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MECHANICAL DESIGN ENGINEERING
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LUNAR HAND TOOLS

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1. Abstract

Our project is to determine and design tools useful for operations and maintenance tasks on the lunar surface. Our primary constraints are the lunar environment, the astronaut's space suit and the strength limits of the astronaut on the moon. We have designed a multipurpose rotary motion tool and a collapsible tool carrier. For the rotary tool, we have specified a brushless motor and controls, chose a material for the housing, recommended bearings and lubrication and designed a planetary reduction gear attachment. For the tool carrier, we have designed primarily for ease of access to the tools and fasteners. We have selected a material and performed structural analysis on the carrier. We have made recommendations about the limitations of human performance and about possible attachments to the torque driver.

2. Problem Statement

2.1. Background

The plans to put a man on the moon for a long period of time presents the design engineer with a vast set of problems. The lunar base will require a wide variety of construction, maintenance, operation and repair tasks that must be performed entirely by space-suited astronauts. The designers are also

presented with the problems of predicting the tasks that an astronaut will encounter on a lunar base, and designing tools to accommodate them - in a way that will maximize their usefulness for a wide variety of tasks, and in a way that will minimize their weight - a critical factor since transporting them to the lunar surface is very difficult and expensive.

2.2 Performance Objectives

Our task is to design a device that will supply rotary motion sufficient for fastening and unfastening tasks, for drilling and cutting tasks, and for other unspecified tasks requiring rotary motion.

We are also designing a carrier to allow the astronaut easy access to various hand tools and easy portability.

2.3 Constraints

The primary constraints are environmental. First, the astronaut is working in 1/6 of earth's gravity, and on a relatively slippery surface. Second, the astronaut's motion and vision are limited by the space suit. Third, the tasks the astronaut expects to encounter are currently undefined. Fourth, the astronaut is limited by the amount of time he can work outside his habitat. Fifth, the temperatures on the lunar surface range from -200 to +260 degrees F. Sixth, the astronaut cannot sense sound and vibration - so he is deprived of an important means of feedback.

3. Description

Our solution to the design problem stated is to provide a handheld torque driver with a gear reduction accessory to provide for high torque operations. We are also constructing a collapsible tool carrier designed for ease of access and for portability.

3.1. Torque Driver (Figure 1.)

3.1.1. Specifications

The torque driver shall provide a maximum no-load speed of rpm. The driver shall provide a maximum torque of 200 inch-pounds at a minimum speed of 300 rpm.

3.1.2 Motor

The motor we have chosen is a high performance DC brushless motor connected directly to the shaft. It will use samarium-cobalt magnets to operate reliably in a high temperature environment. The windings have been encapsulated to protect them from lunar dust. We have selected for purposes of example the Inland Motors Model No. RBE-01205, supplied with high efficiency laminations to meet these requirements. This motor supplies a peak torque of 25 inch pounds and runs at a

no-load speed of 1500 rpm. The motor is described in the vendor literature in Appendix 1.

3.1.3. Controls

Inland Motors supplies a control module which allows for maximum torque to be preset by the user. The motor controls we selected are of the BCL Series. The control module is described in Appendix 2.

3.1.3 Controls, continued

The motor torque shall be preset by a 2.75 inch diameter ring knob around the back of the motor as shown in Figure 2. The motor shall be turned on and off by a spring loaded trigger switch mounted on the handgrip so that it falls naturally under the astronaut's index and middle fingertips. The panel on the back of the motor housing also has lighted pushbutton switches to reverse the direction of the motor and to switch on a small work light mounted at the front of the driver.

3.1.4. Shaft

We have selected a shaft that is 3/8 inches in diameter and 4 1/4 inches long. The shaft shall be made of AISI/SAE 4340 steel, quenched and tempered. The material has an ultimate tensile

strength of 217 kpsi, a yield strength of 198 kpsi and specific gravity of 0.284 lb-in³. The steel has a Brinell hardness rating of 440. For our torque driver application, the steel selected will give a factor of safety of 5.

3.1.5 Bearings

The driver will use two plain type F-2 bronze bushings of type FM-3029, one at the front of the shaft and one capping the back of the shaft. Both bearings are 1/2 inch long and are 1/4 inch thick.

3.1.5.1 Lubrication

Due to problems with outgassing, petroleum based lubricants are unfeasible. Therefore we recommend using molybdenum disulfide for its high temperature performance characteristics.

3.1.6 Housing

The housing shall be made of polyimide plastic molded to shape. The thickness of the housing shall be 3/16 inch, except for the bearing mounts and the internals of the handle. The motor and controls housing shall be cylindrical in shape with the following dimensions: 2 3/8 inches in diameter, and 6 inches in length. The end effector connection shall protrude 3/4 inch from the front of the housing. The handle shall be mounted under the center of

mass of the cylinder at an angle of 78 degrees from the horizontal. The handle shall be oval in shape to conform to the gloved astronaut's hand, and will be 1 inch in thickness and 2 1/2 inches wide at the base, and 1 1/2 inch wide around the hand grip. Mounted at the front of the device below the end-effector connection will be a small halogen lamp on a flexible shaft. There shall be four 3/8 inch diameter holes drilled into the front of the housing to allow attachment of the planetary reduction gear and other tool attachments.

3.1.7 Battery Pack

The batteries shall be of the nickel-cadmium type and shall provide a constant 24 volts and a maximum current of 20 amperes. The batteries shall be rechargeable. The tool shall be supplied with a reserve battery pack so that one pack can be charging as the other is being used in the tool. The pack shall fit into the handle of the torque driver and be hermetically sealed. The battery pack shall weigh 29 ounces.

3.1.8 Finish

The torque driver shall be finished with a reflective paint or plated with chrome to minimize radiative heat transfer from the environment to the tool.

3.1.9 Planetary Reduction Gear Attachment (Figure 3.)

The torque driver shall be supplied with a detachable planetary reduction gear. The gear shall be mounted with 3/8 inch pins onto the front of the driver housing, engaging the shaft. The gear shall provide a 10:1 reduction in speed and a corresponding 10:1 increase in torque. The gear train shall consist of a 3-3/8 inch annulus gear with 162 teeth, a 3/8 inch sun gear with 18 teeth attached to the end of the motor shaft, and 3 - 1 1/2 inch planet gears with 72 teeth attached to a rotating planet carrier that is fixed to the output shaft. The entire gear train shall fit into a housing that is 2 1/2 inches long and 3 7/8 inches in diameter. The bearings, gears, and housing shall be constructed of the same materials as their counterparts in the torque driver. The torque driver shaft connection shall accept the end-effector connection at the end of the torque driver, and the output shaft of the planetary gear attachment shall have the same end-effector connection as the torque driver.

3.2 Tool Carrier

As seen in Figure 4, our tool carrier design is a cylindrical container composed of three major compartments. It is 30 inches in diameter, 25 inches tall when in closed position and 71 inches tall when in open position. The bottom compartment of the tool carrier is comprised of wedges which act as storage bins for

fasteners and smaller tools, Figure 5. Each compartment has an angled bottom surface. The entire section rotates about a central shaft. This exposes an individual opening in the shell when the door is opened.

Located above this section are two telescoping cylinders. Each of these cylinders contains tools clipped along its inner and outer surface areas. To access the tools on the interior surface area, a front over center latch is opened in order to swing the doors back. Figure 6 demonstrates this procedure. Because there are two pivot points, the cylinder becomes three curved walls when in open position. To use the tool box, the astronaut simply places the unit on the lunar surface, lifts the tool box out of the protective shell and locks in open position.

In order to close the tool box, a release mechanism on the handle is activated by slightly raising and twisting the handle. This unlocks the sections and the tool box slides closed. The unit is then locked using the over center locks on the outer shell and the unit is ready for transportation to another location.

The following is a list of fasteners that have been taken into consideration in the design of the tool box and will be provided space in the bottom compartment.

A. Rivets

- 1. Huckbolts**
- 2. Riv-O-Seal**

B. Bolts

- 1. Regular Hexagonal Head**
- 2. Hexagonal Socket Head Cap Screw**

C. Nuts

- 1. Finished hex and hex jam nuts**
- 2. "McLaughlin" clinch nuts**
- 3. Weld nuts**

D. Washers

E. Retaining rings

F. Pins

- 1. Ball-loc quick release pin**
- 2. "Harley" quick release pin**
- 3. "pip" quick release pin**

G. Quick release fasteners

- 1. Camloc quick release fastener**
- 2. Camloc stressed panel fastener**
- 3. Simmons Quick-loc quick release fastener**

H. Clamps

I. Velcro

J. Zippers

K. Rope

4. Analysis

4.1 Torque Driver

4.1.1 Human Performance Limits

Figure 7. gives a visual description of the limits of human performance when performing rotary motion tasks.

The red line on the graph shows the maximum torque the astronaut can resist hand held. We have assumed that the astronaut and pack weigh 420 pounds on the earth, we have assumed a friction coefficient of 0.1 and that the astronaut's arm is extended horizontally above the lunar surface. Assuming that the astronaut is a rigid body, from statics, we show that the astronaut can resist 350 inch pounds of torque before slipping on the lunar surface.

The blue line represents the maximum torque producible by a male with a 2 foot long lever held vertically in front of him. As with the powered torque driver, the astronaut only has 7 pounds of frictional force at his feet to resist any pushing or pulling force. With a 24 inch bar, the astronaut can produce about 170 inch pounds of torque before slipping on the lunar surface. The graph shows a maximum speed of 30 rpm. This is shown only for

comparison purposes. In practice, a 2 foot torque bar would be used only in low repetition operations.

The brown line represents the maximum torque producible by a male with an 8 inch diameter handwheel. From human factors research, it has been shown that a male can produce 35 inch pounds. We have also shown a maximum speed of 30 rpm for purposes of comparison, and have neglected the effects of fatigue. See Reference 15.

The green line represents the expected maximum torque from our torque driver. As shown, it has been designed to operate well within our predicted maximum torque limits.

4.1.2 Motor

We selected a brushless motor to conserve space and to reduce the number of parts in the tool. The tool uses samarium cobalt magnets for improved performance at high temperatures. We are using direct current to simplify power requirements and portability. The elimination of brushes on the motor offers several advantages. First, because there is nothing directly contacting the motor, there is no radio frequency interference from arcing. The brushless motor also provides a high torque to weight ratio, low heat generation, and low maintenance.

Mechanical drag is also eliminated within the motor. The brushless motor provides smooth, jerk free operation, can accelerate and decelerate very rapidly, and can reverse direction easily. The additional weight disadvantage of the commutator will not offset these advantages.

4.1.3 Shaft

We selected 4340 steel because it combines deep hardenability with high ductility, toughness, strength and reliability. The alloy has a wide operating temperature range (-328 to +752 degrees F.). It has high fatigue and creep resistance and is especially immune to temperature embrittlement and will retain its strength.

For a shaft in pure torsion, we designed around the ASME Code for Transmission Shafting from reference 20 assuming no bending moment applied to the shaft. The shaft design formula is:

$$d = \left(\frac{32n}{\pi} \left[\left(\frac{T}{S_y} \right)^2 + \left(\frac{M}{S_e} \right)^2 \right]^{1/2} \right)^{1/3}$$

d=shaft diameter

T=applied torque

S_y=material yield strength

M=bending moment (M = 0 for this application)

S_e=endurance limit

Refer to the shaft calculations program and data in Appendix 3. for the details of our shaft calculations. We have designed around a factor of safety of 5, a maximum torque of 200 inch pounds and 200 kpsi.

4.1.4 Planetary Reduction Gear Calculations

We have designed for a 10:1 increase in torque with corresponding reduction in speed. We have also designed around a sun gear with a 3/8 inch pitch diameter. For a simple planetary gear, the following formula applies:

$$\begin{aligned} 10 &= N_r / N_s + 1 \\ &= d_r / d_s + 1 \end{aligned}$$

N_r = number of teeth in ring gear

N_s = number of teeth in sun gear

d_r = pitch diameter of ring gear

d_s = pitch diameter of sun gear

$$d_p = (d_r - d_s) / 2$$

d_p = pitch diameter of planet gear

We designed using AGMA standard spur gears with 20 degree gear teeth (see reference 10.). For these gears, the minimum number of teeth per gear is 18. Using this figure applied to the sun gear, we have determined from the following formula that the diametral pitch for the planetary gear train is 48.

$$N = Pd$$

N=number of gear teeth

P=diametral pitch

d=gear diameter

Summarized below are the geometries of the gears in the reduction gear housing.

<u>Gear</u>	<u>Pitch Diameter</u>	<u>Number of Teeth</u>
Ring	3 3/8 Inch	162
Sun	3/8 Inch	18
Planet	1 1/2 Inch	72

4.1.5 Housing

We chose the polyimide because it retains high strength at extreme temperatures and their wear resistance. The plastic will also be light in weight and low in thermal conductivity. The handle angle of 78 degrees was chosen to maximize the user's comfort (see reference 15.)

4.1.6 Weight Analysis

Each component of the torque driver weighs the following, and is summed below for a total estimated weight of the driver.

<u>Component</u>	<u>Weight</u>
Motor	1.26 pounds
Shaft	(included in motor weight)
Controls	0.25 pounds (estimated)
Batteries	1.81 pounds
Gear Assy.	3.00 pounds (estimated)
Housing	1.50 pounds (estimated)
Total	6.32 pounds

4.2 Tool Carrier

4.2.1 Operation

To operate the tool carrier, several steps need to be taken in a particular order to open it. First the legs must be lowered and locked into place. They must then be leveled using the one inch adjustment latches. Once the outside over center latches are unlocked , the sections can be raised. The upper section must then be locked in place by turning the handle. The top two sections may now be unlocked and opened to expose the tools on the interior surfaces. The bottom section may be utilized by opening a door to expose one section at a time.

To close the carrier, steps must also be completed in order. First close the door to the bottom compartment and lock this and the upper two sections in closed cylindrical position. Next release the locking mechanism for the bottom compartment which is located on the handle and lower into outer protective shell. To close the top section, turn the handle counterclockwise to unlock and lower in bottom sections and outer shell. Finally, lock over center outside latches , raise extending legs and rotate to lock into closed and traveling position.

4.2.2 Performance

The tool carrier is divided into several parts. The outer shell is 25 inches tall and 30 inches in diameter. It houses all three tool compartments when in closed position and helps support the compartments when in open position. The rotation bin compartment is 12 inches tall and 29 inches in diameter. It contains six wedged sections all of which hold tools. Three of the compartments also hold fasteners in a front section which is 4 inches deep, semi-circular in shape and divided in two. The wedged sections rotate about a central shaft and there is a door in the shell to expose one compartment at a time. The lower folding shell, located above the rotation bin compartment, is 13 inches tall and 24.5 inches in diameter. It holds tools on both the inner and outer surface areas. The tools on the inner surface can be accessed by unlocking a front over center latch. This section also rotates about the central shaft. The upper folding shell is 13 inches tall and 20 inches in diameter. This section does not rotate due to the problem of telescoping but tools on the interior surface are accessed by opening an over center latch on the front of the section.

Each part of the tool carrier must be locked in place whether in open or closed position. There are several different types of locks or latches used in the carrier. First, on the outer shell, 3 overcenter type latches lock the telescoping shells to the outer

shell. The upper folding shell locks in to open position using a tongue and groove locking mechanism. When the shell is fully raised, the handle is turned clockwise to locked position. The rotation bin shell is locked and unlocked using a spring release mechanism. Wires connected to the handle are connected to spring loaded pins at the bottom of the shell. These wires are routed through the central shaft. The head of the pins are slanted to slide over the supports, located inside the outer shell, when opening the carrier. To release the locking mechanism, the handle will be raised and will retract the pins and the shells will fold down into the outer protective shell. The upper and lower folding shells and rotating bin door will lock in closed position using an over center latch. After the leg extensions have been retracted, the legs will swing around 180 degrees on a pin and will lock using a clip latch on the side of the outer shell.

In each of the compartments, there is an approximate maximum number of tools which can fit. In the upper folding shell, 8 to 9 tools can fit on the outer surface taking into account room for the two hinges. On the inner surface, approximately 10 to 11 tools will fit. The lower folding section can hold 10 to 11 tools on the outer surface and 12 to 13 on the inner surface. All approximations take into account a 4 inch clearance for the gloved hand and 2 inches for an average width of a tool. See Appendix 5. The tools in all of the sections will be fastened using

over center latches with adjustable straps to attain best use of surface area. The bottom compartment will contain a variable amount of tools depending on size and orientation.

4.2.3 Structural Analysis

The basic support of the tool carrier consists of three major parts, fig.s-1. Part A is a cylindrical rod of length 16 inches, outside diameter $3 \frac{3}{16}$ inches, and thickness $\frac{1}{16}$ inch. This cylinder provides support and stability for the top telescoping section that will open and close. It will be made of Uespel (see Appendix 9) as will the entire tool box. The strength calculations for this cylinder are given in Appendix 4. The cylinder will extend 3 inches into Part B to provide a flexural stability between the two cylinders.

Part B is also a cylinder of length 25 inches, outside diameter 4 inches, and thickness $\frac{1}{8}$ inch. This cylinder will provide support for not only the lower section that opens up, but also for the rotating wedged section containing fasteners and other tools. A groove will be machined inside the cylinder for the sliding and locking of Part A.

Part C is a circular plate $\frac{1}{8}$ inch thick, 29 inches in diameter, with a 4 inch diameter hole in the center. This hole is the same size as the outside diameter of Part B. These two parts will be

molded together to form the base of the tool box structure. Part C will also act as a base for the rotating compartment.

Structure of Rotating Bins

The support structure of the rotating bins also has three main components, fig. S-2. Part D is a cylinder that is 12 inches long, has an inside diameter of $4 \frac{1}{32}$ inches, and a $\frac{1}{8}$ inch thickness. This provides support for the side of the bin compartments, Part E. There are six sides to separate the different bins. These supports are 12 inches high at the center decreasing to 2 inches at the outer edge. These are $\frac{1}{8}$ inch thick and are molded to both Parts D and F. They are placed so that each compartment forms an angle of 60 degrees. Part F is a circular plate exactly like Part C of the internal support structure except the center hole has a $4 \frac{3}{16}$ inch diameter. It provides the bottom support of this section and rotates on top of Part C.

Structure of Lower Folding Shell

The structure of the lower folding shell consists of three parts, Fig. S-3. Part G is a cylinder exactly like Part D except that it is 13 inches long. It will be able to rotate about the center shaft, Part B. The base support, Part H, will be molded to Part G and provide stability for the outer shells, Part I. Part H spans 90 degrees

around the outer surface, leaving 270 degrees to be spanned equally by the two shells, Part I. Both Parts H and I are 1/8 inch thick. Part I will provide the surface on which to clip the tools. They are hinged to Part H to allow them to rotate open and closed. When closed, the two shells, Part I, and the stationary shell, Part H, will form a cylinder with a 24.5 inch diameter.

Structure of Upper Folding Shell

The structure of the upper shell is very similar to the lower shell, Fig.S-4. Part J is exactly like H except it extends to only 20 inches. It will be molded to Part A at the top of the cylinder to allow for retraction into the lower shell. The outer shells, Part K, are hinged to Part J and form a closed cylinder with a 20 diameter.

Outer Shell

The outer shell is being used as part of the support structure to make full use of every component of the tool carrier. Figure 4 shows the cylindrical shell that surrounds the entire tool box. It is 1/8 inch thick and has a 30 inch diameter. The supports on the inside of the shell are the bracing points for the spring-loaded release mechanism described in the operations section. These also act as a guide when the box is slid into and out of the shell. There is a solid circular plate exactly like Part C except it has a 30

inch diameter with no hole in the middle. It is molded to the shell and acts as a support for the entire tool carrier in its closed position. Three legs support the carrier and act as a leveling devices. A detail of the leg can be seen in Fig. 8 and its function is described in the operations section.

4.2.4 Ergonomic Factors

The lunar tool carrier is designed so as to fall within a comfortable range of motion as best as possible for the astronaut when the carrier is in the open position. An optimum range of access was calculated in order to guide the designed dimensions of the carrier when open. The goal in doing this was to minimize the amount of motion required to access the stored tools.

Calculations were carried out using anthropometric data to identify these optimum open dimensions. It was assumed that the most convenient access would simply involve reaching down at a 45 degree angle while standing straight up. It was assumed that the convenient arrangement for the astronaut would involve him having easy access to the tool carrier by reaching down with arm straightened arm at an angle of 45 degrees from the horizontal while standing upright. Refer to Appendix for a diagram and sample calculations. Given minimum and maximum arm lengths, vertical and horizontal projections were made in

relation to the astronaut's plane of motion. The vertical projection was subtracted from the minimum midshoulder height to obtain a minimum desired height of the lower compartment of 39.84 in. (101.20 cm). A calculation involving the astronaut's field of vision was then made to determine whether the previous specifications would be within the astronaut's field of vision. This field of vision, called the inferior direction, is 70 degrees down from the horizontal. The field of vision is constrained by the inferior direction in the vertical plane which is a field of 70 degrees from the horizontal.

When placing the over-center straps which will fasten the tools securely in the carrier, a minimum clearance of 3.0 in. (7.62 cm) between suspended tools in order to afford gloved hand clearance.

Design of the compartment bins in the lower section should also consider this recommendation.

The final dimensions of the tool carrier design involve some digression from the minimum height calculated. As designed, the height of the lower compartment is 29.0 in. (73.66 cm). This aberration will mean that the astronaut will have to bend over somewhat in order to reach tools in the bottom compartment. Referring to Appendix 6 will provide graphic and mathematical description of this analysis. The analysis involves chiefly relationships given by the

law of cosines. Results are that the fifth percentile anatomy astronaut will have to bend about 70 degrees at the waist and the ninety-fifth anatomy astronaut will bend about 93 degrees if he maintains the distance between himself and the tool carrier calculated previously.

4.2.5 Weight Analysis

The weight of the tool carrier material was calculated by determining surface area, multiplying by thickness to get volume and by density of the material to finally obtain a weight of 28.5 pounds. These calculations are found in Appendix 7.

5. Conclusion

We have designed a hand held rotary motion tool and a tool carrier optimized for conditions on the lunar surface, including the environmental limitations and the motion restrictions placed on the suited astronaut. We have met our design objectives set in the problem statement.

6. Recommendations

6.1 Proposed Attachments to Torque Driver

The torque driver has been designed primarily as a multipurpose tool. Described below are a few of the attachments we suggest be investigated further.

Other planetary reduction gear attachments with different reduction gear ratios could be designed as tasks become more specifically defined.

A linear motion convertor could be added to provide motion for a saber saw or for an impact tool.

An adapter to open and close valves could prove useful. The adapter could be equipped with a mechanical torque limiter to prevent overtorquing the valve stem.

A complement of fastening and unfastening devices should be specified as the tasks and fastener standards are decided upon.

A set of drill bits should be specified to complement the fastener selection.

6.2 Alternative Gearing Methods

We considered the possibility of having the reduction gearing incorporated within the tool. We looked at using a lever operated gearshift to enable the user to select the speed and torque range needed for the task. We also looked at using an automatically

engaging reduction gear operated by a flyweight mechanism on the shaft.

6.3 Task Specific Tools

As specific missions are planned for the lunar surface, tools for maintenance and repair should be planned as part of that mission, with the eventual aim in mind that those tools will be incorporated into a central tool kit. These specific tools can be designed around the torque driver or other general purpose tools.

6.4 Fastener Standards

More extensive research into fastener design on the lunar surface needs to be performed. The research should account for the environmental and human limitations unique to the moon, including the maximum torques that can be applied with hand tools, and the presence of dust and dirt and high thermal gradients. We observe that conventional bolts require a high degree of precision which creates difficulty for the installer. Bolts are not tolerant of the dust on the lunar surface. Conventional bolts also require high torques to install and remove. For these reasons, we recommend that conventional bolts and nuts not be used.

6.5 Additional Torque Driver Design Considerations

The torque driver needs to have some scheme for cooling the motor. The driver motor could be provided with a dynamic braking scheme to conserve battery power. The shaft could be plasma coated to aid in lubrication.

6.6 End-Effector Connection

An unresolved aspect of our design is the method which we connect our end-effectors to the torque driver. Several possibilities have been proposed. We have looked at magnets, which provide the advantage of easy operation and few moving parts, but may not be strong enough to hold the end-effector in place during some operations. We have also looked at a ball lock mechanism, which has the advantage of easy operation, but the disadvantage of low dust tolerance. We have considered using a pin to lock our end-effectors in place, but have rejected the idea because of its difficulty of use with gloved hands.

6.7 Miscellaneous

We have considered using a folding pad work surface or spiked boots to increase the astronaut's coefficient of friction at his feet, to enable him to resist more torque.

7. Acknowledgements

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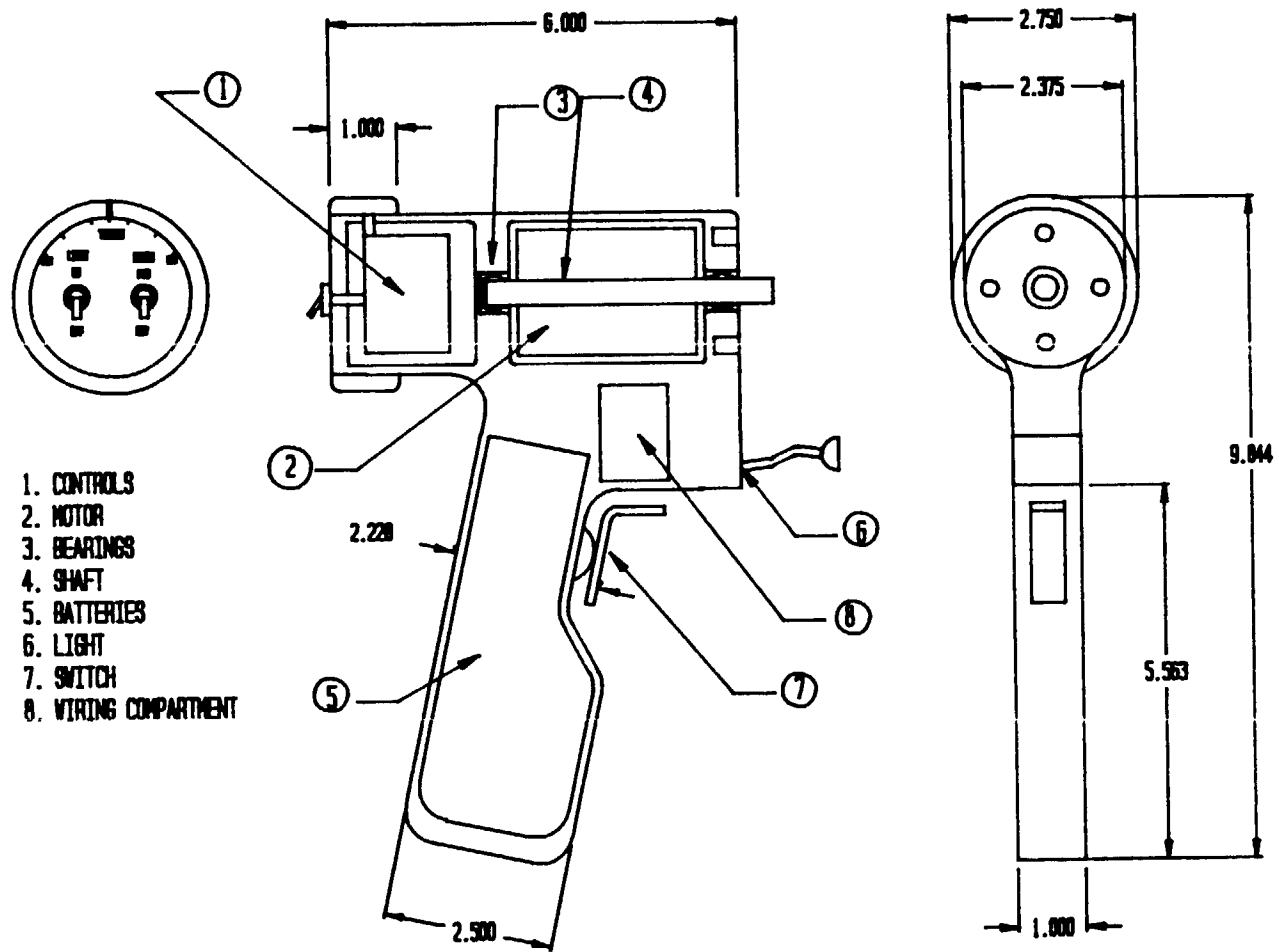
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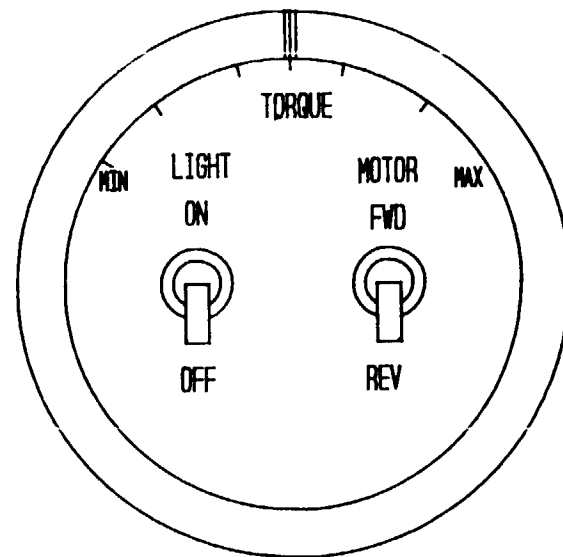
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1. CONTROLS
2. MOTOR
3. BEARINGS
4. SHAFT
5. BATTERIES
6. LIGHT
7. SWITCH
8. WIRING COMPARTMENT

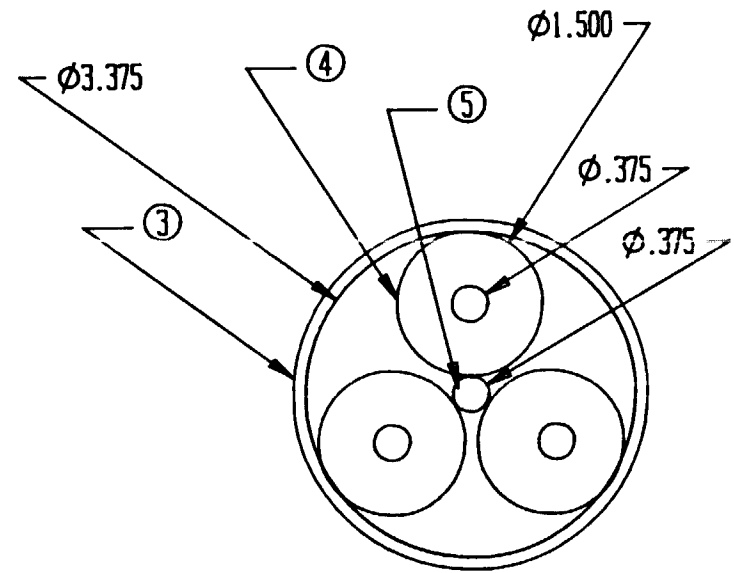
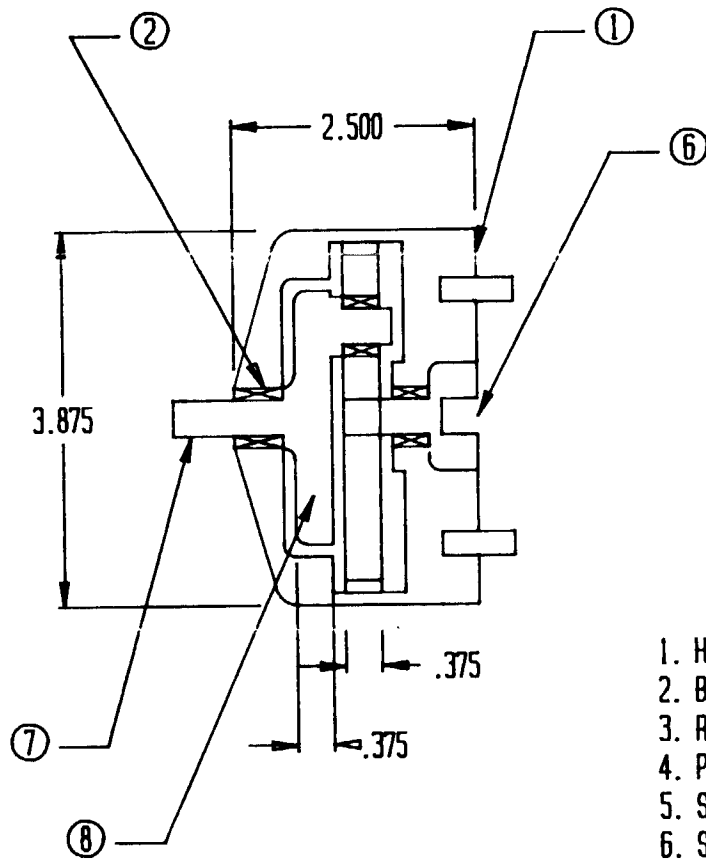
LUNAR TORQUE DRIVER

FIGURE 1.



DETAIL OF CONTROLS PANEL

FIGURE 2.



PITCH DIAMETERS

1. HOUSING
2. BEARINGS
3. RING GEAR
4. PLANET GEAR
5. SUN GEAR
6. SHAFT CONNECTION
7. OUTPUT SHAFT
8. PLANET CARRIER

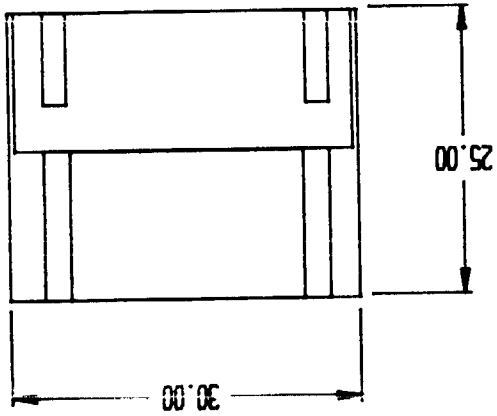
PLANETARY REDUCTION GEAR AND HOUSING

FIGURE 3.

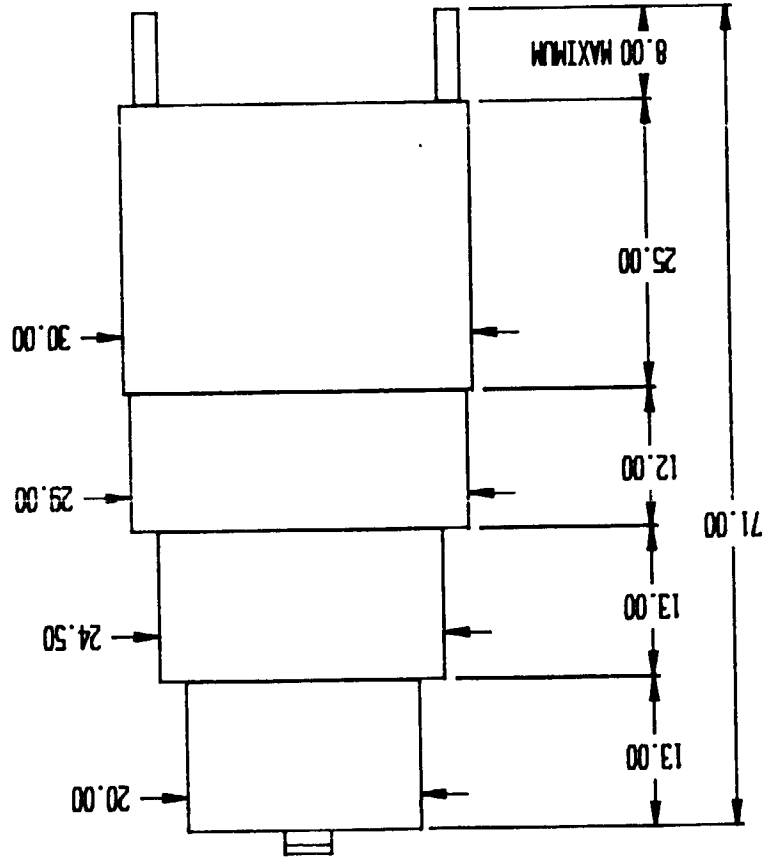
LUNAR TOOL CARRIER

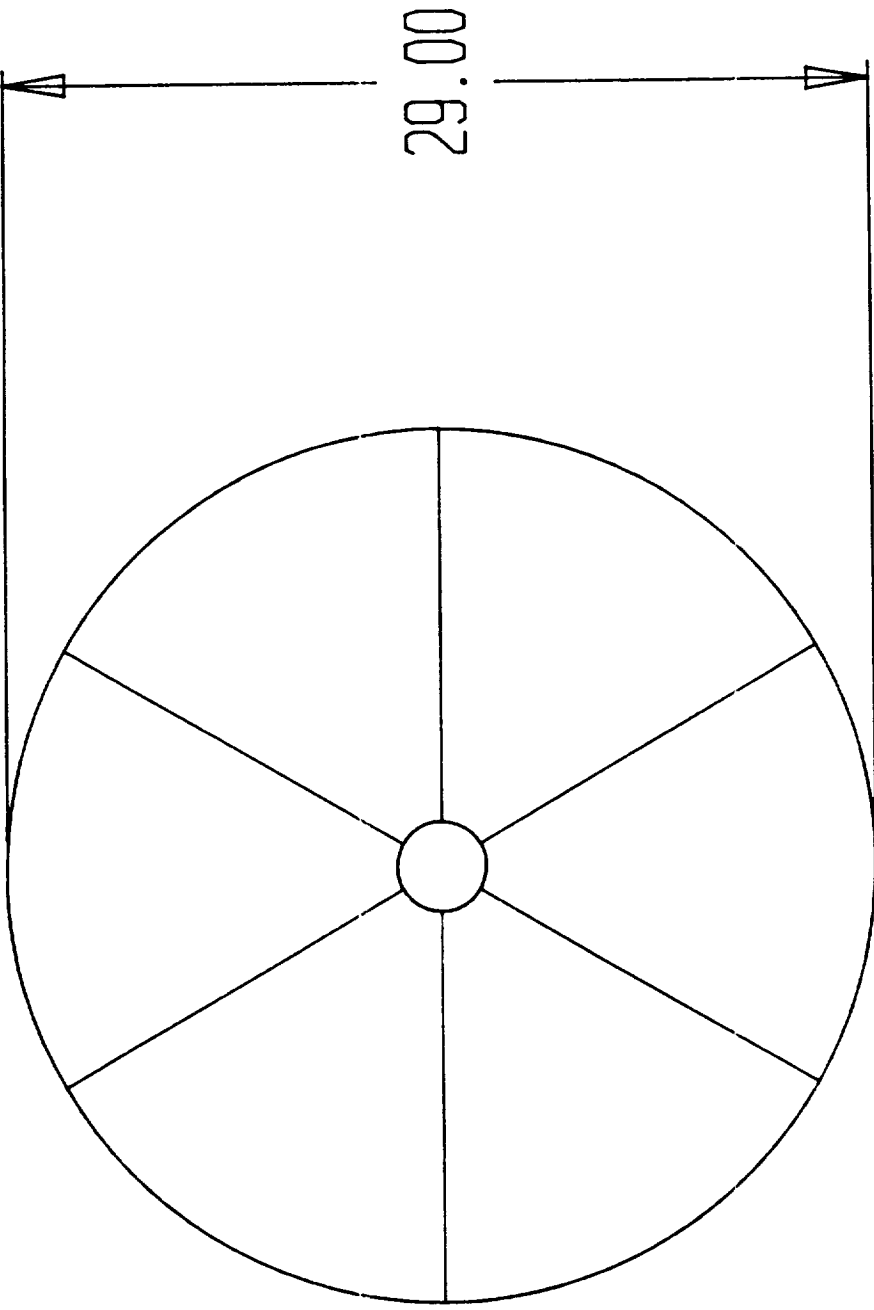
FIGURE A.

SHOWN IN CLOSED POSITION



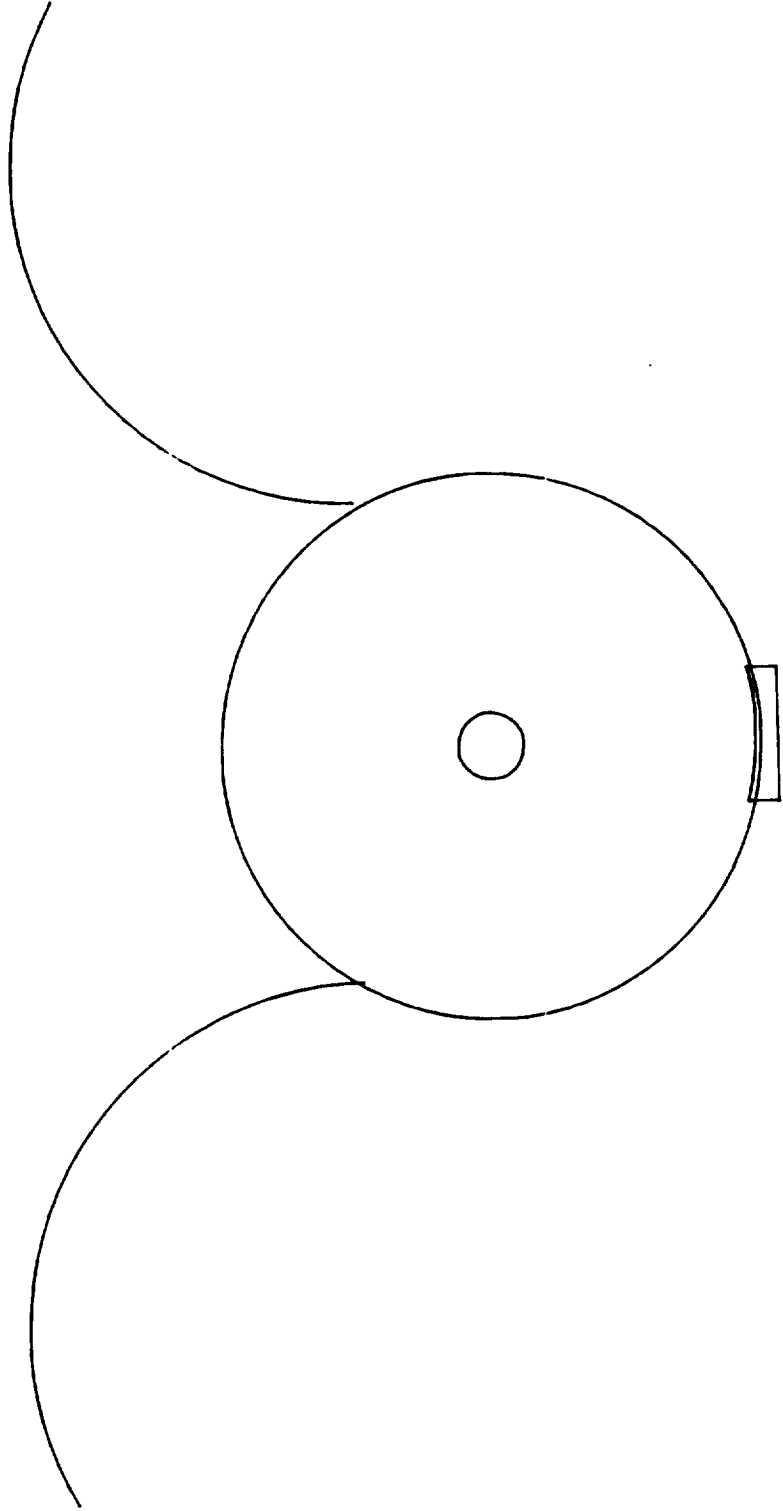
SHOWN IN OPEN POSITION





TOP VIEW OF ROTATING BINS

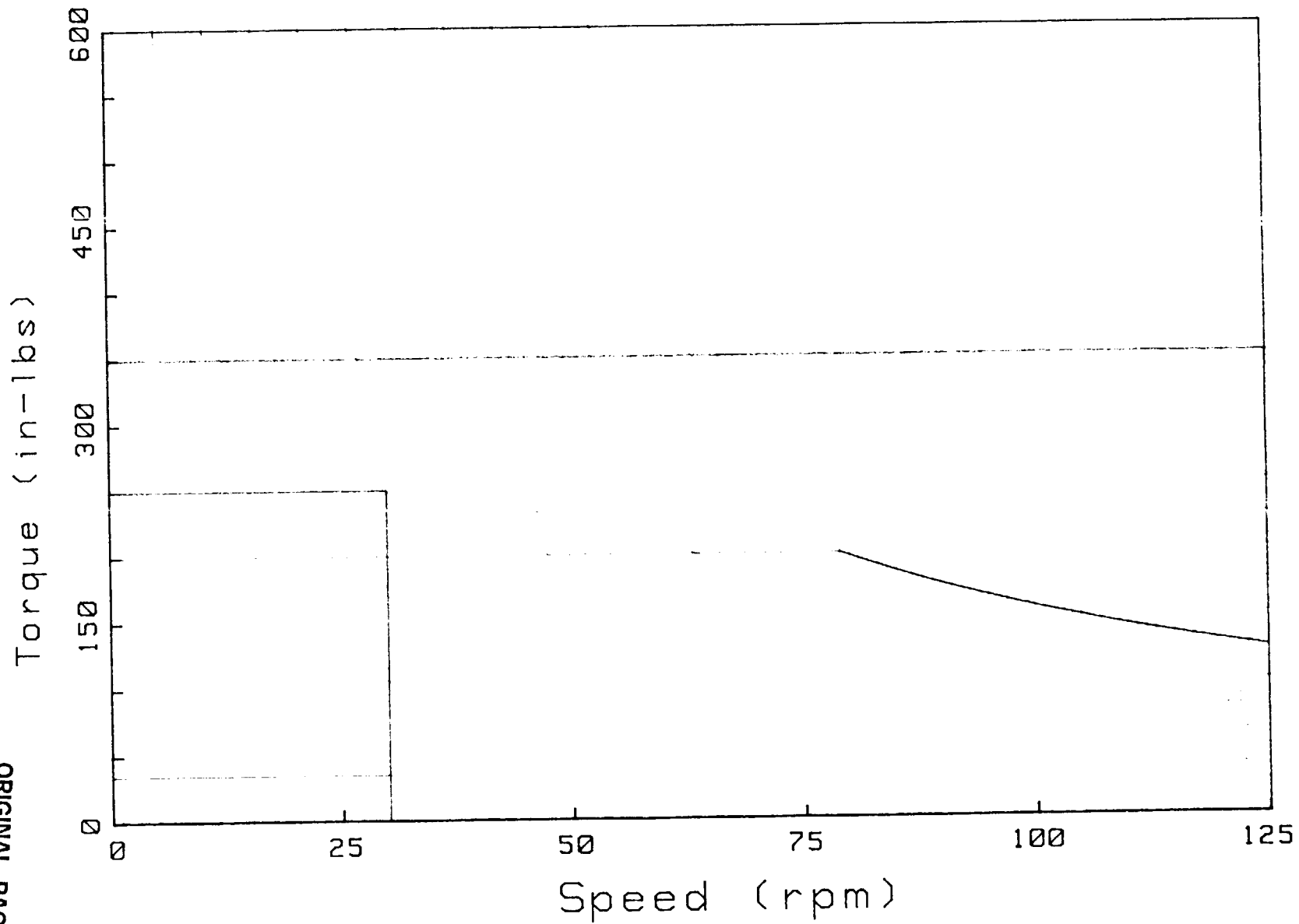
FIGURE 15.



TOP VIEW OF FOLDING SHELL
SHOWN OPEN

FIGURE 6.

Tasks Window for Rotary Motion - Very Low Speeds



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FIGURE 7.

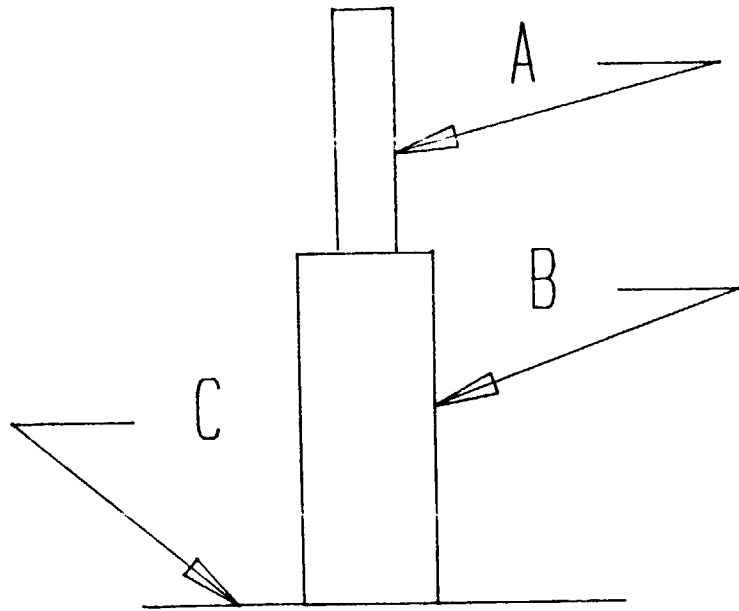


FIGURE S-1.

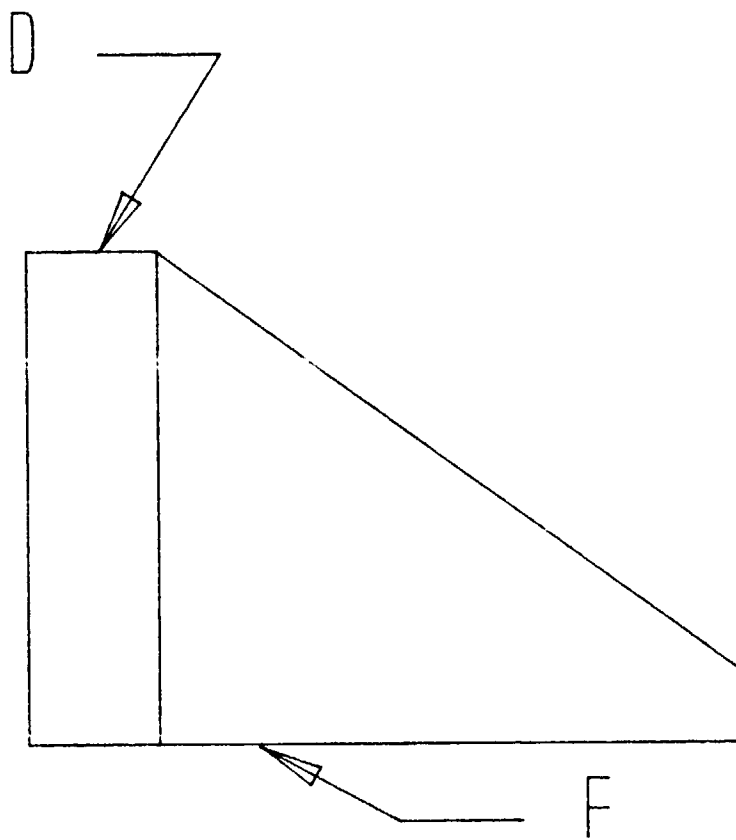


FIGURE S-2.

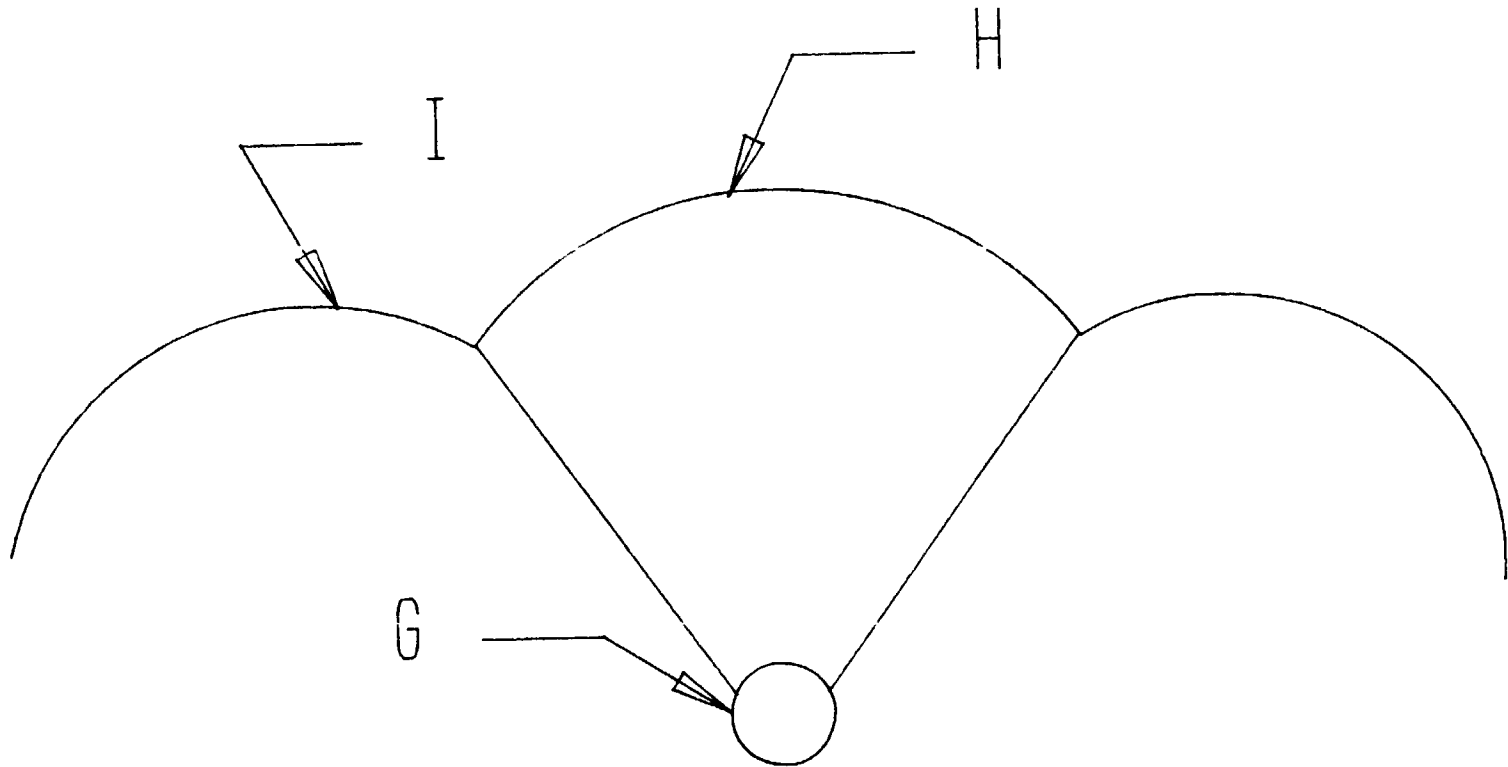


FIGURE S-3.

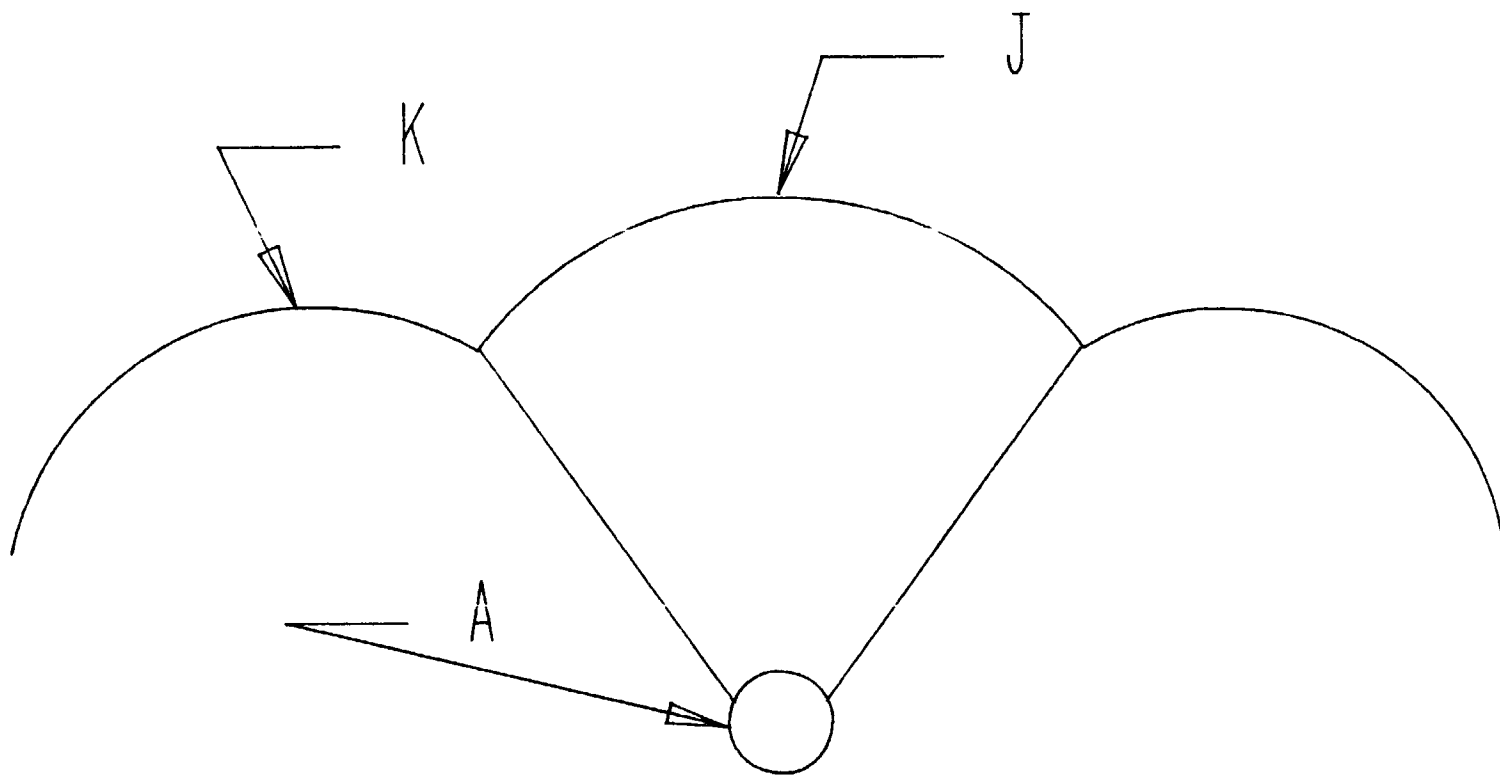


FIGURE S-4.

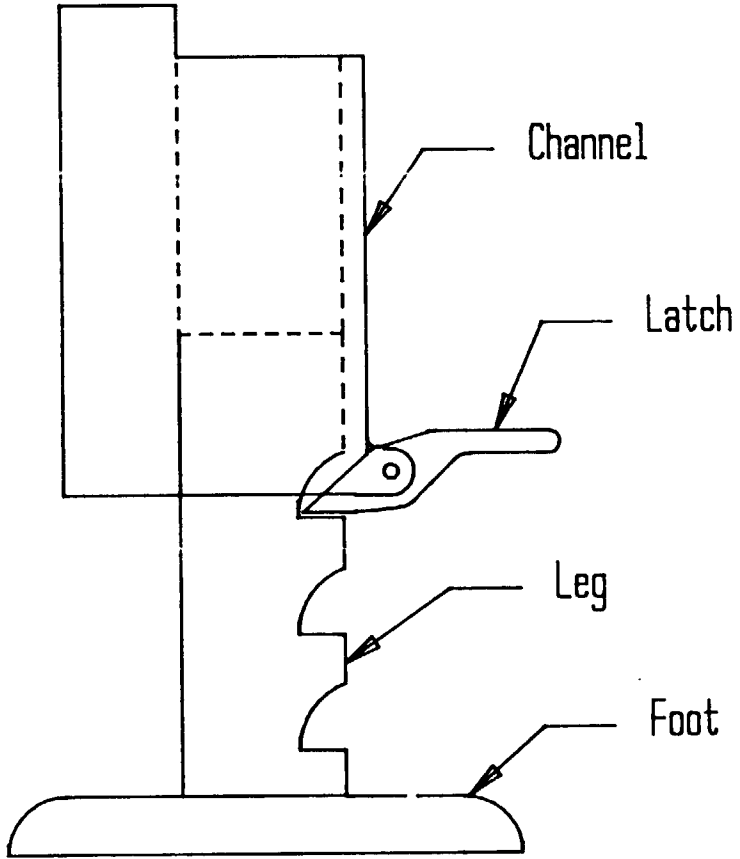


FIGURE 5.

INLAND MOTOR

KOLLMORGEN CORPORATION

SIZE CONSTANTS

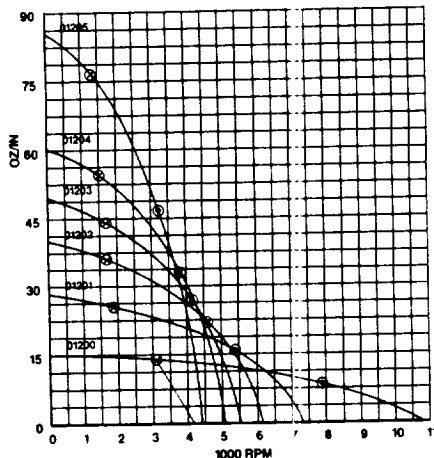
PARAMETERS		MODEL NO.	UNITS	RBE-01200	RBE-01201	RBE-01202	RBE-01203	RBE-01204	RBE-01205
For 100° C Rise in 15 Sec	Max Stall Torque		oz-in N.cm	46 33	95 67	139 98	181 128	227 160	332 234
	Max Stall Power		Watts	124	168	201	229	260	331
Max. Continuous Stall Torque, T _c			oz-in N.cm	15 11	28 20	39 28	49 35	60 42	85 60
Max. Continuous Output Power			Watts	49.8	62.9	72.5	82.5	92.1	113.5
Motor Constant, ± 15%, K _m			oz-in/√W N.cm/√W	4.3 3.0	7.3 5.2	10.0 7.1	12.1 8.5	14.4 10.2	18.3 12.9
TPR, ± 15% †			(°C/W)	5.5	5.0	4.6	4.3	3.9	3.3
Viscous Damping, F _v			oz-in-/RPM N.cm/RPM	1.6x10 ⁻⁴ 1.1x10 ⁻⁴	3.5x10 ⁻⁴ 2.5x10 ⁻⁴	5.3x10 ⁻⁴ 3.7x10 ⁻⁴	6.8x10 ⁻⁴ 4.8x10 ⁻⁴	8.8x10 ⁻⁴ 6.2x10 ⁻⁴	1.3x10 ⁻³ 9.2x10 ⁻⁴
Hysteresis Drag Torque, T _f			oz-in N.cm	0.33 0.23	0.73 0.52	1.10 0.78	1.40 0.99	1.83 1.29	2.72 1.92
Max. Cogging Torque			oz-in N.cm	1.2 0.9	2.0 1.4	2.5 1.8	2.8 2.0	3.0 2.1	4.0 2.8
Rotor Inertia, J _m			oz-in-sec ² gm-cm ²	7.3x10 ⁻⁴ 51.5	1.2x10 ⁻³ 84.7	1.7x10 ⁻³ 120	2.1x10 ⁻³ 148	2.7x10 ⁻³ 191	4.0x10 ⁻³ 282
Unhoused Weight			oz gm	3.1 87.9	5.8 164	8.3 235	10.7 303	13.5 383	20.1 570
No. of Poles				8	8	8	8	8	8

24 VOLT WINDING CONSTANTS

Peak Torque, ± 25%, T _p	oz-in N.cm	58 41	104 73	166 117	229 162	291 206	400 283	
Peak Current, ± 15%, I _p	Amps	7.7	8.3	11.4	15	17.1	20.0	
Torque Sensitivity, ± 10%, K _t	oz-in/Amp N.cm/Amp	7.5 5.3	12.5 8.8	14.5 10.2	15.3 10.8	17.0 12.0	20.0 14.1	
No Load Speed, ± 10%	RPM	4150	2500	2100	2000	1800	1500	
Voltage Constant, ± 10%, K _v	Volts/1000 RPM	5.54	9.24	10.72	11.31	12.57	14.78	
Terminal Resistance, ± 12%, R _m	ohms	3.1	2.9	2.1	1.6	1.4	1.2	
Terminal Inductance, ± 30%, L _m	mH	1.3	1.3	1.2	1.0	1.0	0.9	
Peak Efficiency	%	76	77	78	80	80	80	
	Torque	oz-in N.cm	6.4 4.5	12.0 8.5	17.0 12.0	22.0 16.0	28.0 20.0	39.0 28.0
	Speed	RPM	3700	2250	1975	1900	1700	1450
Max. Continuous Output Power	Power	Watts	31.9	36.1	44.7	54.6	60.9	73.1
	Torque	oz-in N.cm	13.7 9.7	25.6 18.1	35.3 24.9	43.9 31.0	54.3 38.3	76.4 54.0
	Speed	RPM	3144	1903	1710	1681	1515	1293

† TPR assumes housed motor mounted to 3.5 x 3.5 x .25" aluminum heatsink or equivalent.

CONTINUOUS DUTY FOR 75° C RISE



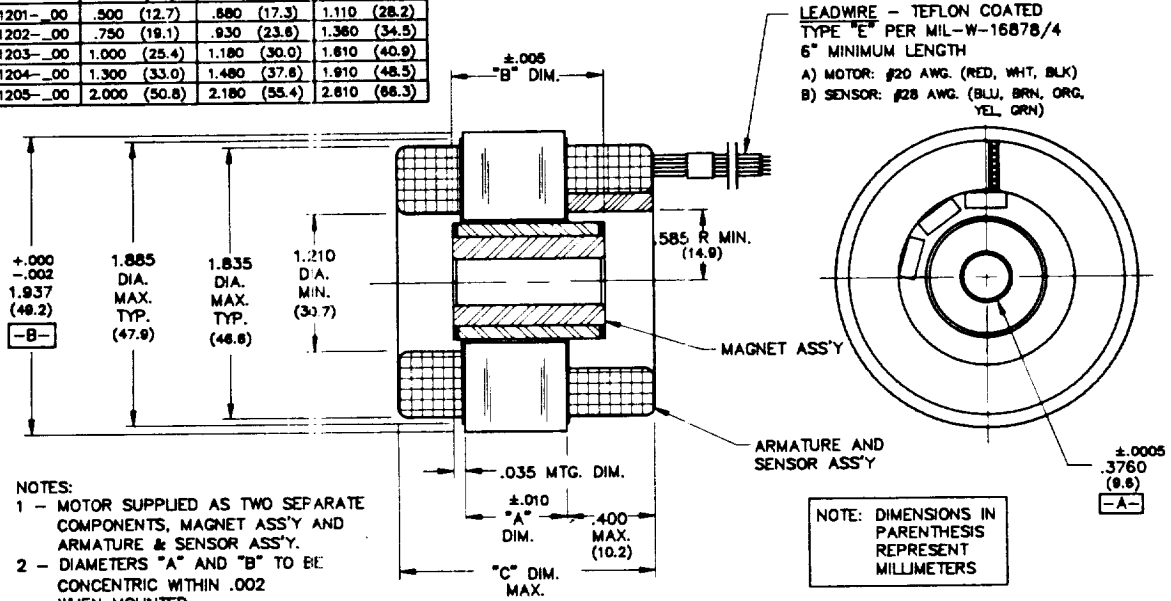
Design Features of Inland DC Brushless Motors

- High torque to weight and inertia ratios
- Samarium cobalt rare earth magnets
- 3 phase delta or wye connection
- Housed or frameless designs
- Stationary outer stator winding
rotating inner permanent magnet rotor
- Stainless steel shafts (housed versions)
- All motors built to MIL-Q-9858A
- Encapsulated windings available for harsh environments
- Built-in Hall effects for electronic commutation

DC BRUSHLESS MOTORS 01200 Series

Frameless Motor

MODEL	"A" DIM	"B" DIM	"C" DIM
RBE-01200-.00	.225 (5.7)	.405 (10.3)	.835 (21.2)
RBE-01201-.00	.500 (12.7)	.880 (22.3)	1.110 (28.2)
RBE-01202-.00	.750 (19.1)	.930 (23.8)	1.360 (34.5)
RBE-01203-.00	1.000 (25.4)	1.180 (30.0)	1.810 (45.9)
RBE-01204-.00	1.300 (33.0)	1.480 (37.8)	1.910 (48.5)
RBE-01205-.00	2.000 (50.8)	2.180 (55.4)	2.810 (71.3)

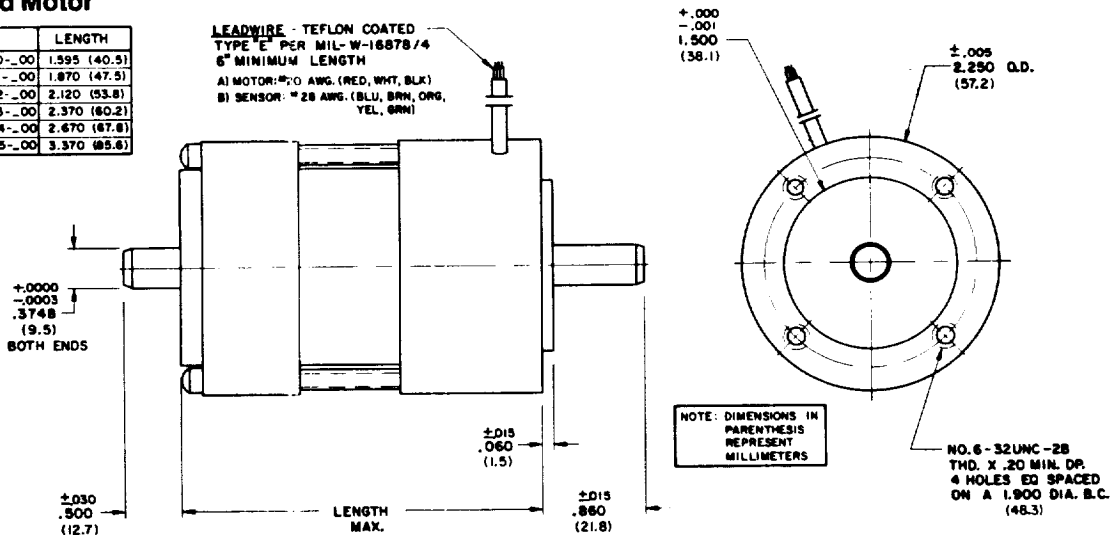


NOTES:

- 1 - MOTOR SUPPLIED AS TWO SEPARATE COMPONENTS, MAGNET ASSY AND ARMATURE & SENSOR ASSY.
- 2 - DIAMETERS "A" AND "B" TO BE CONCENTRIC WITHIN .002 WHEN MOUNTED.
- 3 - MOUNTING SURFACE BETWEEN 1.937 AND 1.885 DIAMETERS ON BOTH SIDES.

Housed Motor

MODEL	LENGTH
RBEH-01200-.00	1.595 (40.5)
RBEH-01201-.00	1.870 (47.5)
RBEH-01202-.00	2.120 (53.8)
RBEH-01203-.00	2.370 (60.2)
RBEH-01204-.00	2.670 (67.8)
RBEH-01205-.00	3.370 (85.6)



INLAND MOTOR
KOLLMORGEN CORPORATION

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HIGH EFFICIENCY LAMINATIONS

The accompanying speed/torque curves represent the performance limits of the Inland DC brushless motors with high efficiency laminations. The advantage of motors with high efficiency laminations is improved performance at higher operating speeds.

Since these motors are best utilized with custom windings, standard windings are not available. To obtain the optimum winding for your application, please contact our application engineers at the factory.

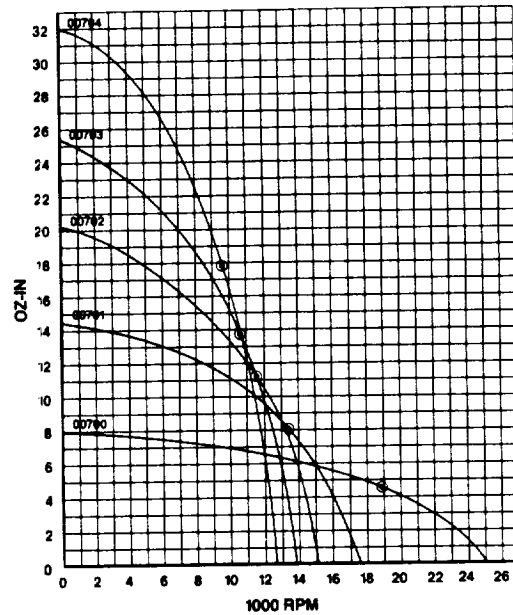
PERFORMANCE PARAMETERS

The physical dimensions of the motors do not change from standard. Please refer to the outline drawings on the DC brushless motor data sheets.

The Size Constants on the data sheets remain the same with the exception of the Viscous Damping (F_v) and Hysteresis Drag Torque (T_d) coefficients which are detailed with the respective curves.

As stated above, standard windings are not available.

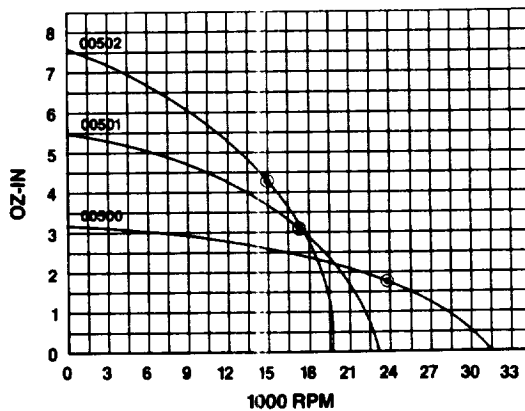
CONTINUOUS DUTY FOR 75°C RISE



PARAMETERS	MODEL NO.	UNIT	RBE-00700	RBE-00701	RBE-00702	RBE-00703	RBE-00704
Viscous Damping, F_v		oz-in/RPM	2.2×10^{-1}	4.4×10^{-1}	6.2×10^{-1}	7.9×10^{-1}	9.9×10^{-1}
		N cm/RPM	1.6×10^{-1}	3.1×10^{-1}	4.4×10^{-1}	5.6×10^{-1}	7.0×10^{-1}
Hysteresis Drag Torque, T_d		oz-in	0.15	0.29	0.42	0.53	0.67
		N cm	0.1	0.2	0.3	0.4	0.5

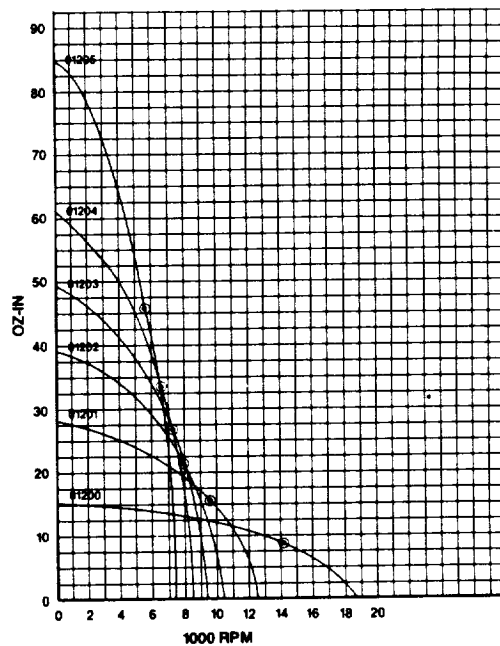
CONTINUOUS DUTY FOR 75°C RISE

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PARAMETERS	MODEL NO.	UNIT	RBE-00500	RBE-00501	RBE-00502
Viscous Damping, F_v		oz-in/RPM	9.1×10^{-2}	8×10^{-2}	2.6×10^{-1}
		N cm/RPM	6.4×10^{-2}	2×10^{-1}	1.8×10^{-1}
Hysteresis Drag Torque, T_d		oz-in	0.06	0.12	0.17
		N cm	0.04	0.08	0.12

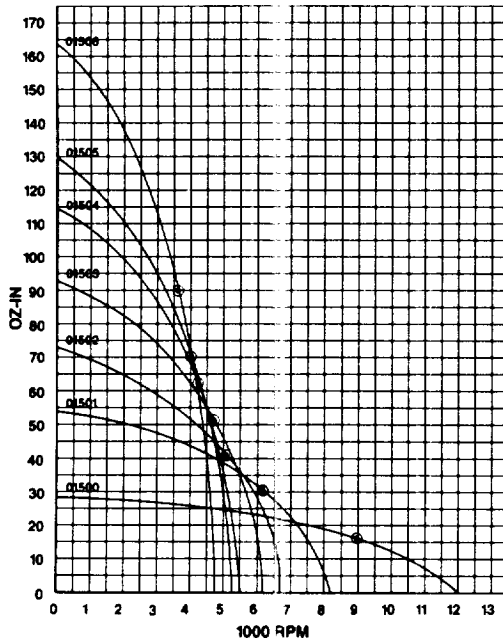
CONTINUOUS DUTY FOR 75°C RISE



PARAMETERS	MODEL NO.	UNIT	RBE-01200	RBE-01201	RBE-01202	RBE-01203	RBE-01204	RBE-01205
Viscous Damping, F_v		oz-in/RPM	5.0×10^{-1}	1.1×10^{-1}	1.7×10^{-1}	2.1×10^{-1}	2.8×10^{-1}	4.1×10^{-1}
		N cm/RPM	3.5×10^{-1}	0.8×10^{-1}	1.2×10^{-1}	1.5×10^{-1}	2.0×10^{-1}	2.9×10^{-1}
Hysteresis Drag Torque, T_d		oz-in	0.25	0.56	0.87	1.03	1.35	2.01
		N cm	0.2	0.4	0.6	0.7	1.0	1.4

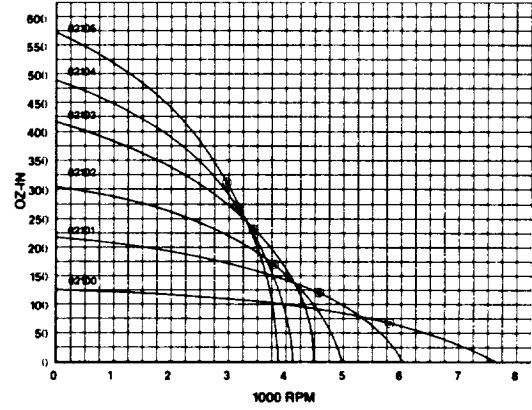
HIGH EFFICIENCY LAMINATIONS / DC Brushless Motors

CONTINUOUS DUTY FOR 75°C RISE



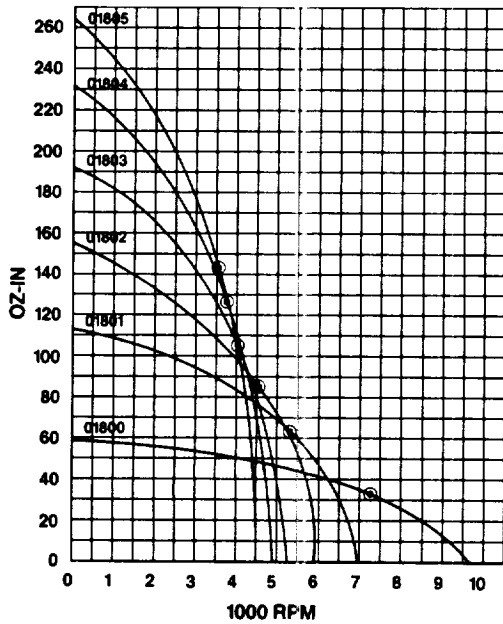
PARAMETERS	MODEL NO.	UNIT	RBE-01500	RBE-01501	RBE-01502	RBE-01503	RBE-01504	RBE-01505	RBE-01506
Viscous Damping, F_v	oz-in/RPM		1.7×10^{-1}	3.6×10^{-1}	5.3×10^{-1}	6.6×10^{-1}	8.4×10^{-1}	9.4×10^{-1}	1.2×10^0
	N-cm/RPM		1.2×10^{-1}	2.5×10^{-1}	3.7×10^{-1}	4.6×10^{-1}	6.0×10^{-1}	6.6×10^{-1}	0.9×10^0
Hysteresis Drag Torque, T_d	oz-in		0.54	1.26	1.70	2.15	2.76	3.10	4.08
	N-cm		0.4	0.8	1.2	1.5	2.0	2.2	2.9

CONTINUOUS DUTY FOR 75°C RISE



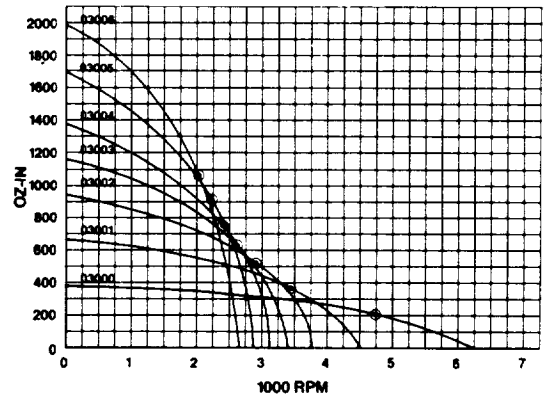
PARAMETERS	MODEL NO.	UNIT	RBE-02100	RBE-02101	RBE-02102	RBE-02103	RBE-02104	RBE-02105
Viscous Damping, F_v	oz-in/RPM		6.9×10^{-1}	1.2×10^0	1.7×10^0	2.210	2.8×10^0	3.3×10^0
	N-cm/RPM		4.9×10^{-1}	0.9×10^0	1.2×10^0	1.6×10^0	2.0×10^0	2.3×10^0
Hysteresis Drag Torque, T_d	oz-in		2.28	3.95	5.71	8.16	9.31	10.94
	N-cm		1.6	2.8	4.0	5.8	6.6	7.7

CONTINUOUS DUTY FOR 75°C RISE



PARAMETERS	MODEL NO.	UNIT	RBE-01800	RBE-01801	RBE-01802	RBE-01803	RBE-01804	RBE-01805
Viscous Damping, F_v	oz-in/RPM		3.4×10^{-1}	6.4×10^{-1}	8.1×10^{-1}	1.2×10^0	1.4×10^0	1.7×10^0
	N-cm/RPM		2.4×10^{-1}	4.5×10^{-1}	6.4×10^{-1}	0.9×10^0	0.99×10^0	1.2×10^0
Hysteresis Drag Torque, T_d	oz-in		1.08	2.16	3.03	3.83	4.89	5.57
	N-cm		0.8	1.5	2.1	2.7	3.3	3.9

CONTINUOUS DUTY FOR 75°C RISE



PARAMETERS	MODEL NO.	UNIT	RBE-03000	RBE-03001	RBE-03002	RBE-03003	RBE-03004	RBE-03005	RBE-03006
Viscous Damping, F_v	oz-in/RPM		1.7×10^0	3.1×10^0	4.5×10^0	5.9×10^0	7.1×10^0	8.6×10^0	1.0×10^1
	N-cm/RPM		1.2×10^0	2.2×10^0	3.2×10^0	4.2×10^0	5.0×10^0	6.1×10^0	0.7×10^1
Hysteresis Drag Torque, T_d	oz-in		5.54	10.44	15.11	19.57	23.67	28.97	34.87
	N-cm		3.9	7.4	10.7	13.8	16.7	20.5	24.6

INLAND MOTOR
KOLLMORGEN CORPORATION

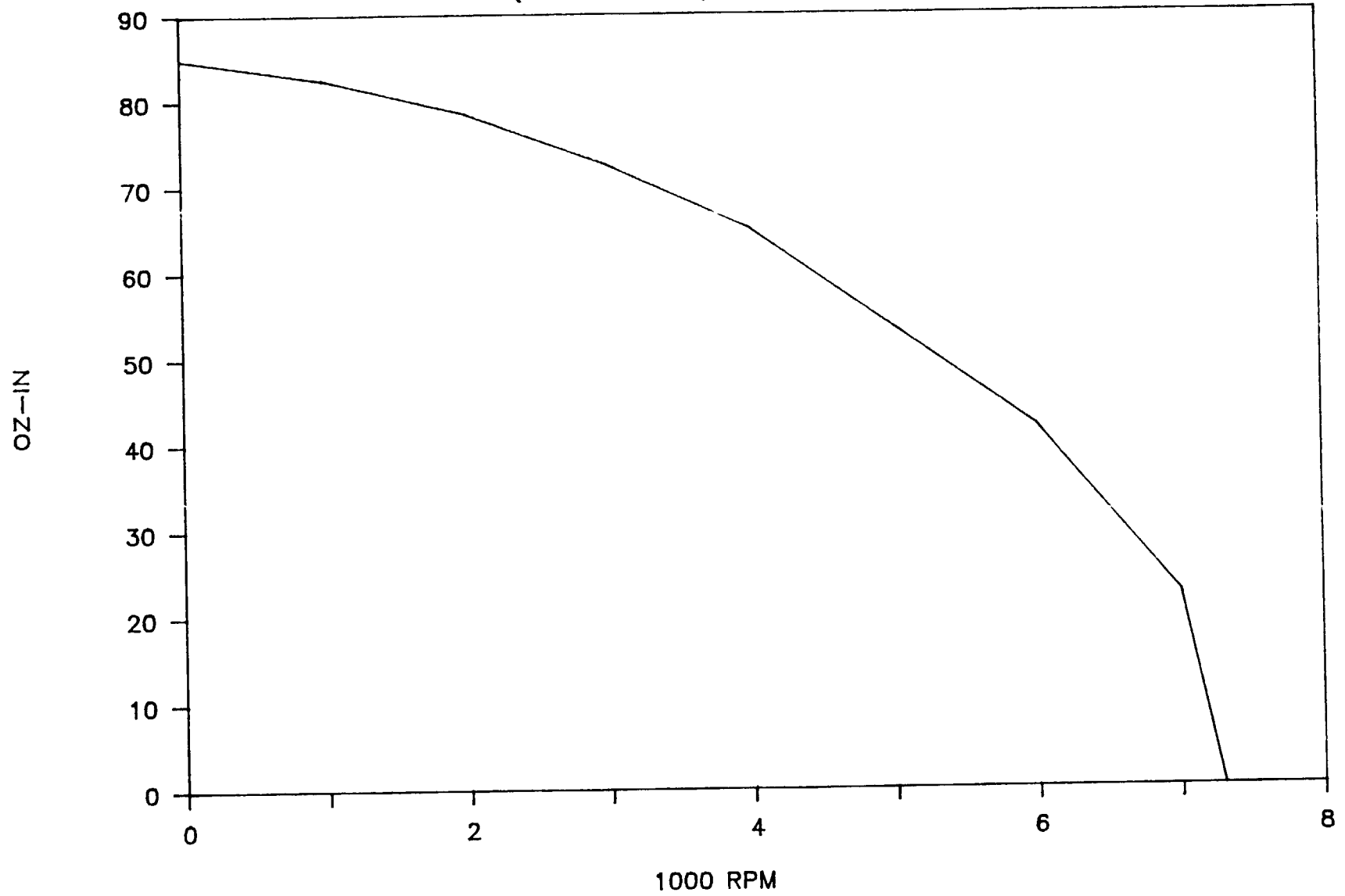
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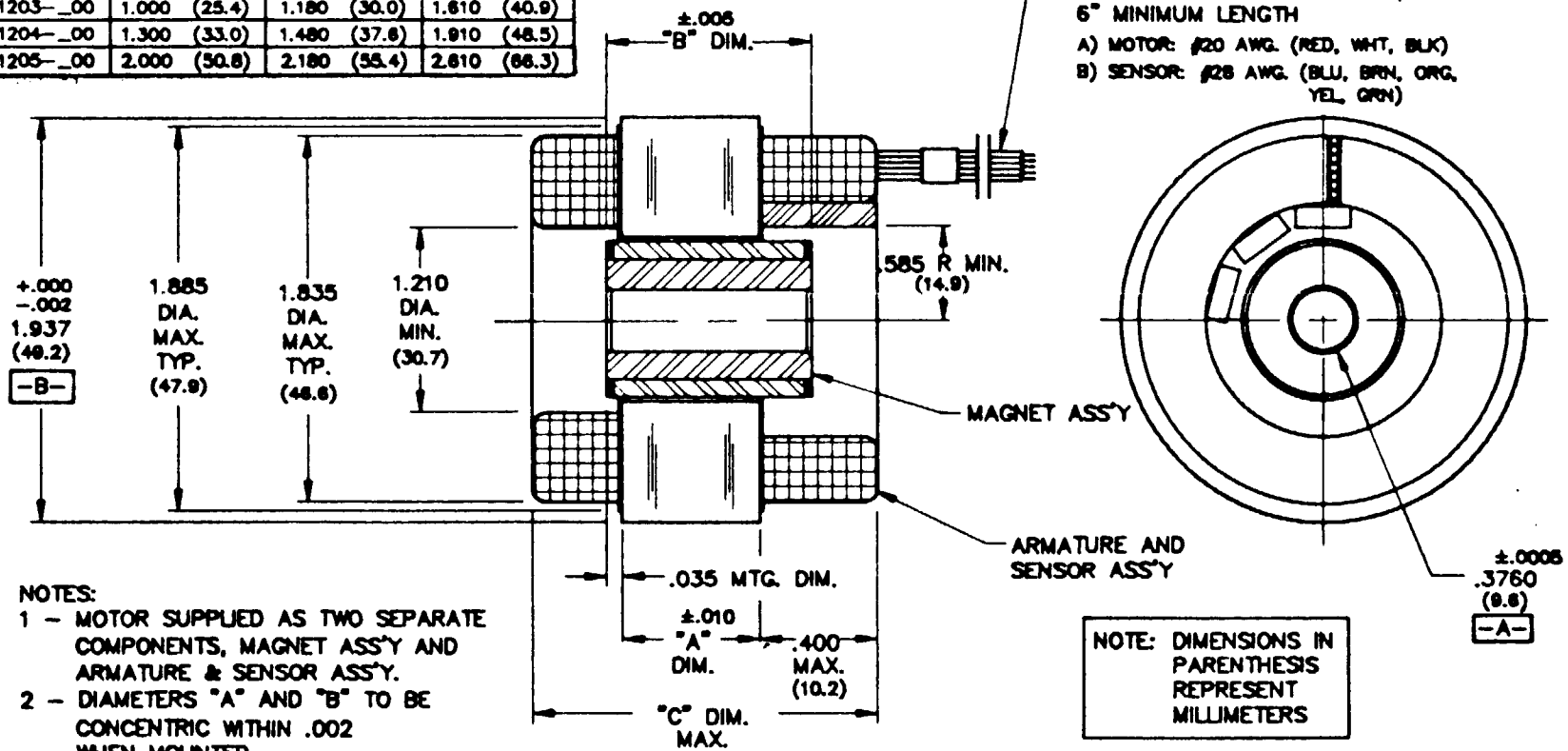
MOTOR TORQUE VS SPEED

(continuous duty for 75°C rise)



Frameless Motor

MODEL	"A" DIM	"B" DIM	"C" DIM
RBE-01200-__00	.225 (5.7)	.405 (10.3)	.835 (21.2)
RBE-01201-__00	.500 (12.7)	.680 (17.3)	1.110 (28.2)
RBE-01202-__00	.750 (19.1)	.930 (23.6)	1.360 (34.5)
RBE-01203-__00	1.000 (25.4)	1.180 (30.0)	1.810 (40.9)
RBE-01204-__00	1.300 (33.0)	1.480 (37.6)	1.910 (48.5)
RBE-01205-__00	2.000 (50.8)	2.180 (55.4)	2.810 (66.3)



- NOTES:
- 1 - MOTOR SUPPLIED AS TWO SEPARATE COMPONENTS, MAGNET ASS'Y AND ARMATURE & SENSOR ASS'Y.
 - 2 - DIAMETERS "A" AND "B" TO BE CONCENTRIC WITHIN .002 WHEN MOUNTED.
 - 3 - MOUNTING SURFACE BETWEEN 1.937 AND 1.885 DIAMETERS ON BOTH SIDES.

APPENDIX 3

INLAND MOTOR

KOLLMORGEN CORPORATION

The BC Drive Series is designed to greatly simplify the engineer's task of applying three phase DC brushless motor technology. Inland's approach has resulted in an amplifier design incorporating all the necessary control, commutation, and power drive electronics.

For maximum flexibility, three configurations of BC Drives are available:

- BCL(S)** Velocity controller using the Hall Effect output pulses for velocity information. The need for a separate tachometer is eliminated. Suitable for velocity control from 300 RPM to maximum motor speed. Not suitable for closed position or rate loop servo systems.
- BC(L)** Current loop controller only. Suitable for use in torque control applications or servo-systems where the rate loop is controlled by the host controller.
- BCL-VL** A closed velocity loop servo-controller utilizing a brushless or DC tachometer for velocity feedback. Suitable for use in applications where the position loop is closed externally.

The design of the BC Drives is enhanced with Power FET technology combined with 20 KHz pulse-width modulation (PWM) for an optimum efficiency drive circuit. With external baseplate heatsinking the need for fan cooling is eliminated.

A high current loop bandwidth ensures fast response to meet the needs of the most demanding applications. The commutation logic uses digital Hall Effect Sensors (built into Inland brushless motors), or optical encoders to determine the commutation sequence for the motor windings.

The BC Drives may also be used in the BCL configuration with Inland's Intelligent Control Modules (ICMs) for complete digital positioning and velocity control systems using incremental encoder feedback.

The versatility of the design, compact size, and ease of installation and use make the BC Drives one of the most complete series of amplifiers available today for small DC brushless motor control systems.

28'0

0 x 6 x 4"

Inland Motor, Sierra Vista, has gained extensive experience in the motion control field and has matured to a complete engineering resource. Capabilities include original application engineering, custom designs (where necessary), component selection, system evaluation, field service and support. For the solution to your next application need call our Application Engineers at (602) 459-1150.

Model	SUPPLY VOLTAGE RANGE (VDC)			OUTPUT VOLTAGE RANGE (VDC)			CONTINUOUS OUTPUT CURRENT (AMPS)	PEAK OUTPUT CURRENT (AMPS)	CONTINUOUS OUTPUT POWER (WATTS)	PEAK OUTPUT POWER (WATTS)
	Min.	Non.	Max.	Min.	Nom.	Max.	Nom.	Nom.	Nom.	Nom.
BC-02806	18	28	32	14	24	28	6	10	144	240
BC-02812	24	28	32	20	24	28	12	12	288	288
BC-02820	24	28	32	20	24	32	20	20	480	488
BC-04810	24	48	50	20	44	46	10	12	440	528
BC-04820	24	48	50	20	44	46	20	20	880	880

CONNECTIONS

J1 Connector

Motor Phase A
 Motor Phase B
 Motor Phase C
 Hall Effect Phase A
 Hall Effect Phase B
 Hall Effect Phase C
 Hall Effect +5VDC Supply
 Signal Ground
 Brushless Tachometer - 1
 Brushless Tachometer - 2
 Brushless Tachometer - 3

J2 Connector

Input Power Positive
 Input Power Ground
 Command Signal Input
 Disable
 Velocity Signal Output
 RMS Current Output
 +15VDC Output, 10mA
 -15VDC Output, 10mA
 +5VDC Output, 250mA
 Signal Ground

SCALING AND ADJUSTMENT POTENTIOMETERS

	<u>Offset</u>	<u>Peak Current Limit</u>	<u>Velocity Feedback Limit</u>	<u>Command Scaling</u>	<u>AC Gain</u>
BCL(S)	*	*	*		
BCL	*	*		*	
BCL-VL	*	*	*	*	*

INLAND MOTOR

KOLLMORGEN CORPORATION

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 FAX 602-458-2161

Represented By:

APPENDIX 3

```

500 REM * PROGRAM TO CALCULATE THE NECESSARY SHAFT DIAMETER GIVEN *
510 REM * SAFETY FACTOR, MAX TORQUE, AND YIELD STRENGTH. *
600 HOME
700 PI = 3.14159
750 REM * FACTOR OF SAFETY *
800 N = 5
850 REM * MAXIMUM TORQUE IN IN*LBS *
900 T = 200
950 REM * SY = YIELD STRENGTH IN PSI *
960 REM * D = DIAMETER IN INCHES *
970 REM * W = WEIGHT IN LB/IN *
980 REM * C = RELETIVE COST COMPARISON IN $ AT $23K./LB SHIPPING COST *
1000 PRINT "N=      T=      SY=      D=      W=      C="
1100 PRINT "      (INLB) (PSI)   (IN)      (LB/IN)   ($)"
1200 PRINT
1250 REM * VARIOUS SY USED TO HELP IN MATERIAL SELECTION *
1300 FOR SY = 150000 TO 250000 STEP 5000
1400 D = ((32 * N * T) / (PI * SY)) (1 / 3)
1500 W = (0.284) * (PI / 4) * (D 2)
1550 REM * USED FOR APPROXIMATE COMPARISON PURPOSES ONLY *
1600 C = 23000 * 8 * W
1650 REM * ROUND OFF ROUTINE *
1700 D = D * 100000:D = INT (D):D = D / 100000
1800 W = W * 100000:W = INT (W):W = W / 100000
1900 C = INT (C)
2000 PRINT N;"      ";T;"      ";SY;"      ";D;"      ";W;"      ";C
2100 NEXT SY
2200 END
2250 REM * NOTE: PROGRAM WAS RUN FOR VARIOUS FACTORS OF SAFTEY AND *
2260 REM * TORQUES TO HELP ANTICIPATE ANY PROBLEMS *

```

]

APPENDIX 3

N=	T= (INLB)	SY= (PSI)	D= (IN)	W= (LB/IN)	C= (\$)
5	200	150000	.40797	.03712	6831
5	200	155000	.40354	.03632	6683
5	200	160000	.39929	.03556	6543
5	200	165000	.39521	.03484	6410
5	200	170000	.3913	.03415	6284
5	200	175000	.38754	.0335	6164
5	200	180000	.38392	.03287	6049
5	200	185000	.38043	.03228	5939
5	200	190000	.37706	.03171	5835
5	200	195000	.37381	.03116	5735
5	200	200000	.37067	.03064	5639
5	200	205000	.36763	.03014	5546
5	200	210000	.36469	.02966	5458
5	200	215000	.36184	.0292	5373
5	200	220000	.35908	.02876	5291
5	200	225000	.3564	.02833	5213
5	200	230000	.35379	.02792	5137
5	200	235000	.35127	.02752	5064
5	200	240000	.34881	.02713	4993
5	200	245000	.34642	.02676	4925
5	200	250000	.3441	.02641	4859

]

APPENDIX B

HOLLOW SHAFT CALCULATIONS

J,s = polar moment of inertia of solid shaft
J,h = polar moment of inertia of hollow shaft
T = maximum torque
d = outside diameter
d,i = inside diameter
Pi = 3.14159
t = maximum shear stress
r = outside radius
Sy = yield strength
n = factor of safety
A = area of cross section of shaft
V = volume of shaft

$$J,s = (Pi)(d^4)/(32) = 0.0019414$$

$$J,h = (Pi)(d^4 - d,i^4)/(32)$$

$$t = T*r/J = Sy/(2*n)$$

$$J = (200 \text{ inlb})(0.1875 \text{ in})(2)(5)/(200 \text{ kpsi}) = 0.001875$$

$$J,h \text{ .GE. } 0.001875 \\ \text{(for a factor of safety of 5)}$$

$$d,i = 0.16129$$

$$A,s = 0.11045$$

$$A,h = 0.09001 \text{ (difference of } 0.020435 \text{ in}^2)$$

Savings in weight would be:

$$(0.020435 \text{ in}^2)(L)(.284 \text{ lb/in}^3) = 0.0406 \text{ lb} \\ \text{(for a FOS of 5 and a 7" shaft)}$$

For a FOS of 3:

$$A,h = 0.03882, \text{ Savings} = 0.14239 \text{ (but very thin, not reliable} \\ \text{on bearings)}$$

APPENDIX A CYLINDER STRESS CALCULATIONS

PART B

$$R = 4''$$

$$E = 300,000 \text{ psi } (\text{ @ } 200^\circ\text{C})$$

$$P = 90 \text{ lb}$$

$$L = 25''$$

$$\rho = 15.6 \text{ lb/in}^3$$

$$I = \frac{P L^2}{\pi^2 E} = \frac{(90 \text{ lb})(25 \text{ in})^2}{(\pi^2)(300,000 \text{ lb/in}^2)} = 19.0 \times 10^{-3} \text{ in}^4 = 0.019 \text{ in}^4$$

$$I = \frac{M}{2} (R^2 + r^2) \quad M = \rho V = \rho \pi L (R^2 - r^2)$$

$$I = \frac{\rho \pi L}{2} (R^2 - r^2) (R^2 + r^2)$$

$$I = \frac{\rho \pi L}{2} (R^4 - r^4)$$

$$\frac{2I}{\rho \pi L} = R^4 - r^4$$

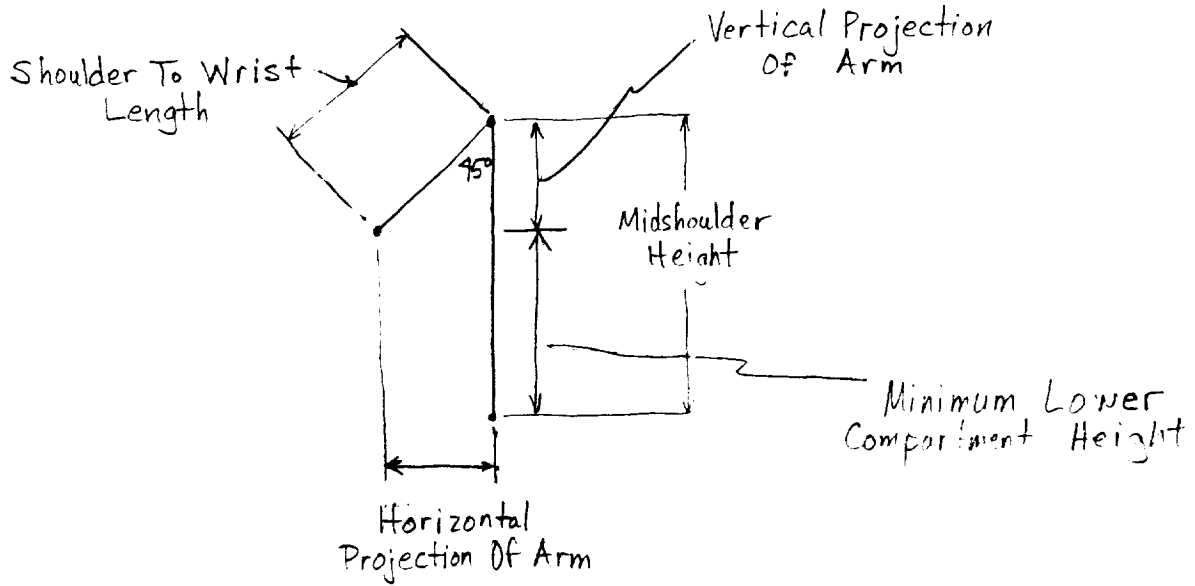
$$r^4 = R^4 - \frac{2I}{\rho \pi L}$$

$$r = \sqrt[4]{R^4 - \frac{2I}{\rho \pi L}}$$

$$= \sqrt[4]{(4)''^4 - \frac{2(0.019)}{(15.6)\pi(25)}}$$

Appendix 5

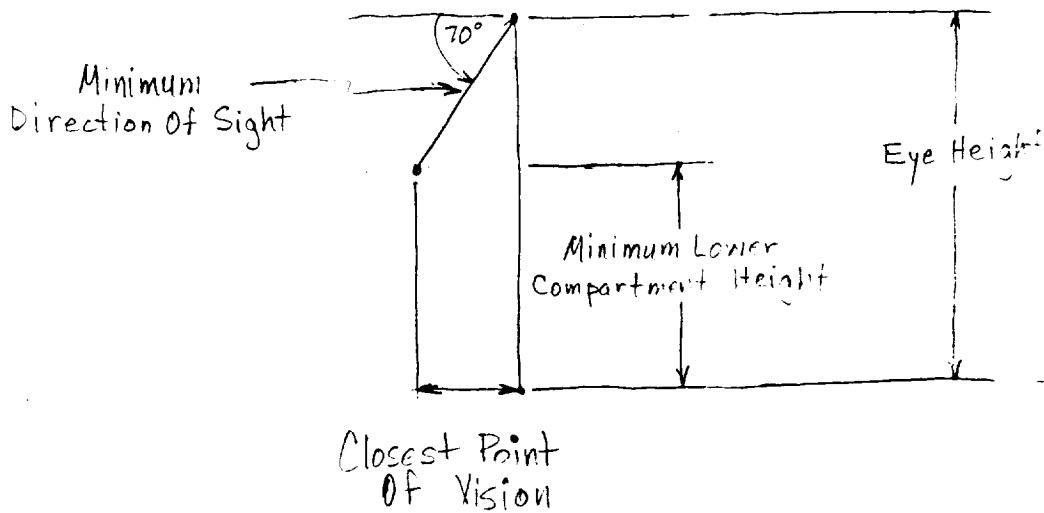
Diagram And Calculations For Determining Minimum Lower Compartment Height



$$\text{Horizontal Arm Projection} = \text{Shoulder To Wrist Length} \times \sin 45^\circ$$

$$\text{Vertical Arm Projection} = \text{Shoulder To Wrist Length} \times \cos 45^\circ$$

Diagram And Calculations For Insuring Vision of Minimum Compartment Height



$$\text{Closest Point of Vision} = (\text{Eye Height} - \text{Lower Compartment Height}) \times \tan 20^\circ$$

Appendix 5

Ergonomic Factors

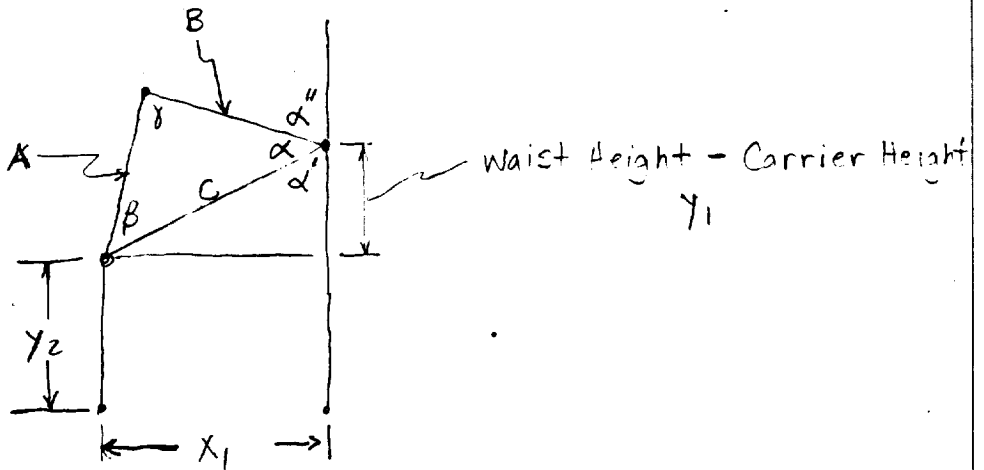
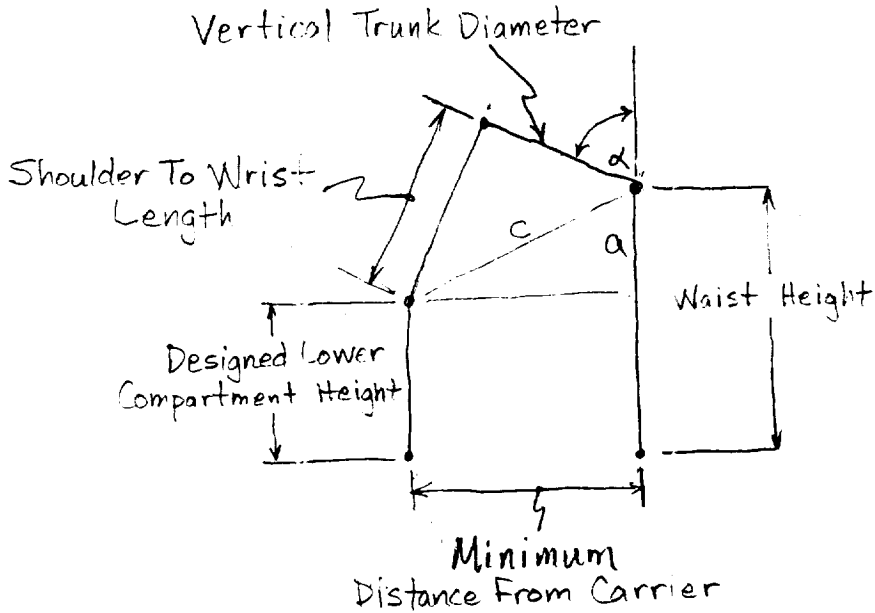
	Minimum (in.)	Maximum (in.)
Chest Breadth	10.55	16.61
Interwrist Span	49.09	61.14
Shoulder To Wrist Length	19.27	22.27
Midshoulder Height	53.97	63.15
Height	64.37	74.29
Eye To Top Of Head	4.00	
Eye Height	60.37	70.29
Vertical Arm Projection at 45°	13.63	15.75
Horizontal Arm Projection at 45°	13.63	15.75
Minimum Lower Compartment Height	39.84	47.40
Eye Height - Minimum Lower Compartment Height Difference	20.53	22.89
Closest Point Of Vision at 70° From Horizontal	7.47	8.33


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Appendix 5

Bend over Angles To Reach Lower Compartment



$$C = \sqrt{x_1^2 + y_1^2}$$

$$\alpha' = \tan^{-1} \left(\frac{x_1}{y_1} \right)$$

$$A^2 = B^2 + C^2 - 2BC \cos \alpha$$

$$\alpha = \cos^{-1} \left(\frac{B^2 + C^2 - A^2}{2BC} \right)$$

$$\alpha'' = 180^\circ - \alpha - \alpha'$$

Appendix 5

Bendover Ergonomic Factors

	Minimum	Maximum
Crotch Height (in.)	28.31	35.91
Crotch To Waist Length (in.)	9.00	—
Waist Height (in.)	37.31	44.91
Vertical Trunk Diameter (in.)	23.54	30.51
Designed Lower Compartment Height (in.)	29.00	—
α (degrees)	54.25	46.88
β (degrees)	82.47	89.75
γ (degrees)	43.28	43.37
α' (degrees)	56.83	40.59
α'' (degrees)	68.92	92.53

Appendix 6

Upper folding Compartment

$$\text{Diameter} = 20''$$

$$\text{Surface Area} = 2\pi\left(\frac{20}{2}\right) = 62.83$$

$$\text{Number of Tools} = \frac{\text{Surface Area}}{2+4}$$

$$= 10.47 \approx 10-11 \text{ tools}$$

$$\text{Outside Area} = \text{Number of tools} - 2 (\text{hinge})$$

$$\text{Outside Area} \approx 8-9 \text{ tools}$$

$$\text{Inside Area} \approx 10-11 \text{ tools}$$

Lower folding Compartment

$$\text{Diameter} = 24.5''$$

$$\text{Surface Area} = 2\pi\left(\frac{24.5}{2}\right) = 76.969$$

Average 2" wide tools and 4" allotment
between tools

$$\text{Number of tools} = 12.82$$

$$\text{Inner Area} \approx 12-13 \text{ tools}$$

Subtract 2 tools for hinge allotment

$$\text{Outer Area} \approx 10-11 \text{ tools}$$

APPENDIX 7

Weight Calculations

$$\pi r^2 \cdot 2\pi r h \rightarrow SA$$

$$\Delta x \rightarrow V \quad \rho \rightarrow \text{weight}$$

$$\rho = .0517 \text{ lb/in}^3$$

cover shell

$$(\pi (15)^2 + 2\pi(15)(25))(.0625) = 191.49 \text{ in}^3$$

2 wh

legs \ominus

$$3[(2)(9)(1)(.125)] + (2)(9)(.75)(.125) = 11.81 \text{ in}^3$$

\oplus

top shell

$$(\pi (10)^2 - \pi (1.906)^2 + 2\pi(10)(13))(.125) = 169.97 \text{ in}^3$$

middle shell

$$(\pi (12.25)^2 - \pi (2)^2 + 2\pi(12.25)(13))(.125) = 91.26 \text{ in}^3$$

bottom shell $(-\pi (2)^2)$

$$(\pi (14.5)^2 + 2\pi(14.5)(12.0))(.125) = 108.82 \text{ in}^3$$

$$((.5)(12)(12.36) + (12.36)(2))(6)(.125) = 37.08 \text{ in}^3$$

$$(\pi (14.5)^2 - \pi (2)^2)(.125) = 90.49 \text{ in}^3$$

$$\text{Total volume} = 550.82 \text{ in}^3$$

$$\text{weight} \rightarrow (V\rho) = 28.47 \text{ lbs}$$

APPENDIX 8

```

10 GINIT
20 PLOTTER IS 705,"HPGL"
30 GRAPHICS ON
31 LORG 4
32 CSIZE 4
40 MOVE 65,95
50 LABEL "Tasks Window for Rotary Motion - Very Low Speeds"
51 LORG 1
60 MOVE 4,30
70 DEG
80 LDIR 90
90 LABEL "Torque (in-lbs)"
100 LORG 4
110 CSIZE 3
112 FOR Y=15 TO 90 STEP (75/4)
113 MOVE 9,Y
114 READ Yaxis$
115 LABEL Yaxis$
116 NEXT Y
117 DATA 0,150,300,450,600
120 MOVE 66,5
130 LDIR 0
131 LORG 4
133 CSIZE 5
140 LABEL "Speed (rpm)"
151 CSIZE 3
152 LORG 6
154 FOR X=10 TO 115 STEP 21
155 MOVE X,14
156 READ Xaxis$
157 LABEL Xaxis$
158 NEXT X
159 DATA 0,25,50,75,100,125
161 VIEWPORT 10,120,15,90
170 FRAME
180 WINDOW 0,125,0,600
190 AXES 25,50,0,0
200 PEN 6
210 FOR Hp=0 TO .5 STEP .05
220 FOR N=1 TO 125 STEP 2
230 T=(63025*Hp)/N
240 PLOT N,T
250 NEXT N
260 PENUP
270 NEXT Hp
280 LORG 4
290 CSIZE 2
360 FOR Hp=.05 TO .5 STEP .05
370 T=(63025*Hp)/122
390 MOVE 122,T
440 LABEL Hp
460 NEXT Hp
470 MOVE 122,5
480 LABEL "HP"
490 PEN 2
500 MOVE 0,350
510 DRAW 125,350
520 PENUP
530 PEN 3

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540 MOVE 0,250
550 DRAW 30,250
560 DRAW 30,0
570 PENUP
580 PEN 4
590 MOVE 0,35
600 DRAW 30,35
610 PENUP
620 PEN 5
630 MOVE 0,200
640 DRAW 78.78,200
650 FOR N=78.78 TO 125
660 DRAW N,15756.25/N
670 NEXT N
680 PENUP
690 END
10 GINIT
20 PLOTTER IS 705,"HPGL"
30 GRAPHICS ON
31 LORG 4
32 CSIZE 4
40 MOVE 65,95
50 LABEL "Tasks Window for Rotary Motion - Very Low Speeds"
51 LORG 1
60 MOVE 4,30
70 DEG
80 LDIR 90
90 LABEL "Torque (in-lbs)"
100 LORG 4
110 CSIZE 3
112 FOR Y=15 TO 90 STEP (75/4)
113 MOVE 9,Y
114 READ Yaxis$
115 LABEL Yaxis$
116 NEXT Y
117 DATA 0,150,300,450,600
120 MOVE 66,5
130 LDIR 0
131 LORG 4
133 CSIZE 5
140 LABEL "Speed (rpm)"
151 CSIZE 3
152 LORG 6
154 FOR X=10 TO 115 STEP 21
155 MOVE X,14
156 READ Xaxis$
157 LABEL Xaxis$
158 NEXT X
159 DATA 0,25,50,75,100,125
161 VIEWPORT 10,120,15,90
170 FRAME
180 WINDOW 0,125,0,600
190 AXES 25,50,0,0
200 PEN 6
210 FOR Hp=0 TO .5 STEP .05
220 FOR N=1 TO 125 STEP 2
230 T=(63025*Hp)/N
240 PLOT N,T
250 NEXT N
260 PENUP

```


Appendix 9

Mechanical Properties of Vespel®

® E. I. Du Pont De Nemours & Co.
Wilmington, DE

Polyimide Polymer

Density: 0.0517 lb/in³

Tensile Strength

-293°F	16 × 10 ³ psi
Room Temperature (RT)	13 × 10 ³
418/508°F	5.7 × 10 ³

Flexural Strength

RT 17 × 10³ psi

Compression Strength

All Temperatures > 40 × 10³ psi

Elongation

RT 6%

Exposure

At Elevated Temperatures
(For Fully Aged Vespel®)

50% weight loss
at 598°F in 600 hours
at 778°F in 45 hours

Tensile Modulus

RT	450,000 psi
418°F	300,000 psi
616°F	230,000 psi

Processing

Precision Sintering,
Machining Articles

Remarks

Vespel® exhibited no drop in mechanical properties after 5000 hours in liquid Nitrogen (-320°F)

Polyimides have very low coefficients of thermal expansion, very low coefficients, and high resistance to energy-rich radiation.

Progress Report

Week 1

As of October 1st, we are still in the process of attempting to contact NASA with reference to previous hand tool research into the background of hand tool design. This includes lunar environment, Computer Aided Design and ergonomics. We have developed our project title, problem statement and tentative time table for the quarter. Also the group distributed job tasks for the upcoming week.

SAE XMA.

Robert Colquhoun

Kathy Walsh

Amy McEntee

Michael C. West

Scott E. Pette

Scott Marshall

Paul Bontz

KD

4. Kathy Dubnik has researched restrictions on device due to lunar environment and has attempted to contact Jim Sommers at Fernbank Science Center.

AM

5. Amy McEntee has researched lunar environment restrictions, and began looking into the NASA Database. Attempted to contact Jim Sommers at Fernbank Science Center.

RC

6. Robert Coleman has used library online system to locate information on pressure switch design. He has also worked on basic limitations of the device.

SHN

7. Sae Na researched types of fasteners expected for a lunar mission, brainstormed on basic hand tool tasks, arranged transportation for field trip to NASA - Huntsville, Alabama.

WSM

8. Scott Marshall has found literature on fastener types and hand tool types from Tech library, and has brainstormed on basic hand tool tasks.

Progress Report

Week 2, October 8, 1987

This week we have completed our problem statement. Each team member has researched the following areas: ergonomics, visual aiming systems and extravehicular tools, lunar environment restrictions, pressure switch design, fastener types and hand tool types.

SP

1. Scott Patton has contacted Dennis Matthews, Vince Cassisi, and Cleate Booher, and ordered manuals on Manned systems Integration Standards and a videotape on manned systems in space. He is still attempting to contact the supervisor of NASA Hand Tool operations.

SKB

2. Karl Bentz has researched ergonomics, and has found several books and further references on the subject. He has begun to locate a sample spacesuit glove from the Textile Engineering Dept.

MCA

3. Michael West has searched for literature on visual aiming systems and extravehicular tools.

Progress Report

Week 3, October 15, 1987

This week we held two informal meetings. During our first meeting broke our project up into smaller areas and assigned two members to each area. We also created a design matrix to help us coordinate design efforts within each area. During our second meeting, we planned our field trip to NASA - Huntsville, Alabama, and we began to brainstorm about possible end-effector configurations.

- SP 1. Scott Patton - continued attempts to get additional information from NASA, and has brainstormed and developed preliminary ideas for the framework.
- KB 2. Karl Bentz - has developed several preliminary ideas for the framework, and has ^set up appointments with professors from ID, PSY, ISYE, EE & TEX.
- MCW 3. Michael West - has begun workshop on I-DEAS software, and has discussed the function of the end-effector mechanism.
- KSD 4. Kathy Dubnik - has continued research on the lunar environment, and has discussed the requirements and design of end-effectors.
- AEM 5. Amy McEntee - ~~has set up trip to Fombank~~, continued to investigate the lunar environment, and is learning CAD on the Apollo system.
- RC 6. Robert Coleman - has talked to Dr. Harvey Lipkin about maintaining static equilibrium with an exoskeletal device, and has discussed possible control systems that are pressure controlled.
- SHM 7. Sae Na - has set up the trip to NASA - Huntsville, Alabama, has revised the IBM VM progress report with Gary, and drew up the design matrix. *learning CAD on the Apollo System*
- WSM 8. Scott Marshall - has worked on power supply and control power supply requirements, and has brainstormed on methods of power supply.

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Progress Report

Week 4, October 22, 1987

This week our group traveled to NASA - Huntsville, Alabama, to talk with engineers about previous hand tool projects, and to determine the feasibility of our initial concepts. During the trip, we tried on several space suit gloves to get a better idea of what kinds of hand tools could be used and how much strength would be required to manipulate tools within the glove. We also watched a videotape of astronauts in a neutral buoyancy tank repairing various structures. We also took a tour of the facilities, and took photos for our report. We also ordered EVA hand tool catalogs and gathered some more information to help us in our design.

After discussing with engineers at NASA, we have decided to scrap our original concept of an "exoskeletal" device attaching to the arm of the suit. Our reasons are as follows. First, after trying on the gloves and observing astronauts in the neutral buoyancy tank, we concluded that the astronaut would have sufficient grip strength to hold on to most tools if the handles and controls were sufficiently large. Second, we cannot attach anything directly to the suit itself due to the risk of wear and damage. Third, we cannot run controls from inside the glove to outside the suit. We have brainstormed on new alternative concepts that could fit the requirements in our problem statement. We have also turned in our database reference search request to the Tech library and we anticipate results by Wednesday, October 21. We have also done some research on the library DATEX CD-ROM system. We have also ordered an EVA hand tools catalog from NASA.

WSM
SWW
LLP
KJD
AEM
ZBS
MCW
RC

Progress Report

Week 5, October 27, 1987

This week, in response to the information gathered at NASA Huntsville, we divided our project into two major tasks. The first task, designing "hand powered tools", we assigned to four team members, and the second task, designing "powered hand tools", we assigned to the other four team members. We have also submitted and received our reference search from the Tech library and are currently reviewing it and gathering our references. We have ordered and are reviewing NASA documents on EVA hand tools and EVA human factors considerations. The "hand powered tools" group made a list of the major fasteners expected to be encountered at a lunar construction site, and began to make recommendations regarding tools needed to implement them. The "powered hand tools" group brainstormed on methods to supply impact from a hand held device, and discussed the design of a torque driver for lunar use. Our group wrote, practiced and critiqued our midterm presentation to be given October 28 before the entire class.

AEM
KJD

~~ESP~~
SAPM

JMB
RC

WSM

MCY

Progress Report

Week 6, November 5, 1987

This week our group got the results of our library on-line database search back, and we've found several helpful references. We've held two informal meetings to further work on our design concepts, organize our tasks, and review our literature.

The "hand powered tools" group has made a list of fasteners best suited to conditions on the lunar surface, and is in the process of designing tools to implement them. They have also decided to design a tool carrier that will allow the spacesuited astronaut easy access to the tools, and that will protect the tools from dust and dirt. The group has found literature on previous design attempts - specifically for lunar geological sampling and surveying tools used on the Apollo missions.

The "power tools" group has brainstormed on several alternatives for torque drivers and impact drivers. From the literature search, the group has begun to determine specifications required for torquing and impact tasks, and has listed desired features for their devices. The group has also subdivided their design tasks into the following categories: housing, power transmission and supply, end-effector connectors, bearings and lubrication, shaft design, and controls.

WSM
HEM
RC
SHM
KSD
SHB
MCW
SGP

Progress Report

Week 7, November 12, 1987

This week our group met informally twice to discuss details of the project. The power tools group has gone through catalogs of various power tool manufacturers. The power tools group has also researched torque, speed and power requirements for various tasks in order to begin sizing the shaft and motor for their rotary tool.

The hand tools group has met to discuss various aspects of their project. Design is continuing on their tool carrier and several concepts for types of tools to be included in their kit have been discussed.

WSM
KJD
RKB
MGN
AEM
RC
SHN
SEP

Progress Report

Week 8, November 19, 1987

This week we have met informally once to discuss our progress and assign tasks to be performed independently. We have come up with a preliminary outline of our final report.

The hand tools group has formulated an idea for a tool carrier design, a workstation and a method for carrying selected tools to the worksite. Before a finalized version is implemented, we plan to discuss our ideas with a professor in the Industrial Design department. We are also in the process of selecting the tools that will be taken to the moon. We plan to modify existing tools and develop several new ideas.

The power tools group has divided up design tasks as follows: Scott Marshall is responsible for coming up with a set of torque and speed requirements and a control scheme for the driver motor. Scott Patton is designing a shaft compatible with our maximum torque specifications and is specifying a motor. Sae Na is designing a connector for the end-effectors and is working on a power supply. Kathy Dubnik is designing the housing and is specifying the bearings, seals and lubricants for the driver.

SHM

SP

WEM
KJD

ASM

RC

MCY

SKB