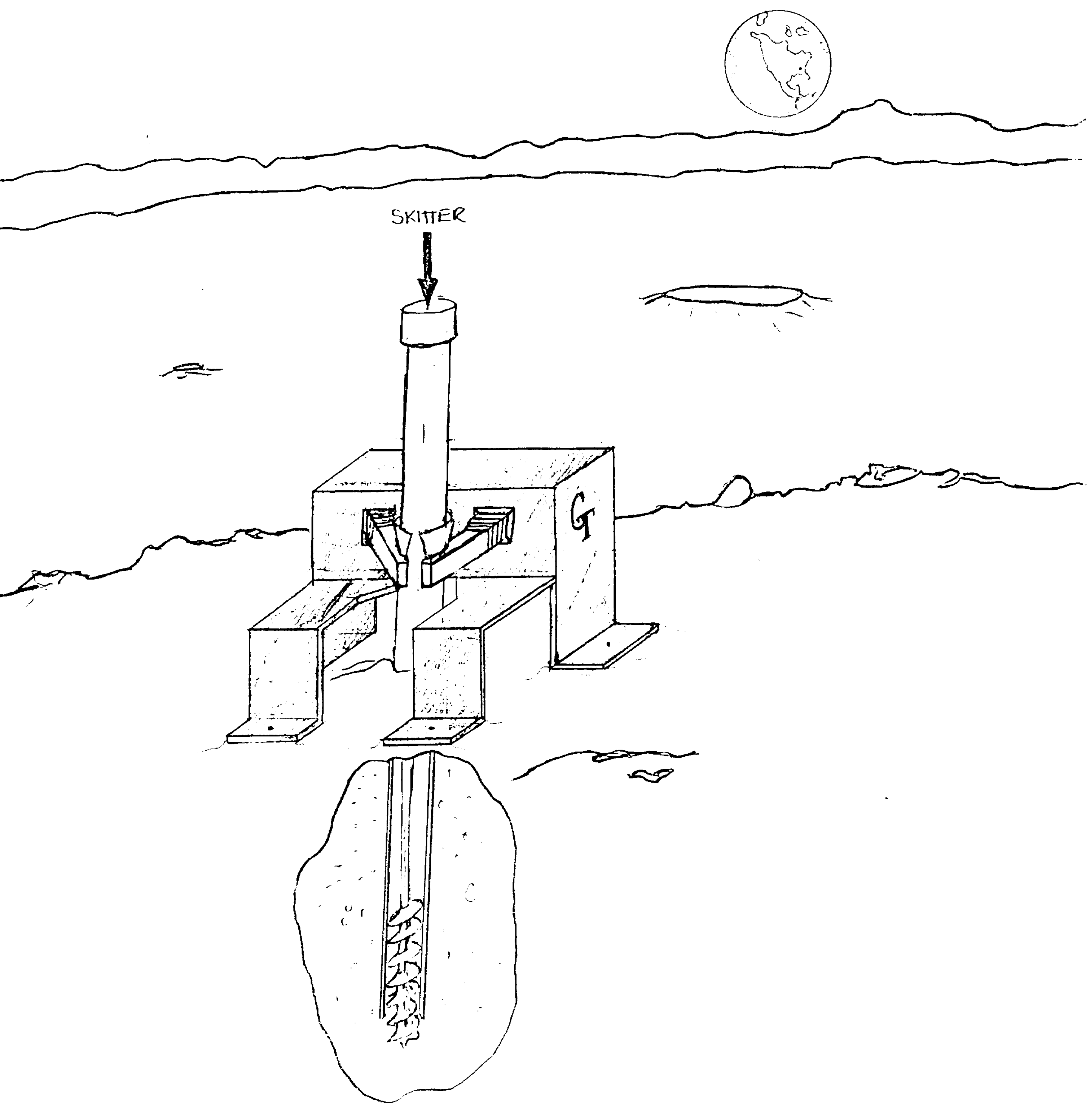

SKITTER
GT

TABLE OF CONTENTS

ABSTRACT

To prevent hole collapse during lunar drilling operations, a casing has been devised of a graphite reinforced polyimide composite which will be able to withstand the lunar environment. Additionally, this casing will be inserted into the ground in segments two meters long which will penetrate the regolith simultaneously with the auger. The vertical action of the mobile platform will provide a downward force to the casing string through a special adaptor, giving the casing the needed impetus to sink the anticipated depth of ten meters. Casing segments will be connected with a simple snap arrangement. Excess casing will be cut off by a cylindrical cutting tool which will also transport the excess casing away from the hole.

A footplate will be incorporated to grasp the auger rod string during rod segment additions or removals. The footplate grasping mechanism will consist of a set of vice-like arms, one end of each bearing threaded to a common power screw. The power screw will be threaded such that one end's thread pitch opposes that of the other end. The weight of the auger and rod string will be transmitted through the arms to the power screw and absorbed by a set of three ball bearing assemblies. The power screw will be driven by a one-half horsepower brushless motor actuated by radio control. The footplate will rest on four short legs and be anchored with pins that are an integral part of each leg.

PROBLEM STATEMENT

Future lunar operations will require a mobile drilling system for mining, geological research, and construction. Previous work at the Georgia Institute of Technology has resulted in a mobile platform design on which to mount various drilling apparatus. A typical drilling operation will require a casing in the hole to prevent hole collapse due to the loose nature of the lunar regolith. The drill will require rod extensions to reach desired depths of greater than two meters. Therefore, a means of supporting the drive bit will be needed.

The goal of this design group is to design a casing which will act in conjunction with the auger-bit assembly such that the hole may be dug and cased simultaneously. The casing must reach a depth of ten meters and allow for hole sizes of fifty and one hundred millimeters.

Complementing the design of the casing method will be a design of a footplate for the drilling apparatus. The objective of the footplate will be to stabilize and support the drilling string once each two meter section has been installed or removed.

Certain constraints must be accounted for in dealing with the environment of the lunar surface. Large temperature gradients, reduced gravitational effect, cost effectiveness of materials, and earth to moon transportation expenses are primary concerns.

DESCRIPTION

SYSTEM OPERATION

The system consists of two major components, the footplate and the casing. These two components work in close harmony to perform the objective.

Initial Casing

Once a drilling site has been chosen, the mobile platform will place and level the footplate in a manner to be determined. The gripper arms of the footplate will open wide enough so that the casing can be placed on the regolith by the rod changer. The rod changer will loosen its grip on the casing. The mobile platform will begin to move vertically pushing the casing into the regolith. Once the casing has penetrated the regolith 0.5 m, the gripper arms will close preventing the casing from falling over. The rod changer will then place the auger inside the first casing segment allowing it to rest on the regolith inside the casing. At this point the auger will rely on the casing to localize it. The chuck for the drive mechanism must locate the end of the auger that is protruding from the casing. Once the chuck has closed on the end of the auger, and footplate arms have retracted, the augering begins.

Augering

The mobile platform will lower while augering allowing the casing adaptor to contact the casing segment. Once the casing and adaptor have met, the mobile platform will push the rotating auger and the casing simultaneously into the regolith. The auger will be rotating at an optimal speed where the regolith will be suspended and moved up the auger flights, and the auger and rod string will not be violently slapping the inside of the casing. Once the mobile platform has pushed the casing below the arms of the footplate the augering will stop.

Removing the Auger

The mobile platform and the footplate will work in conjunction to remove regolith trapped in the casing flights from the hole. After the augering has stopped, the mobile platform will rise up exposing the entire rod to which the drive mechanism is attached as well as enough of the next rod to which the drive mechanism can attach. The gripper arms of the footplate will close grasping the rod string. The chuck of the drive mechanism will disengage from the exposed rod. The rod changer will remove the exposed rod from the rod string. The mobile platform will lower itself so that the chuck can attach on to the rest of the rod string. The gripper arms are retracted beyond the outer diameter of the casing. This procedure is repeated until the chuck attaches to the auger.

Clearing the Auger

The mobile platform must clear the auger of the regolith and restart the the augering process. To do this, the drive mechanism will spin the auger as it pulls the auger out of the hole, relying on centrifugal force to throw off most of the regolith. All but a thin film of regolith will be removed by a rapid up and down motion of the mobile platform. After the auger has been cleared, the mobile platform lowers the auger into the hole. The arms of the footplate close to grip the auger as the chuck disengages from the auger. The mobile platform moves up to allow for the rodchanger to attach a rod to the auger. The chuck reattaches to the rod. After the gripper arms retract releasing the rod string, the mobile platform lowers the auger and rod string. The footplate arms grip the rod string, and the attachment of another rod to the rod string is repeated until the number of rods in the string is one less than the number of casings.

Adding New Casing

Once the rods that have been previously removed are reattached, a new casing segment can be added. The chuck will release the rod string leaving the footplate to hold up the string. After the mobile platform rises out of the way, the rod changer will add another rod to the string. After this is added to the string, the rod changer with the help of the mobile platform will slide the new casing segment over the last rod. The casing will be released by the rod changer allowing the casing to rest in the casing guides and against the last rod. The drive chuck will then attach to the rod string. The Footplate arms will open slowly allowing the casing to slide down the casing guides aligning the new casing segment with the existing casing string. Once the casing guides have aligned the new casing with the existing casing, the mobile platform will squat forcing the casing segments to snap together. After the casing segments have joined, the mobile platform can begin rotating the auger and pushing the casing down until the auger is filled. Once the Auger is filled, the above procedure for clearing the auger is repeated.

Impeding Material

The casing and augering procedures will continue until one of several situations occur. If in the augering process the auger encounters material that causes it to bind or stop, the auger is to be removed and replaced by a rock bit. The rock bit will have a cutting range larger than the outside diameter of the casing. As with the auger, the rotating rock bit and casing will be pushed down at the same rate by the mobile platform. Once the rock bit has passed through this material, it will be replaced by the auger, or the drilling will stop.

Cutting the Casing

If the auger and casing have reached bed rock or a desired depth, any casing that is remaining outside of the hole must be removed. This task is accomplished by the casing cutter. Once the drilling has stopped at a particular site, the rod string and auger or rock bit will be disconnected from the drive. Then rod changer with the help of the mobile platform will slip the casing cutter over the remaining casing. Once the drive chuck has attached to the casing cutter and the cutting blades have been extended, the cutting can begin. The casing is cut by rotating the casing cutter with the blades extended. Once the casing has been cut, the mobile platform will rise up leaving the blades extended removing the excess casing. This excess casing can be removed from the cutter by retracting the blades of the casing cutter. The excess casing will then drop from the cutter, and the cutter will be returned to storage.

CASING

First Casing Segment

Similar to the other segments, the majority of the first segment is fabricated with graphite reinforced polyimide. The smaller casing, 50 mm in inner diameter, 0.4 mm thick, and 1.5 m long, will be strengthened at the tip by a 3 mm angled tip of titanium carbide. They will be joined by a 0.55 long half-lap with a polyimide adhesive. As for the 100 mm inner diameter, 0.75 mm thick, and 1.5 m long casing segment, the half-lap bonding length will require 0.6 mm of polyimide adhesive.

Casing Segments 2-5

The 50 mm ID casing will be 2.0135 m long. The casing will have a wall thickness of 0.4 mm for all sections except joining sections at either end. The casing will be fabricated of a graphite reinforced polyimide composite. The joining method of the casing consists of twelve tabs spaced equally around the circumference of the casing that will flex enough to allow the following section to slide down and be joined as seen in Figure 1.

The 100 mm ID casing will be 2.0246 m long. The casing will have a wall thickness of 0.4 mm for all sections

Interfaces

A casing needs to interface with three other components: the mobile platform, the footplate, and another casing segment. The mobile platform provides a downward force through the casing adaptor. The casing adaptor is fabricated from graphite reinforced polyimide. A conceptual design is illustrated in Figure 2. The drive mechanism operates through the adaptor as shown in the figure. The mobile platform will provide a maximum downward force of 1.33 kN for 0.4mm thick casing and 2.43 kN for the 0.75mm thick casing through a bearing area of 57.1mm^2 and 113.6mm^2, respectively.

When joining the casing, the female end of a new casing will be directed by the footplate's conical casing guide to the male end of the casing, buried in the ground. Figure 3 illustrates the joining ends and their dimensions.

Casing Cutter

The casing cutter consists of a shell and a blade assembly. The shell has a 2 cm clearance over the outside diameter of the casings, a 1 mm thickness and a 2 m length. One end of the shell is tapered to form a rod which has a diameter of 25 mm. The other end will provide a base support for the blade assembly. The blade assembly will have the following components, see Figure 4.

* Two titanium carbide cutting blades inserted in a high strength steel alloy support.
* Two flat springs to which the alloy supports and the blades are mounted and which will provide a feedrate of 5.08 μm per revolution. The unloaded displacement of the spring will be 3 cm from the inner wall of the shell.

* Two radio controlled solenoids powerful enough to retract the blades so that the casing cutter can be slid over the excess casing.
* One battery to provide power to the selenoids.
* Two rollers.

The drive mechanism is assumed to provide a angular speed of 500 RPM. With this, the cutting time expected for the 0.4 mm thick casing is 9.45 seconds and for the 0.75 mm thick casing is 17.72 seconds.

FOOTPLATE

The footplate provides a stable platform by which the rod string may be supported during rod segment addition or removal. The rod string is held in place when disengaged from the main drive unit by a set of gripping arms driven by a single, opposed pitch powerscrew (See Figure 5. The ends of the gripping arms are gnarled for better gripping action. Guides above gripping surfaces direct the casing during casing segment assembly. Footplate stability is enhanced by a stinger attached to each of the four feet. These stingers act as stakes, limiting footplate lateral and vertical motion.

Footplate operation is designed to function remotely. The powerscrew is driven by a battery powered motor actuated remotely via radio control. It will work in conjunction with the walking mobile platform and other associated drilling equipment.

Base Assembly

The base assembly is provided to support and protect the gripper arms, gripper drive unit, power supply and rod string when engaged with gripper arms. The area of ground covered by the footplate is approximately 360 mm^2. As shown in Figure 4, the footplate has a U-shape which lends adequate support and reduces the bulk of the assembly. The support frame will be comprised of 3.18 mm thick Aluminum-Silicon-Magnesium cast alloy channels. A 3.18 mm thickness will provide a shear stress at 400 C of 1.23 MPa, assuming the maximum rod string weight (445 N) is supported by a single channel. The material shear strength is calculated to be 16 MPa, resulting in a safety factor of 13. This provides an adequate ruggedness to protect against unexpected events such as falling debris from the auger.

Housing

The housing of the footplate will be an Aluminum-Silicon-Magnesium cast alloy. Individual 3.18 mm thick plates will be attached to the skeletal structure of the plate using flat head machine bolts of the same material. See Figure 6. The top plate will be attached in a similar manner but replacing the flat head bolts with a machine cap bolt for ease of removal with a ratchet when general maintenance or repairs are required.

Bellows

The gripper arms will penetrate the footplate housing through a bellows assembly which will allow for radial and lateral movement of the arms while providing foreign matter exclusion protection to the components within the housing. The bellows will consist of 5 convolutions of 32 mm inside diameter and 38 mm outside diameter resulting in a span of 3 mm. The bellows free length will be 35 mm, adapted on one end to taper a length of 10 mm to the arm body. The bellows will be attached to the gripper arms and housing by flat head machine bolts. The bellows material will be of a high nickel alloy such as Mech metal with properties maintained in temperature ranges of -350 C to 250 C. See Figure 7 for the bellows arrangement.

Legs/Stabilizers

The base assembly will be supported by four legs, 100 mm tall. This elevation will allow the gripper arms to clear the pile-up of soil which will form by auger expulsion. Each leg will be stabilized by a footpad of 2500 mm^2. To prevent footplate sliding and retard vertical motion, each footpad bottom will be fitted with 100 mm long stabilizing stingers. The entire length of the stinger will be cast from Aluminum-Silicon-Magnesium alloy like the rest of the base assembly. This alloy provides good mechanical strength, light weight, and good casting characteristics.

Powerscrew

The mechanics of the footplate operate around the powerscrew. Emphasis was therefore placed on the material selection for this component. 44M2 high carbon high chromium steel was selected as a sufficient material in regards to environmental considerations and practicality. The powerscrew will have 5 threads per mm, and a helix angle of 7.25 degrees. A outer diameter (d) of 15 mm, major diameter (d_m) of 12.5 mm, and a radial diameter (d_r) of 10 mm will govern the size of the powerscrew. The power screw rests on 3 anti-friction ball bearings placed one at each end and one in the center.

Gripper Arms

The arms of the footplate will be manufactured from 44M2 high carbon, high chromium steel. The thermal expansion coefficient of this material is average relative to other materials. Allowing the powerscrew, roller nut, and arms all to be cast from the same material decreases the chance of any binding or play occurring due to high and low temperatures seen on the lunar surface.

The arms will be 200 mm in length, have a cross sectional area of 435 mm^2 and connect to the powerscrew via a roller ball nut. The arms are allowed to pivot in one plane about this nut. This allows all movement to continue along the center line of the arm and guards against binding at the nut/screw interface.

The gripper arms pivot on points 30 mm from the front of the footplate housing, 100 mm apart from one another. The pivot pins are 10 mm in diameter constructed of the same material as the arms. They are mounted in slots measuring 10 mm by 13 mm to allow for small forward motion of the arms as they move from a closed configuration to an open configuration. Thrust washers are between the gripper arms and both the top and bottom mounting braces of the footplate housing.

To grip the rod, a partial semicircular plate will also be cast with the arms. The radius of this plate depends upon the size of the drill rod. To aid in the gripping of the rods, the inside of the plate will be gnarled. This is to take advantage of the high coefficient of friction resulting from metal to metal contact in a vacuum. A nominal height of 25 mm was chosen for the plates.

Casing Guides

Joined to the gripping end of the footplate arms will be two connical plates. These plates will be cast as an integral part of the gripping plates. The connical geometry allows the casing to slide down the plate as the arms are being retarcted. This methods brings the casing joining segments together in a more gentile manner reducing the potential of brealing the joining tabs.

Ball Bearings

Three Series 02 anti-friction ball bearings will support the powerscrew and aid in its rotational motion. These bearings will be located one at each end of the powerscrew and one in the middle. The bore of the bearing is 15 mm which corresponds with the outer diameter of the powerscrew.

Ball Bearing Screw/Nut Assembly

The Lunar Drill Footplate Device will have a linear actuator to provide the necessary force for the clamp mechanism. The actuator will be a power transmission screw utilizing a ball bearing nut assembly, see Figure 8. The balls circulate in hardened helical races and roll relative to the screw and nut and the forces are transmitted only through the balls themselves because there is no direct contact between the screw and the nut. The actuator will have both left-handed and right-handed threads connected together to provide the clamping action as the screw is rotated and also the nuts are self-locking. Bearing for the shaft will be provided at the center point and at both ends. The end bearings will also be bear axial thrust load. The bearings, screw, balls, and nuts will all be constructed of a high carbon, high chromium 44M2 steel alloy. At a shaft speed of 200 rpm, the clamp will go from an open configuration to a closed configuration in 1.5 seconds. Life expectancy of these actuators at the relatively low loads placed upon them will be in the neighborhood of 254,000 meters of travel.

ANALYSIS

CASING

Frictional Force

First, the casing was designed to avoid buckling under the worst case. The worst case occurred when the fifth casing segment was about to enter the regolith, because once the casing has entered the lunar surface the lateral support of the soil will keep the casing from buckling under greater loads. The maximum force on the casing was calculated using a surface friction force equation [1] found in the NASA Apollo 15 Preliminary Science Report with three assumptions. The first assumption was that the regolith interacted with all materials considered in the same manner. This prevented constant iteration to find out if the original tan(∂) value was correct. The second assumption was that the suspended soil between the flights of the auger created a lateral soil pressure equivalent to that of the soil on the exterior of the casing. The third assuption is that the wall thicknesses for the 100 mm ID and 50 mm ID casings are 5 mm and 2.5 mm, respectively. With these three assumptions, the frictional force acting on the casings was calculated as 4.86 kN and 2.43 kN for the 100 mm ID and the 50 mm ID, respectively which can be seen in Appendix A. From this result another assumption follows. The mobile platform must be able to exert a vertical force in the downward direction of at least 5 kN.

Material Selection

Several criteria go into the material selection process of the casing. First and foremost is the weight. This restriction dictates what material is chosen in the end, since the cost of transporting the equipment to the moon is expected to be quite high, about $55,134/kg. The cost of the two meter segment itself is only a small fraction of this, no matter what material is used. Therefore, the lightest material gets the highest priority.

The casing needs to be light and strong resisting buckling under eccentric loading. When the casing is in the regolith, it is not expected to buckle due to the side support along the outside circumference of the casing. Buckling for a given load would occur first when the casing is above the ground. For this reason, a two meter segment prior to entering the soil is considered for possible buckling. The analysis given by Griffel is used to calculate the minimum buckling stress under eccentric loading. The eccentric load was arbirarily chosen to occur at the inner diameter of the casing, since the outer diameter had not been calculated yet. This stress is compared to that of the material's compressive yield strength. A factor of safety of five was applied. Here obviously a strong and light casing is preferred.

Another criteria in material selection for the casing is thermal expansion. Due to the large temperature gradients that exist on the lunar surface, stresses and strains will develop in the casing from expansion and contraction. Excessive thermal expansion is not desirable, since it decreases the strength of the material. Therefore, a material whose properties are affected little by large temperature gradients is needed.

Another important casing consideration is flexibility, since the snap joining method requires a material that can not only bend without breaking, but is strong and light. Further, when the hole has been cased to the desired depth, the excess casing must be removed. The disposing method requires cutting, so this material must be soft enough to be cut easily requiring little power.

After a detailed analysis of several materials, the graphite reinforced polyimide was chosen for the casing. All materials considered failed the worst-case buckling load criterion except graphite reinforced polyimide. Most metals and their alloys would have no problems if larger thicknesses were considered, but they are relatively dense. Most polymers would melt in the extreme temperatures experienced on the lunar surface. High strength composites lack the flexibility required for the joining method. However, since density is a high priority, a high temperature polymer composite of graphite reinforced polyimide is chosen. This material is strong enough to resist buckling under the worst case but is flexible enough to bend for the joining requirements. The material can be cut relatively easily allowing a long cutting tool life. The thermal expansion of the material is small enough to be insignificant and the material retains its properties up to 315 C. Overall, this material meets all of the requirements with a minimal cost.

First Casing Segment

A strong, heat and wear resistant material, titanium carbide, is chosen to accommodate for the initial impact due to the force applied to the casing. This force is also reduced by the tip angle of 30 degrees, allowing easier penetration into the soil. The inward direction of the lip allows for better soil removal. This tip is bonded with a poyimide adhesive in a half-lap joint to reduce the compression stress by loading the shear stress on the bond area, since adhesives perform better in shear than in compression.

The length of this segment is 1.5 m instead of 2 m. This accounts for the 0.25 m lead the auger needs in the regolith and the 0.25 m above the ground clearance required by the footplate.

<u>Casing Joint</u>

The casings were design to snap together under a force equivalent to the weight of a casing segment multiplied by a Friction Effect coefficient. This force was assumed to be distributed evenly about the circumference of the casing. The Friction Effect term in the calculations is an arbitrary value which is large enough take into account the added force needed to overcome the friction between the leading edge of the tab and the other casing. The term also accounts for any additional force that the mobile platform must exert to deflect tabs of reasonable dimensions.

In order to make the analysis of the deflection of the tabs within the ability of the design group, the tabs were modeled after a wide cantilevered beam. This analysis ignors the radius of curvature of the tabs. With a Friction Effect coefficient of 20, the number of tabs and the tab dimensions were found through iteration, and the results can be seen in appendix A.

Casing Adaptor

Since simplicity is one of the main goals, the mobile platform will be utilized to provide the necessary downward force on the casing. This force is transmitted to the casing through the adaptor conceptualized in Figure 2. This adaptor will be made out of the same material as the casing, namely graphite reinforced polyimide, to minimize the effect of thermal expansion. The adaptor has an inner diameter of 51.2mm, and 102mm for the 50mm and 100mm casings, respectively. These diameters provide enough clearance for the tab, so that the adaptor pushes on the bearing surfaces of the casing.

Casing Cutter

Compatibility and simplicity forces one to use a shell type casing cutter. This design allows for easier storage and easier use. Utilizing the drive mechanism and the displaced spring force, the titanium carbide blades will be used to cut the 0.4 mm thick casing in 9.45 seconds and the 0.75 mm thick casing in 17.72 seconds. All this is accomplished with the assumption that this tool can be manufactured to operate at specified characteristics, described above. Further study and analysis is limited due to time constraints.

Cost

Not surprisingly, the cost of these casings are astounding. Due to the high cost of transportation to the moon, the 50mm casing will cost about $60,000.00 per two meter segment each, while the 100mm casing segments will cost about $220,000.00 each. Together with the casing cutter, the total cost will come to approximately $400,000.00 for the 50mm casing setup, and $1,300,000.00 for the 100mm casing setup.

FOOTPLATE

Powerscrew

A load normal to the powerscrew was calculated to be approximately 1135 N. This was determined using a coefficient of friction (between the rod and gripper arms) of 0.3, an approximation based on smooth metals in contact in a vacuum, and coefficients of friction found on earth. Using a pitch value of 5 mm, a coefficient of friction of 1 (between powerscrew and roller nut), and an outside diameter of 15 mm, the helix angle for this configuration is 7.25 degrees. The bearing stresses seen on the threads was determined to be 1.45 MPa. This satisfies our requirement for clamping with a factor of safety of 517. These calculations can be found in Appendix A.

Gripper Arms

To choose a material for the gripper arms, a static force balance was used. A deflection of 1 mm was assumed and from previously defined arm geometry and load characteristics, a minimum modulus of elasticity was deduced. A satisfactory material for the arms is 44M2 high carbon high chromium steel, which a modulus of elasticity of 207 GPa. Computation of the load required for a deflection greater than 1 mm is 10,273 N. See Appendix A for calculations.

Ball Bearings

Selection of bearings resulted using Timken Engineering Journal standards. A design life of 200 rpm at 1000 hours use and 99% reliability of the bearing results in a load rating of 1313 N. Referring to the Timken Charts we find that a 15 mm bore anti-friction bearing can withstand a load of 5870 N. With this load, a factor of safety of 4.5 results. See Appendix A for calculations.

Ball Bearing Screw/Nut Assembly

The rollerball bearing screw actuator will provide the force and motion for the Lunar Drill Footplate. A screw type actuator is used because of the ease of converting rotary power (from a motor) to the linear action needed for the footplate. The high coefficient of sliding friction for metal to metal contact in a vacuum eliminated the consideration of using a sliding screw and nut assembly. The ball bearing screw provides the rolling efficiency of the balls in place of the inefficient sliding action of the conventional power screw threads. Efficiency of the roller ball actuator is close to 90% whereas the efficiency of the screw is only about 30%. The major diameter of the shaft is 15 mm, the minor (root) diameter is 12.46 mm and the pitch of the thread is 2 threads per centimeter. The shaft will be 200 mm long, the nut 40 mm long and at a shaft speed of 200 rpm the clamp will go one half-cycle (open to shut) in 1.5 seconds. The helix angle of the thread is 7.25 degrees and with a coefficient of friction over .13, the nut will be self locking. The power needed at the shaft to clamp the rod string is .028 Horsepower or 20.88 watts. The screw, nut, and ball bearings will be constructed of a high carbon, high chromium 44M2 steel alloy commonly used in the production of bearings. The power screw and bearing nuts will weigh a total of .874 kg on Earth. This power screw assembly is very durable and reliable. Its life expectancy ensures almost no chance of failure of this component as long as the threads and balls are kept lightly lubricated.

Weight

The components cast of Aluminum-Silicon-Magnesium components comprise 49% of the total weight of the footplate. These components include the skeletal frame and housing of the footplate. Another 35% of the weight is found to lie in the powerscrew, ball bearings, and rollerball bearing screw, each of which are made up of 44M2 high carbon high chromium steel. The remaining 16% of the weight is accounted for in the power supply of the footplate. This includes batteries and a maximum 1/2 horsepower motor used to power the footplate. The total terrestrial weight of the footplate is 6.41 kg. Appendix A contains a breakdown of these calculations.

Cost

See Appendix A for footplate cost summary. Based on a 6.4 kg total earth weight for the entire footplate assembly, cost for transportation to the moon will be approximately $350,000. Construction cost should be approximately $3500, 1% of the transport cost.

CONCLUSION

The preceding report concerning footplate and casing concepts provides a reliable means for lunar drilling operations. These designs are very compatible with existing mobile platform technology.

The casing is not only lightweight, but exhibits unsurpassed durability. The method suggested for casing a hole simultaneously with auger operation is uncomplicated and requires no additional drive mechanisms. The casing operation is further simplified by the clever casing segment joining method. The scheme for cutting away excessive casing works well within the given capabilities of the mobile lunar platform as well as providing a method for evacuating casing excess from the drill sight.

The footplate provides a simple platform of few moving parts. The materials selected for construction allow for design limits with very conservative safety margins. The given structure allows for rough usage with a long life. The problem of foreign matter contaminating the footplate mechanism is overcome via the housing and metal bellows arrangement. An added benefit to the given design is the ease in which maintenance can be performed on the unit.

For the above reasons, the design team responsible for this report recommend the given footplate and casing as a practical solution to the problem of rod string support and hole collapse.

RECOMMENDATIONS

CASING

Due to the limited time and lack of concrete information, many assumptions were made as previously noted. Further study and research are needed to provide better estimates for the design criteria. The following lists several recommendations to further improve the design.

1) Lunar soil experiment to study its behavior and characteristics at depths up to 150 m.

2) Further study is needed to investigate the casing cutter design to meet specified criteria.

3) Investigations are needed in possible fabrication techniques for the casing to provide specified material properties.

4) Further tests are needed to determine the power requirement to cut the casing.

5) Further investigation is needed to determine if the friction created when pulling up the auger will cause the casing to be pulled up as well.

6) Further investigation is needed to determine if exposing the joining tabs is more beneficial than exposing them to the casing adaptor. This would occur if the casing were inserted.

7) Further investigation is needed to determine if the joining tabs can be modeled as a wide cantilevered beams.

FOOTPLATE

Following is a list of recommendations for further study into the footplate design:

1) Resistance added to stingers of footplate feet in the form of screw anchors to provide extra security of footplate to the lunar soil.

2) Since cost analysis indicates that footplate construction cost is insignificant compared to the transportation costs, future studies should investigate ways to reduce footplate weight to significantly reduce the overall cost.

3) Internal footplate lubrication at this time seems feasible using vacuum pump-type lubricants. However, more in depth studies need to be conducted in this area.

4) Due to the limited background knowledge of the design team in the area of motor design and operation, further research is required for an adequate power screw driver.

ACKNOWLEDGEMENT

The casing and footplate group would like to gratefully thank the following people who provided us with valuable information and insight into the problem at hand.

Mr. James W. Brazell, Mechanical Engineering, Georgia Tech.

Mr. Kirk Smith, Precision Drilling, Marietta, Georgia.

Dr. Robert Bachus, Civil Engineering, Georgia Tech.

Dr. Ashok Saxena, Materials Engineering, Georgia Tech.

BIBLIOGRAPHY

Brantly, John Edward. History of Oil Well Drilling. Gulf Publishing Company: Houston, 1971.

Defrayne, Garry. High-Performance Adhesive Bonding. Society of Manufacturing Engineers Marketing Services Department: Dearborn, Michigan, 1983.

Dieter, George E. Engineering Design: A Materials and Processing Approach. McGraw-Hill Book Company: New York, 1983.

Espe, Werner. Materials of High Vacuum Technology. Pergamon Press: Oxford, 1966.

Fielder, G. Geology and Physics of the Moon. Elsevin Publishing Company: New York, 1971.

Griffel, William. Handbook of Formulas for Stress and Strain. Frederick Ungar Publishing Company: New York, 1966.

Gunia, Russel B. Source Book on Materials Selection. Vol I and II. American Society for Metals: Metals Park, Ohio, 1966.

Jeffries, William. Tool Design. Prentice-Hall, Inc.: New York, 1955.

Kane, George. Modern Trends in Cutting Tools. Society of Manufacturing Engineers: Dearborn, Michigan, 1982.

Lee, Johnathan A. and Mykkanen, Donald L. Metal and Polymer Matrix Composites. Noyes Data Corporation: Park Ridge, New Jersey, 1987.

Lindsey, John. Lunar Stratigraphy & Sedimentology. Elsenvier Scientific : Amsterdam, 1976.

Lyndon B. Johnson Space Center. Apollo 15: Preliminary Science Report. National Aeronautics and Space Administration: Washington D. C., 1972.

Miner, Douglas F.and Stone, John B. Handbook of Engineering Materials. John Wiley & Sons Inc.: New York, 1955.

Mutch, Thomas. Geology Of The Moon: A Stratigraphic View. Princeton University: Princeton, 1970.

-----. Apollo 17: Preliminary Science Report. 1973.

Parker, Earl R. Materials Data Book. McGraw-Hill Book Company: New York, 1967.

Register, Bridget Mintz. "The Fate Of The Moon Rock". Astronomy: New York, 1985.

Ross, Robert B. Metallic Materials Specification Handbook. E & FN Spon Ltd : London, 1972.

Rotkiewicz, J.A. Power Transmission Design Handbook. Michigan, 1981.

Schwartz, M. M. Composite Materials Handbook. McGraw-Hill Book Company: New York, 1984.

Shigley, J.E. and Mitchell, L.D. Mechanical Engineering Design. McGraw Hill : New York, 1983.

Swinehart, Haldon J. Cutting Tool Material Selection. American Society of Tool and Manufacturing Engineers: Dearborn, Michigan, 1968.

APPENDICES

APPENDIX A

CALCULATIONS

1. Casing Worst Case Buckling
2. Casing Thickness Evaluation, 50mm
3. Casing Thickness Evaluation, 100mm
4. Joining Dimensions, 50mm & 100mm
5. Tab Deflection Analysis
6. Casing Thermal Stresses
7. Bonding Length, 50mm & 100mm
8. Cutting Times for 50mm & 100mm
9. Bearing Thicknesses
10. Powerscrew
11. Bearings
12. Gripper Arms
13. Torque/HP
14. Base Plate Stress
15. Weight Analysis
16. Cost Analysis

FRICTION FORCE FOR WORST CASE BUCKLING

$$f_s = \frac{\gamma K D \tan\delta}{2} \qquad [1]$$

$\gamma \equiv$ SOIL WEIGHT (ρg)

$D \equiv$ DEPTH

$K \equiv$ LATERAL EARTH PRESSURE $(\simeq 0.5)$

$\delta \equiv$ FRICTION ANGLE BETWEEN THE SOIL

$\rho \equiv$ DENSITY $(1500\ kg/m^3)$

$g \equiv$ GRAVITY $\left(\left(\frac{1}{6}\right)(9.8\ m/s^2)\right)$

$\tan\delta \equiv$ FRICTION COEFFICIENT BETWEEN THE SOIL AND THE CASING $(\simeq 0.5)$

$$\gamma = \rho g$$
$$= (1500\ kg/m^3)(1/6)(9.8\ m/s^2) = 2.45\ kN/m^3$$

$$f_s = \frac{(2.45\ kN/m^3)(0.5)D(0.5)}{2} = 0.306\ kN/m^3\ D$$

ASSUMPTION(S): THE CASING IS MOST VULNERABLE TO BUCKLING WHEN SKITTER IS PUSHING THE LAST CASING SEGMENT INTO THE REGOLITH. THE FORCE SKITTER MUST OVERCOME IS THE FRICTION FORCE OF THE SOIL/CASING INTERFACE. THIS FORCE ACTS ON THE OUTSIDE OF CASINGS ONE THROUGH FOUR AND ON THE INSIDE OF CASING ONE.

— THE OUTSIDE DIAMETER WAS ASSUMED TO BE 110 mm.

INTEGRATE SURFACE FRICTION OVER THE ENTIRE LENGTH OF CASING EXPOSED TO THE REGOLITH.

ORIGINAL PAGE IS OF POOR QUALITY

FRICTION FORCE FOR WORST CASE BUCKLING (CONTINUED)

$$F_{S100} = \int_{\text{SURFACE}}^{\text{BOTTOM}} \pi(\text{DIAMETER})(f_s)\,D\,dD + \int_{\text{TOP OF 1ST CASING}}^{\text{BOTTOM OF 1ST CASING}} \pi(\text{DIAMETER})(f_s)\,D\,dD$$

$$= \int_0^8 \pi(.110\,m)(0.306\,kN/m^3)\,D\,dD + \int_6^8 \pi(0.110\,m)(0.306\,kN/m^3)\,D\,dD$$

$$= \frac{\pi(0.110\,m)(0.306\,kN/m^3)}{2}\left[\frac{D^2}{2}\Big|_0^8 + \frac{D^2}{2}\Big|_6^8\right]$$

$$= \frac{\pi(0.110\,m)(0.306\,kN/m^3)}{4}(8^2 + 8^2 - 6^2)$$

$$= 2.43\ kN$$

NOTE: AT FIRST F_c WAS CALCULATED INCORRECTLY RESULTING IN A VALUE OF 1.15 kN. A SAFETY FACTOR OF 5 WAS [illegible] BUCKLING ANALYSIS WHICH WOULD INCREASE THE INCORRECT F_c WELL ABOVE THE CORRECTED F_c FOR A SAFETY FACTOR OF $\frac{5(1.15)}{2.43}$ OR 2.37.

$$F_{S50} = \int_0^8 \pi(0.060\,m)(0.306\,kN/m^3)\,D\,dD + \int_6^8 \pi(0.060\,m)(0.306\,kN/m^3)\,D\,dD$$

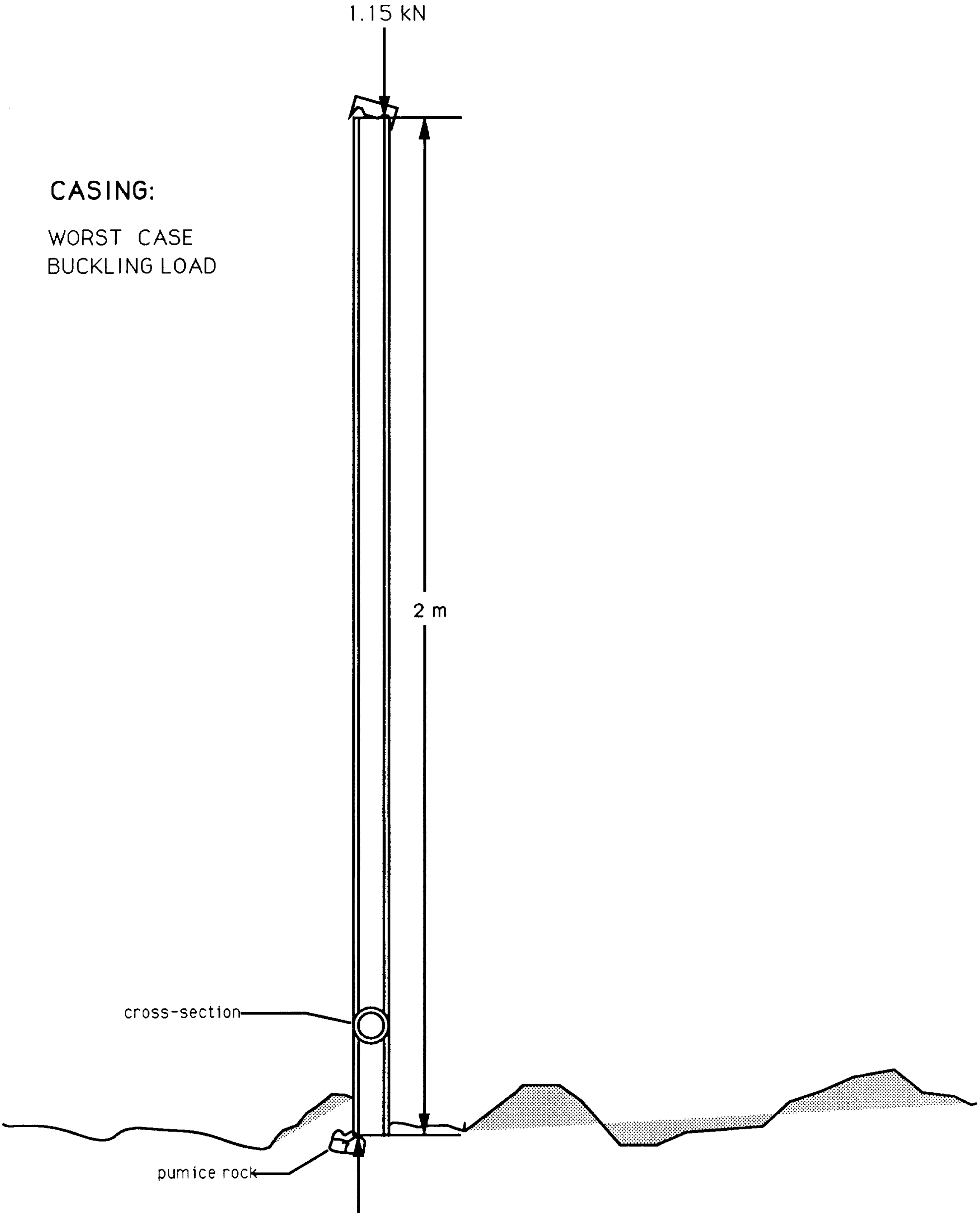
$$= \frac{\pi(0.060\,m)(0.306\,kN/m^3)}{4}(8^2 + 8^2 - 6^2)$$

$$= 1.33\ kN$$

ORIGINAL PAGE IS OF POOR QUALITY

CASING:

WORST CASE
BUCKLING LOAD



ME 4182 EVALUATION OF CASING THICKNESS WITH ECCENTRIC BUCKLING CONDITIONS
GROUP 5 VL-EM-RP-TH-BW-EM

CONSTANTS ANALYSIS FOR 50mm CASING

Pcr =	1235	N
I.D. =	0.05	m
length =	2	m
ecc =	0.025	m
T Cost=	55134.39	$/kg

MATERIAL	Y Modulus (Pa)	Scy (Pa)	Density (kg/m^3)	Temp. Req	Scy-stress (must>0)	THICKNESS t (m)	l*lambd	(M/Pe) *	Dmean (m)	I (m^4)	M (N-m)	area (m^2)
COPPER	1.17E+11	6.90E+07	9190	YES	-2.73E+07	0.0004000	1.448	1.00	0.050	2.01E-08	30.88	6.33E-05
SiC RIF ALUMINUM	8.04E+10	1.66E+08	2850	YES	6.92E+07	0.0004000	1.748	1.00	0.050	2.01E-08	30.88	6.33E-05
BORON RIF ALUMINUM	1.66E+11	1.86E+08	2630	YES	8.97E+07	0.0004000	1.218	1.00	0.050	2.01E-08	30.88	6.33E-05
PMR RIF POLYIMIDE	1.00E+11	1.28E+08	1938	YES	3.17E+07	0.0004000	1.567	1.00	0.050	2.01E-08	30.88	6.33E-05
ALUMINUM	6.90E+10	1.03E+08	2710	YES	6.74E+06	0.0004000	1.887	1.00	0.050	2.01E-08	30.88	6.33E-05
PVC	2.41E+09	1.03E+08	1537	NO	8.35E+07	0.0004000	10.089	--	0.050	2.01E-08	0.00	6.33E-05
FIBER GLASS	1.03E+10	1.03E+08	1926	YES	-4.70E+07	0.0004000	4.874	1.70	0.050	2.01E-08	52.49	6.33E-05
GRAPHITE RI POLYIMIDE	1.00E+10	2.55E+08	1660	YES	9.73E+07	0.0004000	4.956	1.80	0.050	2.01E-08	55.58	6.33E-05
MG ALLOY (AZ31B)	4.48E+10	1.66E+08	1770	YES	6.97E+07	0.0004000	2.342	1.000	0.050	2.01E-08	30.88	6.33E-05

This analysis derived from Griffel's Handbook of Formulas for Stress and Strain, pp. 209-211.

ME 4182 EVALUATION OF CASING THICKNESS WIT
GROUP 5 VL-EM-RP-TH-BW-EM

CONSTANTS
Pcr =
I.D. =
length =
ecc =
T Cost=

MATERIAL	Stress (Pa)	Mass (kg)	Mat Cost ($/kg)	Total Cost ($/ 2m seg)
COPPER	9.63E+07	5.82	0.55	320909.67
SiC RIF ALUMINUM	9.63E+07	1.81	10.36	99538.12
BORON RIF ALUMINUM	9.63E+07	1.67	134.92	92061.95
PMR RIF POLYIMIDE	9.63E+07	1.23	130.00	67832.77
ALUMINUM	9.63E+07	1.72	1.00	94632.46
PVC	1.95E+07	0.97	0.70	53671.33
FIBER GLASS	1.50E+08	1.22	1.50	67256.00
GRAPHITE RI POLYIMIDE	1.58E+08	1.05	22.05	57988.88
MG ALLOY (AZ31B)	9.63E+07	1.12	8.81	61816.67

ME 4182 EVALUATION OF CASING THICKNESS WITH ECCENTRIC BUCKLING CONDITIONS

GROUP 5 VL-EM-RP-TH-BW-EM

CONSTANTS

Pcr = 5759.1 N

I.D. = 0.1 m

length = 2 m

ecc = 0.05 m

T Cost= 55134.39 $/kg

ANALYSIS FOR 100mm CASING

MATERIAL	Y Modulus (Pa)	Scy (Pa)	Density (kg/m^3)	Temp. Req	Scy-stress (must>0)	THICKNESS t (m)	l*lambd	(M/Pe)	Dmean (m)	I (m^4)	M (N-m)	area (m^2)
COPPER	1.17E+11	6.90E+07	9190	YES	-1.47E+08	0.0007500	0.808	1.00	0.100	3.01E-07	575.91	2.37E-04
SiC RIF ALUMINUM	8.04E+10	1.66E+08	2850	YES	-5.00E+07	0.0007500	0.975	1.00	0.100	3.01E-07	575.91	2.37E-04
BORON RIF ALUMINUM	1.66E+11	1.86E+08	2630	YES	-2.95E+07	0.0007500	0.680	1.00	0.100	3.01E-07	575.91	2.37E-04
PMR RIF POLYIMIDE	1.00E+11	1.28E+08	1938	YES	-8.75E+07	0.0007500	0.875	1.00	0.100	3.01E-07	575.91	2.37E-04
ALUMINUM	6.90E+10	1.03E+08	2710	YES	-1.12E+08	0.0007500	1.053	1.00	0.100	3.01E-07	575.91	2.37E-04
PVC	2.41E+09	1.03E+08	1537	NO	-3.99E+08	0.0007500	5.630	2.500	0.100	3.01E-07	1439.78	2.37E-04
FIBER GLASS	1.03E+10	1.03E+08	1926	YES	-1.12E+08	0.0007500	2.720	1.00	0.100	3.01E-07	575.91	2.37E-04
GRAPHITE RI POLYIMIDE	1.00E+10	2.55E+08	1660	YES	3.95E+07	0.0007500	2.765	1.00	0.100	3.01E-07	575.91	2.37E-04
MG ALLOY (AZ31B)	4.48E+10	1.66E+08	1770	YES	-4.95E+07	0.0007500	1.307	1.000	0.100	3.01E-07	575.91	2.37E-04

This analysis is derived from Griffel's Handbook of Formulas for Stress and Strains, pp. 209-211.

ORIGINAL PAGE IS
OF POOR QUALITY

ME 4182 EVALUATION OF CASING THICKNESS WITH ECCENTRIC

GROUP 5 VL-EM-RP-TH-BW-EM

CONSTANTS

Pcr =

I.D. =

length =

ecc =

T Cost=

MATERIAL	Stress (Pa)	Mass (kg)	Mat Cost ($/kg)	Total Cost ($/ 2m seg)
COPPER	2.15E+08	21.82	0.55	1202814.34
SiC RIF ALUMINUM	2.15E+08	6.77	10.36	373082.79
BORON RIF ALUMINUM	2.15E+08	6.24	134.92	345061.07
PMR RIF POLYIMIDE	2.15E+08	4.60	130.00	254246.70
ALUMINUM	2.15E+08	6.43	1.00	354695.70
PVC	5.02E+08	3.65	0.70	201167.65
FIBER GLASS	2.15E+08	4.57	1.50	252084.91
GRAPHITE RI POLYIMIDE	2.15E+08	3.94	22.05	217350.43
MG ALLOY (AZ31B)	2.15E+08	4.20	8.81	231697.53

This analysi

ORIGINAL PAGE IS
OF POOR QUALITY

ME 4182 EVALUATION OF JOINING DIMENSIONS FOR 50mm CASING
GROUP 5 VL-EM-RP-TH-BW-EP

CONSTANTS

MATERIAL	GRAPHITE REINFORCED POLYIMIDE
YOUNG'S MODULUS (Pa)	1.00E+10
TENSILE YIELD STRENGTH (Pa)	1.00E+09
DIAMETER (m)	0.05
FRICTION EFFECT	20.00
NUMBER OF TABS	12.00
SLOPE (rad)	0.52
LENGTH (m)	2.00
DENSITY (kg/m^3)	1660.00
CASING THICKNESS (m)	0.0004
SERVICE TEMPERATURE (C)	300

WEIGHT (N)	APP FORCE (N)	Bo (m)	P (N)	W (N)	TAB LEN (m)	TAB THIC (m)	WID/Bo	I (m^4)	DEFLECTION (m)	STRESS (Pa)	DIFFERENCE (Pa)
0.2103	4.2054	0.0042	0.3505	0.1511	0.0100	0.00040	0.9	2.000E-14	0.00025	1.511E+07	9.849E+08
0.2103	4.2054	0.0042	0.3505	0.1511	0.0110	0.00040	0.9	2.000E-14	0.00034	1.662E+07	9.834E+08
0.2103	4.2054	0.0042	0.3505	0.1511	0.0120	0.00040	0.9	2.000E-14	0.00044	1.813E+07	9.819E+08
0.2103	4.2054	0.0042	0.3505	0.1511	0.0130	0.00040	0.9	2.000E-14	0.00055	1.964E+07	9.804E+08
0.2103	4.2054	0.0042	0.3505	0.1511	0.0140	0.00040	0.9	2.000E-14	0.00069	3.116E-07	9.788E+08
0.2103	4.2054	0.0042	0.3505	0.1511	0.0150	0.00040	0.9	2.000E-14	0.00085	2.267E+07	9.773E+08
0.2103	4.2054	0.0042	0.3505	0.1511	0.0100	0.00045	0.9	2.848E-14	0.00018	1.194E+07	9.881E+08
0.2103	4.2054	0.0042	0.3505	0.1511	0.0110	0.00045	0.9	2.848E-14	0.00024	1.313E+07	9.869E+08
0.2103	4.2054	0.0042	0.3505	0.1511	0.0120	0.00045	0.9	2.848E-14	0.00031	1.433E+07	9.857E+08
0.2103	4.2054	0.0042	0.3505	0.1511	0.0130	0.00045	0.9	2.848E-14	0.00039	1.552E+07	9.845E+08
0.2103	4.2054	0.0042	0.3505	0.1511	0.0140	0.00045	0.9	2.848E-14	0.00049	1.672E+07	9.833E+08
0.2103	4.2054	0.0042	0.3505	0.1511	0.0150	0.00045	0.9	2.848E-14	0.00060	1.791E+07	9.821E+08
0.2103	4.2054	0.0042	0.3505	0.1511	0.0100	0.00050	0.9	3.906E-14	0.00013	9.671E+06	9.903E+08
0.2103	4.2054	0.0042	0.3505	0.1511	0.0110	0.00050	0.9	3.906E-14	0.00017	1.064E+07	9.894E+08
0.2103	4.2054	0.0042	0.3505	0.1511	0.0120	0.00050	0.9	3.906E-14	0.00022	1.161E+07	9.884E+08
0.2103	4.2054	0.0042	0.3505	0.1511	0.0130	0.00050	0.9	3.906E-14	0.00028	1.257E+07	9.874E+08
0.2103	4.2054	0.0042	0.3505	0.1511	0.0140	0.00050	0.9	3.906E-14	0.00035	1.354E+07	9.865E+08
0.2103	4.2054	0.0042	0.3505	0.1511	0.0150	0.00050	0.9	3.906E-14	0.00044	1.451E+07	9.855E+08
0.2103	4.2054	0.0042	0.3505	0.1511	0.0160	0.00050	0.9	3.906E-14	0.00053	1.547E+07	9.845E+08
0.2103	4.2054	0.0042	0.3505	0.1511	0.0170	0.00050	0.9	3.906E-14	0.00063	1.644E+07	9.836E+08

ME 4182 EVALUATION OF JOINING DIMENSIONS 100mm CASING

GROUP 5 VL-EM-RF-TH-BW-EM

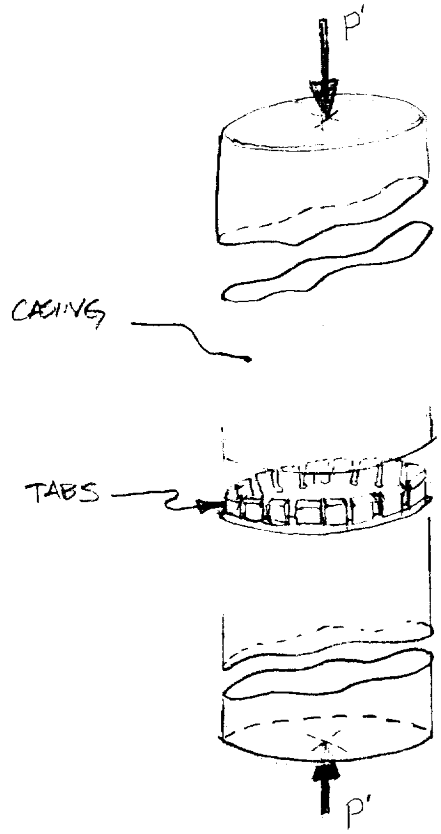
CONSTANTS

MATERIAL	GRAPHITE REINFORCED POLYIMIDE	
YOUNG'S MODULUS (Pa)		1.00E+10
TENSILE YIELD STRENGTH (Pa)		1.00E+09
DIAMETER (m)		0.10
FRICTION EFFECT		20.00
NUMBER OF TABS		15.00
SLOPE (rad)		0.52
LENGTH (m)		2.00
DENSITY (kg/m^3)		1660.00
CASING THICKNESS (m)		0.00075
SERVICE TEMPERATURE (C)		300

WEIGHT (N)	APP FORCE (N)	Bo (m)	P (N)	W (N)	TAB LEN (m)	TAB THIC (m)	Wid/Bo	I (m^4)	DEFLECTION (m)	STRESS (Pa)	DIFFERENCE (Pa)
0.7881	15.7625	0.0067	1.0508	0.4531	0.0200	0.00075	0.9	2.109E-13	0.00057	1.611E+07	9.839E+08
0.7881	15.7625	0.0067	1.0508	0.4531	0.0210	0.00075	0.9	2.109E-13	0.00066	1.692E+07	9.831E+08
0.7881	15.7625	0.0067	1.0508	0.4531	0.0220	0.00075	0.9	2.109E-13	0.00076	1.772E+07	9.823E+08
0.7881	15.7625	0.0067	1.0508	0.4531	0.0230	0.00075	0.9	2.109E-13	0.00087	1.853E+07	9.815E+08
0.7881	15.7625	0.0067	1.0508	0.4531	0.0240	0.00075	0.9	2.109E-13	0.00099	1.933E+07	9.807E+08
0.7881	15.7625	0.0067	1.0508	0.4531	0.0200	0.00080	0.9	2.560E-13	0.00047	1.416E+07	9.858E+08
0.7881	15.7625	0.0067	1.0508	0.4531	0.0210	0.00080	0.9	2.560E-13	0.00055	1.487E+07	9.851E+08
0.7881	15.7625	0.0067	1.0508	0.4531	0.0220	0.00080	0.9	2.560E-13	0.00063	1.558E+07	9.844E+08
0.7881	15.7625	0.0067	1.0508	0.4531	0.0230	0.00080	0.9	2.560E-13	0.00072	1.628E+07	9.837E+08
0.7881	15.7625	0.0067	1.0508	0.4531	0.0240	0.00080	0.9	2.560E-13	0.00082	1.699E+07	9.830E+08
0.7881	15.7625	0.0067	1.0508	0.4531	0.0250	0.00080	0.9	2.560E-13	0.00092	1.770E+07	9.823E+08
0.7881	15.7625	0.0067	1.0508	0.4531	0.0260	0.00080	0.9	2.560E-13	0.00104	1.841E+07	9.816E+08
0.7881	15.7625	0.0067	1.0508	0.4531	0.0270	0.00080	0.9	2.560E-13	0.00116	1.912E+07	9.809E+08
0.7881	15.7625	0.0067	1.0508	0.4531	0.0200	0.00085	0.9	3.071E-13	0.00039	1.254E+07	9.875E+08
0.7881	15.7625	0.0067	1.0508	0.4531	0.0210	0.00085	0.9	3.071E-13	0.00046	1.317E+07	9.868E+08
0.7881	15.7625	0.0067	1.0508	0.4531	0.0220	0.00085	0.9	3.071E-13	0.00052	1.380E+07	9.862E+08
0.7881	15.7625	0.0067	1.0508	0.4531	0.0230	0.00085	0.9	3.071E-13	0.00060	1.442E+07	9.856E+08
0.7881	15.7625	0.0067	1.0508	0.4531	0.0240	0.00085	0.9	3.071E-13	0.00068	1.505E+07	9.849E+08
0.7881	15.7625	0.0067	1.0508	0.4531	0.0250	0.00085	0.9	3.071E-13	0.00077	1.568E+07	9.843E+08
0.7881	15.7625	0.0067	1.0508	0.4531	0.0260	0.00085	0.9	3.071E-13	0.00086	1.631E+07	9.837E+08
0.7881	15.7625	0.0067	1.0508	0.4531	0.0270	0.00085	0.9	3.071E-13	0.00097	1.693E+07	9.831E+08
0.7891	15.7625	0.0067	1.0509	0.4531	0.0280	0.00085	0.9	3.071E-13	0.00108	1.756E+07	9.824E+08
0.7881	15.7625	0.0067	1.0508	0.4531	0.0290	0.00085	0.9	3.071E-13	0.00120	1.819E+07	9.818E+08

ORIGINAL PAGE IS OF POOR QUALITY

CASING JOINING DESIGN - TAB DEFLECTION ANALYSIS



g ≡ ACCELERATION OF LUNAR GRAVITY

P' ≡ TOTAL FORCE ON CASING

F_E ≡ FRICTION EFFECT COEFFICIENT

r_o = CASING OUTER RADIUS

r_i = CASING INNER RADIUS

P ≡ UNIT TAB FORCE

ρ ≡ DENSITY OF GRAPHITE REINFORCED POLYMIDE

t ≡ CASING WALL THICKNESS

l ≡ LENGTH OF CASING SEGMENT

n ≡ NUMBER OF TABS

$$P' = F_E \rho l \pi (r_o^2 - r_i^2) g$$

$$P'_{50} = (20)(1660\ kg/m^3)(2m)\pi\left((0.0254m)^2 - (0.0250m)^2\right)g$$

$$= (4.205\ kg)(1/6)(9.8\ m/s^2) = 6.869\ N$$

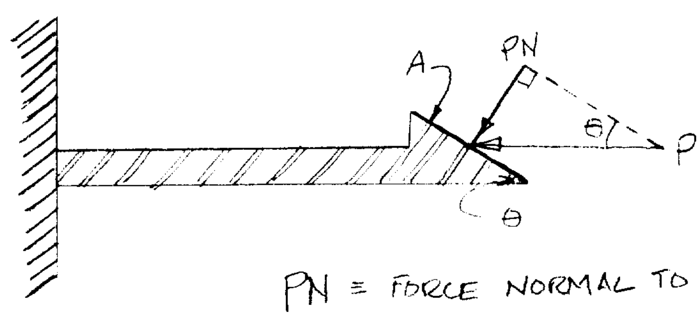
$$P'_{100} = (20)(1660\ kg/m^3)(2m)\pi\left((0.05075m)^2 - (0.05000m)^2\right)g$$

$$= (15.762\ kg)(1/6)(9.8\ m/s^2) = 25.745\ N$$

$$P = P'/n$$

$$P_{50} = \frac{6.869\ N}{12\ \text{TABS}} = 0.572\ N/\text{TAB}$$

$$P_{100} = \frac{25.745\ N}{15\ \text{TABS}} = 1.716\ N/\text{TAB}$$



PN ≡ FORCE NORMAL TO A

θ ≡ SLOPE OF A

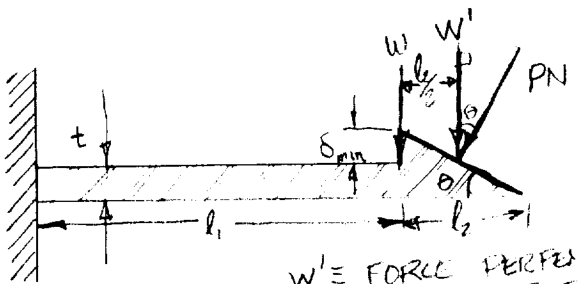
$$PN = P\sin(\theta)$$

$$PN_{50} = (0.572\ N)\sin(30°) = 0.286\ N$$

$$PN_{100} = (1.716N)\sin(30°) = 0.858\ N$$

ORIGINAL PAGE IS OF POOR QUALITY

CASING JOINING DESIGN - TAB DEFLECTION ANALYSIS (CONTINUED)



$W' \equiv$ FORCE PERPENDICULAR TO BEAM AT CONTACT POINT

$W' = PN\cos(\theta)$

$W'_{50} = (0.286\ N)\cos(30°) = 0.247\ N$

$W'_{100} = (0.858\ N)\cos(30°) = 0.743\ N$

$W \equiv$ FORCE PERPENDICULAR TO BEAM JUST BEFORE WEDGE OF TAB

$l_1 \equiv$ LENGTH OF CANTILEVER MODEL

$l_2 \equiv$ LENGTH OF WEDGE

$t \equiv$ THICKNESS OF TAB

$\delta_{min} \equiv$ MINIMUM REQUIRED DEFLECTION

$$l_2 = (t+\delta)\cot(30°)$$

$$W = \frac{l_1}{l_1+l_2}$$

$\delta \equiv$ CALCULATED DEFLECTION

$E \equiv$ YOUNG'S MODULUS

$I \equiv$ MOMENT OF INERTIA OF THE CROSS SECTION

$\sigma \equiv$ TENSILE STRESS

$b \equiv$ WIDTH OF THE TAB

$$I = \frac{t^3 b}{12}$$

CASING JOINING DESIGN - TAB DEFLECTION ANALYSIS (CONTINUED)

$$\delta = \frac{W l_1^3}{3EI}$$

$$\sigma = \frac{6W l_1}{b t^2}$$

ITERATIONS FOUND ON TABLE

50 mm ID CASING

$t = 0.00040$ m

$\delta_{min} = 0.00044$ m

$l_1 = 0.0120$ m

100 mm ID CASING

$t = 0.00075$ m

$\delta_{min} = 0.00075$ m

$l_1 = 0.022$ m

THERMAL STRESS ON CASING

$$\sigma = \frac{1}{2} E \alpha \Delta T / (1-\nu) \quad (*)$$

$$= \frac{(10^{10} Pa)(1.7 \times 10^{-6} m/K)(250-100)K}{(2)(1-0.33)}$$

$$\sigma = 4.44 \times 10^{6} Pa$$

$\sigma_t = 4.4$ MPa << ANY PROPERTIES OF GRAPHITE REINFORCED POLYMIDE

$\alpha_{GRAPHITE\ REINFORCED\ POLYMIDE} = 0.8 - 1.7\ 10^{-6}$ m/m K

$\nu = 0.33$ (ASSUMPTION)

$E = 1 \times 10^{10}$ Pa

$\sigma_{y,c} = 300$ MPa

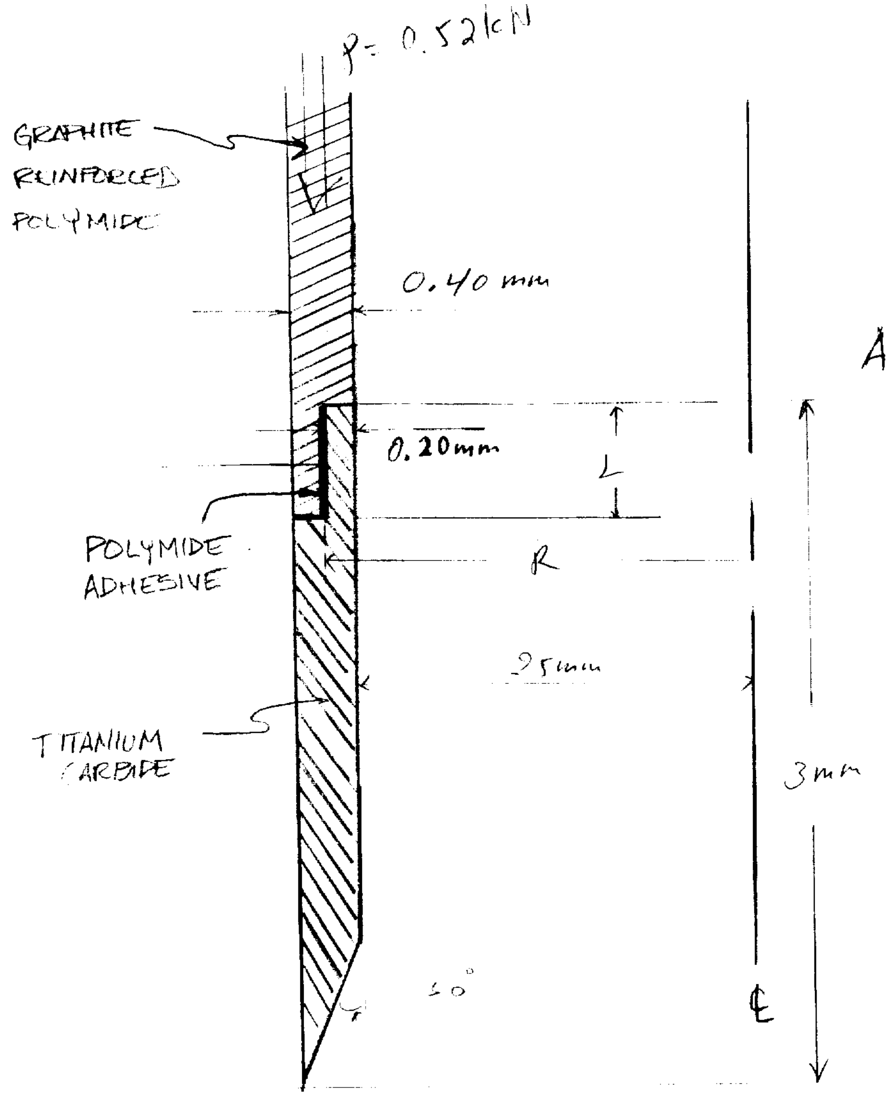
$\sigma_{y,t} = 1.0$ GPa

$\sigma_{SHEAR,LAMINAR} = 50$ MPa

* REED, RICHARD P., MATERIALS AT LOW TEMPERATURES, AMERICAN SOCIETY FOR METALS, MEDFORD PARK, OHIO, 1983, PAGE 112.

ORIGINAL PAGE IS
OF POOR QUALITY

BOND LENGTH FOR 50 mm ID CASING



ASSUME BOND STRENGTH

$\tau_{SHEAR} = 30$ MPa

$P = 0.52$ kN

$A_{SHEAR} = \pi D L$

$R = 25 + 0.20$

$R = 25.2$ mm

$D = 2(25.2) = 50.4$ mm

$n = 5$

$P_{MAX} = 0.52\ kN(5)$

$= 2.6$ kN

$$\tau_{shear} > \frac{P_{max}}{A}$$

$$\pi D L = A > \frac{P_{max}}{\tau_{SHEAR}}$$

$$L > \frac{P_{max}}{\pi\, \tau_{SHEAR}\, D} = \frac{2.6 \times 10^{3}\ N}{\pi \left(30 \times 10^{6} \frac{N}{m^2}\right)\left(50.4 \times 10^{-3}\ m\right)}$$

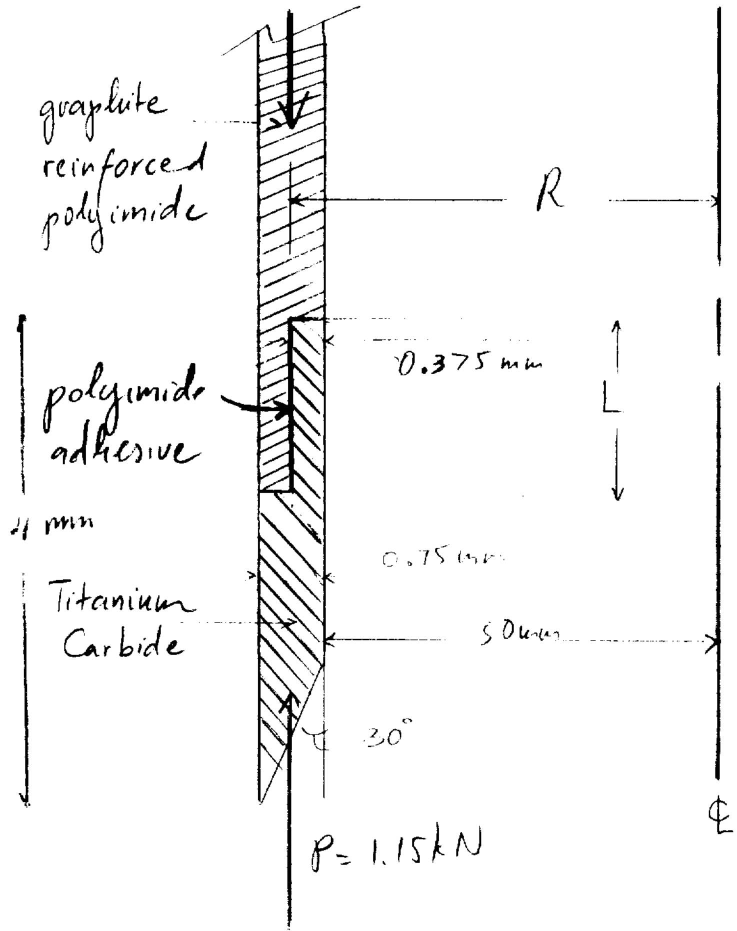
$$L > 5.5 \times 10^{-4}\ m$$

0.00055 mm

USE $\boxed{L = 0.55\ mm}$

ORIGINAL PAGE IS
OF POOR QUALITY

BONDING LENGTH FOR 100mm Di CASING



ASSUME $\tau_{SHEAR} = 30$ MPa

$P = 1.15$ kN

$R = 50\text{mm} + 0.375\text{mm}$

$R = 50.375$ mm

$D = 2R = 2(50.375\text{mm})$

$D = 100.75$ mm

n = SAFETY FACTOR = 5

$P_{max} = Pn = 1.15\text{kN}(5)$

$= 5.75$ kN

$$\tau_{SHEAR} > \frac{P_{max}}{A} = \frac{P_{max}}{\pi D L}$$

$$L > \frac{P_{max}}{\pi D \tau_{shear}} = \frac{(5.75 \times 10^{3}\ N)}{\pi (100.75 \times 10^{-3} m)(30 \times 10^{6} \frac{N}{m^2})}$$

$$L > 6.05 \times 10^{-4}\ m = 0.000605\ m$$

$$L > 0.605\ mm$$

$$\boxed{\text{USE } L = 0.61\ mm}$$

ORIGINAL PAGE IS OF POOR QUALITY

TIME TO CUT THE 50mm D_i CASING

$D_i = 50\text{ mm}$

t_c = Casing thickness = 0.4 mm = 0.01575 in

$D_{mean} = 50.2\text{ mm}$

RPM = 500 RPM

V = Surface Speed = 0.262 (RPM) D_{mean}

$$= 0.262 \left(500 \frac{\text{rev}}{\text{min}}\right)\left(50.2 \times 10^{-3}\text{ m}\right)\left(\frac{1\text{ ft}}{0.3048\text{ m}}\right)$$

V = 21.58 sfpm

F = Feed Rate assummed provided by spring = 0.0002 in/rev

D = Depth of cut = 0.04 mm = 0.001575 in

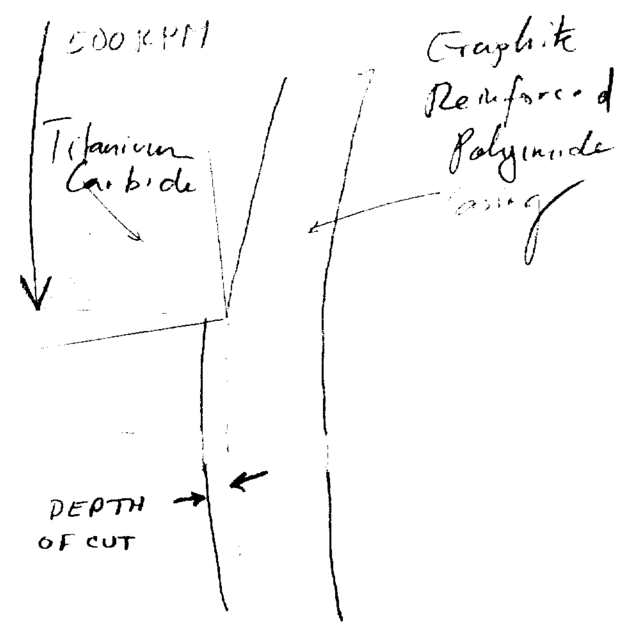
T = Time to cut casing

$$T = \frac{t_c}{RPM(F)}$$

$$= \frac{0.01575\text{ in}}{\left(500 \frac{\text{rev}}{\text{min}}\right)\left(0.0002 \frac{\text{in}}{\text{rev}}\right)}$$

T = 0.1575 min

$$\boxed{T = 9.45 \text{ seconds}}$$



TIME TO CUT THE 100mm D_i CASING

D_i = 100 mm

t_c = Casing thickness: 0.75 mm = 0.029528 in

D_{mean} = 100.375 mm

RPM = 500 RPM

V = Speed, sfpm = $(0.262)(500)\left(100.375 \times 10^{-3}\,m\right)\left(\frac{1\,ft}{0.3048\,m}\right)$

$$V = 43.14 \text{ sfpm}$$

F = feed rate provided by spring assummed = $0.0002 \frac{in}{rev}$

D = depth of cut = 0.04 mm = 0.001575 in

T = Time to cut casing

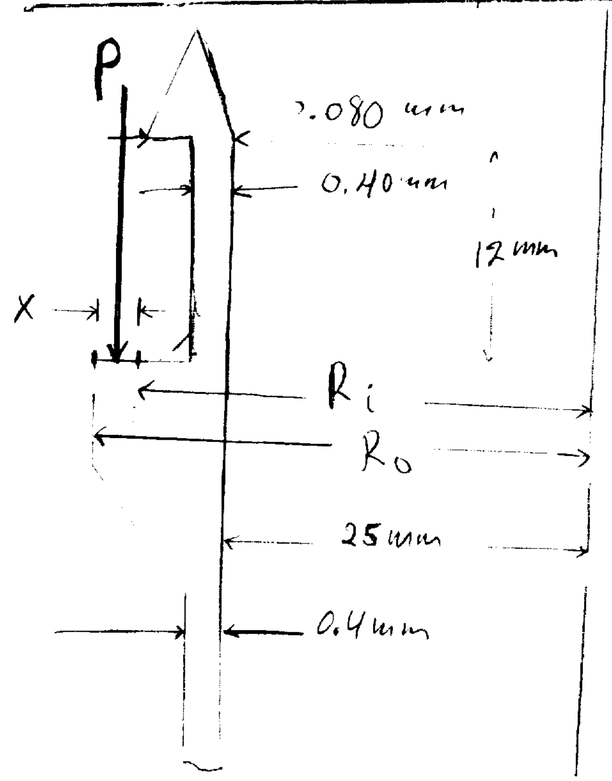
$$T = \frac{t_c}{RPM\,(F)}$$

$$= \frac{0.029528 \text{ in}}{\left(500 \frac{rev}{min}\right)\left(0.0002 \frac{in}{rev}\right)}$$

$$T = 0.29528 \text{ min}$$

$$\boxed{\text{Time} = 17.72 \text{ seconds}}$$

BEARING THICKNESS — 50 mm ID



LAMINAR SHEAR STRESS OF GRAPHITE REINFORCED POLYMIDE = 50 MPa

$$A_{LOAD} = \pi (R_o^2 - R_i^2)$$

n = SAFETY FACTOR = 5

$\therefore$ Max load = n P
$= 5(0.52\ kN)$
$P_{max} = 2.6\ kN$

$R_i = 25\ mm + 0.8\ mm$
$R_i = 25.8\ mm$

X = BEARING THICKNESS

$$\tau = \frac{P_{max}}{A} = \frac{P_{max}}{\pi (R_o^2 - R_i^2)}$$

$$R_o = \sqrt{\frac{P_{max}}{\pi \tau} + R_i^2}$$

$$= \sqrt{\frac{(2.6 \times 10^3)}{\pi\ 50 \times 10^6} + (25.8 \times 10^{-3})^2}$$

$$R_o = 2.612 \times 10^{-2}\ m$$

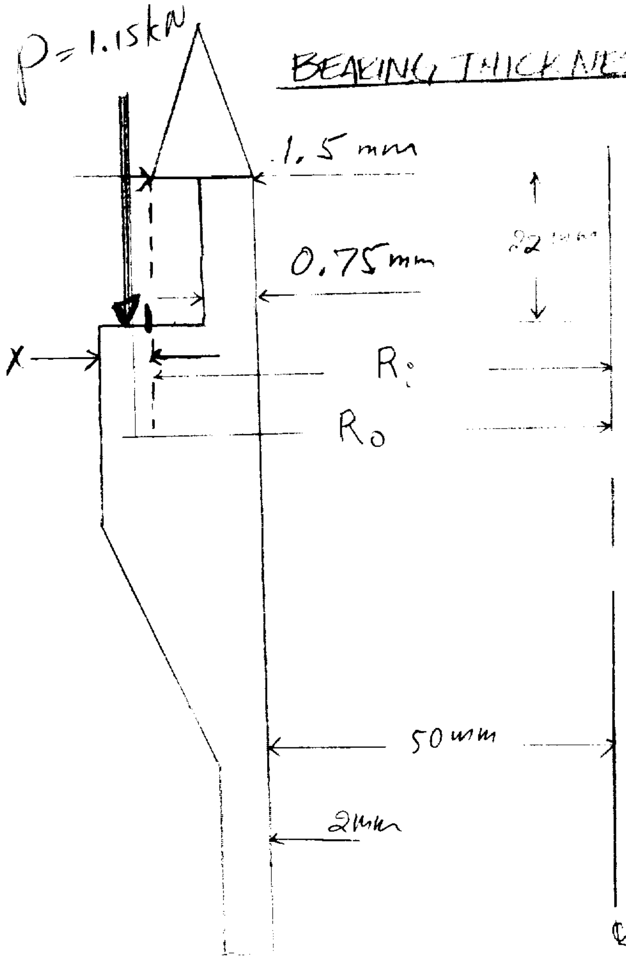
$$= 0.02612\ m$$
$$= 26.12\ mm$$

$\therefore X = R_o - R_i = 26.12\ mm - 25.8\ mm$

$X = 0.32\ mm$ (MINIMUM)

$$\boxed{\text{Use}\quad X = 0.35\ mm}$$

ORIGINAL PAGE IS OF POOR QUALITY



BEARING THICKNESS - 100 mm ID CASING

$\tau_{SHEAR} = 50\ MPa$

$A_{LOAD} = \pi(R_o^2 - R_i^2)$

$n_{SF} = 5$

$P_{max} = P_n = 1.15kN(5) = 5.75kN$

$R_i = 50mm + 1.5mm = 51.5mm$

X = BEARING THICKNESS

$$\tau = \frac{P_{max}}{A} = \frac{P_{max}}{\pi(R_o^2 - R_i^2)}$$

$$R_o = \sqrt{\frac{P_{max}}{\pi\,\tau} + R_i^2}$$

$$= \sqrt{\frac{5.75 \times 10^3}{\pi\,(50 \times 10^6)} + (51.3 \times 10^{-3})^2}$$

$R_o = 51.85 \times 10^{-3}\ m$

$R_o = 51.85\ mm$

$X = R_o - R_i = 51.85\ mm - 51.5mm = 0.35\ mm$ (MINIMUM)

$\boxed{\text{Use } X = 0.35\ mm}$

ORIGINAL PAGE IS OF POOR QUALITY

POWERSCREW

THREAD STRESSES

ASSUMING AN AVERAGE BEARING STRESS ON EACH THREAD OF THE POWERSCREW.

$$\sigma = \frac{4F \cdot \frac{1}{P}}{\pi h (d^2 - d_r^2)} = \frac{4 \cdot 1135 N \cdot 5mm}{\pi \cdot 40mm \cdot (15^2 - 10^2)_{mm^2}} = \underline{1.45 \ MPA}$$

$$S_y = 0.75 \ S_{UT}$$

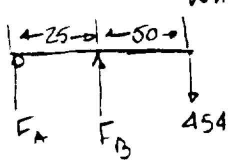
A NOMINAL VALUE FOR 44 MZ HIGH CARBON, HIGH CHROMIUM STEEL ULTIMATE TENSILE STRENGTH IS 1.0 GPA. CHOOSING COMPARISON OF YIELD STRENGTH TO ACTUAL LOAD σ WE FIND THE FACTOR OF SAFTY INCORPORATED INTO THE POWERSCREW,

$$S_y = 750 \ MPA$$

$$n = \frac{S_y}{\sigma} = \frac{750 \ MPA}{1.45 \ MPA} = \underline{517}$$

BEARINGS

ASSUMING STANDARD L_{10} Life of 3000 hrs of operation AT 500 RPM WITH BEARING RELIABILITY OF 90%



$$\Sigma F_x = 0$$

$$\Sigma F_y = F_A + F_B = 454 N$$

$$\Sigma M_A = F_B(25) = 454 \cdot 75$$

$$\therefore F_B = \underline{1362 \ N}$$

PLACING 3 BEARING ALONG POWERSCREW SHAFT

ORIGINAL PAGE IS OF POOR QUALITY

Bearing

Upon distributing the load on each bearing:

$$F_{BEARING} = \frac{1135\,N}{3} = \underline{378\,N}$$

Design Criteria

$L_D = 1000$ hrs

$N_D = 200$ rpm

$R = 99\%$

$$C_R = F\left\{\left(\frac{L_D \cdot N_D}{L_R \cdot N_R}\right) \cdot \frac{\left(\frac{1}{4.48}\right)}{\left[\ln\left(\frac{1}{R}\right)\right]^{1/1.15}}\right\}^{3/10}$$

$$C_R = 1135\,N\left\{\left(\frac{1000 \cdot 200}{3000 \cdot 500}\right) \cdot \left(\frac{\left(\frac{1}{4.48}\right)}{\left[\ln\left(\frac{1}{0.99}\right)\right]^{1/1.15}}\right)\right\}^{3/10}$$

$$C_{R_D} = 1313\,N$$

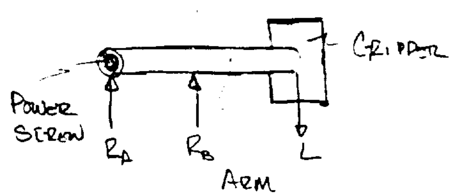
Choice bearing from 02-series ball bearings

Bore ≡ 15 mm

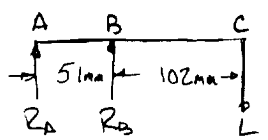
$C_R = 5870\,N$

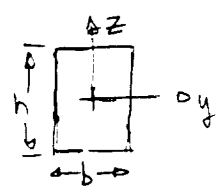
$$n = \frac{C_R}{C_{R_D}} = \frac{5870}{1313} = \underline{4.5}$$

Material Determination For Arms



Deflection for simple supports, overhanging load.





$b = 19$ mm

$h = 25$ mm

$$y = \frac{L \cdot \overline{BC}^2}{3E \cdot I}(\overline{AC}) \qquad I = \frac{1}{12}bh^3 = \frac{1}{12}(19)(25)^3 = 24{,}739.6\,mm^4$$

$$E \geq \frac{(1135\,N)(102\,mm)^2(153\,mm)}{3(1\,mm)(24{,}739.6\,mm^4)} = \underline{2.4\ GPA}$$

MATERIAL DETERMINATION FOR ARMS

ASSUMING A MAXIMUM DEFLECTION OF 1 millimeter, WE DESIRE A MATERIAL WITH A MODULUS GREATER THAN 2.4GPA. 44M2 HIGH CARBON, HIGH CHROMIUM STEEL SPORTS A MODULUS OF 206.7 GPA. TO OBTAIN YIELDING IN THE ARMS, THE FOLLOWING LOAD MUST BE APPLIED.

$$F = \frac{3EI \cdot y}{\overline{BC}^2 \cdot \overline{AC}} = \frac{(3)(206.7\,GPA)(24{,}739.6\,mm^4)(1\,mm)}{(102\,mm)^2\,(153\,mm)}\left(\frac{1\,m}{1000\,mm}\right)^2$$

$$\underline{F = 9637.4\ N}$$

THIS ALLOWS A FACTOR OF SAFTY OF:

$$n = \frac{9637.4}{1135.0} = \underline{8.50}$$

TORQUE REQUIREMENTS

TORQUE REQUIRED BY THE FOOTPLATE IS EVALUATED BY THE EQUATION:

$$T = \frac{PL}{2\pi E}$$

$P \equiv$ LOAD $=$ 1135 N
$L \equiv$ LEAD $=$ 5 mm
$E \equiv$ EFFICIENCY $\cong$ 0.90

$$T = \frac{(1135N)(0.005m)}{(2\pi)(0.90)} = \underline{1.0\ Nm}$$

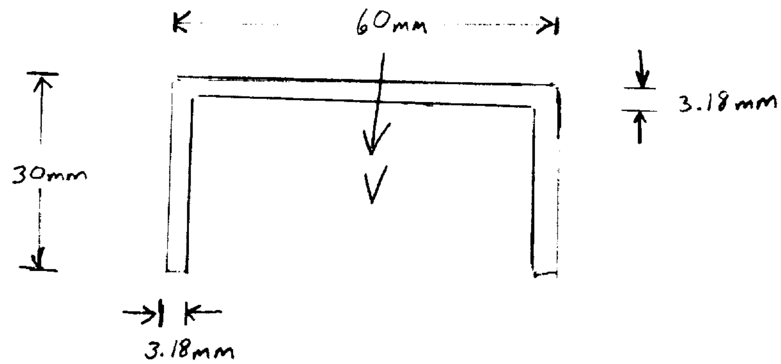
HORSEPOWER REQUIREMENTS

$w = 20.88$ rad/sec

$$H = T \cdot w = (1.0\,N{\cdot}m)(20.88\,\tfrac{rad}{s})$$

$$\underline{H = 20.88\ W}$$

BASE PLATE STRESS



$$A = (60\,mm)(3.18\,mm) + 2(30\,mm - 3.18\,mm)(3.18\,mm) = 361\,mm^2$$

$$V = 445\,N \quad (= 100\,lbf)$$

$$\frac{V}{A} = \frac{445\,N}{361\,mm^2} = 1.23\,\frac{N}{mm^2}$$

AT 400°C, ALUMINUM-SILICON-MAGNESIUM TENSILE STRENGTH IS APPROXIMATELY $37\,\frac{N}{mm^2}$

AT 400°C, SHEAR STRENGTH IS $(37\,\frac{N}{mm^2})(.75)(.577) = 16.0\,\frac{N}{mm^2}$

$$\text{SAFETY FACTOR} = \frac{16.0}{1.23} = 13$$

ORIGINAL PAGE IS OF POOR QUALITY

Luner Drill Footplate
Weight Analysis

Part	Volume (mm^3)	Material
Base Arm 1	126000	Aluminum - Silicon - Magnesium
Base Arm 2	126000	"
Crossmember	95200	"
Leg 1 (Vert.)	42000	"
Leg 2 (Vert)	42000	"
Foot 1	21000	"
Foot 2	21000	"
Leg 3 (Vert.)	38500	"
Leg 4 (Vert.)	38500	"
Foot 3	21000	"
Foot 4	21000	"
Box Side 1	38500	"
Box Side 2	38500	"
Box Front	56000	"
Box Rear	56000	"
Cover	61600	"
Brace 1	50400	"
Brace 2	50400	"
Power Screw	35345	Chromium Steel
Gripper Arm 1	95000	"
Gripper Arm 2	95000	"

		Weight
Aluminum:	$1170400\ mm^3 = 0.0011704\ m^3$ @ $2700\ kg/m^3$ =	3.16 Kg
Chromium Steel:	$225,345\ mm^3 = 0.0002253\ m^3$ @ $7700\ kg/m^3$ =	1.75 Kg
Motor		= 1.0 Kg
Bearings		= .5 Kg

Total Weight (on Earth) = 6.41 Kg
(on Moon) = 1.07 Kg

ORIGINAL PAGE IS
OF POOR QUALITY

COST ANALYSIS

Footplate:

Description	Cost
Gripper Arms (2)	$75
Base Assembly (1)	$400
Power Screw (1)	$150
Bearing (3)	$70
Ball Bearing Nut (2)	$200
Bellows (2)	$100
Half Horsepower Motor (1)	$500
Power Supply (1)	$500
Control System	$1000
Cap Screws, Pins	$50
	$3495

Transport Cost: $55135/kg * 6.4kg = $352,864

APPENDIX B

FIGURES

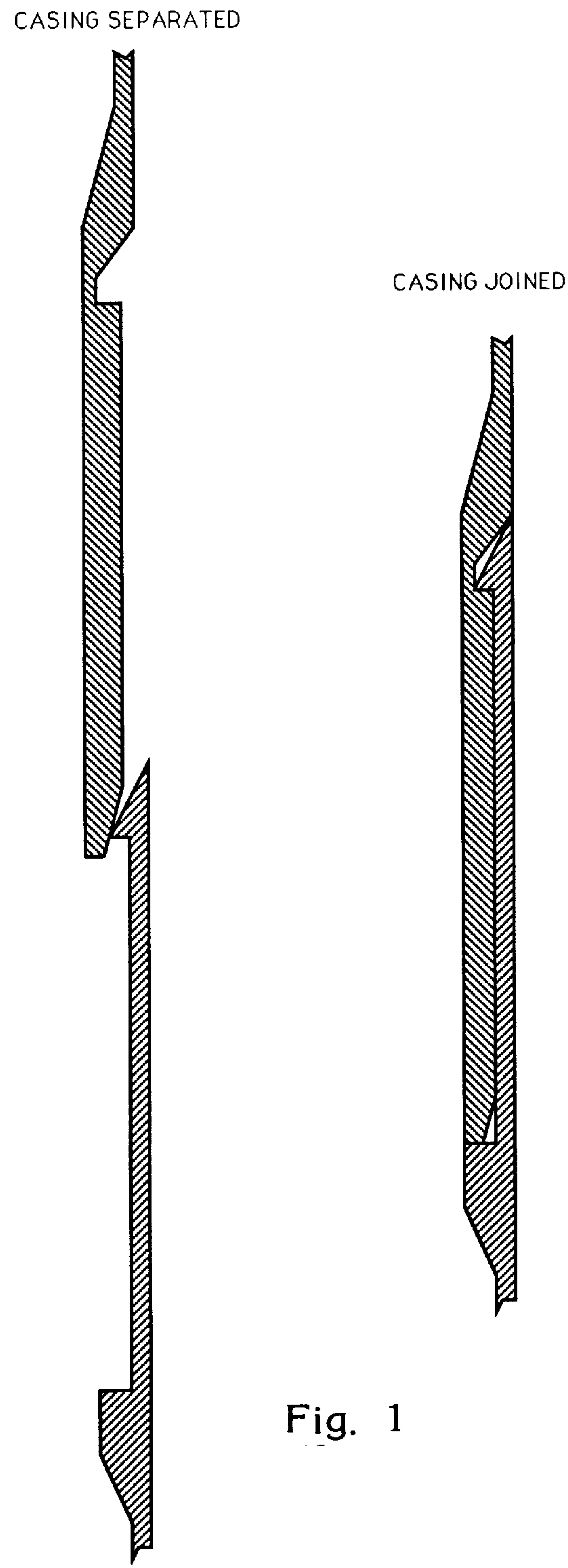


Fig. 1

CASING ADAPTER - CONCEPTUAL DESIGN

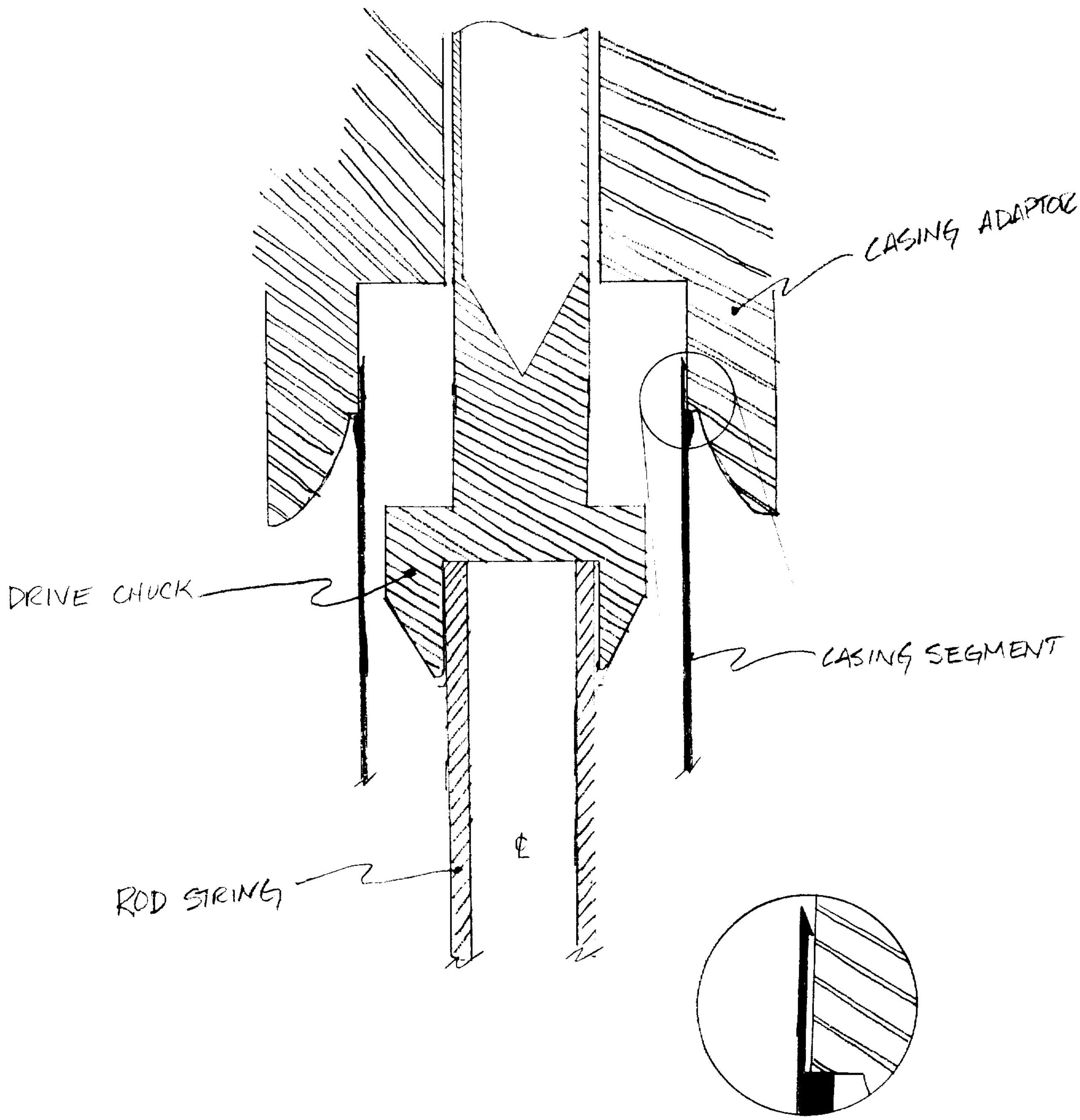


Fig. 2

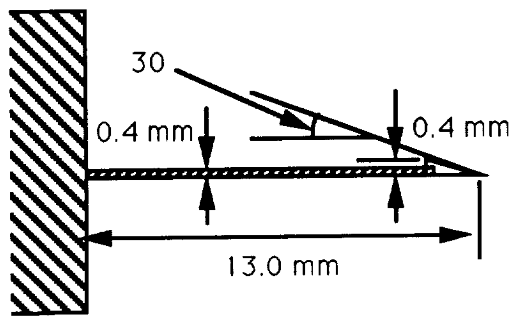
ORIGINAL PAGE IS
OF POOR QUALITY

CASING JOINING METHOD ANALYSIS

The tab was modeled as a cantilever beam

casing I.D. = 50 mm

TAB CROSS SECTION



TAB TOP VIEW

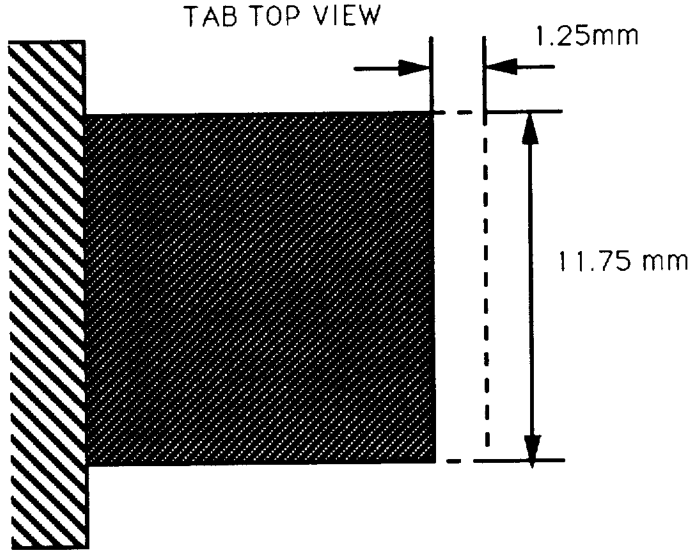


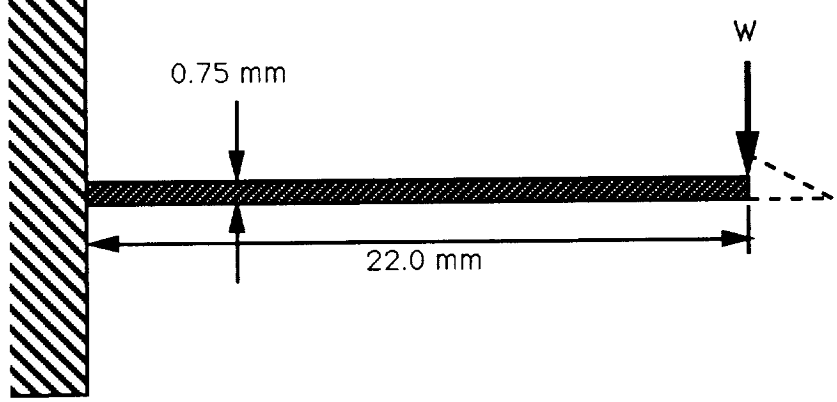
Fig. 3a

CASING JOINING METHOD ANALYSIS

The tab was modeled as a cantilever beam

casing I.D. = 100 mm

TAB CROSS SECTION



TAB TOP VIEW

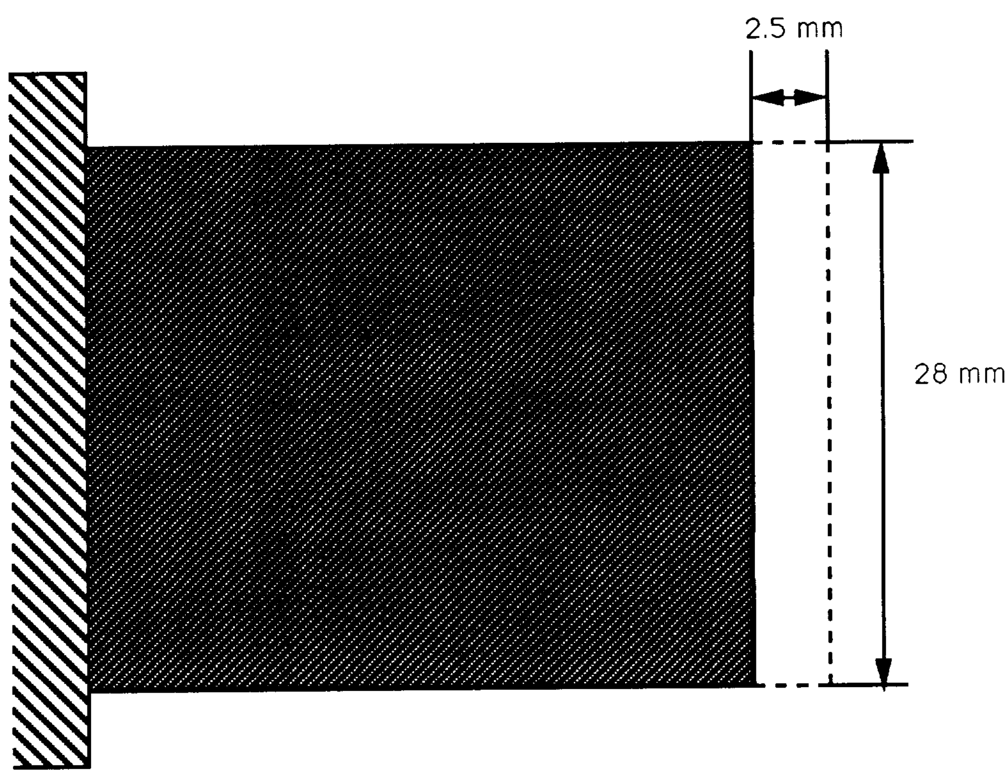


Fig. 3b

CASING CUTTER - CUTTING MECHANISM CONCEPTUALIZATION

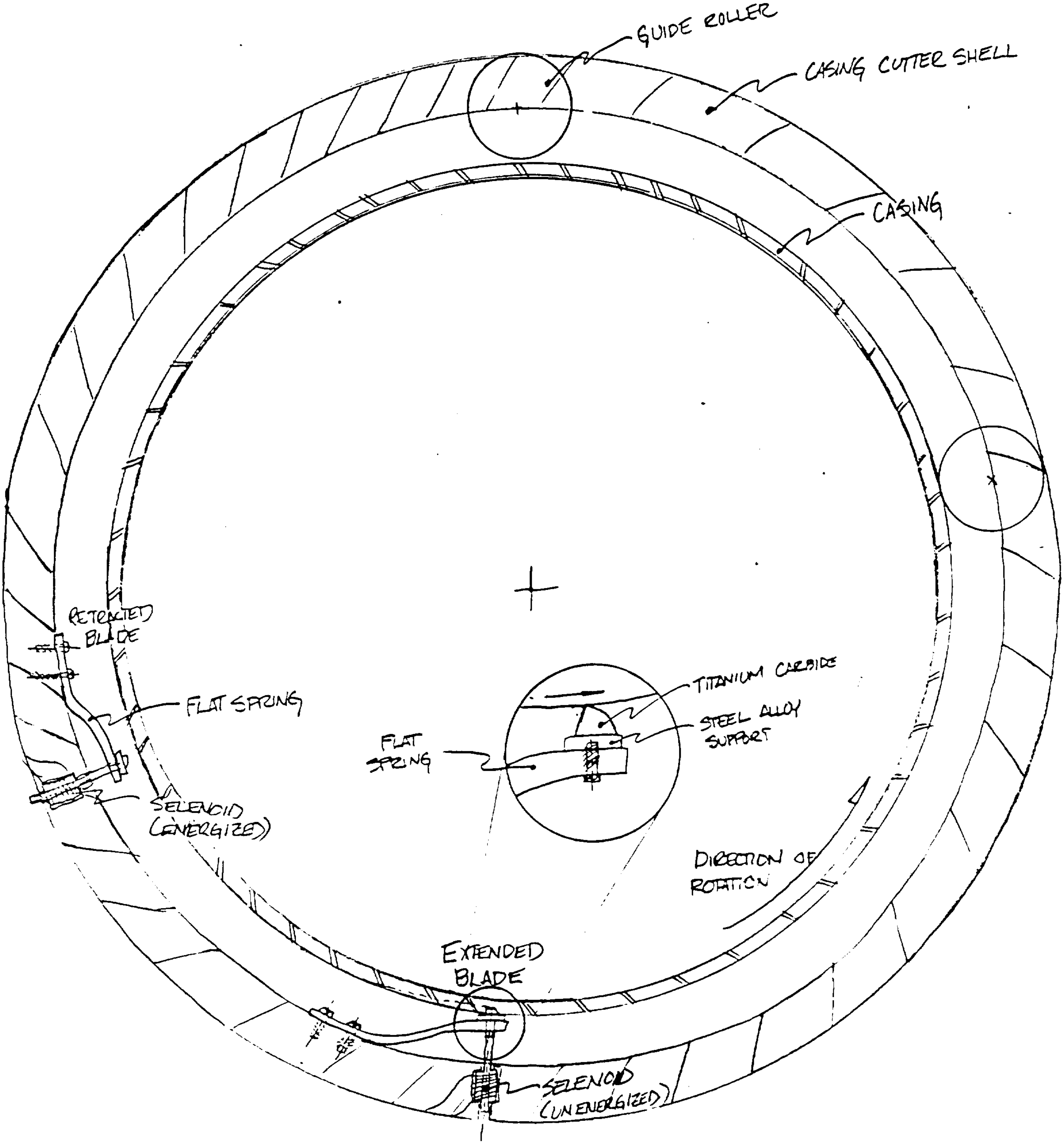


Fig. 4

ORIGINAL PAGE IS
OF POOR QUALITY

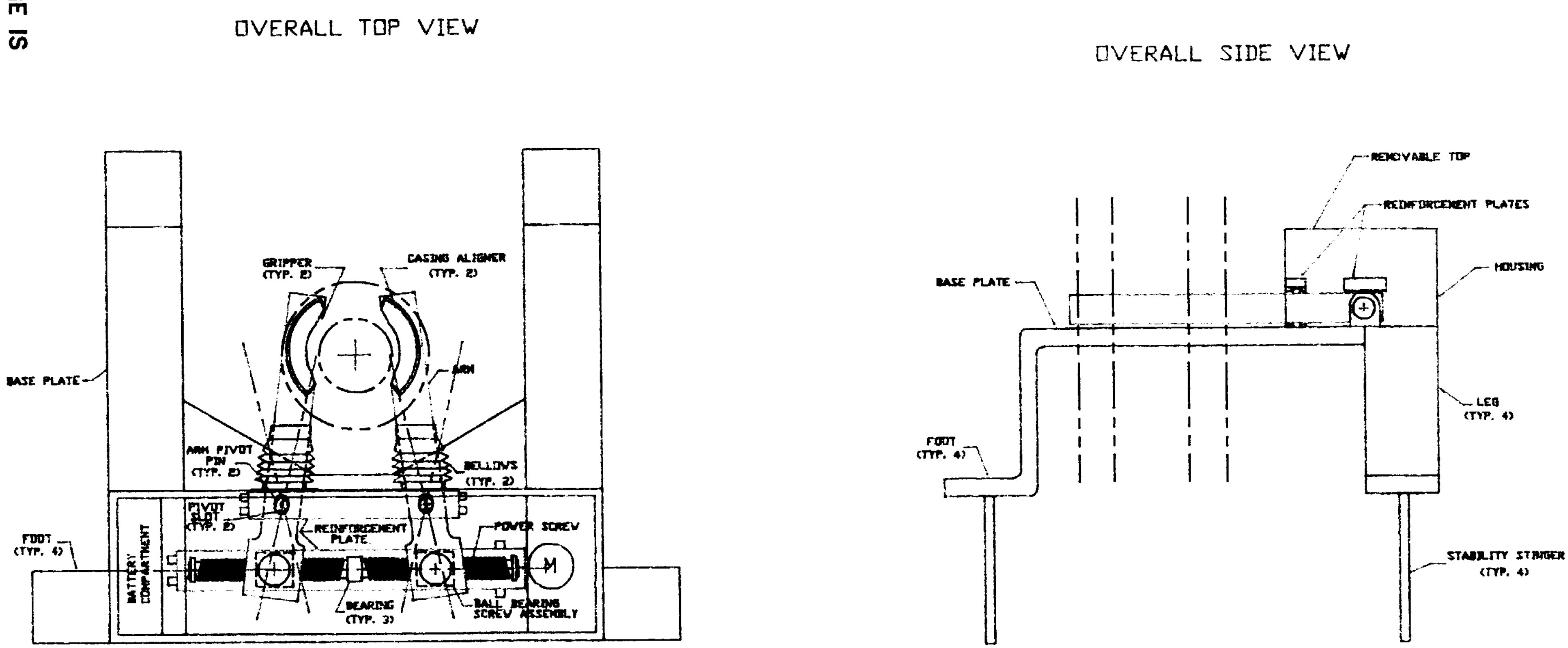


FIG. 5

OVERALL BASE PLATE STRUCTURE

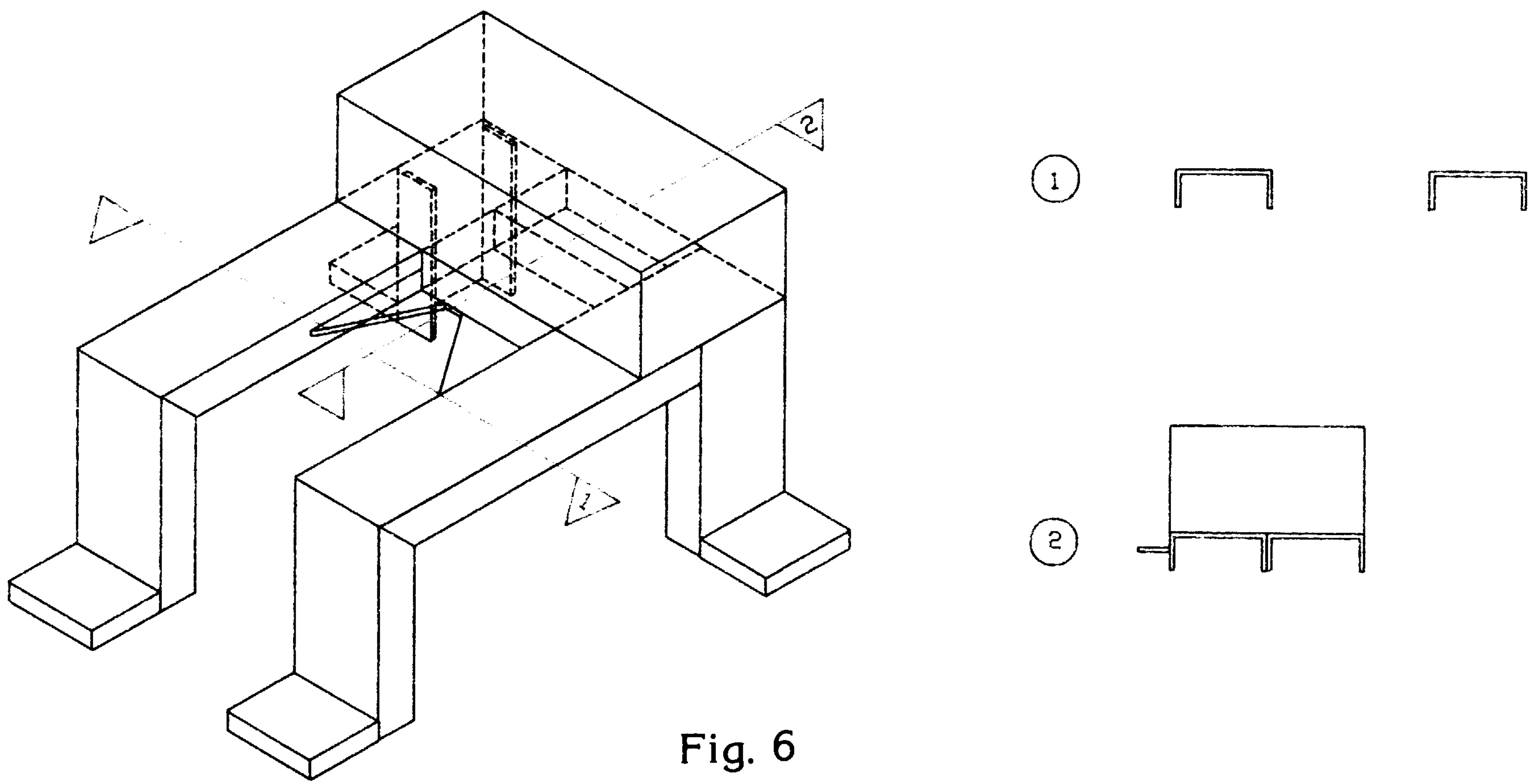


Fig. 6

BELLOWS DETAIL

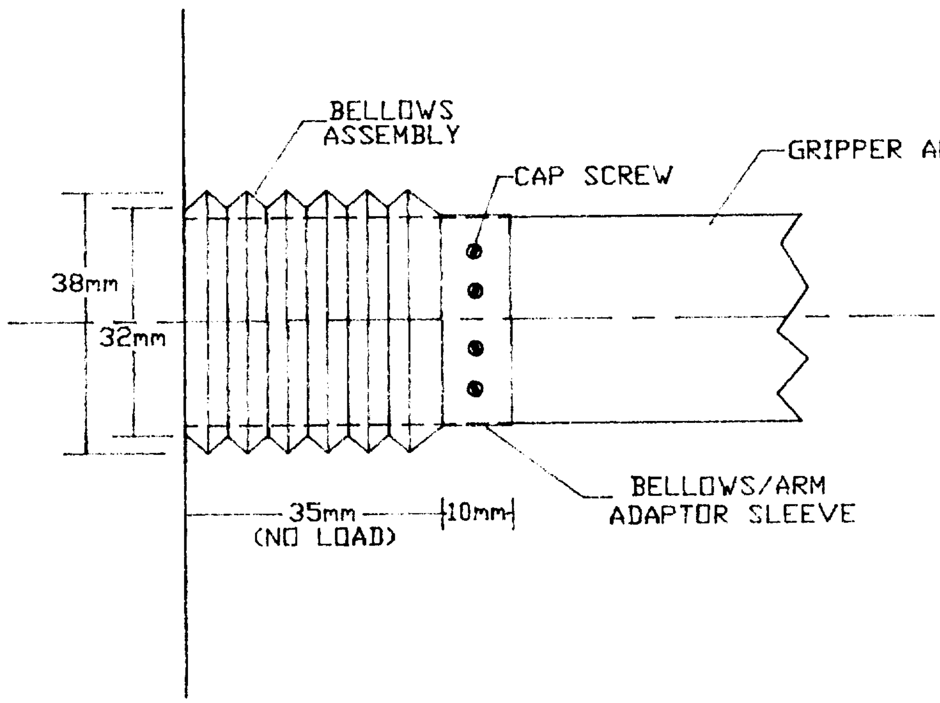


Fig. 7

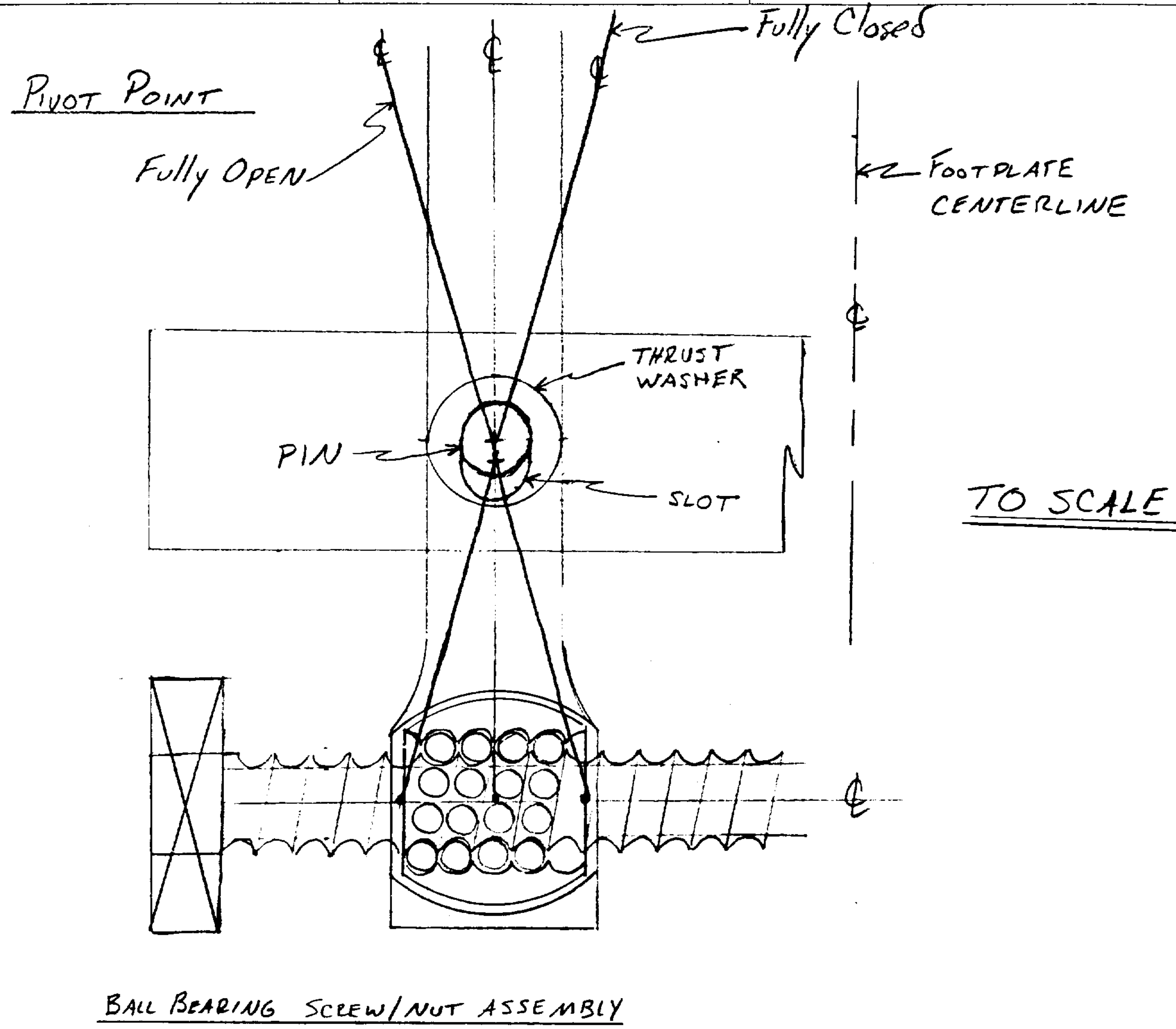

Fully Closed
Pivot Point
Fully Open
Footplate Centerline
Thrust Washer
Pin
Slot
To Scale
Ball Bearing Screw/Nut Assembly

Fig. 8

APPENDIX C

PROGRESS REPORTS

ME 4182
Mr. Brazell
Group 5

LUNAR DEEP DRILL FOOT PLATE AND CASING DEVICES

Future lunar operation will require a mobile drilling system for mining, geological research, and construction. Previous work at the Georgia Institute of Technology has resulted in a mobile platform design on which to mount various drilling apparatus concepts. A typical drilling operation will require a casing in the hole to prevent hole collapse due to the loose nature of the lunar regolith. The drill will require rod extensions to reach desired depths of greater than two meters. Therefore, a means of supporting the rod string while disengaged from the drive unit will be needed.

The goal of this design group is to design a casing which will act in conjunction with the auger-bit assembly such that the hole may be dug and cased simultaneously. The casing must reach a depth of 50 meters and allow for hole sizes of 50 and 100 millimeters.

Complementing the design of the casing method will be the design of a footplate for the drilling apparatus. The objective of the footplate will be to stabilize and support the drilling string once each 2 meter drilling increment has been installed or removed.

Certain constraints must be accounted for in dealing with the harsh environment encountered on the lunar surface. Large temperature gradients, reduced gravitational effects, cost effectiveness of materials, and earth to moon transportation expenses are primary concerns.

LUNAR DEEP DRILL APPARATUS FOOTPLATE AND CASING DEVICES

ME 4182 GROUP 5
PROGRESS REPORT
11 JAN. 1989
MR. BRAZELL

Preliminary research has been done on two subjects, lunar geology and drilling technology. The first subject was undertook by Erik Maassen, Vu Le, and Eddie Wayne Morrison. The second subject was investigated by Tommy Hendrix, Bruce Works, and Rod Phillips.

In the subject of drilling operations, calls were made to a local drilling company, in which information was provided and an offer was made to visit their facilities and view the actual drilling procedures. A preliminary data search was conducted using GTEC and INSP databases in the areas of lunar geology, lunar surfaces and drilling operations. Previous ME 4182 design reports and prototypes were utilized.

Equal work was performed by each member of the group.

WEEKLY PROGRESS REPORT
ME4182 - GROUP 5
JANUARY 18, 1989

On Monday January 16, the entire group visited Precision Drilling of Marietta and talked with Kirk Smith concerning drilling operations as they applied to this company. Mr. Smith was very helpful in explaining percussion drill operations as they apply to granite quarrying. We were shown drilling components such as bits, couplings, and centralizers, mounted upon the main drilling assembly. While we do not feel that these particular components will directly apply to our design problem, Mr. Smith was gracious enough to provide us with further contacts in Tennessee concerning caisson drilling techniques.

On Tuesday, we drafted our initial problem statement and discussed assumptions which apply to our project. Also, further research was done on an individual basis concerning the casings and footplates.

ERIC MASSEN	VU LE	EDDIE MORRISON
TOMMY HENDRIX	BRUCE WORKS	ROD PHILLIPS

WEEKLY PROGRESS REPORT
ME4182 - GROUP 5
JANUARY 25, 1989

We met twice for two brainstorming sessions. We discussed the possibility of ribbon casing. Other ideas on casings were slotted flexible segments which could be directly fitted around the rod and whole sections of casings to act with the rod as an integral unit while attaching to the drill string. Another consideration was using Skitter to physically drive the casing down while augering the hole. We discussed possible joining methods for casing such as direct butt contact, tongue-and-groove, threading, gluing, pinning, and laser welding.

Footplate technology was investigated. We referenced the Longyear vendor catalog in the VSMF. Possible configurations for the footplate include a screw actuator scissor device, variations of the gravity activated slips, a forked auger gig, and an offset radial slip jaw configuration.

Further research into soil mechanics produced little information due to the limited available data. Discussion was held on the possibility of utilizing the data base off-campus.

ERIC MASSEN VU LE EDDIE MORRISON

TOMMY HENDRIX BRUCE WORKS ROD PHILLIPS

PROGRESS REPORT
MR. BRAZELL
GROUP 5
2-1-89

For this week, we concentrated on the principles upon which the casing assembly could be inserted into loose regolith. On Saturday, January 28, the group purchased certain hardware items to simulate the hole casing operation on a reduced scale. Items purchased from The Home Depot and Builders Square were:

* An 11/16 in. wood auger
* A 2 foot length of 3/4 in. type M copper tubing
* A 3/4 in copper coupling
* A plastic utility bucket
* 1 50 lb. bag of playsand
* 2 hacksaw blades

Modifications were made to the wood auger in removing the cutting blades and reducing the outside diameter to allow for free movement within the copper tubing. A method of allowing the casing to sink itself simultaneously with drilling action of the auger. Another creative method investigated was pushing the casing in conjuction with the turning of the auger. These methods proved encouraging and further study will be pursued.

We discussed scaling our model to get a better idea of the actual drilling operation. Procuring a hand held auger and four inch PVC casing, we will investigate this casing placement operation under motorized drive conditions and deeper soil depths.

WEEKLY PROGRESS REPORT
ME4182 - GROUP 5
FEBRUARY 8, 1989

In preparation for the progress report, we have narrowed our casing designs down to two possibilities. Both involve simultaneous insertion of casing and augering of the hole. One involves direct axial pushing by Skitter into the regolith. The other method utilizes the rotational work supplied during drilling to twist a threaded casing into the regolith. The threading could consist of spiral fins cast or attached to the outside of the casing.

A new footplate conception was discussed. It consists of two pinned arms that may pivot parallel to the ground with minimal angular displacement. Driving action could be provided by a collar internally geared to mesh with the pivot pins. The same collar could be driven by a power pinion which Skitter actuates.

A final report on the Apollo Lunar Drilling System was found and provided a better model for the soil than what we have conceived. Many other avenues of thought were opened to us by this article published by Martin Marietta.

ERIC MASSEN VU LE EDDIE MORRISON

TOMMY HENDRIX BRUCE WORKS ROD PHILLIPS

PROGRESS REPORT
MR. BRAZELL
GROUP 5
2-15-89

The group was divided in two work groups. Two members are concentrating on the casing design, while the remainder of the group is pursuing the footplate design problem.

Some frictional forces on the casing were estimated. A civil engineering professor, Dr. Baccus, was consulted as to the accuracy of these estimations. The external friction on the casing was a reasonable assumptiom. However, the estimation for the internal friction was off by several orders of magnitude. Alternate methods for casing the hole were discussed.

One solution to the casing cutting problem is installing an electrically activated blade installed within a rotating cylinder which will fit over a casing segment to be driven by SKITTER's drive system.

Lubricating possibilites were investigated with regard to the internal footplate gearing. These possibilities included solid lubricants and plastic type bearings which require no additional lubrication. Further discussion were held in the area of joint protection by a corrugated bellows arangement.

Vu Le	Erik Maassen	Tommy Hendrix
Rod Phillips	Bruce Works	Eddie Morrison

PROGRESS REPORT
GROUP 5 ME4182
FEBRUARY 22, 1989 MR. BRAZELL

The week was spent performing calculations with regard to the casing and the physical effects such as buckling analysis and skin friction. These methods of computation were performed on a spreadsheet setup from a software package QUATTRO. Investigation was continued and is being narrowed down in the determination of a material to manufacture the casing from through the use of the spreadsheet.

The footplate design has reached final stages such that a format is being developed to perform computations for shear and normal stresses, torsional constaints, gearing specifications, and power requirements. Lubrication for the footplate is still being pursued. The rod gripping mechanism of the footplate will consist of a set of opposing vise-type jaws driven by a common worm gear with opposite threading on opposing ends. Contemplation of a pin set in these jaws to aid in the slippage of the rod string has materialized on paper. Running through a few simple calculations, the torsional force required to shear these pins from the jaws is larger than will be delivered by Skitter.

In the Tuesday meeting, acknowledgement of the final report was mentioned. A brief organization of the report was outlined mentally.

Tommy Hendrix	Bruce Works	Rod Phillips
Eric Maassen	Vu Le	Eddie Morrison

PROGRESS REPORT
GROUP 5 ME4182
MARCH 1, 1989 MR. BRAZELL

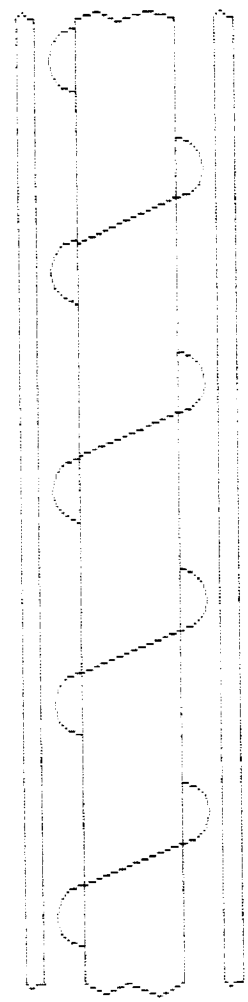
This week Group 5 concentrated on finalizing several design criteria. The mating material was [illegible] located at the [illegible]. From this [illegible] calculations [illegible]. Several [illegible] the calculations [illegible].

The final footplate design consists of [illegible] with four supporting legs which contain anchors to be screwed into the ground. The [illegible] in conjunction with [illegible] plane. Metal bellows will protect the mechanical operation from the environmental wear. Calculations have been performed relating to the [illegible] the footplate will [illegible], as well as [illegible] from the [illegible] to the [illegible] the footplate [illegible]. The [illegible] of the footplate were modeled [illegible]. Flexure was analyzed and [illegible] for the operation of the footplate.

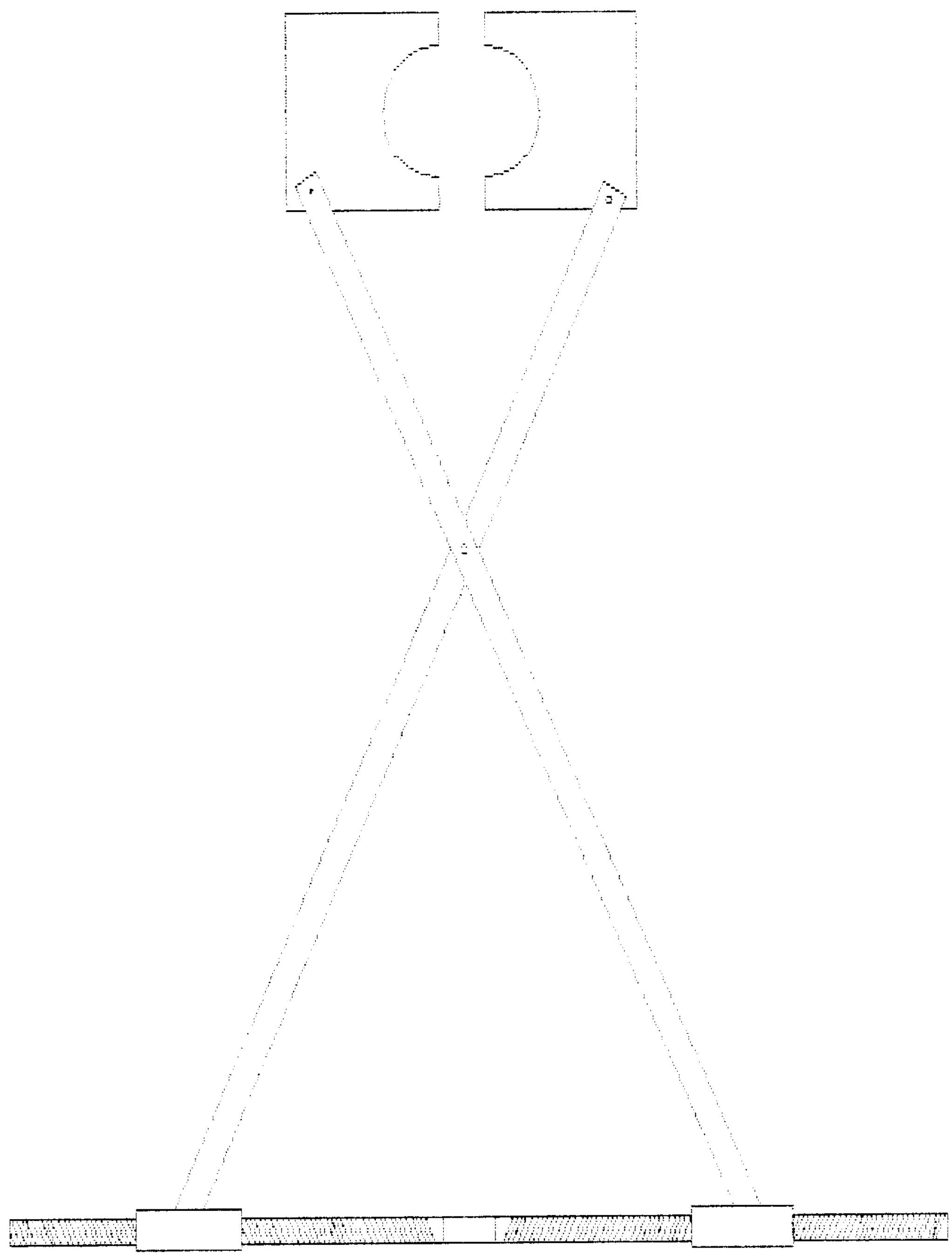
[illegible] [illegible] Rod Phillips

[illegible] [illegible] [illegible]

SKETCH 1



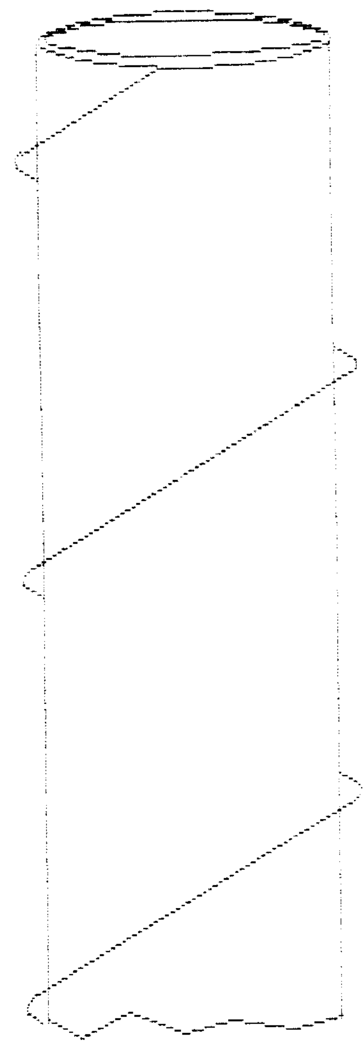
CLAMP1



ORIGINAL PAGE IS
OF POOR QUALITY

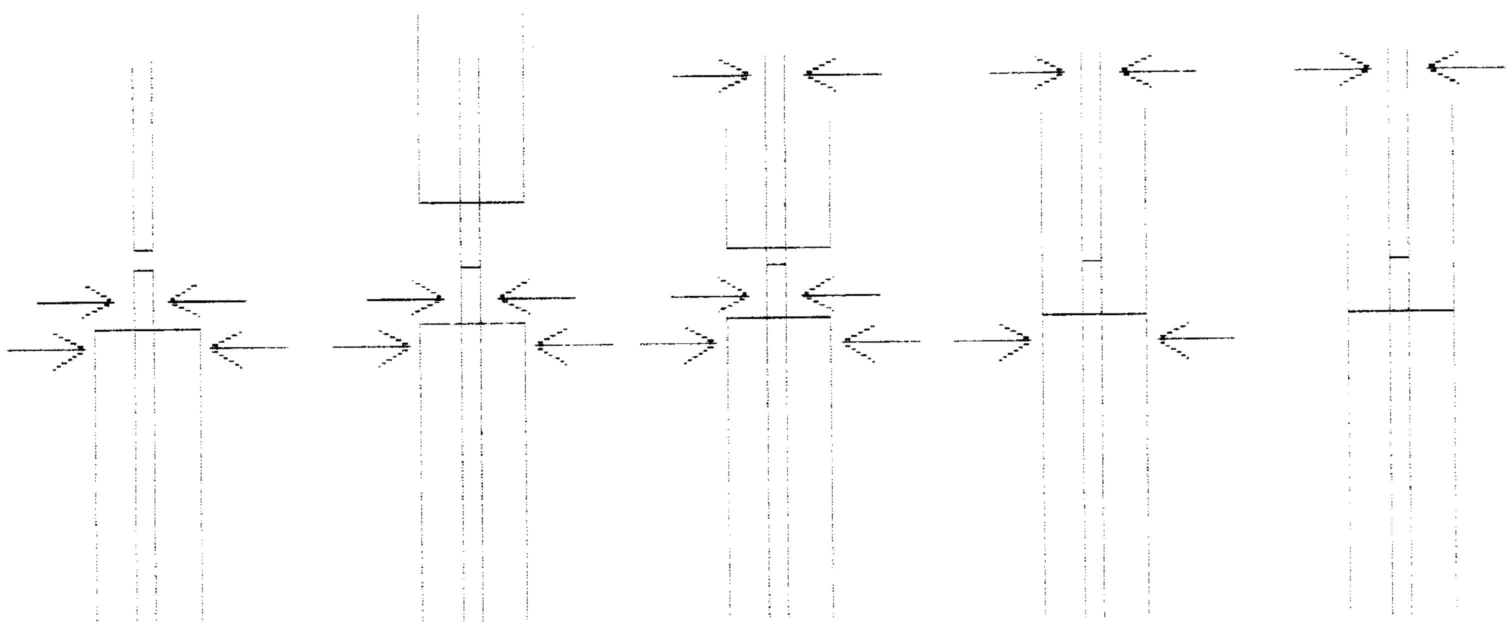
SKETCH 3

THREADED CASING

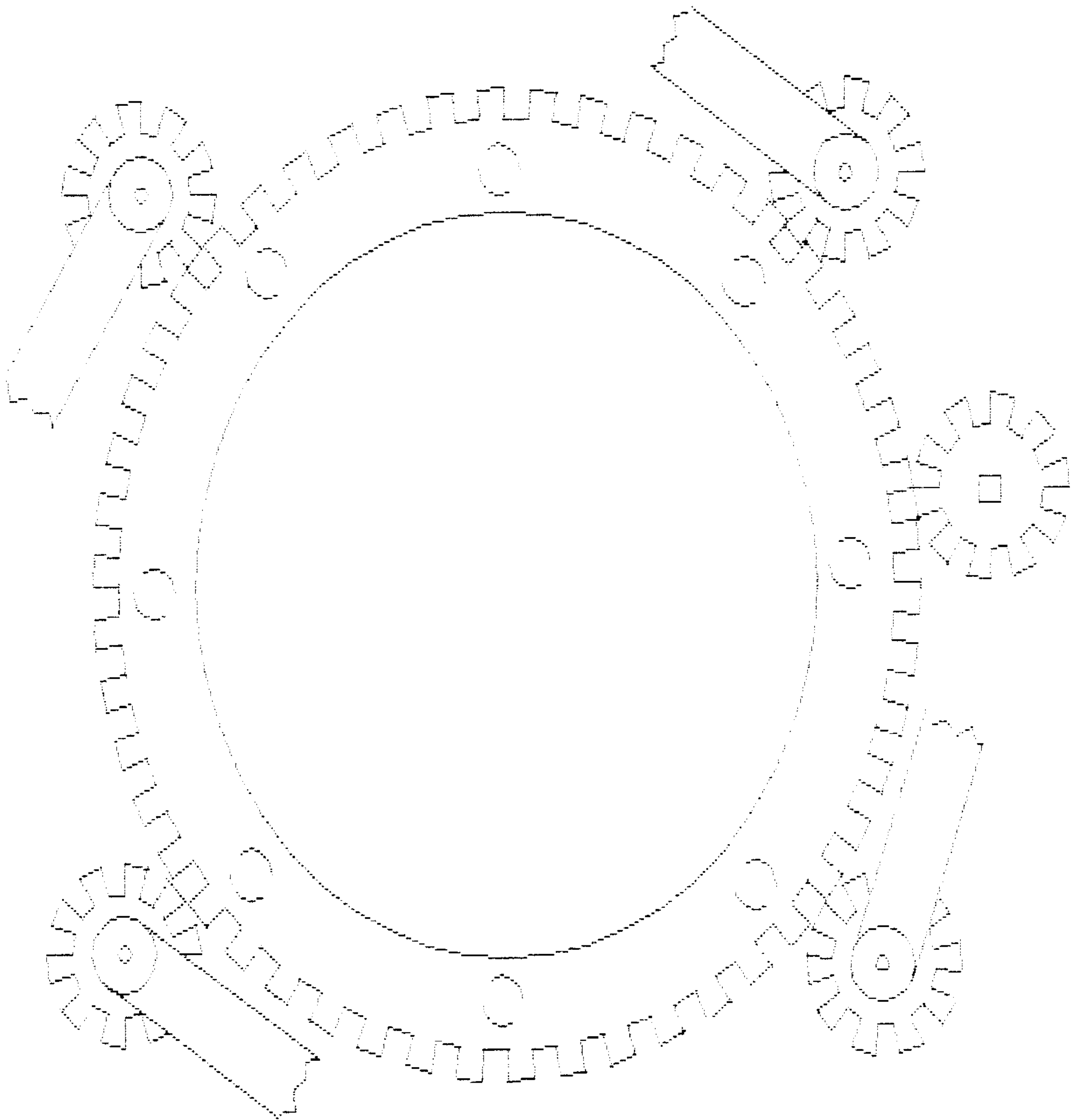


ORIGINAL PAGE IS
OF POOR QUALITY

FLOW PROCESS FOR ADDING CASING AND ROD TO STRING



CLAMP 2



ORIGINAL PAGE IS
OF POOR QUALITY