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# SPACE TOOL KIT

# SURVEY, DEVELOPMENT AND EVALUATION

# PROGRAM

## FINAL REPORT

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Prepared under Contract No. NAS 9-3161 by

MARTIN COMPANY
Baltimore, Maryland

for

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

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## SUMMARY

The Space Tool Survey Development and Evaluation Program was conducted by Martin Company's Baltimore Division for NASA Manned Spacecraft Center, Crew Systems Division. Black and Decker Company, under a subcontract, designed and fabricated a power tool motor and three zero reaction attachments.

The program objectives were to define the minimum requirements for tools and accessories for maintenance, repair and assembly tasks in orbit and on the lunar and martian surfaces, to design tools and accessories capable of being used in the various environments to perform the tasks defined and to fabricate and evaluate the prototype tools and accessories (Figure 1).

Initially, a survey of current literature revealed almost no effort had been previously accomplished in developing space power tools or in defining the requirements for an integrated tool kit. Studies were undertaken to define the expected maintenance task functions and tool performance range as well as the environmental and human factors constraints which would affect tool design. Tradeoffs between the various elements were made and the basic tool and accessory areas defined for development were:

1. Power torquing;
2. Power sawing;
3. Power drilling;
4. Hand hammering;
5. Hand screwdriving;
6. Personnel restraint;
7. Work site illumination;
8. Small parts manipulation and retention; and
9. Work site layout and tool and accessory packaging.

Following definition of the basic areas, work proceeded concurrently on tools and accessories. A power tool consisting of a basic 12 V dc permanent magnet motor and handle unit was developed along with attachments for torquing, sawing and drilling. All attachments work with essentially zero reaction on the astronaut worker so that they are capable of being used in an orbital maintenance situation. The wrench will torque up to 1/2-inch bolts to 85-95 foot-pounds with less than 6 inch-ounces torque on the handle. The saw will cut typical spacecraft skin materials at rates up to 4 feet/minute with hands off during the cutting. The drill will cut holes up to 21/32 in .040 aluminum or .028 titanium in less than 30 seconds with only an initiating impulse of .5 foot-pound energy required. Actual hole cutting in zero gravity requires no holding by the operator.



Figure 1. Final Tool Kit and Contents

Standard hammer and ratchet screwdrivers were adapted with special handles to work with the spacesuit glove for more efficient operator performance.

A study of restraint requirements led to the development of an adhesive attachment 1.13 inches in diameter which is electrically bonded to the skin in about 30 seconds to produce restraint capabilities of greater than 50 pounds at the work site. The attachment is designed to be applied when and where needed to minimize the structural complications of mechanical attachment points. A feasibility model of a "rigid rope" restraint device was fabricated to investigate a solution to the problem of providing high strength restraint devices with maximum flexibility of worker movement. Work site illumination requirements resulted in a design of ellipsoidal reflector work lights mounted on the helmet at eye level to provide optimum illumination intensity and direction with minimum power input. The problem of small parts manipulation and restraint which is aggravated by the pressurized glove was solved by the design of a small brush holder into which the nuts and screws can be pushed. The bristles on the brushes retain the parts to keep them from drifting away.

The various tools and accessories were integrated into a complete package which is roughly a triangular parallelepiped which fits between the worker and the work surface. The tool kit is 14 x 14 x 10 inches with a volume of .7 cubic feet. The kit is fabricated from a drawn metal shell with a rigid foam liner into which pockets have been cut for storage of the tools and accessories. In the kit is a 5-pound, 163 watt hour, 12 V dc silver zinc rechargeable battery providing power for the tools, adhesive applicator and work lights.

Evaluation of the tools, accessories and tool kit has shown conclusively that it is possible to design tools which can be used by an astronaut with minimum restraint.

Brief evaluations of the performance of the motor in a 5 psia 100% oxygen environment indicates that the explosion and contamination hazard is less than had been anticipated.

This program has demonstrated the basic feasibility of developing tools for space use. Prior to the time when a complex maintenance, assembly or repair task is actually performed in space, refinement in concepts, and analysis of performance and required functions, beyond what has already been accomplished in this program will be necessary.

## INTRODUCTION

As spacecraft develop to the point where long stays in orbit or on the Moon or Mars become desirable, the ability of man to extend the useful life of his equipment by appropriate maintenance and repair techniques becomes increasingly significant. Although maintenance and repair techniques have developed in sophistication and efficiency under normal earth-bound conditions, the requirements of extraterrestrial operations create many new problems. Even simple mechanical tasks such as removing and replacing a bolt or nut are complicated by the facts that the task must be done (1) in a pressurized spacesuit with restricted mobility and dexterity, (2) under gravity conditions which distort the mechanical relationships a worker is accustomed to using on earth, (3) under visual conditions of extreme contrast and glare and (4) under logistic restrictions which preclude replacement of items accidentally dropped or damaged during maintenance.

A limited amount of work has been accomplished to provide an adequate maintenance and repair capability in space. All manned space programs to date have officially adopted a no-maintenance policy. However, the desirability of a maintenance capability as an alternate to even higher equipment reliabilities is being more widely recognized as the nation's Space Program progresses toward longer manned missions. Most of the initial efforts have been concentrated upon defining the maintenance tasks expected and analyzing the capability of existing hand tools to accomplish specific tasks under the extraterrestrial environmental restrictions. A primary area of interest has been the use of tools in orbit where the mechanical advantage of the workers weight and traction are missing and "zero reaction" tools are necessary. All such tools employ a closed loop of mechanical forces so that components of the active force applied to the work do not react on the worker, causing undesirable motion of the worker relative to his work. Many earthside tools incorporate such features in use, for instance, tightening a tubing fitting with two open end wrenches involves a closed loop of force through the arms and shoulders and under zero gravity conditions could be done by an astronaut. Squeezing with a pair of pliers involves a closed loop of force through the fingers and thumb. On the other hand drilling a hole, sawing, or wrenching a nut on an attachment stud involve forces which are open loop and which will therefore cause the astronaut to rotate about the work or to push himself away as he pushes on the tool. Many concepts for tools which will overcome this problem have relied upon a special fitting to which the tool is attached to react the active force. Although this solves the initial problem it creates another just as large because of the design complexity of introducing a special non-standard fitting requirement in the structure in all potential maintenance areas.

In addition to the force problems and the general environmental problem, the question of physiological efficiency is important because increased metabolic loads from tool use will require more life support capability. Power tools have been developed on earth because they are quicker and easier to use. The same applies to space tools with increased emphasis. Although the technical problems of providing power to the tool are greater, steps in that direction have been made.

In 1964 a zero reaction battery operated power tool was developed for the USAF Aero Propulsion Laboratories.[1] The tool provides an output torque for fastening and loosening nuts of 75 foot-pounds with a reaction on the handle of 6 inch-ounces. With such a low handle reaction the astronaut does not rotate during the approximately 3 seconds the tool requires to tighten a nut. The tool also has a capability of drilling, grind-

ing and tapping when used with an appropriate maintenance restraint.

The space tool survey, design development and evaluation program was oriented toward further investigation of the potential of power tools for space maintenance use. It was planned that basic tool functions such as cutting, torquing and hammering would be evaluated as potential space maintenance functions and that prototype tools performing these functions be developed with consideration of the space environment and human factors restrictions. In addition, the necessary accessory equipment would be defined and developed. The complete set of equipment would be packaged in a tool kit embodying the approaches defined by the space operation considerations.

## Program Objectives

The principle objectives of the Space Tool Survey Development and Evaluation Program were threefold: (1) to define the requirements for design of space tools; (2) to define the interface requirements between the tool systems and the worker; and (3) to fabricate a prototype tool kit based upon the definitions of (1) and (2). In general the development of a space maintenance capability is dependent upon a somewhat circular logic typified by these two questions; "Should this specific maintenance function be performed?" and "Is it possible to perform it?". It is difficult to answer one without the answer to the other. Yet if one is not answered the other is a meaningless question. For the purposes of this program the emphasis was placed upon answering the second, "Is it possible to perform a maintenance function?". The answer to whether or not a specific maintenance function should be performed is dependent not only upon the existence of a capability to do it, but a detailed study of other factors such as mission objectives, equipment design, logistic capability, etc. The present status of the manned space program limits the activities in that area. However, answers to the second question, is it possible to perform a particular maintenance function, can be developed so that at a later date the trade off for a particular mission can be made from a base of sound experience.

With this emphasis the detailed objectives were:

(1) to define broad maintenance functions rather than a detailed maintenance task analysis;

(2) to define the range of operation of a tool, e.g., torque fasteners of 1/2 inch diameter or smaller, saw metal skins up to 1/4 inch thick, etc.;

(3) to define the environmental conditions under which a tool would be used;

(4) to define the capabilities of a spacesuited astronaut to exert the necessary tool forces in appropriate positions so that the difference between his capability and the requirements of the task can be made up by the tool itself;

(5) to define the tools and accessories constituting a minimum demonstration of the feasibility of space maintenance within the constraints of number (4);

(6) to design and fabricate prototype tools and accessories which demonstrate the feasibility of performing the defined maintenance functions under the restrictions of astronaut performance and expected environments.

These objectives have been met. Because the space programs are adding new knowledge and developing new equipment at a rapid pace, the approaches embodied in this program should be considered as the best that can be done within program limitations and not as the definitive answer to the space maintenance problem. Far more effort must be expended toward the same objectives to develop a complete comprehensive maintenance system oriented toward some specific mission.

## References

1. Daisey, Floyd K.: Space Power Tool. AF Aero Propulsion Laboratory, WPAFB, Ohio, RTD-TDR-63-4227, April 1964.

## REQUIREMENTS DEFINITION

### General

To adequately define the design characteristics of a space tool kit it was necessary to define the requirements of three major areas: (1) Maintenance Functions; (2) Environment; and (3) Human Performance Capabilities. Efforts were concurrently undertaken in all three areas. In addition, feasibility studies and definition of existing tool capabilities were undertaken so that at the midpoint of the program detailed tradeoffs could be made. A brief survey of current literature was made as a starting point.

### Literature Survey

In order to assure that a broad technological base was included in the development of the prototype space tool kit a brief literature search was made to ascertain the existence of space tools applicable to the program. Although there had been efforts by several companies such as Martin, American Machine and Foundry, Ling Temco Vought, North American Aviation and even the Soviets,[1] little was discovered which was applicable to the program. Efforts in the past few years have been oriented toward hand tool development and definition of hand tool requirements (ref. 2, 3) for orbital use. No effort has been made for defining the possible requirements of lunar maintenance.

The hand tool approaches have followed two paths. One is the use of standard hand tools along with a restraint system (ref. 2, p. 190) and the other is development of special fittings which react the tool forces (ref. 2, p. 189). In both cases the closed loop of reaction forces is attained for orbital operations. The tradeoff between the two approaches for this program is made later in the discussion of restraint system design. Recognition of the basic requirements of spacecraft design for lightweight equipment is evident although many of the special hand tool concepts show the need for design refinement before they would be considered satisfactory for actual space use.[4]

Although the physiological loads of hand tool operation versus power tool operation is significant for space operations, literature considering this tradeoff was not discovered. Those items which did examine the basic problem of metabolic loads[5] were generally developed from old style pressure suits. Since pressure suit design is progressing rapidly the data was considered as potentially misleading and, in view of the orientation of this program, of slight significance. Definition of this area in the general literature will most likely occur after spacesuit development and space tools have reached an initial plateau of development and some actual space experience has been gained.

There is only one reference to power tool development for space use. This is the report (ref. 6) on the initial feasibility development of a powered space tool.

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5. Pierce, B.: Effects of Wearing a Full Pressure Suit on Manual Dexterity and Tool Manipulation. Human Factors Vol. 5 No. 5, Pergamen Press, New York, October 1963.

6. Daisey, F. K.: Space Power Tool. AF Aero Propulsion Laboratory, WPAFB, Ohio, RTD-TDR-63-4227, April 1964.

## Environmental Constraints

General. The utilization of tools in space will occur in a number of different environments. The major areas requiring consideration are intravehicular and extravehicular orbital operations and intravehicular and extravehicular lunar surface operations. In addition, consideration was given to martian surface operations. The different areas all require consideration of the relative importance of major characteristics of the environment as well as the micro-environment of a particular tool. The characteristics considered were:

- Gravity
- Temperature (surface of the work site)
- Vacuum
- Gaseous Environment

The illumination characteristics were also defined. Although they did not greatly affect tool design they constitute a major work problem. Figure 2 represents a summary of the worst case environmental constraints.

Gravity. The most pressing problem of gravity is the effect of apparent weightlessness in orbit. Because orbital mechanics, in effect, eliminate the gravity vector the mechanical relationships involved in tool use are changed. The relationship of tool, work and operator is discussed in the Human Factors section. Zero gravity effect upon tool mechanisms is minimal except insofar as a design requires a closed loop of work forces. The only major consideration is in the elimination of gravity-dependent mechanisms such as latches. Since tools are normally designed to operate in any position this was not considered a problem.

The possible range of gravities is 0-G in orbit, 1/6-G on the lunar surface and 2/5-G on the martian surface. Although the lunar and martian gravities will influence the use of tools, the design and test of tools was oriented toward 0-G. The only point where the gravity of 1/6-G or greater will be more significant than 0-G is in the design of load bearing components which may be required to support the tool weight. Since the tools are all hand held this is not considered a major problem. Utilization of the tools in test under 1-G considerations will bracket the expected range and demonstrate the capability of a tool to operate under the full range from zero to one earth gravity.

Temperature. Operation of the tools is required under extremes of temperature which are greater than those normally encountered on earth. Detailed definition of temperatures for the tools was beyond the scope of this program because of the relative complexity of the problem and the extreme dependence of the exact definition of work geometry, tool geometry and the dynamics of heat flow between the work, the tool and the worker's gloves.

The temperature environment was therefore defined as a function of a basic operational sequence which is felt to cover the worst case condition.

- The tools would be stored in the spacecraft cabin and at the start of a maintenance operation would be at cabin temperature of approximately 75°F. (standard room temperature).
- The tools would be stored in an insulated tool kit except when actually in use.

# GENERAL ENVIRONMENTAL CRITERIA

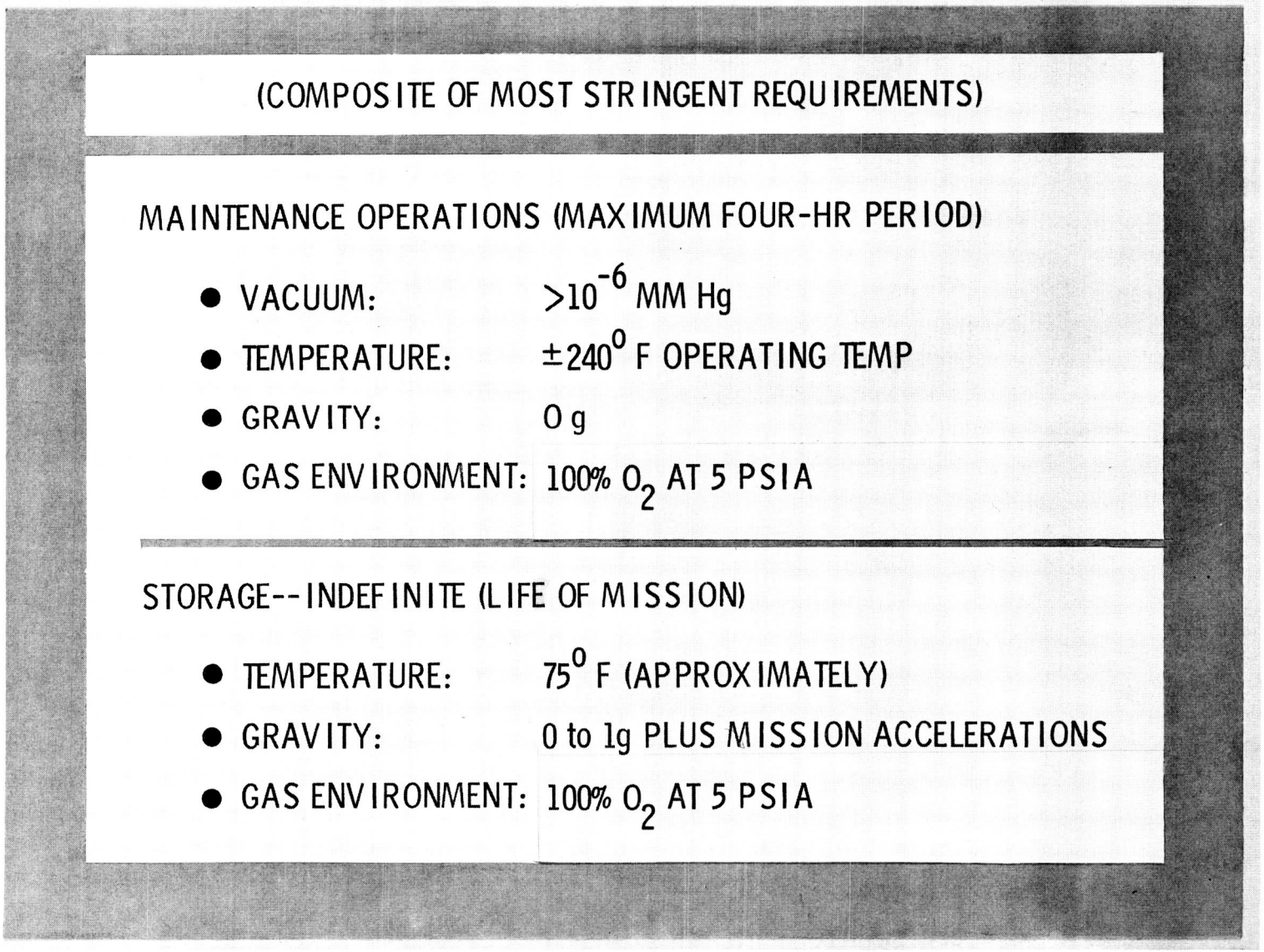



Figure 2. General Environmental Criteria

- The tools in use would contact the work surface with only a small area in intimate contact.
- The work surface temperature would be dependent upon the mission characteristics. The two major temperatures considered were temperatures on the lunar surface and in orbit around the earth or the moon.

Lunar Surface Temperatures (1). The maximum average temperature on the surface of the moon occurs at the point where the sun is directly at the zenith which is known as the subsolar point. It has been observed that the maximum temperature occurs when the heat flux is a maximum, that is, no phase lag is apparent. The theoretical and experimental determination of this temperature are in agreement within less than 1°K. at an average value of 374°K. (213.8°F). The actual temperature at a particular subsolar point may be as high as 390°K. (242.6°F) or as low as 330°K. (134.6°F). The variation of subsolar temperature is due to the fact that the solar constant (average $1.37 \times 10^{-6}$ ergs/sec-cm$^2$) varies slightly depending on the moon's distance from the sun, and the actual albedo ranges from less than 5% to more than 40%. The lunar temperature distribution about the subsolar point of 374°K. (213.8°F) decreases with increasing radii from that point.

As soon as the sun sets at any point on the lunar surface, the temperature drops within a few days to its minimum value of 120°K. (-243.4°F). However, the initial temperature drop upon "black out" of solar radiation can occur very rapidly. One observer measured a temperature drop from 374°K. (213.8°F) to 200°K. (-99.4°F) over a period of one hour during an eclipse.

Reference 2 indicates that short term experiments should be designed for satisfactory operation within the range of 220°K. (-63.4°F) to 360°K. (188.6°F).

Earth Orbital Temperatures. The variations in temperature for low orbits around the earth or the moon are dependent upon the amount of heat flux impinging upon the tools and tool kit. Since the geometry of the situation is highly variable the establishment of a worst case condition will require some arbitrary assumptions. Reference 3, page 90, shows that temperatures in low 50 to 100 mile orbits will result in surface temperatures varying from -292°F. to +208°F. for surfaces with an $\alpha/e$ ratio of 1.0.

| Temperature °F. | Dark Side | Sunlit Side |
|---|---|---|
| Facing Earth | -62 | +92 |
| Facing Moon | -292 | +163 |
| Facing Sun | -- | +208 |

The actual temperatures which the tools will experience will depend not only upon the heat transfer from the work surfaces but also upon the generation of heat within the tool and the transfer of heat through the spacesuit gloves.

It is not likely that the tool kit contents would cool rapidly to the lowest extreme indicated by the surface temperatures because the thermal lag in the tool mechanisms will result in the heat from the interior of the tool reaching the surface of the tool and dissipating to other items after it has been replaced in the tool kit. In addition, heat transfer from the gloves will assist in maintaining a higher temperature.

Temperature Constraint. Because the tool kit is intended for use in a variety of situations the worst possible condition should be defined as the constraint. Since the lunar surface temperatures would affect the tool kit by conduction when the kit is placed on the surface near the work site while other situations would affect it by radiation, it was decided that a temperature swing of approximately $\pm$ 240°F. would be appropriate as the worst case. However, it is felt likely that a detailed analysis or empirical study would show a much lower variation of temperatures from the storage ambient of 75°F. during a three hour maintenance excursion.

Vacuum Vacuum operation of tools has two major effects. One is the change in thermal characteristics caused by absence of a coolant fluid to transfer heat and the other is the evaporation of lubricants and changes in material characteristics caused by an ambient pressure below the basic vapor pressure of the material.

The temperature effects become significant at vacuums of greater than $10^{-3}$ Torr when substantially all the air which could carry away heat is no longer present.

The effects upon materials span the whole range of pressures but they become increasingly significant at greater than $10^{-6}$ Torr (4) (5). The significant mechanisms are loss of lubricants due to evaporation, particularly when this is accelerated by high temperatures and increased wear resulting from the lubricant loss.

Cold welding of tool parts which experience relative motion is expected to be significant only on the cutting edges of tools where atomically clean surfaces are placed in intimate high pressure contact. On other surfaces selection of lubricants sufficient for the operating times expected will minimize the vacuum effects.

Gaseous Environment. Operation of tools within the spacecraft is an expected requirement. It is assumed that the gaseous environment in all future spacecraft will approximate an environmentally controlled workshop as far as temperature ($\sim$75°F) and humidity ($\backsim$40-50%) are concerned. The parameters of pressure and gas composition will differ. The present spacecraft designs of 100% oxygen at 5 psia (3); if they change, will change toward higher pressure mixed gas systems. It was therefore decided to use 100% oxygen at 5 psia as the worst case condition.

Two problems required consideration. The explosion hazard of pure oxygen and the generation of contaminants in a closed environment. Specification of the allowable contaminants has not proceeded to the point where design guides for developers of equipment such as tools are readily available. (6, pg. 18) As a criteria for preliminary evaluation of the contaminant problem a "table of recommended maximum concentrations of atmospheric contaminants" (3, pg. 153) was used. The cabin volume of 200 to 240 cubic feet (3, pg. 402, pg. 377) which is typical of Apollo Command Module and LEM are considered as the minimum in which tools would be operated.

The explosion hazard was considered the more serious problem. Pure oxygen has been notorious as a breeding ground for accidents. Although oxygen is a highly reactive environment, propagation of burning requires a fuel as well. It would be expected that the initial energy for an explosion caused by a power tool would be developed by arcing at the commutator/brush assembly as the motor operates. A second source would be in the trigger switch.

In both instances the micro-environment would require the presence of combustibles at the point of arcing in order to generate a self-sustaining rapid reaction which would then propagate out through the motor case to the general cabin volume. Therefore the oxygen environment is not a completely unreasonable environment for the operation of a properly designed electric motor.

Illumination. Illumination requirements do not directly affect the design of tools although their proper use is highly dependent upon being able to see the work site adequately. In the cabin environment the lighting would be expected to be adequate for maintenance work, although emergency conditions of a lighting system failure would require supplementary illumination. Areas of limited accessibility might also require supplemental illumination for some detailed tasks.

Extravehicular activities require consideration of two major conditions, minimum dark side illumination and shadow detail visibility under bright side high glare illumination. In the latter the worst case would occur when the work site is in the interior of the spacecraft and the subject is peering through a minimal area access in a highly reflective skin surface. The solar brightness of $7 \times 10^8$ mL (7) will be well above the maximum tolerance of $1.6 \times 10^4$ mL (8). This will create an extreme contrast which is further aggravated by the necessity for providing a protective filter of relatively low transmissibility to maintain the highlights below the threshold where retinal damage occurs.

It is assumed that the design of a protective filter would provide a highlight brightness inside the helmet of no more than the maximum tolerance of $1.6 \times 10^4$ mL with maximum contrast because of the absence of atmospheric scattering. This worst case condition would occur during orbital maintenance activities on the side of the spacecraft exposed directly to the sun. In other situations the highlight illumination would be at the same level but earth or moon albedo would provide a source of "fill" light in the shadow areas to reduce the need for supplementary illumination. It is probable that multiple reflections from the suit and the spacecraft components will provide a limited scattering of light in the immediate work area. Empirical definition of the actual work site illumination levels appears to be the only practical approach.

The minimum level of illumination will occur on the dark side of a spacecraft which is not oriented to take advantage of either earth or moon albedo. Only the sky will provide natural light at the extremely low level of $1 \times 10^{-6}$ mL (6).

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## Human Factors Constraints

General. — In order to properly design special tools and tool kits for use in extra-terrestrial maintenance, repair and assembly situations considerable knowledge of the human operators capability to perform under extremes of environmental conditions was required. The areas of particular interest for tool and accessory design are:

(1) reach, mobility and dexterity limitations of a spacesuited worker;

(2) force capabilities of a man in reduced gravity fields; and

(3) Illumination requirements for visual identification of maintenance work items.

Five individual studies were conducted to help define the key parameters involved in manned space maintenance tasks. They included:

(1) Operator Performance and Hand Tool Evaluation;

(2) Anthropometric Study;

(3) Operator Force Capability Study;

(4) Illumination Thresholds Study; and

(5) Light Source Location Study.

The primary tools employed during testing and evaluation were the five-degree-of-freedom frictionless simulator, a supplementary Extraterrestrial (Lunar) Gravity Simulator, a 30 x 40-inch work panel and the International Latex Corp. State-of-the Art and Gemini G-1C pressure suits which were obtained on loan from MSC.

Task simulation. —The five-degree-of-freedom frictionless simulator provides a capability for evaluating the gross mechanics of zero gravity task performance. A subject in the simulator is restrained except for head and arms and balanced so that forces exerted by him will cause him to pitch, roll, yaw or horizontally translate with respect to the work site. The breakout forces are low; pitch - 20 inch-ounces, roll - 70 inch-ounces, yaw and horizontal translation - almost unmeasurable, depending upon floor levelness. Yaw and horizontal translation are accomplished by a tripod support of three low pressure air film bearings fed by an electrically driven blower on the structure so that only a small electric cable is present to create a minimal amount of drag.

Lunar gravity simulation. —With a modified cradle, which provides for torso restraint and head, arm and leg mobility, the five-degree-of-freedom simulator is used with the Extraterrestrial Gravity Simulator (fig. 3) to evaluate the problems of operating on the lunar surface. The basic principle of the device depends upon the mechanical relationships involved in performing work with tools. The diagram illustrates the relationships.

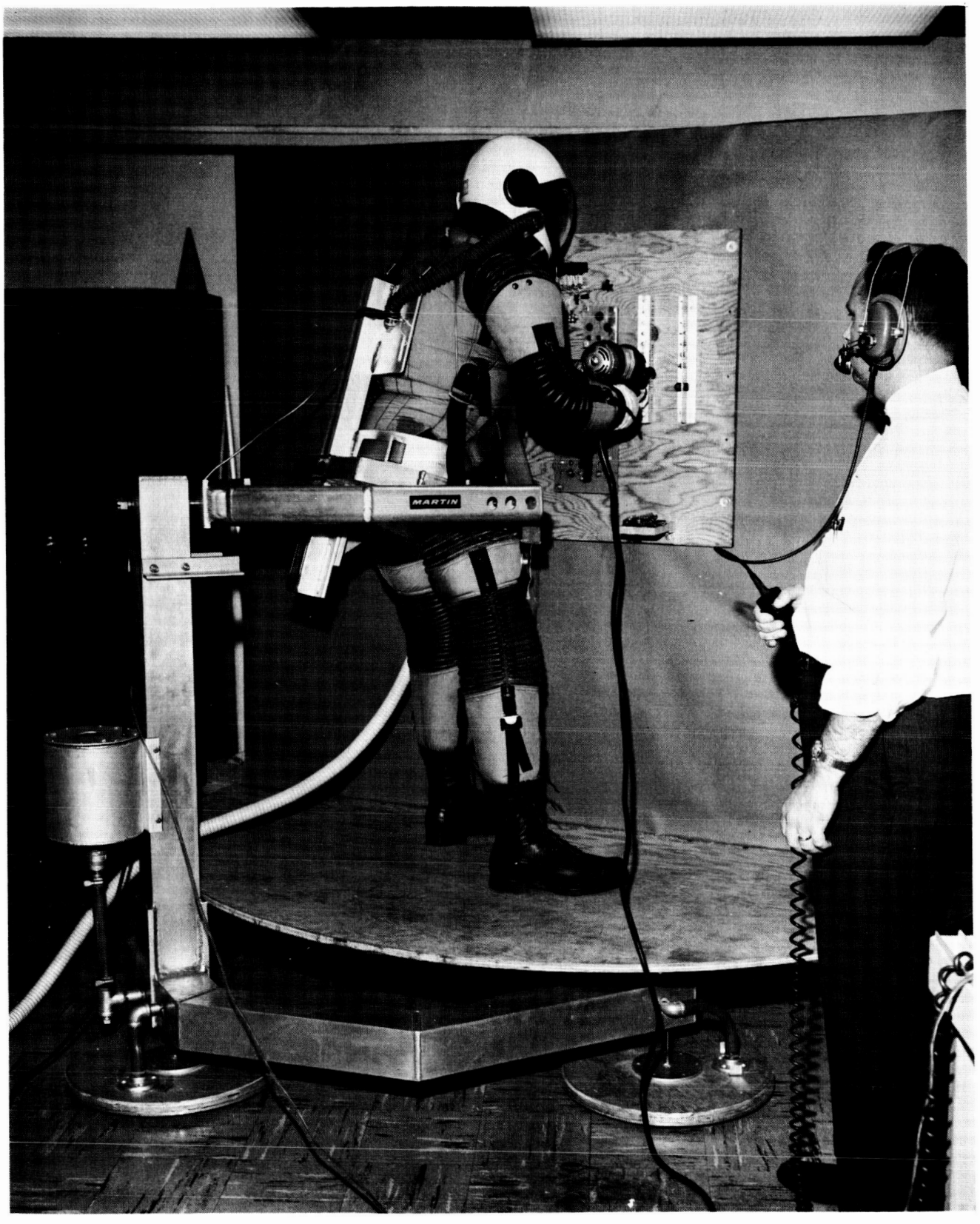
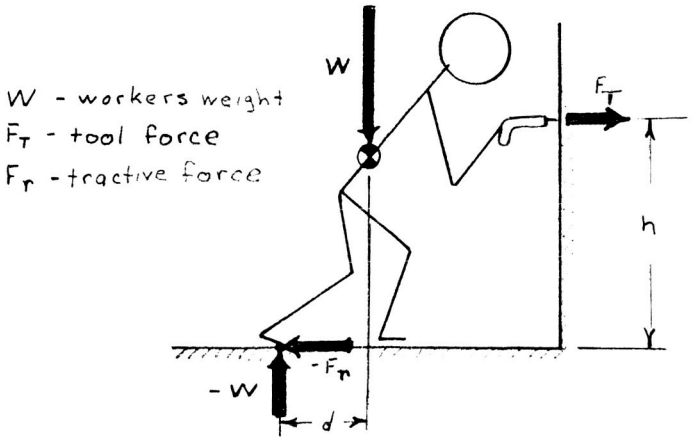
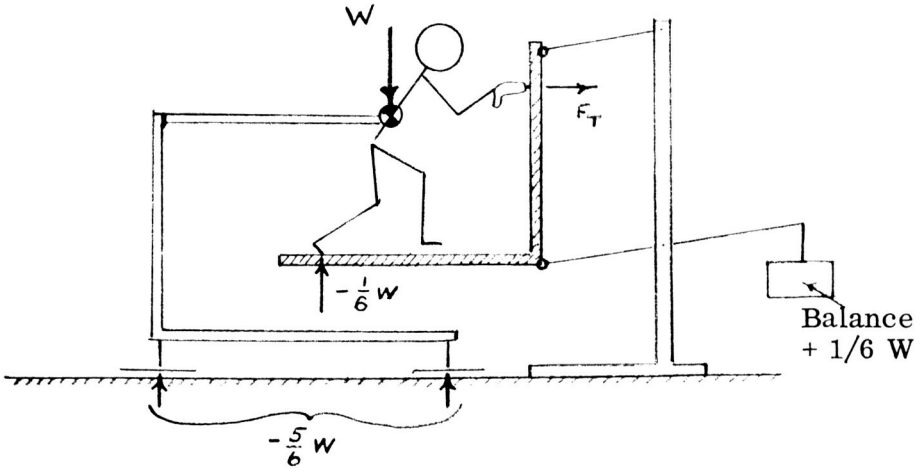



Figure 3. Extraterrestrial (Lunar) Gravity Simulator




As the worker pushes against the work, the push force is developed as a moment around the point of contact of the worker's foot so that $Wd = F_t h$ where W is acting through the worker's center of gravity. To change the force applied by the tool, the position of the center of gravity relative to the foot must be changed. In most cases this also involves a change in the height of the CG from the floor. The tractive force can be resolved into two components equal to -W and $-F_r$ with the relationship that, with the coefficient of friction of the foot on the surface ($\mu$), $\frac{-W}{-F_r} = \mu$ for impending slipping. Although a man's mass is constant, W is proportional to the gravity field. Therefore for a given $\mu$, $F_r$ will be affected by the gravity field. To simulate the effect of reduced gravity it is necessary to reduce the effective magnitude of the vector W. This is accomplished in the simulator as follows. The subject is supported in the simulator so that his center of gravity is at a fixed height from the lab floor, but he has five-degrees-of-freedom of motion. The floor of work site is supported on a moveable counterweighted parallelogram structure (fig. 4) so that it can move vertically and it presses upward against the worker's feet with a force equal to 1/6 of his earth weight. As he changes position his CG stays fixed vertically while the floor moves. The work piece is fixed in relation to the simulator floor so that the relationships between work, subject CG and floor are the same as if the worker were standing unrestrained in a 1/6 G environment. The force relationships are the same as shown above except that the reaction to W now has two components.




Only the -1/6 W component is effective in determining the tool forces involved.

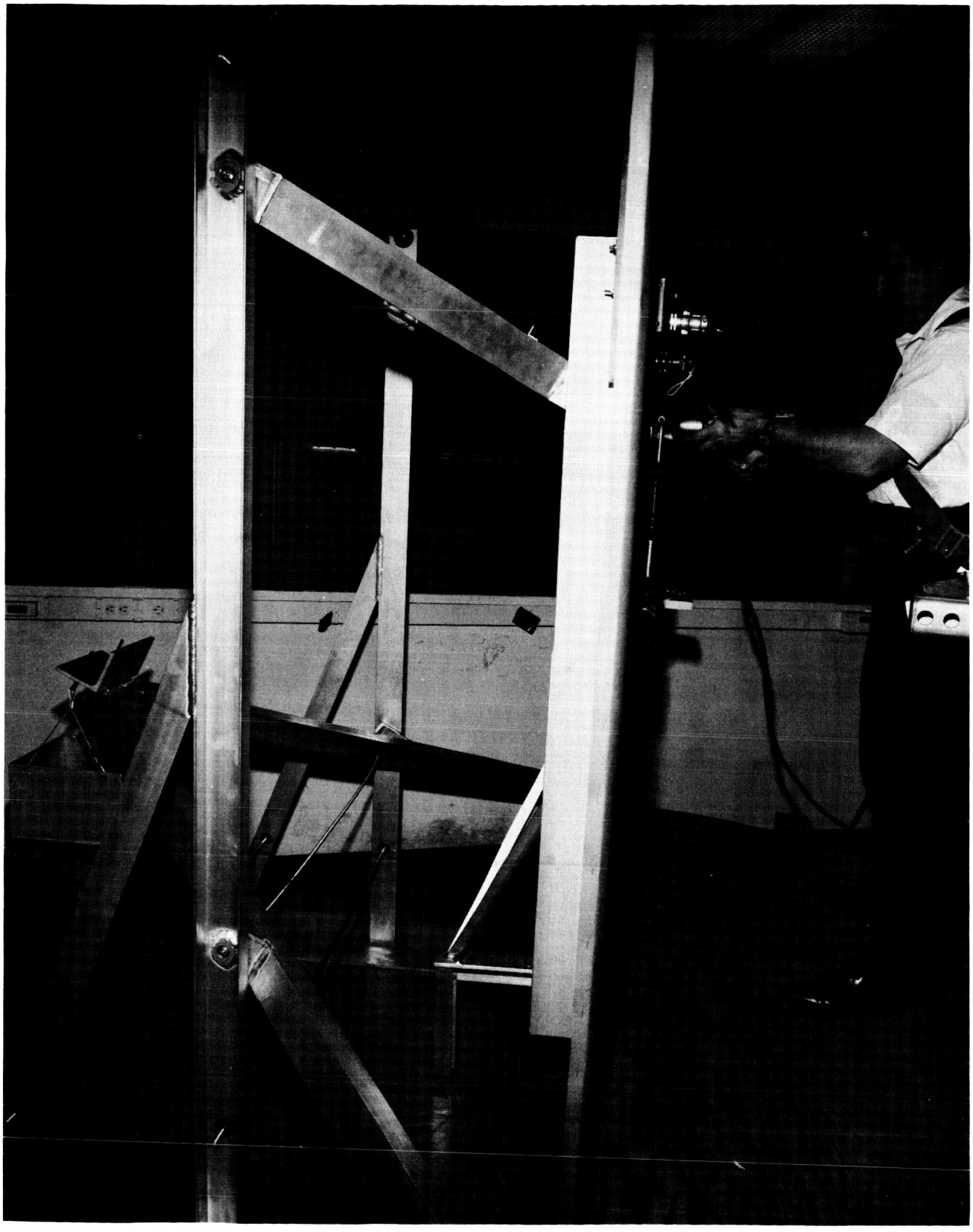


Figure 4. Extraterrestrial (Lunar) Gravity Simulator (Side View)

A visit was made to NASA Langley Research Center for the purpose of obtaining a comparison of the Martin Lunar Gravity Simulation technique and the technique developed at Langley.(1)

The two techniques are essentially complementary and, in the areas where direct comparison was possible, similar results were obtained. In the Langley simulator it is quite easy to perform tasks involving predominantly in-plane motions, such as walking, running, jumping, etc. Because the arms are left free, it is also possible to perform maintenance-type tasks in which the major forces are in-plane. Tests were made with two subjects pushing and pulling in-plane at angles of $0^{o}$, $+45^{o}$, and $-45^{o}$ from the simulator horizontal. In directions where the applied force vector would cause sliding or lifting of the feet, forces in the range of 10 to 20 pounds represented the maximum capabilities. In those positions where the feet could be braced to form a closed system, the maximum forces were a function of subject strength only. These results agree with the tests performed in the Martin Lunar Gravity Simulator.

Films of studies of locomotion in the Langley Simulator, particularly the initial attempts to accelerate to running speed, also tended to confirm our previous conclusions -- that the body positions assumed under 1/6-G will be more extreme and that traction will be a significant determinant of the forces an operator is capable of exerting while performing maintenance.

One interesting aspect of 1/6-G operations which had not been noted in the Martin simulator was the results of falling and the implications of this as far as dropping parts or tools while working. Because the accelerations are lower, the problem of catching an item may not be as severe. It is possible that a worker would have time, even in a pressure suit, to retrieve an object before it had fallen too far. Of course, because the accelerations are different, the problem of grasping a falling object may nullify this advantage. Experience in working in a 1/6-G environment will be required before these subtler questions can be resolved.

For six-degree-of-freedom zero gravity testing the work panel can be moved forward to the edge of the simulator floor to provide relative vertical motion which the zero G configuration of the simulator is lacking.

Operator performance and handtool evaluation study.—The study was performed for the following purposes:

(1) to evaluate operator performance in a shirtsleeve condition, utilizing an unpressurized ILC spacesuit condition, and a pressurized ILC spacesuit condition at 3.75 psi;

(2) to evaluate operator performance in 1-G, Zero-G and 1/6-G environments;

(3) to determine operator performance limitations for suit conditions and environments;

(4) to establish levels of task difficulty among suit conditions and environments;

(5) to evaluate the design of commercial handtools for space maintenance applications, and to investigate the hand-glove interface.

Discussion and results.—While much work has been done concerning the topic of "operator performance" and "work decrement", only a small fraction of it has been oriented toward extravehicular, lunar and martian surface maintenance by the human operator. Studies in the past have successfully demonstrated that an operator's clothing plays an important role in defining his capabilities and limitations in performing psychomotor tasks.

As new protective and complex garments are developed for advanced technological applications, the need arises to investigate the effects of such apparel on its wearer. This does not mean that the human element was never considered during the development of a given article, but instead that technical tradeoffs were necessary which the human operator had to accept. The results of such tradeoffs place limitations on an operator's ability to perform certain types of work (refs. 2, 3).

Two subjects participated in the study. A working pressure of 3.75 psi was selected for the ILC suit. A representative sample of maintenance tasks that a spaceworker will encounter was compiled. These tasks were mounted on a work panel which was mounted to the rear of the 1/6-G simulator platform during testing (see figures 3 and 5). Only certain components on the panel were utilized during this test, since others were employed for different areas of testing. No power tools were used in this study, and the handtools that were used are shown in the top row of Figure 6.

Two subjects were given two trials each of the following tasks in the shirtsleeve, unpressurized and pressurized suit conditions. See Figure 7 for component identification.

Task a - Remove component 4-A-3, place on retaining device, and replace using 1/8-inch screwdriver.

Task b - Remove component 1-A and place on 1-B. No tools needed for this task.

Task c - Remove component 2-D with 3/8-inch socket & ratchet, and replace.

Task d - Remove and replace component 7-B using 1/4-inch screwdriver and ignition wrench.

Task e - Remove and replace component 9 using 7/16-inch socket on ratchet wrench. Note: There are 4 washers associated with the 4 nuts.

Task f - Disassemble component 8A-F and re-assemble in reverse order using crescent wrench.

Time was the only quantitative criterion, and a counterbalanced order of suit conditions was employed. The clock was stopped when a component was dropped until it was retrieved. When two components were dropped in succession the trial was aborted.

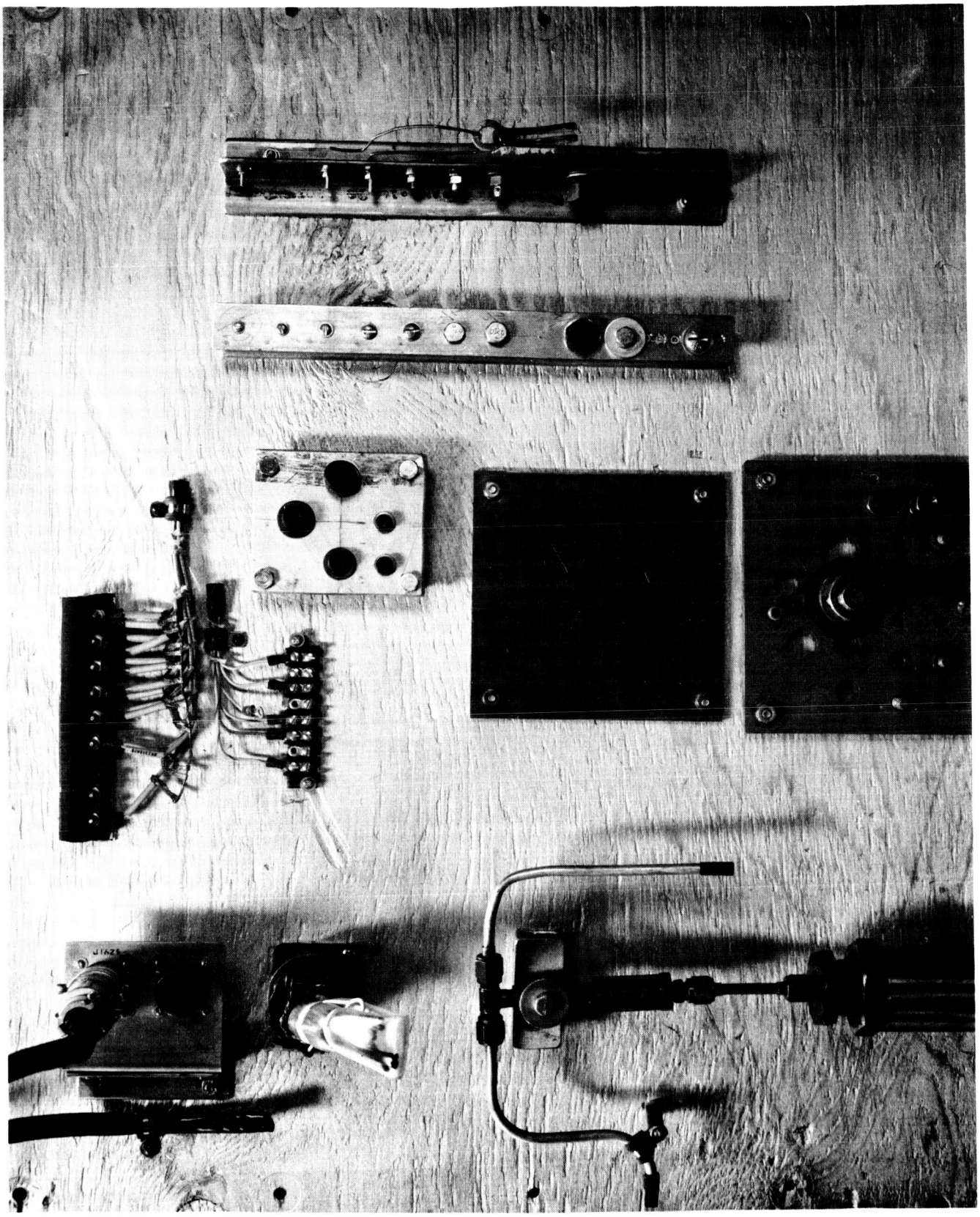


Figure 5. Work Panel



Figure 6. Tools Used in Studies

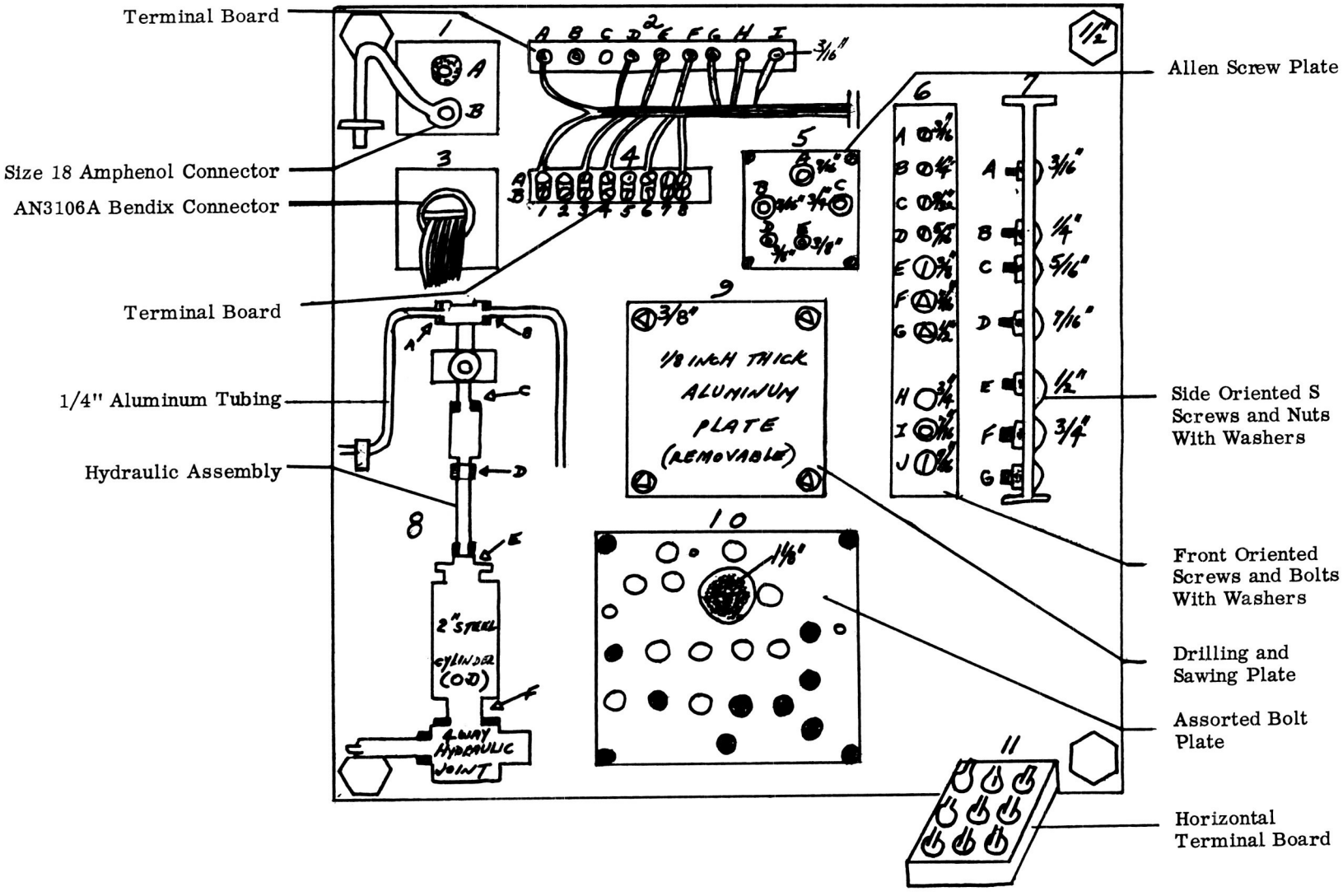



Figure 7. Work Panel Identification Sketch

In addition to the time criterion, experimenter observations and subject comments were recorded on the data sheets. Following is a list of significant observations and remarks:

(1) Little or no difficulty was encountered by any of the subjects while performing the tasks in a shirtsleeve environment.

(2) Subjects felt that the retaining clip, employed for small parts retention, was optimally located on the right side of the panel.

(3) There is very limited tactile sense wearing the gloves of the ILC pressure suit.

(4) Fingers get tired because of the glove configuration.

(5) Cannot rotate screwdriver around wrist axis in ILC suit.

(6) Optimum working height above floor is approximately 45 inches.

(7) Is difficult to reach component 2-D because it is located near the upper reach limit on the panel, namely 57 inches. This comment is associated with performance in the suit.

(8) Subject dropped retaining clip while performing in the suit.

(9) Subject dropped washer associated with task-a under both suit conditions.

(10) Trial aborted for task-a under pressurized condition.

(11) Evaluate handles of screwdrivers as they are too small in diameter.

(12) Evaluate nut-driver in performing task-c.

(13) All subjects reported a large amount of fatigue while performing work in the suit.

A complete set of data was collected for two subjects for all suit conditions in the 1-G environment and for the shirtsleeve condition in the 1/6-G simulator. It was decided to investigate anthropometric characteristics of the ILC and Gemini suits with the remaining time.

Following data collection, a statistical analysis was performed. Mean times for two trials of each task per condition per subject in 1-G and 1/6-G were obtained. Means were then established for each task per suit condition in each environment (Table 1). Total task times and means were established for each of the three conditions in 1-G and for the shirtsleeve condition in 1/6-G. The percentage of increase in times (or work decrement) required to perform the tasks was found for each condition using the shirtsleeve 1-G condition as the basis for comparison. By expressing the 1-G shirtsleeve data as 100 percent of the combined subjects' capability, we established a constant base of comparison, despite the variability in individual performance.

TABLE 1

MEAN TIMES PER TASK PER CONDITION

| CONDITIONS: | MINUTES | SECONDS |
|---|---|---|
| 1. No Suit - 1-G Environment | | |
| Task-a | 1 | 7 |
| Task-b | | 22 |
| Task-c | | 46 |
| Task-d | | 36 |
| Task-e | 1 | 2 |
| Task-f | 2 | 00 |
| Total | 5 | 53 |
| Mean Task Time Per Condition | | 59 |
| 2. ILC Suit Unpressurized - 1-G | | |
| Task-a | 2 | 13 |
| Task-b | | 48 |
| Task-c | 1 | 29 |
| Task-d | 1 | 1 |
| Task-e | 2 | 2 |
| Task-f | 2 | 33 |
| Total | 10 | 6 |
| Mean Task Time Per Condition | 1 | 41 |
| 3. ILC Suit Pressurized - 1-G | | |
| Task-a | 4 | 40 |
| Task-b | | 44 |
| Task-c | 1 | 10 |
| Task-d | 1 | 2 |
| Task-e | 4 | 2 |
| Task-f | 3 | 46 |
| Total | 15 | 24 |
| Mean Task Time Per Condition | 2 | 34 |
| 4. No Suit - 1/6-G Environment | | |
| Task-a | 1 | 44 |
| Task-b | | 23 |
| Task-c | | 49 |
| Task-d | | 28 |
| Task-e | 1 | 8 |
| Task-f | 1 | 52 |
| Total | 6 | 24 |
| Mean Task Time Per Condition | 1 | 4 |

Results indicate that it takes an operator 42 percent longer to perform a maintenance task in 1-G wearing an unpressurized ILC suit than in a shirtsleeve environment and 68 percent longer in a pressurized suit. It was also found that it took 8 percent longer for an operator to perform a task in 1/6-G over 1-G in a shirtsleeve environment. These decrements of 42 and 68 percent are in general agreement with those of other experimenters (refs. 2, 3).

The reason for the increase in time, or lower performance, while working in the suit is a combination of factors. Among the more significant are the results of fatigue, the glove-tool interface, and the general mobility limitations of the suit. The pressurized suit is more difficult to work in because of the stiffening factor of the suit, and especially the glove fingers. Consequently, more muscular effort is required to perform the task resulting in a reduction of the subject's dexterity.

While it was not possible to investigate the zero-G environment and only the shirtsleeve condition in 1/6-G, results indicated little difference between the shirtsleeve condition for 1-G and 1/6-G - a work decrement of 8 percent. The writer does not feel this data is valid in itself, since a shirtsleeve condition is unrealistic on the lunar surface. The condition was intended to be used for statistical analysis and comparative purposes.

Levels of task difficulty were established based upon trial times and the number of dropped items during the trials. All tasks were easy in the shirtsleeve condition for both 1-G and 1/6-G. Tasks performed in the suit ranged from easy to nearly impossible. Removal and replacement of the connector was easy under all conditions.

Task-a was the most difficult of the six test items. The subject was required to remove a 1/16-inch screw and lock washer with a screwdriver and retaining clip, place the retaining clip on the retaining wire located on the extreme right side of the panel, remove the terminal, and replace the terminal in reverse order. The task would have been impossible in the pressurized ILC suit if the retaining device had not been utilized. Small parts such as 1/16-inch screws and lock washers cannot be picked up and manipulated by the operator. Some means must be provided to assist the maintenance worker in manipulating and retaining small parts during such activity in the suit. Disassembly and assembly of the hydraulic components (task-f) was time consuming due to the six task elements involved, but was within the capability of the operators under all conditions. The other tasks were difficult in the suit, but feasible.

A separate study was conducted to define the "impossible" tasks in the ILC suit. Six nuts, bolts, and washers were arranged in three horizontal rows. The nuts and washers ranged in size (outside diameter) from 1/16 through 1/2-inch, and the bolts ranged in length from 1/16 through 1/2-inch. Thread diameter and head size increased proportionally with the length. Three subjects participated in the test. The study only concerned itself with the unpressurized and pressurized suit. The subjects picked up each component from a flat surface and placed it in a box to their right. The components were arranged in decreasing order of size from right to left. The test was similar in structure to the Purdue Pegboard Manual Dexterity Test. Components were picked-up using each hand independently, and then one item with each hand simultaneously. The test indicated that it was impossible to pick-up the washers in the suit. It was also extremely difficult to pick-up nuts

and bolts less than 1/4-inch in diameter or length. The washers were impossible to handle because of their round, smooth, thin shape. Tasks were also more difficult in the pressurized suit than in the unpressurized suit. It appears reasonable to conclude that a maintenance worker could not be expected to handle components similar to washers, and nuts and bolts of less than 1/4-inch in size, in the suit without auxiliary means.

Handtool evaluation.—A major problem area revealed during the performance study was the glove-tool interface. A problem is apparent when the subject employs a screwdriver with a standard size handle. It was not possible for the subjects to bend the fingers around the handle due to the metal bar in the glove and pressurization in the fingers. Following performance testing, handles on the two screwdrivers were increased from 3/8 and 5/8-inches to a 1-inch diameter by means of tape. The subjects repeated several tasks with and without the gloves pressurized. Empirical evaluation clearly pointed out that the increased handle size permitted a better grasp and gave the operator more control of the tool. Various handle diameters were studied, and a 1-inch diameter appears optimum. A T-bar screwdriver was investigated, and found most desirable. The T-bar extending through the index and second fingers provides maximum control of the tool. A ratchet T-bar handle is recommended for screwdrivers, socket wrenches, and other tools of this type.

Tools such as crescent wrenches will be more effective if a round handle of 1-inch diameter is provided.

Conclusions and recommendations.—Within the limitation imposed upon this study the following conclusions and recommendations seem warranted:

(1) The pressure suit significantly affects human performance required for efficient space maintenance.

(2) It takes approximately 42 percent longer to perform maintenance in an unpressurized suit than in a shirtsleeve condition, in a 1-G environment.

(3) It takes approximately 68 percent longer to perform maintenance in a pressurized suit (3.75 psi), than in a shirtsleeve condition, in a 1-G environment.

(4) It would take approximately 8 percent longer to perform maintenance in a shirtsleeve condition at 1/6-G than at 1-G if such a condition were feasible on the lunar surface.

(5) Conclusions No. 2 and 3 are in general agreement with those of other experimenters.

(6) The primary reasons for the lower performance level in the suit are the results of fatigue, the glove-tool interface, the stiffening factor of the suit when pressurized--especially the glove fingers, loss of the tactile sense with the glove, and the overall mobility limitations of the suit.

(7) It was not possible to obtain data with the pressure suit under simulated zero-G and 1/6-G environments due to suit availability.

(8) Some additional means must be provided for the maintenance operator wearing the pressure suit, for small parts handling and retention.

(9) Handles on standard tools, such as screwdrivers and socket wrenches, are too small in diameter to permit a firm grasp with the pressure suit gloves.

It is recommended that a ratchet handle be employed for tools of this type.

Anthropometric study.—The purpose of this study is to determine the maximum working envelope for the maintenance operator wearing a pressure suit of the Gemini or ILC type. Additionally, the study defined reach limitations imposed upon an operator wearing such garments.

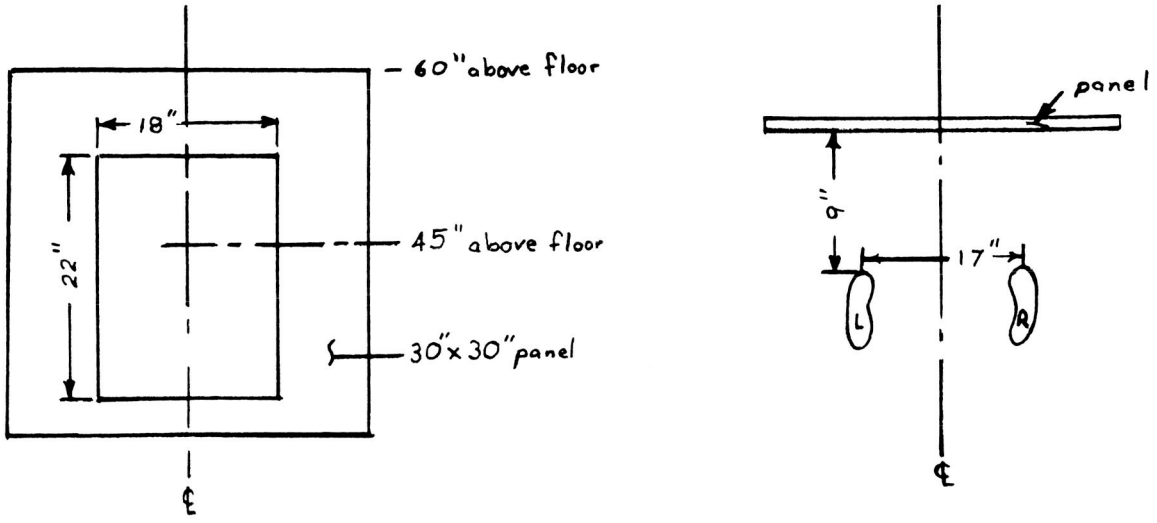
Discussion and results.—Two subjects participated in this study. The first subject, approximately a 70th percentile operator, donned the Gemini Pressure Suit and was pressurized to 3.75 psi. To obtain reference data for later testing the following measurements were recorded:

| | | |
|---|---|---|
| 1. | Eye height (standing operator) | 57 inches |
| 2. | Shoulder height (standing position) | 54 inches |
| 3. | Eye height (squatting or semi-kneeling position) | 48 inches |
| 4. | Shoulder height (squatting or semi-kneeling position) | 45 inches |

The subject was then asked to step onto the 1/6-G simulator platform ("bottomed out" for this study) located 14-inches above the standing surface. While this task was possible, the subject reported it to be fairly difficult due to limited suit mobility. The work panel (30-inches square) is located 30-inches above the simulator platform.

The subject was directed to align himself with the centerline of the panel, to assume a square stance, and to select a comfortable working distance. Given a screwdriver, the operator was told to establish right, left, upper and lower reach limits on the panel at which such a task as screwdriving could be performed. The subject was free to move his torso and other bodily members, but not his feet. The experimenter marked the limits on the panel as defined by the subject. The following data produced a reach envelope approximately 18 inches wide by 22 inches high as shown below.

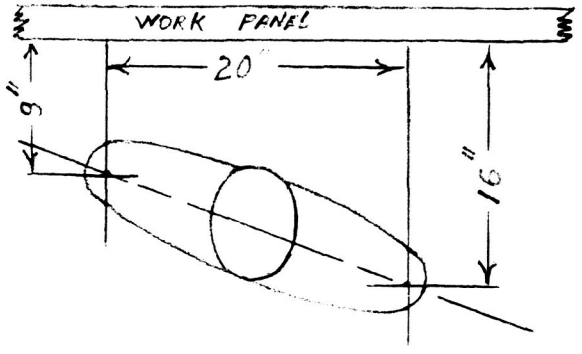
| | | |
|---|---|---|
| 1. | Distance to the right of panel centerline | 9 inches |
| 2. | Distance to the left of panel centerline | 9 inches |
| 3. | Upper limit from standing surface | 54 inches |
| 4. | Lower limit from standing surface | 32 inches |
| 5. | Optimum toe distance from panel plane | 9 inches |
| 6. | Optimum distance between feet (inside measurement) | 17 inches |
| 7. | Minimum distance attainable between feet (inside measurement) | 10 inches |



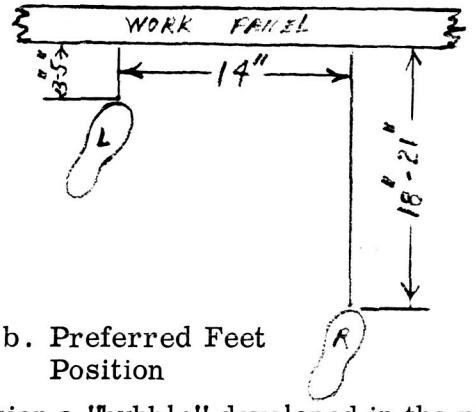

At this point in the testing program it was revealed that a square stance was not the most desirable or optimum working position for those tasks requiring the operator to apply forces perpendicular to the body. Screwdriving, drilling, and sawing tasks would fall into this category. The square stance does not permit the operator to assume a natural position where he can "lean into" his work. The subject assumed what he considered to be an optimum working position with respect to the work panel. He was then asked to remove a terminal strip from the panel. The evaluation favored a stance, as shown below, for the right-handed operator, which permitted him to "lean into" the work. The following measurements were recorded:

| | | |
|---|---|---|
| 1. | Optimum distance from panel plane to center of left shoulder | 9 inches |
| 2. | Optimum distance from panel plane to center of right shoulder | 16 inches |
| 3. | Maximum working distance from panel to center of left shoulder | 15 inches |
| 4. | Maximum working distance from panel to center of right shoulder | 18 inches |
| 5. | Optimum toe distance from panel plane (left foot) | 3-5 inches |
| 6. | Optimum toe distance from panel plane (right foot) | 18-21 inches |
| 7. | Optimum distance between feet | 14 inches |

Preferred Working Position




a. Preferred Shoulder Position




b. Preferred Feet Position

Prior to the next portion of the testing session a "bubble" developed in the right armpit of the Gemini suit. Further testing was accomplished at the reduced pressure of 2.0 psi.

To gain more of an appreciation for the mobility limitations imposed by the pressure suit the subject was directed to assume a square stance in front of a large, smooth surfaced wall. A reference mark was placed at 57-inches (eye level) above the standing surface for vertical body alignment. The subject was given a piece of white chalk and told to define the limits of his reach by drawing circles first with the right hand, then the left hand, and finally with both hands together. Measurements were recorded with the toes 9, 12, 15, and 18 inches from the wall under each of two conditions. The first condition required the subject to keep his feet and torso still, while the second condition permitted him to move the torso, but not the feet. Measurements were not taken at toe distance of less than 9 inches because preliminary testing indicated that the subject could not assume a natural standing position due to helmet interference with the wall. Results of this segment of testing are shown in Figures 8 and 9.

Figure 8 depicts the condition with the feet and torso still or rigid. The outside reach decreases significantly as the subject moves from 9-15 inches out from the wall. At 18-inches the subject had difficulty reaching the wall, and consequently no measurements were recorded at this distance. Below is a list of the more pertinent anthropometric data:

| | | |
|---|---|---|
| 1. | Maximum reach to right of vertical body axis (right hand) | 27.5 inches |
| 2. | Maximum reach to left of vertical body axis (left hand) | 31.0 inches |
| 3. | Maximum reach above standing surface (right hand) | 60.0 inches |
| 4. | Maximum reach above standing surface (left hand) | 61.0 inches |
| 5. | Minimum reach above standing surface (right hand) | 29.5 inches |

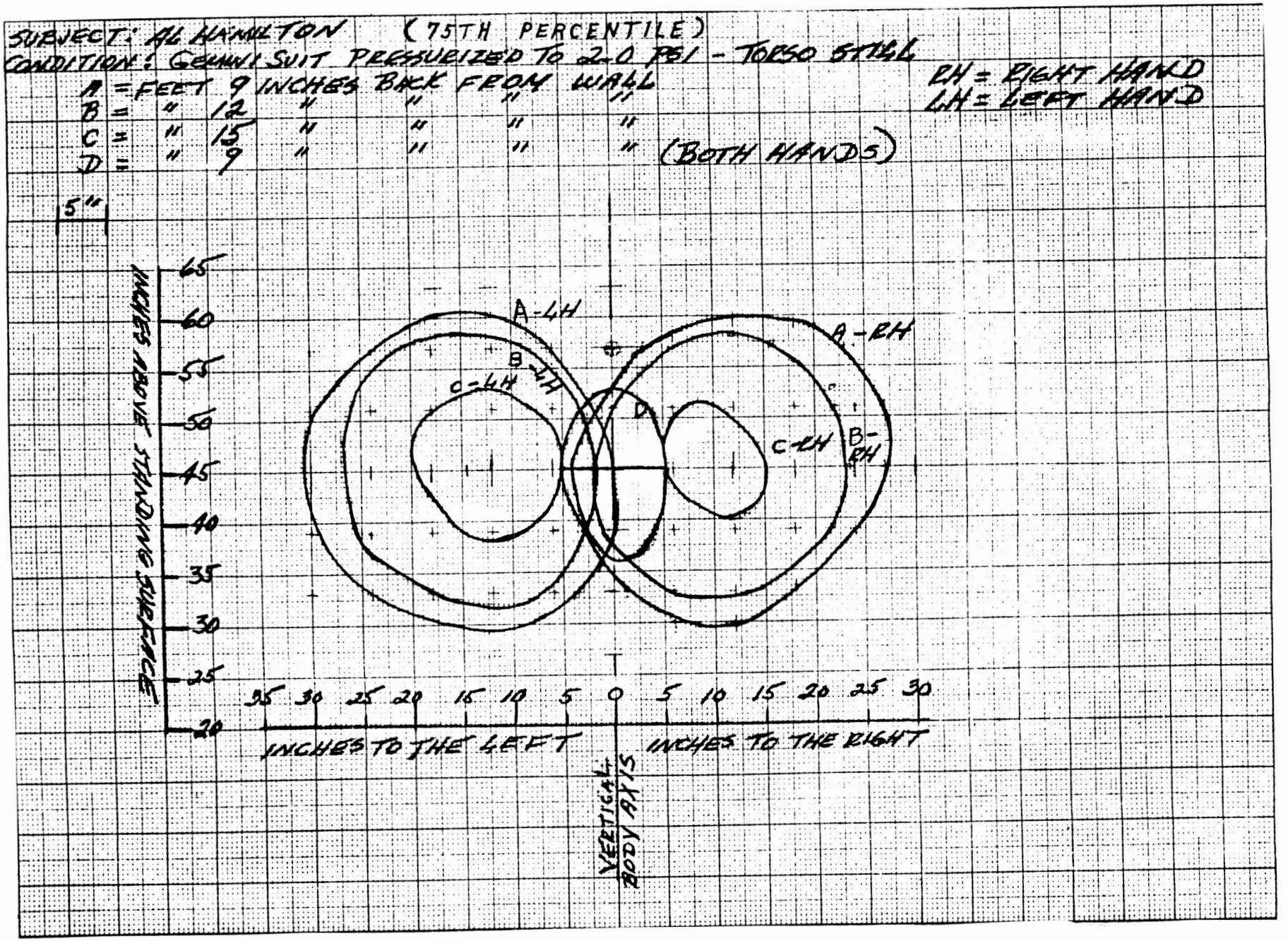



Figure 8. Pressure Suit Reach Envelope

# PRESSURE SUIT REACH ENVELOPE

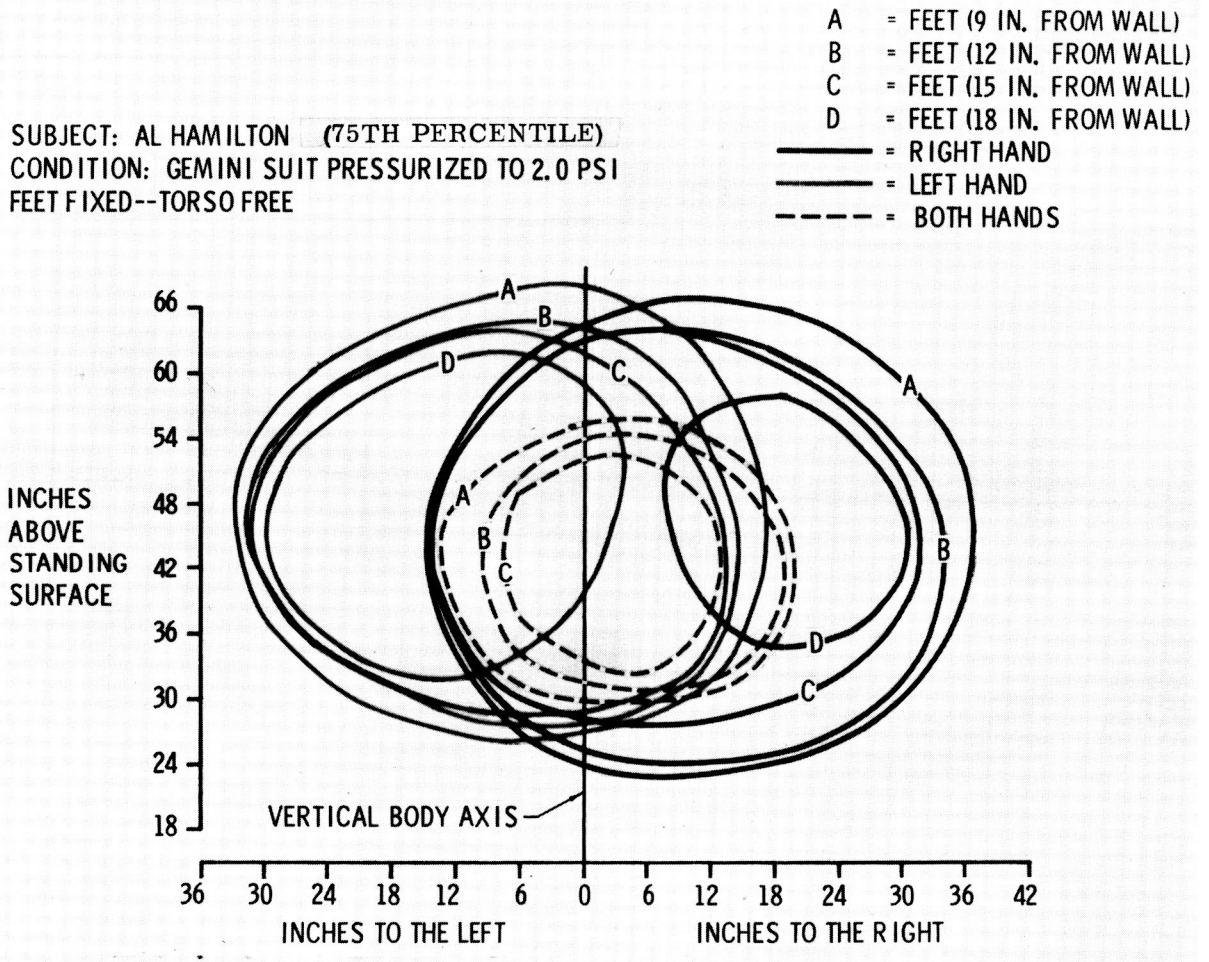



Figure 9. Pressure Suit Reach Envelope

| | | |
|---|---|---|
| 6. | Minimum reach above standing surface (left hand) | 29.0 inches |
| 7. | Maximum reach to right of vertical body axis (both hands) | 5.0 inches |
| 8. | Maximum reach to left of vertical body axis (both hands) | 5.0 inches |
| 9. | Maximum reach above standing surface (both hands) | 53.0 inches |
| 10. | Minimum reach above standing surface (both hands) | 36.0 inches |

The results indicate that the operator, standing 9-inches from the work surface, can span 58.5 inches horizontally and approximately 30-inches in the vertical plane when reaching with the left and right arms independently. In a shirtsleeve environment the subject could span 64-inches horizontally and 52-inches vertically. However, more significant is the fact that when the operator is required to employ both hands in close proximity to one another the same dimensions are reduced to 10 and 17-inches as compared to 45 and 46-inches in the shirtsleeve environment. Since the large majority of maintenance tasks required both hands it is these latter measurements which are of primary concern in establishing a working envelope. Such an envelope is located from 36-57 inches above the standing surface or from 4-17 inches below the operator's eyes. This will impose limitations on the operator utilizing larger handtools and power tools through an access opening of this size.

Results of the free torso condition are presented in Figure 9. Following are the more relevant findings:

| | | |
|---|---|---|
| 1. | Maximum reach to right of vertical body axis (right hand) | 37.0 inches |
| 2. | Maximum reach to left of vertical body axis (left hand) | 33.0 inches |
| 3. | Maximum reach above standing surface (right hand) | 67.0 inches |
| 4. | Maximum reach above standing surface (left hand) | 68.0 inches |
| 5. | Minimum reach above standing surface (right hand) | 23.0 inches |
| 6. | Minimum reach above standing surface (left hand) | 26.0 inches |
| 7. | Maximum reach to right of vertical body axis (both hands) | 19.0 inches |
| 8. | Maximum reach to left of vertical body axis (both hands) | 14.0 inches |
| 9. | Maximum reach above standing surface (both hands) | 56.0 inches |
| 10. | Minimum reach above standing surface (both hands) | 30.0 inches |

These results show that an operator standing 9-inches from the wall (work plane), with the torso free to move, can span 70-inches horizontally and approximately 45-inches in the vertical plane when reaching with the left and right arms independently.

The same measurements in a shirtsleeve environment were 86-inches horizontally and 74-inches vertically. The reach increase for the torso free condition over the torso still condition, with the left and right arms reaching independently, is not nearly as significant as it is with the hands together or in close proximity to each other. Here the limits were increased from 10 and 17-inches to 33 and 26-inches. In a shirtsleeve environment the operator could span 68-inches horizontally and 68-inches vertically employing both hands together. By constructing a rectangle within the reach envelope in Figure 11 a two-dimensional working envelope 24-inches wide and 18-inches high resulted. The envelope is located from 33 to 51-inches above the standing surface or from 6 to 24-inches below the operator's eyes.

To define the depth of the working envelope an adjustable access panel was mounted to the front of the work panel. The subject was asked to assume a square stance and reach through an access opening 24-inches wide by 18-inches high with both hands. A maximum reach of 6-9 inches was recorded at the corners and center of the working envelope. The subject was then given a screwdriver and directed to remove a terminal strip from the panel in the preferred working position. The working panel was located 9-inches behind the access panel. While able to perform the task the subject stated he was working at a maximum depth. Results of the first subject indicate a maximum working envelope of 24-inches wide by 18-inches high by 9-inches in depth. A maximum working envelope of this size is also somewhat optimum for an operator to perform two-handed tasks with larger handtools and power tools. It permits the operator greater arm and hand mobility in addition to better visibility into the work area. Due to suit availability and the absence of prototype power tools it was not possible to obtain minimum aperture sizes for various one and two-handed tasks.

The results also point out the need for an operator restraint system, to be used for extravehicular maintenance which will maximize torso mobility.

Apollo suit performance data was limited because of a malfunction of the shoulder joint. What data it was possible to obtain substantiates the Gemini suit data with the exception that maximum depth of reach appears to be greater up to 24-inches.

Conclusions and recommendations.—Within the limitations imposed upon the current study the following conclusions and recommendations seem warranted:

A. General

1. A stance which permits an operator, wearing a pressure suit, to stand at an angle to the work surface is preferred over a square stance for tasks which require forces to be applied perpendicular to the body (Figure 10).

2. An operator's reach capability decreases as he moves away from the work surface. The optimum toe distance from the work surface for maximum reaching, by an operator wearing a pressure suit, is 9-inches.

3. There is an overall reach decrement for all conditions tested in the Gemini and ILC suits compared to the shirtsleeve environment. The most significant decrement is associated with reaching with both hands together or in close proximity to one another.

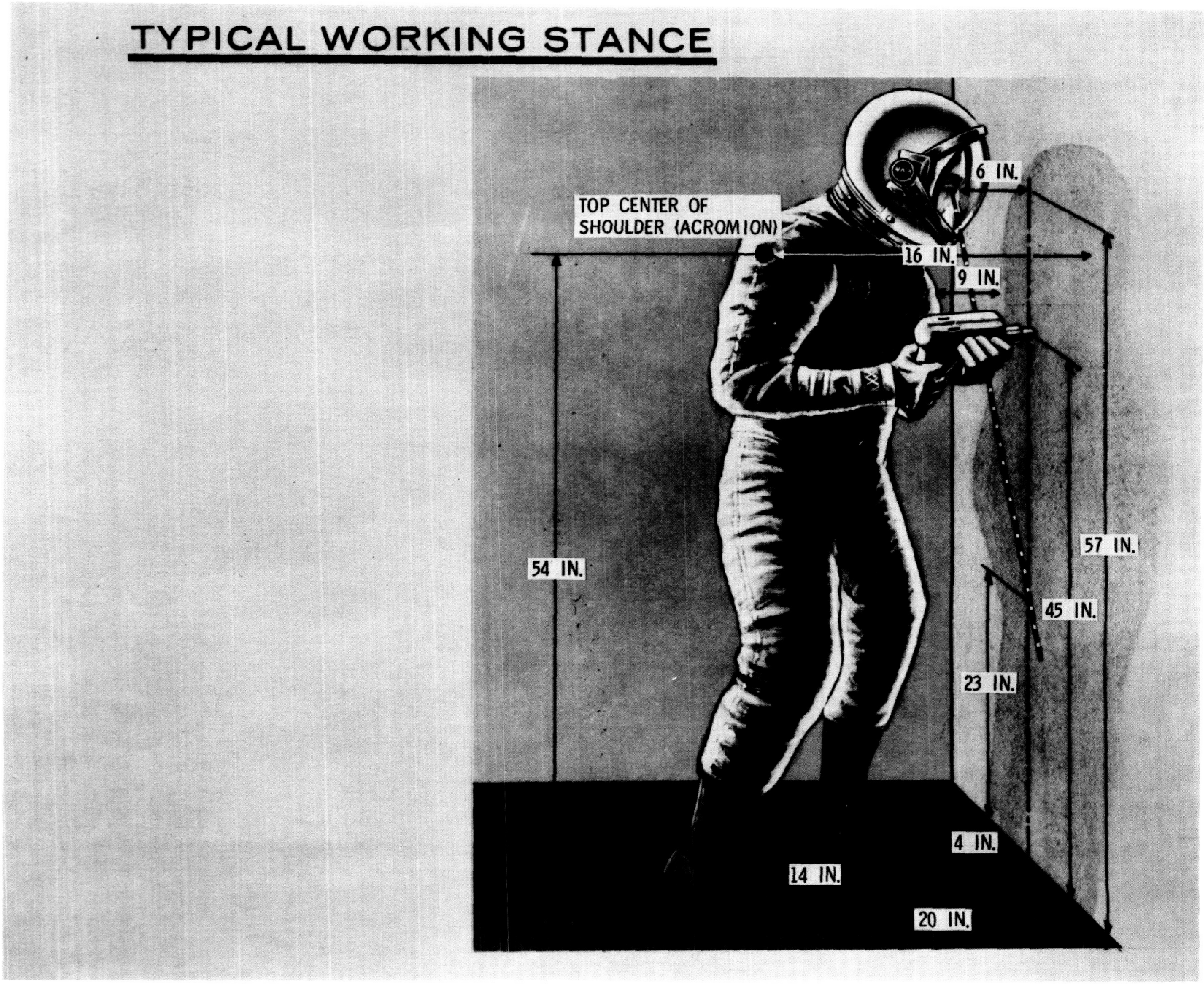



Figure 10. Typical Working Stance

4. An operator's overall reach capability is greatly increased when the torso is free to move compared to when it is kept rigid. It is recommended that a restraint system for extravehicular maintenance be considered which maximizes torso mobility.

5. An access opening 10-inches wide and 17-inches high is adequate for visual inspection and small tool (screwdriver, etc.) manipulation, but is inadequate for arm and hand manipulation of larger handtools (wrenches, etc.) and power tools.

6. A working envelope 24-inches wide, 18-inches high, and 9-inches deep is maximum for two-handed operation. The envelope is located from 33-51 inches above the standing surface or from 6 to 24-inches below the operator's eyes.

7. The working envelope described above is somewhat optimum in terms of operator visibility and arm and hand mobility required for two-handed tasks utilizing larger handtools and power tools.

8. In view of the development of improved space suits (4, p. 167) the data obtained in this study should be applied with caution.

Operator force study.—The purpose of this study is to investigate operator forces required to utilize power tools on the lunar surface, and to study the effects of a 1/6th G environment on an operator's ability to exert maximum forces employing standard handtools.

Discussion and results.—To perform extravehicular maintenance in zero-G an operator must be restrained so he can exert forces to do tasks. Since the restraint development deals with the zero-G environment, the present investigation will look at the 1/6th G conditions.

In general, an unrestrained operator working on the lunar surface should be able to exert a maximum force of 1/6th his body weight on earth before losing contact with the surface. It will not be feasible in many cases for the worker to exert such forces due to slipping on the lunar surface, pressure suit mobility and dexterity limitations, anthropometric restrictions, environmental limitations, and individual human capabilities and limitations. Where leverage cannot be obtained and large forces are required the operator will need to be restrained to the lunar surface or work area in some manner. An operator will be able to exert large forces where leverage can be gained between the surface and the work.

Three average size subjects participated in the study. The apparatus employed was the fully functional 1/6-G simulator. Pressure suits were not available for this study. The weight of the ensemble may enable the operator to exert larger forces. On the other hand, the force gained may be offset by the mobility limitations of the suit. Operator forces associated with screwdriving, torquing, drilling, sawing, and pulling were determined.

A 30 inch-pound torqueometer was used to measure the maximum force that could be applied to a 5/16-inch slotted screw with a 1/4-inch wide screwdriver blade. Two nuts were secured on the screw and placed inside of a socket which attached to the

torqueometer. The torqueometer was secured to the drilling and sawing plate in the middle of the work panel (see Figure 11). With the simulator balanced and in operation, the maximum force that could be exerted on the screw by two operators was 17 inch-pounds. This force was found adequate to tighten the screw.

Three subjects torqued a 1-1/8-inch hex nut located on the nut plate of the work panel (see Figure 12). An 18-inch torque wrench was employed. With the subject facing the work panel, maximum sustained forces were recorded at points located at 0, 90, 180 and 270 degrees around the nut. It must be noted that torquing is one of the primary functions of the zero reaction power space tool. This test was selected as a general measure of operator force capability.

Measurements were recorded for the left hand, right hand, and both hands together. After forces were measured at points around the nut in a clockwise direction, forces were measured in the counterclockwise direction. The subjects were required to keep their feet on the floor of the simulator at all times. Results indicate that the range of maximum torque was from 20-35 foot-pounds or from 13.3-23.3 pounds. Measurements taken at 270 degrees in clockwise direction and at 90 degrees in the counterclockwise direction permitted the subjects to "bottom out" the platform and exert large forces equivalent to those possible on earth. These forces, which exceeded the 140 foot-pound wrench capability, were excluded from the other data. There was a tendency for the subjects to exert slightly more force when both hands were utilized together. There was no apparent force difference between the clockwise and counterclockwise conditions.

The next part of the torquing task placed the operator 90 degrees to the panel. Measurements were recorded in line with the sagittal plane of the body, utilizing the same nut and torque wrench. The wrench was first pulled from the vertical position down to the horizontal. Next, it was pushed from the horizontal position downward to the vertical position. Left hand, right hand, and both hand conditions were employed. Results indicate a range of maximum torquing forces along the vertical axis of the body from 20-35 foot-pounds or 13.3-23.3 pounds.

The next task was aimed at determining maximum forces an operator could exert from a fixed point without slipping or losing his footing. Forces were measured to the right of the reference point (large cylinder of hydraulic assembly) with the right hand, to the left with the left hand, downward with both hands, and backward with both hands. The backward force away from the panel also represents a pushing reaction force. No measurements were made in the upward direction since an operator could "bottom out" the simulator. A strain gauge, calibrated in pounds, was hooked around the hydraulic cylinder on the panel -- located approximately 36 inches above the standing surface. The subject held onto the ring at the opposite end of the gauge, and forces were recorded for the various conditions. Results indicate a maximum pulling force to the right with the right hand of 12-pounds, to the left with the left hand of 10-pounds, away from the panel with both hands of 30-pounds, and down from the panel of 35 pounds. Lateral forces to the left and right of the body are relatively small. It is likely that the application would not be encountered often under the conditions tested. Instead, an operator would orient himself off-center to the work.

The power drilling study utilized a specially constructed jig in conjunction with the 30 inch-pound torqueometer. The assembly was attached to the center of the work panel (see Figure 13). A 1/8-inch piece of aluminum was inserted in the jig.

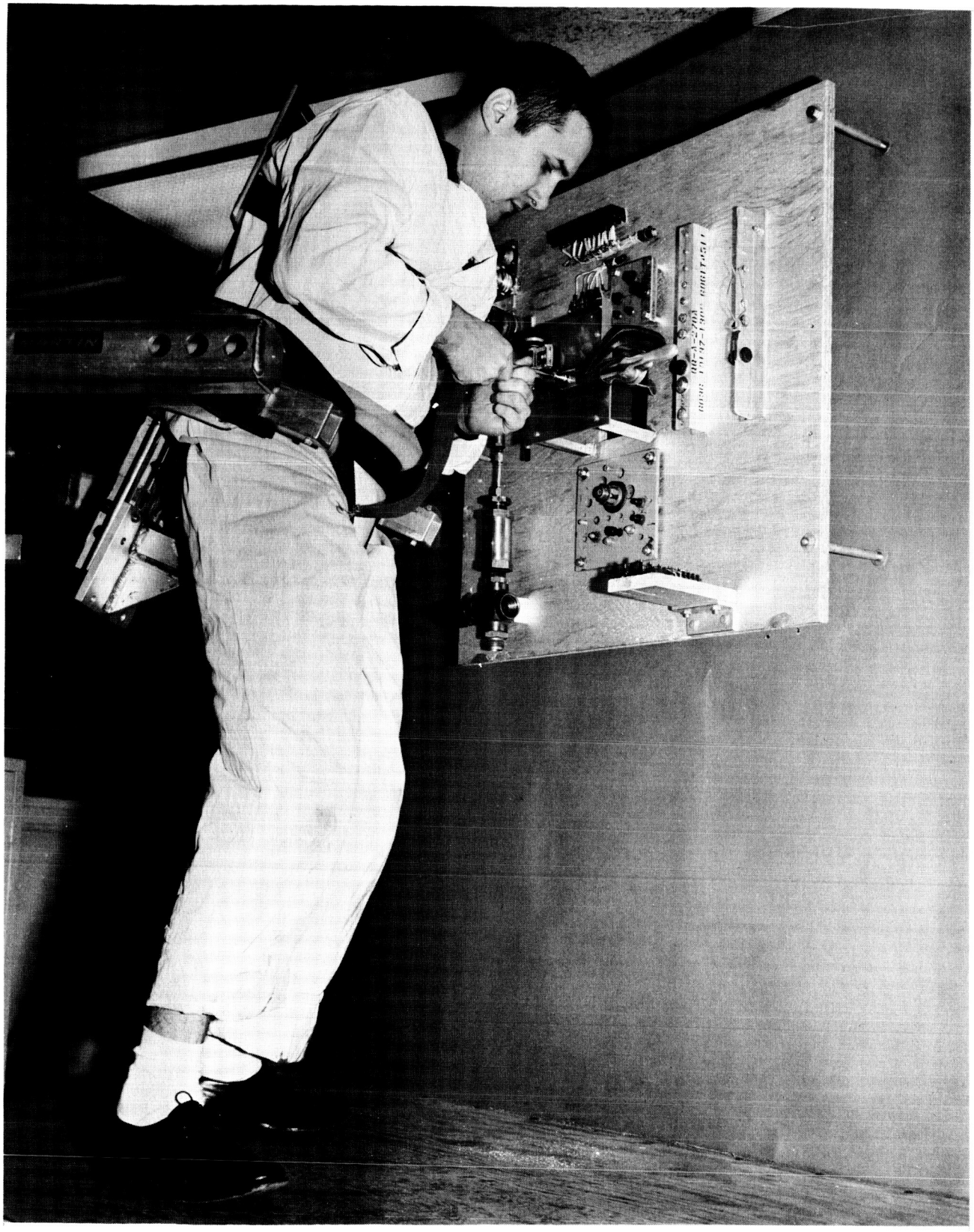


Figure 11. Torque Measurement in 1/6-G Condition




Figure 12. Torquing Nut in 1/6-G Condition

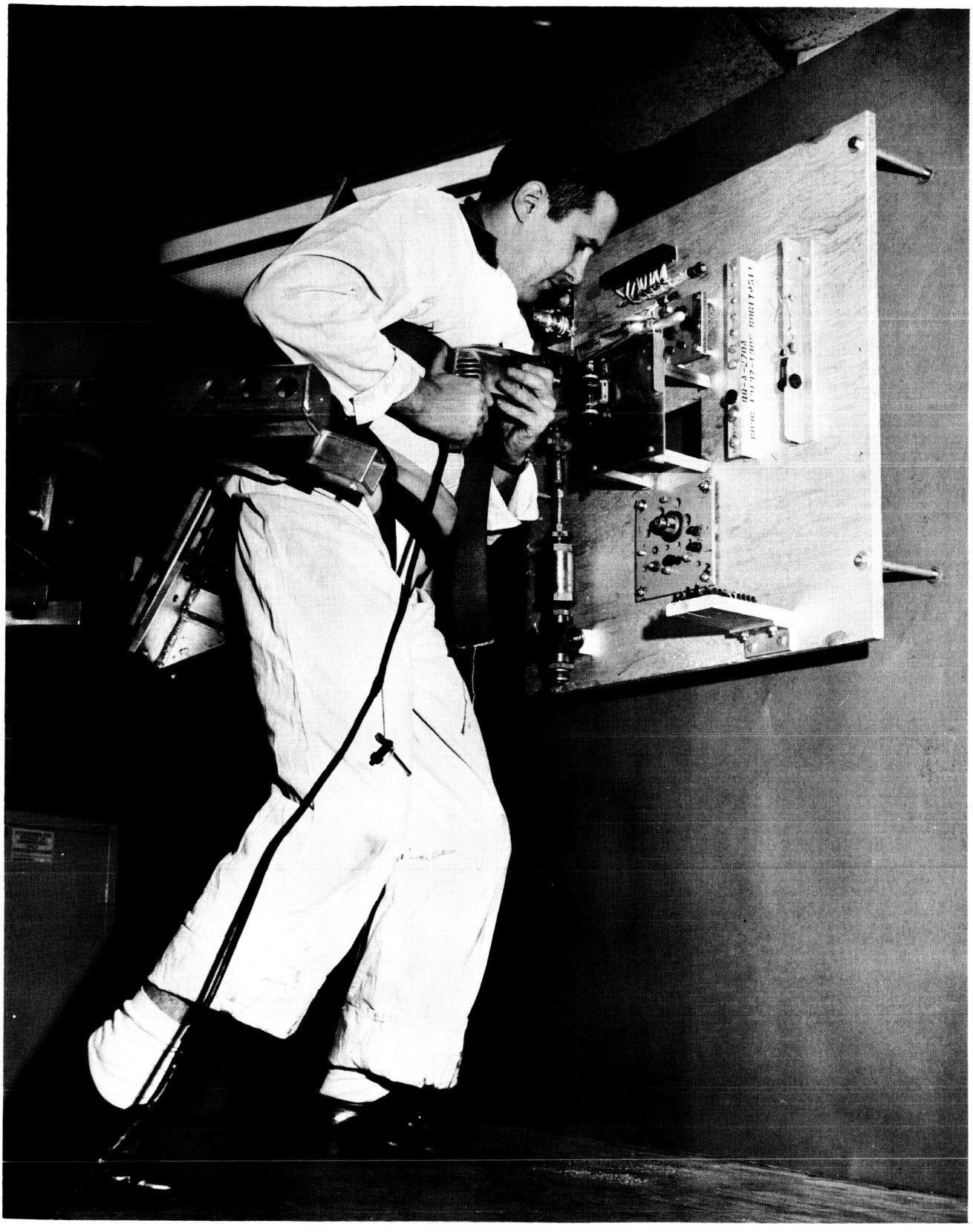


Figure 13. Drilling in 1/6-G Condition

A drill mark was centerpunched one inch away from the centerline of the torqueometer so force readouts were in pounds and not inch-pounds. The subjects drilled a 1/4-inch hole through the work sample. Results indicate that 13-16 pounds of force are required. The subjects had more of a tendency to lose their footing over the 15-20 second task duration than in previous tasks. The sharpness of the drill bit was found to be significant in performing a drilling task. A sharp drill reduced the operator force and also the task time. The subjects reported that they had to "lean into the work" more in a 1/6th G environment than in 1 G. Note in figure 15 that the subject's left foot is almost raised from the floor indicating that maximum force is applied by the right foot which is further from the work surface. The subjects encountered minor trouble in maintaining the drill level, probably because of the changed body position. A certain amount of learning will be required in drilling on the lunar surface.

While the drill used in this study is not space oriented or qualified, it appears reasonable to conclude that power drilling on the lunar surface is one of the tasks that an operator can be expected to perform. It is recommended that different types of drill bits be studied to reduce the operator force as much as possible. A force requirement of five pounds or less would be considered optimum. It is recommended that a study similar to the current one be conducted with a prototype power space tool and drilling attachment to better define the operator force requirements.

The final portion of this study is concerned with forces required for sawing. Essentially, two forces are employed when using this tool. A combined pushing force is required to hold the saw against the work surface and along a given course. The force was studied employing a work sample and strain gauge as seen in Figure 14. Results indicated that the subjects exerted a force of approximately 15 pounds while sawing through a 1/8-inch sheet of aluminum with a commercial Black and Decker sabre saw. While the task was possible, an operator force in the neighborhood of 5 pounds would be desirable. The subjects had a tendency to lose their footing when sawing a circular pattern. A sabre saw is preferred to a reciprocating type for its ability to handle larger thicknesses of material in less time. A maintenance operator will need to exercise the same safety precautions in space, and on the lunar and martian surfaces, as he does on earth when working with tools. Optimum saw blades should be selected for various materials to be cut and included in the tool kit.

Conclusions and recommendations.—Within the limitations of the present study, the following conclusions and recommendations seem warranted:

1. An unrestrained operator working on the lunar surface can be expected to exert large forces when leverage can be obtained between the surface and the work. The work must be associated with a stable mass capable of reacting such leverage forces.

2. A maximum force of 17 inch-pounds can be exerted on a screw by an average operator with a manual screwdriver. The force appears adequate to tighten or loosen components of this type.

3. The maximum torque range an average operator can exert around a point in front of him is 20-35 foot-pounds or 13.3-23.3 pounds linear force. The force range represents approximately 50-75 percent of the subjects 1/6th G weight.

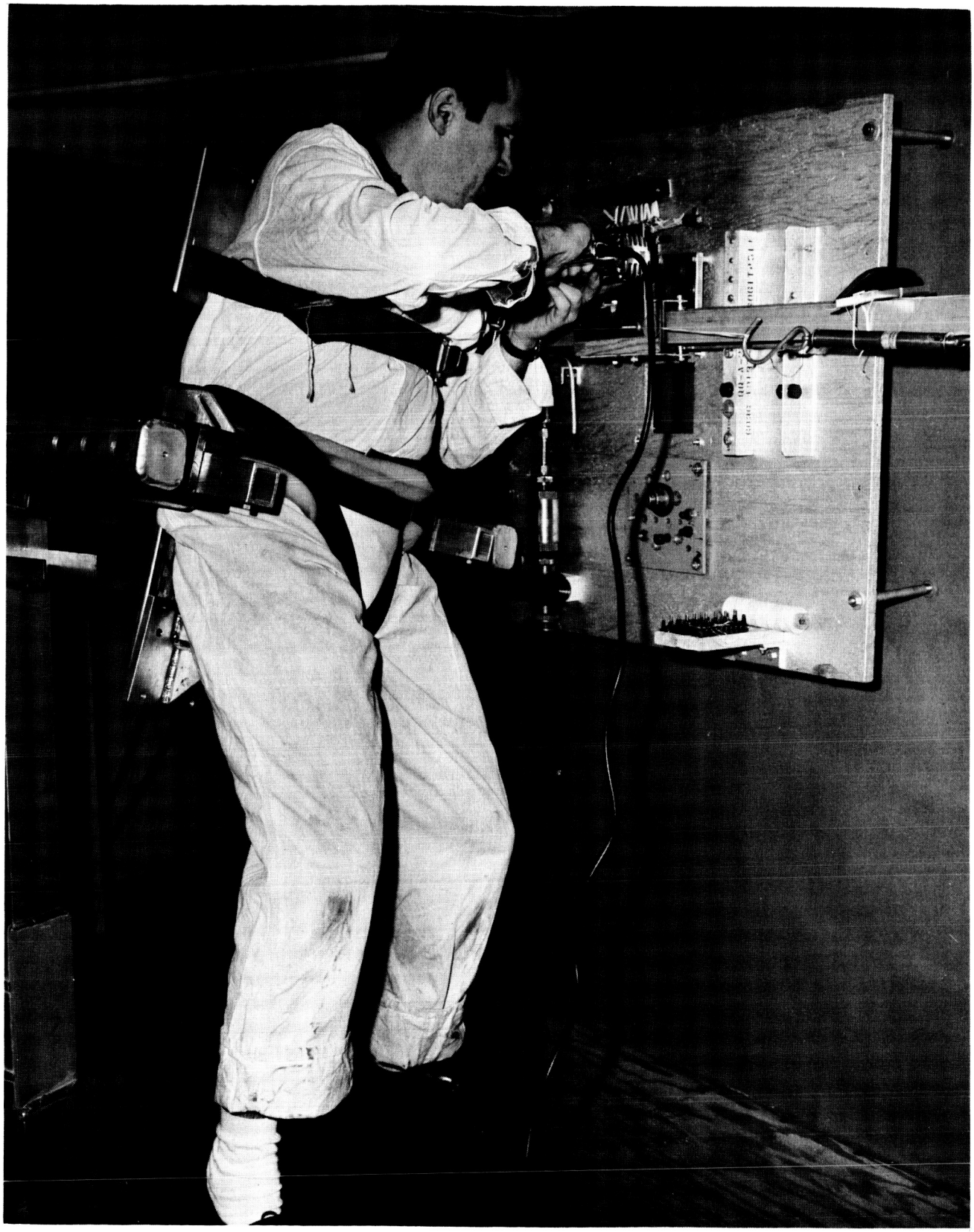


Figure 14. Sawing in 1/6-G Condition

4. The maximum torque range for pulling and pushing in line with the vertical axis of the body is 20-35 foot-pounds or 13.3-23.3 pounds linear force.

5. Thirteen to sixteen pounds of force was required to drill a 1/4-inch hole through a 1/8-inch thick piece of aluminum with a commercial electric drill.

6. An operator force of 15-pounds was required to saw through a 1/8-inch thick piece of aluminum with a Black and Decker commercial sabre saw.

7. Some difficulty was encountered with drilling and sawing tasks due to operator forces and tool manipulation. It is recommended that a study be made to determine the most effective drill bits and saw blades for various types of materials to be drilled and sawed with the power space tool. The study should be aimed at reducing the force required to perform sawing and drilling operations.

8. An operator force of 5 pounds could be considered optimum for drilling and sawing operations with the power space tool.

Illumination threshold study.—The purpose of this study is to establish optimum intensity levels for an auxiliary illumination system, to be employed by an operator performing extravehicular maintenance, under the following conditions:

(1) When the sun is behind an operator and reflecting off the surface of a space vehicle (bright condition)

(2) When an operator and space vehicle are in the shadow of the earth and not illuminated by reflection off of the moon (low illumination condition).

Discussion and results.—Whether or not the maintenance worker will require artificial lighting to illuminate his work area depends not only on the illuminance of natural light sources such as the sun, but also the shadows cast across the work area by himself or other objects. Because there is no atmosphere to scatter and diffuse incident light, shadows will be very prominent and of high contrast. If it is necessary or desirable for the worker to perform in the shadow areas and through access openings into maintenance compartments, it will be necessary to use artificial lights or have reflectors at appropriate locations to adequately illuminate his work. It has been estimated by Baker and Steedman (Ref. 5) that, while in the shadow of the earth, a satellite lighted by the moon's reflection would have a luminance of 0.01 foot lambert. The luminance of the sun, as seen from the earth is 440,000 lamberts at earth's surface, and 700,000 lamberts outside the atmosphere (Ref. 6). A space maintenance worker would be blinded if he directly observed a glare source such as the sun. Thus, some protective means must be incorporated into the visor of a space helmet if the operator is to locate targets under such conditions.

The upper limit of visual tolerance for the human operator is 16,000 millilamberts (Ref. 7). This is equivalent to 14,870 foot lamberts. New snow on a clear day or the upper surface of clouds at noon have a brightness of approximately 9,000 millilamberts (Ref. 8). This value is equivalent to 8,364 foot lamberts, since one foot lambert equals 1.076 millilamberts. The present study investigates a "worse condition" than just described, as 10,000 to 12,000 foot lamberts will be used to illuminate the white fore panel.

Four subjects, three with 20/20 vision and the other with corrected 20/20 vision, participated in this study. The experimental configuration for the bright illumination study involves basically a two panel arrangement with two independent light sources. The white fore panel is intended to simulate a highly reflective surface like the side of a space vehicle illuminated directly by the sun. It was not possible at this time to obtain enough high intensity lamps to evenly flood the visual field at a normal viewing distance. Instead, the viewing distance was reduced so the visual field could be reduced. By so doing, it was possible to flood the required area to the needed intensity levels with a single 2,000 watt directed, incandescent spotlight. The spotlight was located at a 45 degree angle to the right of the viewing aperture, and at a distance of 37 inches. The center of the 10-inch lens on the light was located at the same height as the center of the aperture from the standing surface, 55 inches.

Two apertures were utilized during testing. The smaller 3 by 3 inch opening was employed for obtaining data with the right eye alone (monocular vision). The eye was located 5 inches in front of the aperture which is equivalent to looking at a 12-inch panel at 20 inches or a 30-inch panel at 50 inches.

A chin rest was secured to a metal stand by means of a C-clamp to provide support and alignment for the subject. As the subject looked through the aperture he saw the work panel used in the other studies (see Fig. 15). The aft panel, which is 30 inches square, was placed on a table in an adjoining dark room lined with black horsehair. The lower edge of the panel was 32 inches above the standing surface, and the distance at the center of the aft panel was 53 inches behind fore panel. A secondary reason for placing the aft panel at this distance was to eliminate illumination spill from the spotlight onto the panel. A 100-watt light, with variac control, was located at an angle to the panel. The distance from the light to the left side of the aft panel was 26 inches, and to the right side 22 inches (see Fig. 15). The aft panel was at a slight angle to the fore panel to reduce highlights resulting from high variac settings. The distance from the center of the aperture at the fore panel to the left side of the aft panel was 51 inches, and to the right side 60 inches.

The subjects tasks basically involved visual identification of a number of representative maintenance tasks under different intensity levels. Each of four subjects were instructed to identify the following tasks (see Fig. 5):

(1) To which terminal, if any, is the connector secured in the upper left-hand corner of the panel?

(2) On which sleeve does lettering appear on the upper terminal strip?

(3) Which of the terminals is not secured to the lower terminal strip?

(4) Which of the Allen screws is missing from the Allen panel?

(5) What is the slot orientation of the first five screws (top to bottom) facing you to the right of the Allen screw panel?

(6) In which hole is the black Allen screw located on the nut plate at the bottom of the panel?

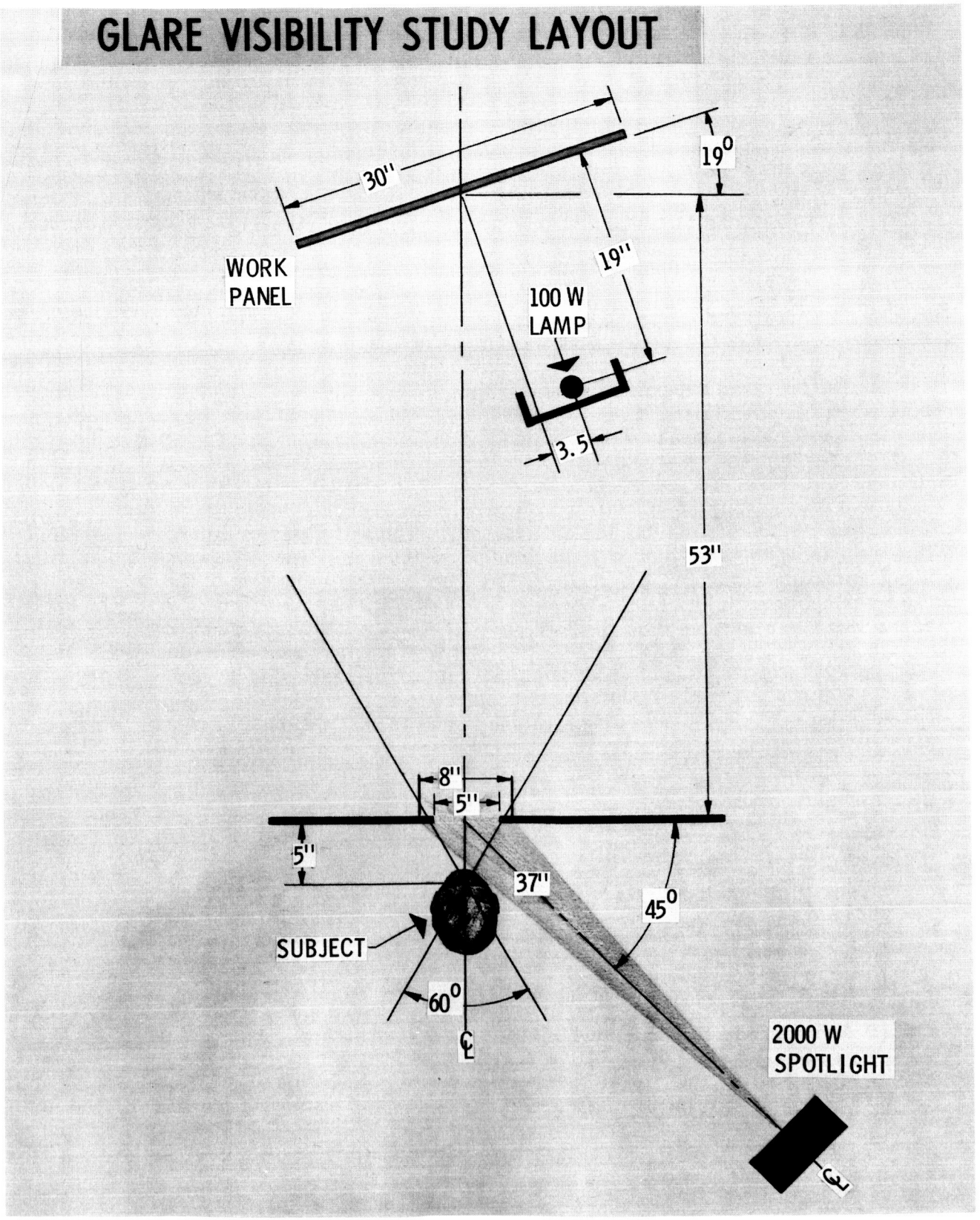



Figure 15. Glare Visibility Study Layout

Prior to testing minimum intensity levels were established to determine variac settings for testing. The variac settings used for the first test condition were 40, 60, 85, 110 and 135 volts. Brightness (luminance) measurements were made with a photometer of each component to be identified on the rear panel. These measurements, in foot lamberts, were recorded at each variac setting. Additionally, foot lambert readings were recorded at each of the four sides of the aperture in the front panel. These latter measurements ranged from 10,000 to 12,000 foot lamberts as previously mentioned.

The current experimental design has inherent limitations. For example, it is unrealistic that an operator will be required to recognize and identify components at a distance of 53 inches with monocular vision. In light of such limitations, a "worst condition" is being studied.

The experimenter, located in the adjoining room, randomly oriented the panel components used for the test and noted their position. The variac was then set to zero and the panel area darkened. The subject was told to rest his chin on the rest and align the right eye with the center of the aperture in the fore panel. The left eye was covered with a mask. The spotlight was turned on, and the subject received a one-minute adaptation period. The experimenter then raised the variac control to the first of five settings. The subject was asked to identify as many of the tasks as possible. The variac setting was increased in increments until all tasks were correctly identified. Settings were recorded for correct identification of each task. One of the ground rules established was not to guess, and to make only positive identifications. Actually, the subjects were asked to make identifications at an intensity level where they felt they could perform a maintenance task such as inserting a screwdriver in the slot of a screw, removing a connector and placing it on another terminal, etc.

Identification of all the tasks constituted one trial. Three trials were given each subject. Between trials the subject closed his eyes while the experimenter rearranged the panel and lowered the variac setting to zero. Following the one-minute adaptation period, the subject proceeded with the next trial. It should be noted that three subjects participated in the first part of the study. As an added source of information the subjects were asked to rank each task from the easiest to the most difficult following testing.

The second part of the study was the same as the first except that an additional subject became available, and binocular vision was tested employing a 3-inch high by 5-inch wide aperture at the fore panel. A larger visual angle and field resulted. Due to spill onto the rear panel from the spotlight, it was decided to maintain the panel distance and eliminate the spill.

The last portion of the study utilized the same four subjects and was concerned with establishing low illumination intensity levels. Once again, the worst condition (total darkness) was tested. An auxiliary illumination system will be required when performing extravehicular maintenance through access openings and internal compartments that must be reached from the outside. An alternative would provide for built-in illumination sources, which to this writer would not be practical.

The test procedure employed was basically the same as for the previous tests. The spotlight was not a part of this test. Minimum brightness thresholds were

established for panel recognition, from which variac settings of 20, 25, 30 and 35 volts resulted. Foot lambert measurements from the front panel were made and recorded off of each test item at the variac settings. The test utilized binocular vision and the same aperture size and panel distances as for the last test.

Results of the study appear in Table 2. Mean variac readings for three trials per task, under each condition, appear in the columns to the right of the task numbers. Subject means were totaled, and a variac mean for each task was established. The task means were then converted to foot lambert brightness levels and appear in the last column on the right of the page. The luminance values for each task under each condition were plotted on a graph (see Figure 16). The task numbers along the abscissa appear in rank order of increasing difficulty from left to right. The four subjects ranked the tasks in the same order.

As can be seen from Figure 16 the most unrealistic condition (monocular vision under bright light) required significantly more illumination for visual identification than the other conditions.

As to be expected, the larger components like the connector, the Allen screw in the Allen panel, and the Allen screw in the lower nut plate required lower brightness levels for identification than did the smaller components. Screw slot identification was the most difficult task. This is due not only to the size of the small slot, but also low contrast with the surrounding surface. It appears reasonable to conclude that the brightness levels established for the representative maintenance tasks depict unfavorable conditions. In reality, such tasks would be performed at closer distances where less illumination would be required due to larger retinal images. Panels and components located nearer the access opening would receive illumination from the outside ambient source, in addition to reflected light off of an operator's spacesuit. Minimum or non-reflective outer surfaces of space vehicles would significantly reduce the overall problem.

An auxiliary illumination system with a brightness of from 18-20 foot lamberts (apparent foot-candles) would illuminate 73-80 percent of the tasks investigated in this study under the three conditions. Assuming a reflectance factor of 50 percent, 36-40 foot-candles of illumination intensity appears optimum for an auxiliary system. Excluding the monocular test condition, 40 foot-candles of light would illuminate all of the components for operator identification and maintenance. The present study clearly indicates that the bright condition establishes the requirements for illumination levels, and will be more than adequate for performing maintenance under low illumination.

Conclusions and recommendations.—Within the scope and limitations of the current investigation the following conclusions and recommendations appear warranted:

1. An auxiliary illumination system will be required while performing extravehicular maintenance through access openings, and in internal compartments accessible from the outside of space vehicles.

2. Auxiliary illumination will be needed to fill in shadows cast upon a work area when the sun is behind an operator and reflecting off the surface of a space vehicle.

## Table 2. Illumination Study Results

MEAN VARIAC READINGS FOR 3 TRIALS PER SUBJECT

CONDITION NO. 1 - BRIGHT ILLUMINATION - RIGHT EYE - 3" x 3" APERATURE

N=3

| Task No. | Subject #1 HAMILTON | Subject #2 HOLMES | Subject #3 THILL | TOTAL | MEAN/TASK | FOOT LAMBERTS |
|---|---|---|---|---|---|---|
| No. 1 | 85.00 | 85.00 | 101.66 | 271.66 | 90.55 | 7.55 |
| No. 2 | 110.00 | 118.33 | 126.66 | 354.99 | 118.33 | 24.33 |
| No. 3 | 93.33 | 93.33 | 118.33 | 304.99 | 101.66 | 25.66 |
| No. 4 | 85.00 | 68.33 | 85.00 | 238.33 | 79.44 | 16.54 |
| No. 5 a. | 101.66 | 101.66 | 110.00 | 313.32 | 104.44 | 33.65 |
| b. | 110.00 | 110.00 | 110.00 | 330.00 | 110.00 | 34.00 |
| c. | 101.66 | 118.33 | 93.33 | 313.32 | 104.44 | 24.22 |
| d. | 101.66 | 110.00 | 110.00 | 321.66 | 107.22 | 25.22 |
| e. | 93.33 | 110.00 | 101.66 | 304.99 | 101.66 | 18.50 |
| No. 6 | 135.00 | 135.00 | 135.00 | 405.00 | 135.00 | 12.00 |

TOTAL - 1052.74/105.274 = MEAN

CONDITION NO. 2 - BRIGHT ILLUMINATION - BOTH EYES - 3" x 5" APER.

| N=4 Task No. | Subject #1 HAMILTON | Subject #2 HOLMES | Subject #3 THILL | Subject #4 COSTICK | TOTALS PER TASK | MEAN/TASK | FT. LAMBTS |
|---|---|---|---|---|---|---|---|
| No. 1 | 13.33 | 68.33 | 40.00 | | 121.66 | 30.42 | .70 |
| No. 2 | 85.00 | 110.00 | 10.00 | 85.00 | 390.00 | 97.50 | 13.75 |
| No. 3 | 60.00 | 68.33 | 60.00 | 76.67 | 265.00 | 66.25 | 5.50 |
| No. 4 | 60.00 | 60.00 | 60.00 | 60.00 | 240.00 | 60.00 | 4.50 |
| No. 5 a. | 85.00 | 93.33 | 85.00 | 93.30 | 356.66 | 89.17 | 19.00 |
| b. | 85.00 | 85.00 | 76.67 | 85.00 | 331.67 | 82.92 | 12.17 |
| c. | 76.67 | 85.00 | 76.67 | 85.00 | 323.34 | 80.84 | 9.62 |
| d. | 85.00 | 85.00 | 85.00 | 93.33 | 348.33 | 87.08 | 12.33 |
| e. | 93.33 | 85.00 | 85.00 | 85.00 | 348.33 | 87.08 | 10.62 |
| No. 6 | 126.67 | 110.00 | 118.33 | 126.67 | 481.67 | 120.42 | 9.08 |

TOTAL - 801.68/MEAN = 80.12

CONDITION NO. 3 - LOW ILLUMINATION - BOTH EYES - 3" x 5" APERATURE

| N=4 Task No. | Subject #1 HAMILTON | Subject #2 HOLMES | Subject #3 THILL | Subject #4 COSTICK | TOTAL | MEAN/TASK | FT. LAMBTS |
|---|---|---|---|---|---|---|---|
| No. 1 | 20.00 | 20.00 | 20.00 | 20.00 | 80.00 | 20.00 | .50 |
| No. 2 | 30.00 | 30.00 | 28.33 | 30.00 | 118.33 | 29.58 | 1.90 |
| No. 3 | 25.00 | 21.67 | 21.67 | 23.33 | 91.67 | 22.92 | 2.79 |
| No. 4 | 20.00 | 20.00 | 20.00 | 20.00 | 80.00 | 20.00 | 3.50 |
| No. 5 a. | 28.33 | 33.33 | 28.33 | 33.33 | 123.32 | 30.83 | 4.50 |
| b. | 28.33 | 30.00 | 25.00 | 31.67 | 115.00 | 28.75 | 3.50 |
| c. | 26.67 | 28.33 | 28.33 | 28.33 | 111.56 | 27.92 | 5.25 |
| d. | 31.67 | 28.33 | 25.00 | 30.00 | 115.00 | 28.75 | 3.50 |
| e. | 30.00 | 25.00 | 26.67 | 26.67 | 108.34 | 27.09 | 2.00 |
| No. 6 | 28.33 | 25.00 | 25.00 | 30.00 | 108.33 | 28.08 | .85 |

TOTAL - 262.92/MEAN = 26.29

# FIGURE ILLUMINATION STUDY

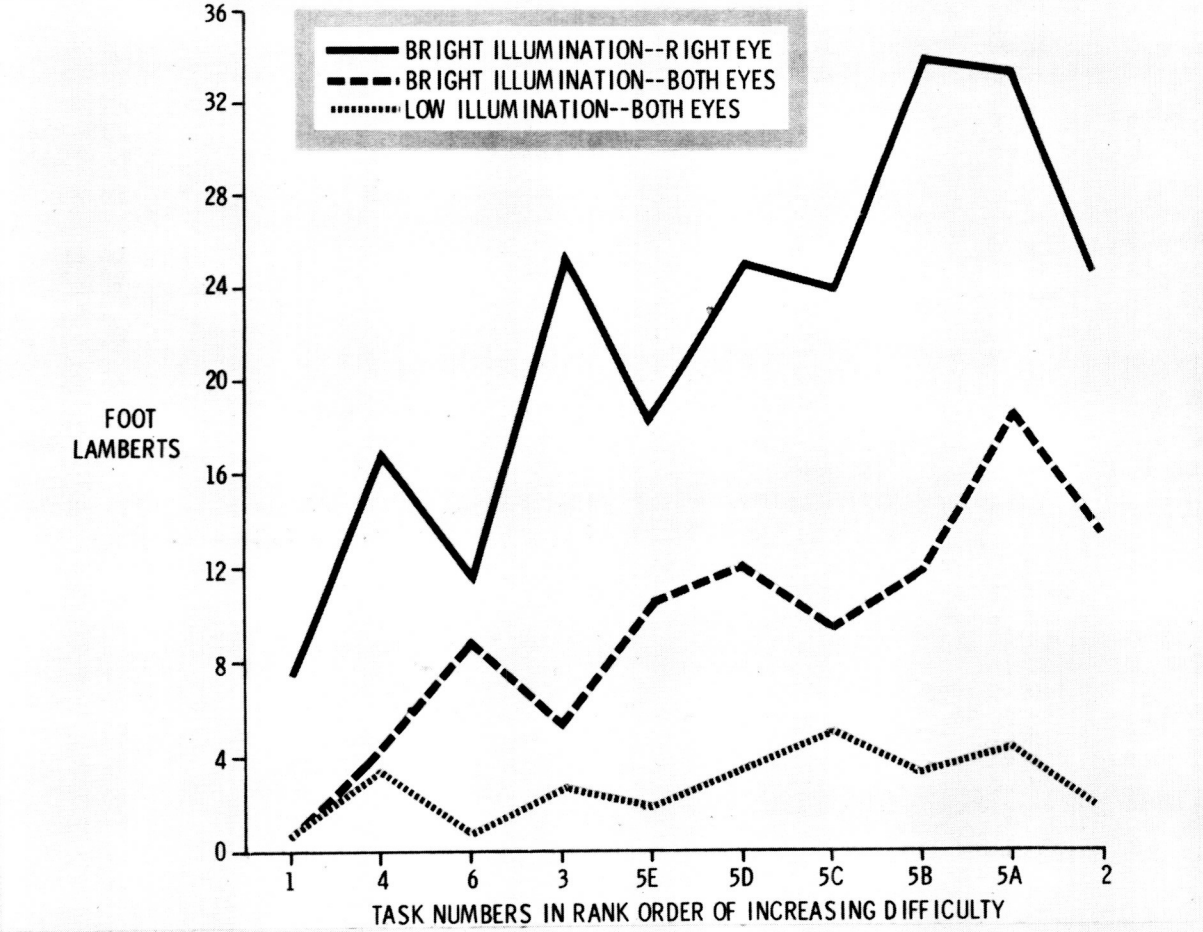



Figure 16. Illumination Study

3. The present study has investigated the worst conditions that a space maintenance operator is likely to encounter.

4. Larger components require less illumination for operator identification and maintenance than smaller components due to larger visual images being formed on the retina of the eye.

5. The solar luminance condition (bright) studied establishes the illumination requirements for an auxiliary system. These requirements will adequately suffice for maintenance under low illumination conditions.

6. An illumination system with an intensity level of 40 foot-candles (assuming a reflectance factor of 50 percent) appears optimum for extravehicular maintenance performed under conditions similar to those investigated.

7. The visor of a space helmet containing a filter, or similar means, to protect an operator's eyes when looking into the sun will affect the illumination levels defined in the current study.

Light source location.—The purpose of this study is to define an optimum auxiliary illumination system and configuration that can be utilized during maintenance extravehicularly and on the lunar and martian surfaces.

Discussion and results.—One subject was placed in the 1/6th G simulator in a shirtsleeve environment. A Navy flight helmet, used primarily as a safety precaution during simulator operation, was donned by the subject. The simulator was counterbalanced to 1/6 the weight of the subject for a functional test. The simulator was started, and the subject was given a brief adaptation period. Following adaptation the subject was directed to locate himself in front of the work panel. He was given a screwdriver and told to remove a terminal wire from the vertical panel, to remove a terminal wire from the horizontal panel protruding from the right lower section of the vertical panel, and to look through the square in the center of the panel. The metal plate, used for sawing, was removed for this test. The room was darkened and the following conditions were evaluated by means of penlight flashlights while performing the above mentioned tasks (see Fig. 17):

A. Light source located on the operator:

1. One light located in the middle of the helmet on top of the head (similar to miner's light configuration).

2. One light located at eye height on the left side of the helmet.

3. One light located at eye height on the right side of the helmet.

4. Two lights located at eye height on either side of the helmet.

5. One light located at chest height in front of the subject.

6. One light located on the back of the left wrist for the right handed operator.

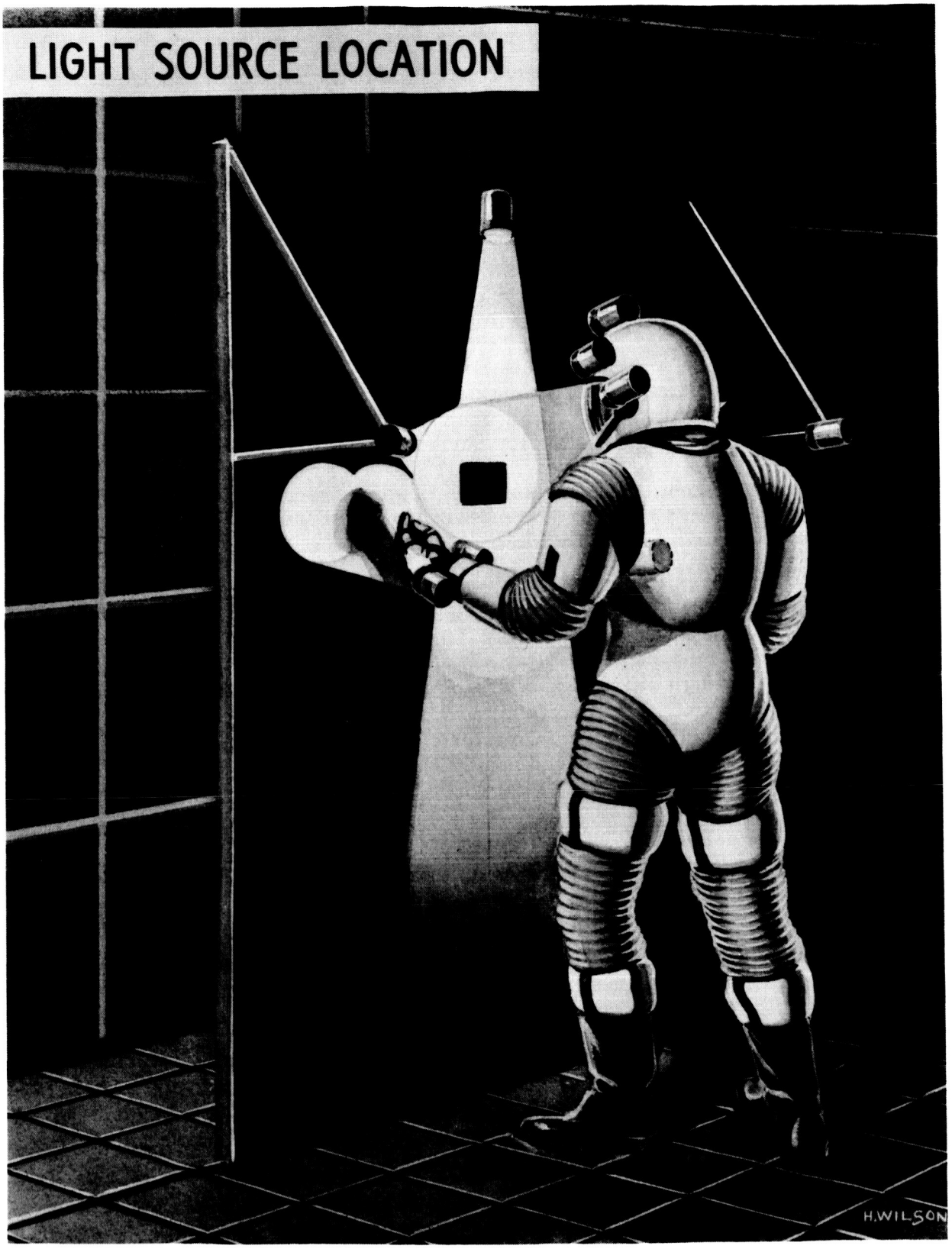



Figure 17. Light Source Locations

7. One light located on the underside of the left wrist for the right handed operator.

B. Light source located off the operator:

1. One light located 45 degrees (with respect to operator and panel) to the right and 10 inches away from the work surface.

2. One light located 45 degrees to the left and 10 inches away from the work surface.

3. One light located above and 10 inches away from the work surface.

The experimenter held the directed light source at the various locations while the subject performed the tasks, except where the source was associated with the helmet. Here, masking tape was employed to secure the light(s) to the helmet. The single light located on top of the helmet was used as the basis of comparison.

Evaluation by two subjects (trained maintenance operators) and the three experimenters quickly revealed that a light source on the helmet is preferred to any other location. The source remains fixed with respect to the operator's visual field and prevents him from having to locate additional light sources, or hold such, when looking through access openings or into a tool box located to the side. Additionally, a light source located off the operator must be secured to a structure which at times could be inconvenient or difficult.

Results of the study indicate that two lights, one on either side of the helmet, located at eye level is the optimum configuration. One light located on the opposite side of the head, at eye level, from the preferred hand was favored over a single source on top of the head.

These results indicate that the nearer the light sources are to the eye level the more directly do they illuminate the visual field. The advantage of two lights over a single source is that they provide reduced contrast by illuminating shadow areas on the working surface resulting from hand and arm movements. The two source side location is less apt to interfere with head movements through access openings, and to provide a redundant system should one of the lamps malfunction.

The two lights should evenly illuminate (excluding illumination falloff) a minimum of 30 degrees of the operator's central visual field. The lights must be mounted on the helmet in such a manner that the beams will not reflect off the visor and into the operator's eyes.

The auxiliary illumination system would ideally be integrated into the suit with an electrical outlet at waist level for hookup to the power supply in the tool kit.

Conclusions and recommendations.—As a result of this study the following conclusions and recommendations appear warranted:

1. The optimum configuration for an auxiliary illumination system consists of two light sources, one on either side of the helmet of a pressure suit, located at eye level.

2. The optimum illumination system should evenly illuminate (excluding fall-off) a minimum of 30 degrees of the operator's central visual field.

3. The two light sources must be designed so that the beams will not reflect off the visor of the helmet and into the operator's eyes.

4. Ideally, the auxiliary illumination system would be an integral part of the suit with an electrical outlet at waist level for hookup to the power supply in the tool kit.

5. An alternate illumination system would consist of one light source being located at eye level and on the opposite side of the head from the preferred hand.

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## Maintenance Analysis

General.—The development of a space tool kit is highly dependent upon the maintenance tasks expected for a particular mission. The present program is not oriented toward a particular mission or program. Instead it is intended to demonstrate a basic feasibility so that planning of space programs can be conducted, in this particular area of concern, on a more positive knowledge of the feasibility of possible maintenance tasks. For this reason emphasis in the maintenance analysis was an identification of typical functions rather than detailed task analysis.

The performance of a maintenance analysis for future spacecraft systems is a somewhat nebulous task since the detailed hardware design of these systems has not been defined. The practical approach to an early maintenance analysis of these systems is most logically accomplished by analyzing each major subsystem to the lowest definable level, using the current equipment of the Gemini and Apollo as a basis for the analysis. Therefore, this analysis will consider the structure, environmental control, propulsion, guidance/controls, telemetry, communications and power systems as the major subsystems of the spacecraft. A discussion of the task requirements anticipated for each subsystem is presented along with sample task analysis data sheets to depict the repair methods of various simulated problems. These repair tasks are based on practical maintenance experience with existing weapon systems and a knowledge of the spatial environmental and mission factors rather than on contractor supplied anticipated MTBF data.

Typical Spacecraft Assemblies.—In order to establish an effective maintenance concept it is feasible to examine existing or developmental spacecraft systems and then infer the trends for future systems.

Obviously, certain spacecraft equipment must be mounted within the spacecraft pressurized crew compartments, while others can be effectively mounted in unpressurized areas. The location of these equipments determines the necessity for space-suited maintenance and therefore greatly affects the tool design requirements.

The following description of the Apollo, MOL, and LEM spacecraft is presented to indicate the trends in spacecraft construction, materials and equipment location in order to establish an approach to the overall maintenance tool concept.

Apollo Spacecraft Assembly.—The Apollo spacecraft consists of two modules, the crew carrying command module and the service module. The modules will be discussed in relation to the types of structure and structural materials involved and the subsystems contained in each.

The Command Module.—The command module consists of an inner compartment and outer shell. The inner-compartment, which is essentially a "cockpit" for the crew, is a pressure vessel constructed of aluminum honeycomb which varies in thickness from about 3/4-inch to 1 1/2-inches. All controls and display devices for the Apollo vehicle are contained within the module. Also within the command module are the waste disposal and personal hygiene equipment, scientific equipment, food preparation equipment, water supply, space suit storage, etc.

The outer shell of the command module is the heat shield structure. It is constructed in 3 sections of stainless steel honeycomb which vary in thickness from about 0.6 inches to 2 inches. The first section, or aft heat shield, held in place by 59

bolts, is constructed of 2-inch thick stainless steel honeycomb covered with up to 2 1/2-inch thick ablative filled plastic honeycomb. The crew compartment and forward heat shield are constructed of 0.6-inch stainless steel honeycomb covered with 1/3-inch to 1 3/4-inches of ablative filled honeycomb. The face sheets of the stainless steel honeycomb vary in thickness from .008-inch to .040-inches at points of weldments or stress. The crew compartment heat shield is fastened by an I-beam and channel locking network and the forward heat shield is held in place by latches.

The command module will contain a probe and drogue type rendezvous and docking system for mating with the LEM. After initial contact the astronaut normally latches the two hatches together.

There are 12 RCS engines with ablative nozzles on the command module. Each develops 91 lbs. of thrust.

The Service Module.—The service module is a 154-inch diameter cylinder about 155 inches long, which houses most of the spacecraft equipment necessary to flight, but not necessary to reentry.

The service module contains the environmental control system, fuel cells, reaction control system and the service module propulsion system.

This unpressurized module is constructed with six radial beams as the inner structure and is covered with a 1-inch thick aluminum honeycomb sandwich. The honeycomb faceplate varies in thickness from 0.020 to 0.125 inches.

Excess heat accumulated in the environmental control system is carried to ten radiators in the outer skin by means of a Glycol-water circulation system. The waste heat is radiated into space.

The service module engine is an ablative thrust type with a 10hf-10w-3y columbium alloy expansion nozzle. It is positioned by a 28-volt dc motor driven gimbal assembly with redundant gearing. The engine compartment is covered by a 0.025 inch thick titanium alloy skirt section. Fluid lines are up to one inch in diameter.

LEM Assembly.—The lunar excursion module (LEM) is composed of two stages, the descent stage and the ascent stage.

The descent stage contains the descent engine and fuel and oxidizer tanks. Hydrogen and oxygen for fuel cell operation, water tanks, helium tanks and lunar scientific equipment are also stowed in the descent stage. Landing struts and shock attenuation system are an integral part of this stage. The shock attenuation system is anticipated to be a crushable honeycomb type. The descent stage will be left behind on the surface of the moon.

The ascent stage contains the crew cabin and ascent engine. On board systems are the ECS, Inertial Measurement Unit Space Suit Atmosphere Revitalization Systems and ECS water tanks. Hydrogen, oxygen and helium storage tanks are outside the pressurized cabin.

A reaction control system is included for rendezvous and docking with the Apollo spacecraft. Two hatches are provided, a top hatch for entry from the Apollo and a

side hatch for docking and exit from the LEM. Since there are no air-locks in the LEM the atmosphere must be dumped before ingress or egress.

OSS Configuration.—The OSS will consist of 4 modules, the Gemini capsule, the mission module, the unpressurized module and the Titan III transtage.

The Gemini capsule will act as a crew compartment during the launch and boost phase and will be the return vehicle for the crew.

The mission module will be pressurized and will serve as the laboratory area.

The unpressurized module is used for storage of systems necessary to maintain the spacecraft.

The Titan III transtage houses the engines for maneuver or orbital changes.

OSS Mission Module.—The OSS mission module structure will consist of an aluminum pressure skin, reinforced with aluminum stringers and an outer meteoroid bumper and radiator panel shell. These will be separated by insulation to prevent radiation back to the inner skin. There will be a separation space between the insulation and the bumper to provide a more punctureproof assembly.

Heat from the ECS and other systems will be conducted to the radiator panels by a Glycol-water mixture circulated through "half tubes" welded to the outer shell.

Compartments will be separated by aluminum sandwich walls and pressure hatches.

All equipment pods in the pressurized mission module will swing out from the walls to provide access to the walls for puncture repair.

The mission module will contain all experimental equipment, personal hygiene equipment, food preparation storage equipment, recreational equipment storage, ECS, etc.

An access hatch to the unpressurized module will probably be provided to permit maintenance without extravehicular activity, although a pressurized space suit will be required.

The unpressurized module will contain the fuel cells, hydrogen and oxygen supplies, attitude control propellant tanks, nitrogen tanks and waste water accumulator.

The Titan III transtage will contain fuel and oxidizer supplies and engine for orbital maneuvers.

System Analysis.—To determine the type and nature of space maintenance to be performed, it is necessary to analyze the various spacecraft systems for failure probability and maintenance feasibility. An empirical analysis follows which relates to systems currently used on the Gemini and Apollo spacecraft and considers the impact of potential innovations for future systems. A discussion of maintenance considerations follows for the structure, propulsion system, various electronic/electrical systems (guidance, controls, telemetry, communications, and power systems), and the environmental control system.

Structures.—Space environment effects which can result in structural damage include metallic sublimation, radiation and micrometeoroid bombardment. The probability of damage occurrence which would result in mission failure is remote, by best prediction, but real. Other structural damage may be incurred during rendezvous and docking, lunar or planetary landings, excessive flight maneuver loading, or internal explosion.

Penetration damage to the space cabin of several inches diameter will cause the cabin pressure to decay to intolerable values in approximately the same interval (about 5 minutes) required for an astronaut to don his spacesuit. Safety requirements of current spacecraft programs and future programs dictate that at least one astronaut be suited at all times. Therefore, if a large diameter penetration should occur, the spacesuited astronaut(s) must immediately locate the damaged area and perform a temporary repair, possibly employing a thick silicone rubber patch to reduce the spacecraft leakage rate until the other astronauts have donned and pressurized their spacesuits.

The actual repair of structural damage from within the spacecraft should not be too difficult. The cabin atmosphere will present little or no problem with the use of tools and adhesives. One problem envisioned for internal spacecraft repair is accessibility to the damaged area. This criterion should be considered in spacecraft design to allow rapid removal of equipment mounted against the pressurized bulkhead.

External damaged areas large enough to affect the structural integrity of the spacecraft will require repair; smaller punctures which do not penetrate the pressure wall can probably be ignored.

Any jagged edges resulting from a puncture will be removed, probably using an oscillatory or saber-type saw or a power file. A patch could be cut of stock aluminum and shaped to cover the puncture. The patch could be attached by adhesive bond or other bonding methods, or by mechanical means such as expandable or standard type screws. In addition, repair of heat shield damage will be mandatory for deep penetration or excessive damage. Heat shield repair will probably be accomplished by shaping the puncture and forming a "plug" of heat shield stock material for insertion into the puncture.

It appears impractical to provide spares for structural elements which may require repair during flight or on lunar or planetary surfaces. A more feasible approach is to provide stock materials, such as sheet aluminum, channel sections, ablative heat shield material, silicon sheet rubber, bonding adhesive, etc.

Propulsion System.—The propulsion system is an area which will require continuing investigation. Future spacecraft may use one or more of the more sophisticated types of engines, such as ion propulsion, nuclear, plasma jet, and so forth. Chemical rockets will probably still be used for planetary landing and launch units due to their high thrust capabilities.

Maintenance, as it is presently envisioned, will consist chiefly of replacement of small components and minor adjustments of thrust vectoring equipment. No work on major components, such as thrust chambers, is anticipated at this time.

Electrical/Electronic System.—All electronic equipment for future space missions will probably be of the thin-film integrated-circuit type. Repair will probably be by module replacement or redundant switchover. Within the spacecraft, most replacement tasks can be accomplished by removal of quick-release fasteners or captive fasteners on the front of the panel. For replacement of units in unpressurized locations, extravehicular work will be required and standard wrenches or impact wrenches will be required for loosening and tightening of panel fastening bolts. Soldering processes may be required in a few isolated cases. Harness repair may be accomplished by utilization of materials in a kit provided for this purpose.

Environmental Control System.—Environmental control systems hardware includes electromechanical fluid and air pumps, filters, air purifiers, water separators, and refrigeration and heat dissipation equipment, in addition to electrical and electronic control systems, air samplers and analyzers, and air flow regulators.

Although similar commercial equipment demonstrate high reliability characteristics, the accelerations and vibrations during boost or lunar/planetary landings will influence the need for a maintenance capability in this area.

Repair and replacement of pumps, filters, fluid lines, and seals will probably be required. Much of the ECS will probably be located in unpressurized compartments and will require spacesuited operations. Routine tasks such as replacement of lithium hydroxide canisters and particle filters will be accomplished without special tools.

Task Function Analysis.—From the foregoing discussion of systems maintenance, conclusions may be drawn regarding the type of maintenance tasks most likely to be encountered on spacecraft systems. The following is a general discussion of the type of maintenance tasks and the performance requirements imposed on the tools and equipment; no attempt is made to define the specific details of the maintenance procedure, nor are troubleshooting and fault-finding tasks considered.

The maintenance tasks are defined as basic operations, such as drilling, sawing, hammering, wrenching, screwdriving, and so forth. A repair might consist of a combination of tasks. The capability requirements of the tools required is predicted by the task function and scope. For example, what type and thickness of material will be sawed, drilled, or bonded; what sizes of screws or fasteners will be encountered and what are the torquing requirements?

Wrenching.—Fluid lines up to 1-inch diameter with standard fittings are likely to be encountered. Open-end or adjustable wrenches with capacity up to 1-1/2 inches will be required.

Large components, piping, ductwork, etc. will be fastened with numerous size bolts in the general range of #10 to 3/8-inch, with most in the 1/4 to 5/16-inch size. A relatively low torque requirement is imposed on wrenching/torquing device. A powered wrench would be necessary to reduce maintenance time and effort to an acceptable level compatible with spacesuit operation.

On present systems hexagonal head bolts are used for most applications. Recent trends are toward the use of spline heads over hexagonal heads. Standardization of bolts to one or two common sizes seems unlikely since design must consider strength requirements and weight saving methods. A wide range of wrenches or several adjustable wrenches will be required.

Screwdriving.—Terminal strip connectors as presently used on Apollo and Gemini spacecraft require standard screwdrivers for slotted-head screws. Chassis and panels held in place by quick-release fasteners will probably require a screwdriver for removal. Phillips or hexagonal head screwdrivers (spin tight) may be required.

Bonding.—Structural repair and reinforcement may be accomplished by adhesive bonding. Prepared epoxy-backed aluminum sheets and patches may be provided as part of the on-board maintenance supplies. One- to two-inch patches would be adequate for most punctures and leaks. Rapid curing could be accomplished through electrically or chemically activated heating elements, possibly contained in the patch.

Shearing.—At present, little use is seen for shearing because of the relatively large forces required, and secondly, because this operation can better be accomplished by sawing.

Sawing.—Saws would be required for some classes of structural repair. The saw should be capable of cutting 1/4-inch aluminum, 1/16-inch stainless steel, and large diameter (3-inch) tubular or honeycomb structures. The saw should be capable of starting a cut within an area and making short-radius (3/4-inch) turns. Cutting large diameter bolts or rods is not a likely task. A saber saw device is recommended.

Filing.—A file will be chiefly used to remove burrs or ragged edges from holes or sawed edges before patches are applied. A file attachment for rotating or oscillatory power devices would be time-saving and simplify work in confined areas.

Drilling.—Hole drilling will be performed primarily for structural repairs. Holes would be required for self-tapping screws or bolts up to 1/4-inch diameter. Larger holes, up to 3-inch diameter, could be cut using a hole saw or similar device.

Tapping.—The requirement for thread tapping is not anticipated to any large degree, but may be required in repairing damaged structural elements (struts, stringers, etc.). Up to 1/4-inch holes may be threaded in aluminum up to 1/4-inch thick.

Hammering.—Hammering may be required for some operations such as freeing stuck parts, minor bending or straightening, or final positioning of components with tight fits. Hammering, riveting or chipping is not expected. Riveting, if required, would make use of blind fasteners which can be inserted and secured from one side of the work.

Jacking.—Jacking operations may be used on lunar or planetary surfaces as a straightening device for damaged landing structures, but little use of the jack is envisioned in space maintenance.

Welding.—Welding techniques for extraterrestrial material joining are presently under study. Space construction and maintenance tasks indicate a definite need for welding capability.

Prying.—Prying action will be used to separate mating surfaces, flanges, connectors, or fittings. For the majority of tasks, a screwdriver will suffice. A heavier pry bar may be required for heavy structural repairs on lunar or planetary surfaces.

Soldering.—Soldering techniques may have applications in repairing damaged electrical wiring or repairing small leaks in fluid containers and tubing.

Ancillary Functions.—An addition to the specific operations required to effect maintenance and repair of the spacecraft, certain additional functions must be considered.

Lighting.—Because there is no atmosphere to scatter and diffuse incident light, the maintenance worker will require artificial lighting under conditions when the spacecraft is in the shadow of the earth or when shadows are cast across his work by himself, the spacecraft or other objects. Extension lights or portable lights will be required.

Scavenging.—In the performance of various maintenance tasks, such as drilling, sawing, and cutting, and when repairing fluid-containing systems, such as the ECS, debris and fluid residue may be released into the space cabin environment. Absolute control of these elements is necessary to prevent their escape in the zero-G environment with attendent hazards of electrical equipment malfunction, toxic effects, etc. Within a pressurized spacecraft, vacuum can be used to remove such material. Under unpressurized conditions, electrostatic devices and other methods of particle attraction must be developed. Maintenance tasks which impart velocity to debris, such as sawing or drilling, should be shielded to restrain the particles to the working area.

Parts Handling and Control.—As a result of the zero-G environment and the limited finger and hand dexterity of the spacesuited astronaut, methods of tool handling and parts containment must be used to preclude the loss of "dropped" items. Lanyards may be provided for tools. Special devices, such as adhesives, pliable retainers, or "tents" may be utilized for parts-handling and retention.

Restraint.—Space maintenance tasks will require the capability to apply forces in a zero-G environment. Counter-reaction forces transmitted to the unrestrained space worker will seriously hamper his capabilities. Where hand-holds and braces are not available, such as on the exterior of the spacecraft, restraint devices will be required. Adhesive-backed plate attachments and restraint belts may prove to be a satisfactory means of restraint.

Maintenance Task/Tool Functions.—Table 3 indicates the projected usage rate of the various tool functions required to effect maintenance and repair operations on the associated systems of a typical spacecraft. Table 4 indicates typical materials and structures expected and table 5 shows the general trends in screw sizes and material thicknesses.

Conclusion.—In considering the successful completion of extended orbital and planetary missions of the future, strong emphasis must be directed toward providing the astronauts with a maintenance and repair capability. The attainment of this capability requires proficiency in maintenance and repair, the development of which can be achieved by providing the spacecraft crew with certain basic items.

Figure 18 represents these basic items. Essentially, they are:

(1) Tools, both hand and powered, to perform a wide variety of required maintenance tasks.

(2) Spare parts to replace failed items.

(3) Repair kit for the purpose of effecting repairs not of the component or black-box replacement type. Included in this kit would be material for:

a. Leak repair of skin and radiators,

b. Repair of electrical harness,

c. Repair of on-board systems containing fluids,

d. The purpose of effecting emergency repairs of a station or craft.

(4) Reference data to include alignment procedures, schematic diagrams, drawings, and general directions necessary to effect repair.

(5) Test equipment for calibration and troubleshooting.

(6) Ancillary equipment to support crew maintenance operations. For work within the station, these include, among others, extension lights for viewing, scavenging devices for control of debris, and absorbent material for control of fluids. External operations require tether lines for safety, restraint devices for control of reaction, illumination devices, and possibly a shield for protection of the maintenance worker from micrometeoroids and radiation.

(7) Training to include theory of equipment operations, maintenance techniques and procedures.

(8) Maintenance techniques and plan describing in detail the procedures and processes by which space maintenance operations will be performed.

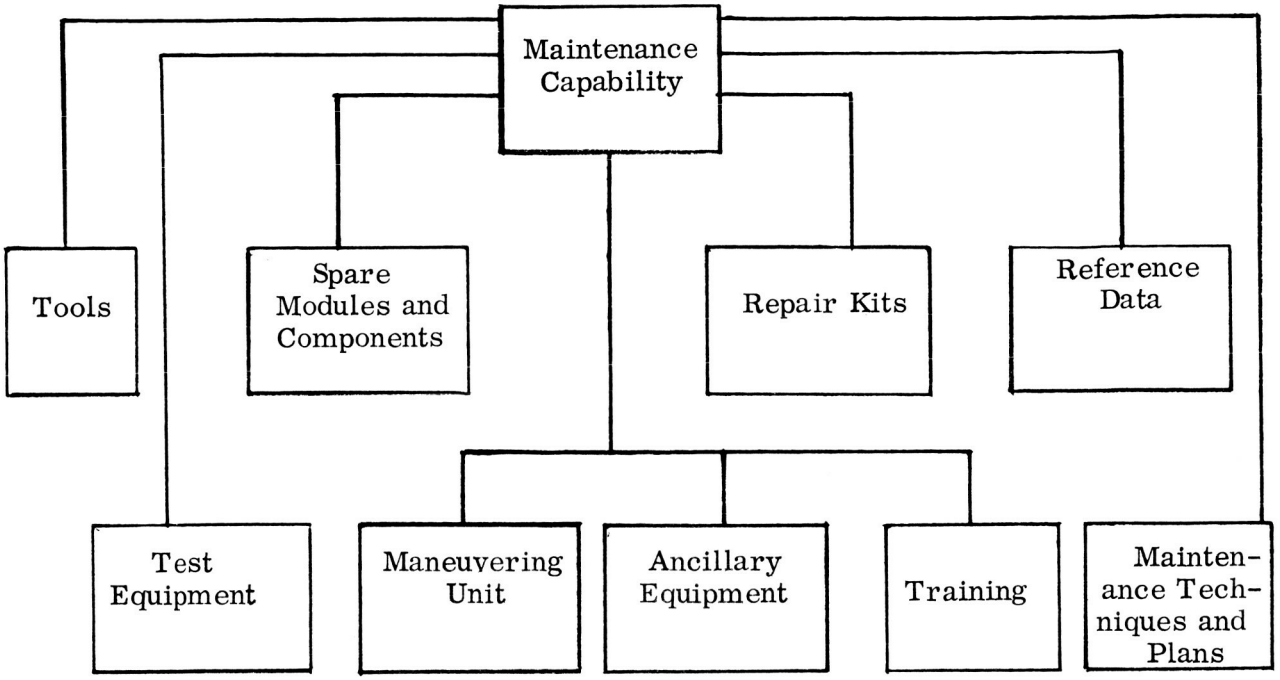



FIGURE 18. BASIC ITEMS OF A MAINTENANCE CAPABILITY

# Table 3. Maintenance Task/Tool Capability Matrix

| Maintenance Task / Tool Function | 1 External Structure (on lunar or planetary surface) | | 2 External Structure (in free space) | | 3 Propulsion | | 4 Exterior Electrical/Electronic | | | 5 Interior Structure | 6 Environmental Control System (ECS) | | | 7 Interior Electrical/Electronic | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 1 Pressure Hull Repairs | 2 Landing Gear Repairs | 1 Pressure Hull Repairs | 2 Mechanical Adjustments | 1 Component Change | 2 Mechanical Adjustments | 1 Chassis/Panel Remove and Replace | 2 Wiring Repair | 3 Fuel Cell or Cryogenics Replacement Repair | 1 Pressure Hull | 1 Pneumatic Repairs | 2 Fluid System Repairs | 3 Electro-Mechanical System Repairs | 1 Chassis/Panel Remove and Replace | 2 Wiring Repair |
| A. Wrenching | | | | | | | | | | | | | | | |
| a. Power | C | B | C | B | B | -- | B/C | -- | B | C | B | C | B | C/B | -- |
| b. Hand | D | B | C | B | B | A | B/C | C | B | C | C | B | B | B/D | C |
| B. Screwdriving | B | C | B | D | D | C | C/B | B | C | B | C | -- | B | B | B |
| C. Bonding | B | C | B | -- | C | -- | -- | -- | C | B | B | -- | D | - | -- |
| D. Cutting | | | | | | | | | | | | | | | |
| a. Shearing | D | D | D | -- | -- | -- | -- | -- | -- | C | -- | - | -- | -- | -- |
| b. Sawing | B | B | B | -- | -- | -- | -- | -- | -- | B | C | C | -- | -- | -- |
| c. Filing | B | C | C | -- | -- | -- | -- | -- | -- | C | C | C | -- | -- | -- |
| d. Drilling | B | B | B | -- | -- | -- | -- | -- | -- | B | B | -- | -- | -- | -- |
| e. Tapping | C | B | C | -- | -- | -- | -- | -- | -- | C | -- | -- | -- | -- | -- |
| E. Impacting | | | | | | | | | | | | | | | |
| a. Hand Hammer | B | B | B | C | C | C | -- | -- | C | B | B | C | C | -- | -- |
| b. Power Hammer | -- | -- | -- | -- | -- | -- | -- | -- | -- | -- | -- | -- | -- | -- | -- |
| F. Jacking | -- | C | -- | -- | -- | -- | -- | -- | -- | -- | -- | -- | -- | -- | -- |
| G. Welding | C | B | C | -- | B | -- | -- | -- | C | C | C | B | D | -- | -- |
| H. Prying | C | B | C | C | B | C | B/B | D | B | C | C | B | B | C | -- |

Frequency of Task

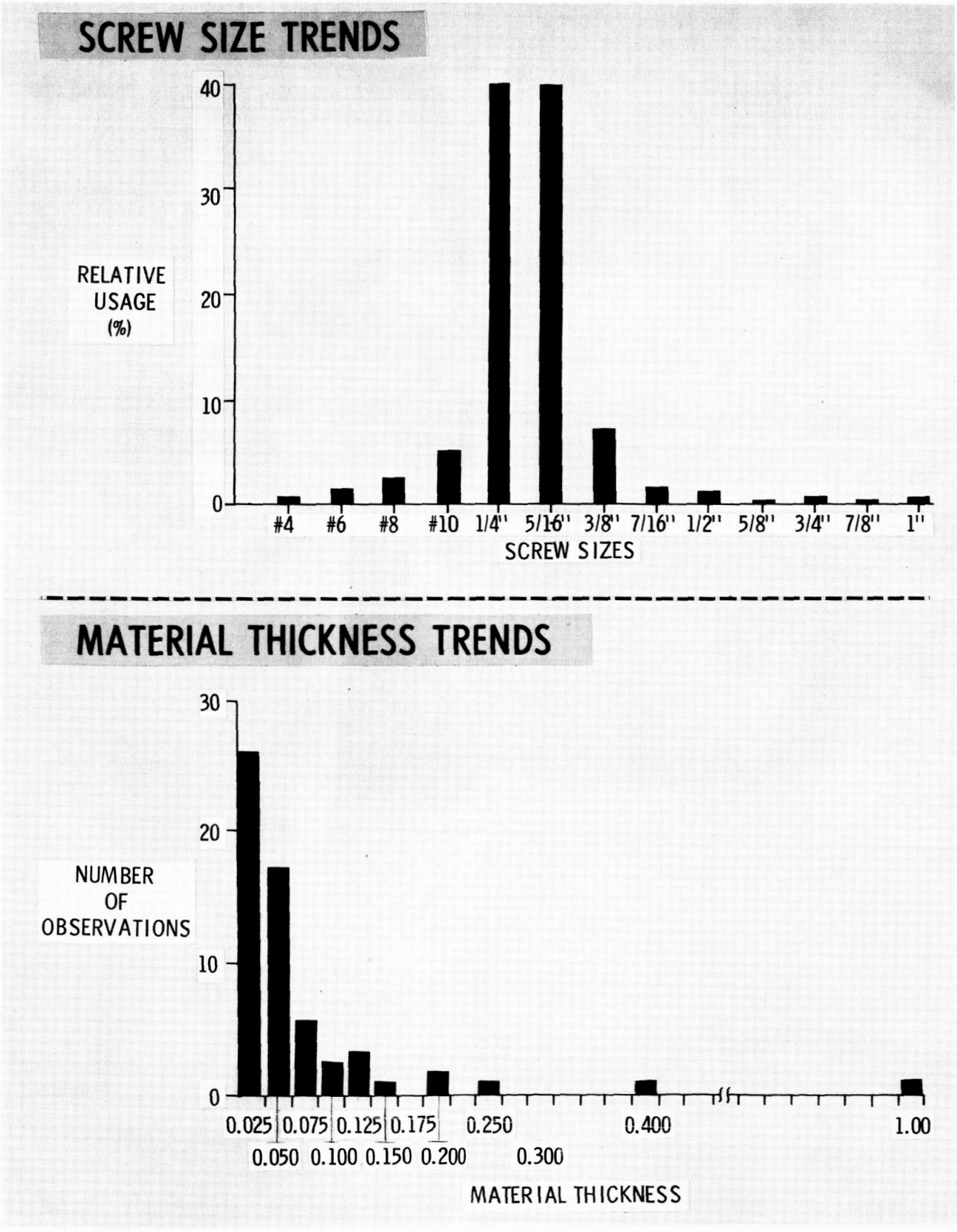
A Always
B Probable
C Seldom
D Possible but not probable
-- Never

TABLE 4

TYPICAL MATERIALS AND STRUCTURES

| MATERIAL | APPLICATION | THICKNESSES |
|---|---|---|
| Aluminum | | |
| 2014, 2024, 6061 | Multiwall Structures | Predominantly .016"-.063" (1" Maximum) |
| 2219 | Single Wall Structures | .100"-.250" |
| | Web and Structure, Supports | .010"-.150" |
| | Radiator Heat Rejection | .030" Wall |
| 1060, 1100, EC, 2014, 2017, 2024, 3003, 4043, 5052, 5456, 6061, 13, 43, 356, 360 | External and Internal Applications | .050" Wall |
| Titanium | Corrosion Resistant Structures and Frames | |
| Pure | Inner Wall Structures | .010" |
| | Re-entry Vehicles | |
| | Ribs in Re-entry Vehicles | |
| C-120 AU | Fuel Containers | .020" |
| | Re-entry Heat Shields | |
| | Radiation Storm Shelter Composite Walls | |
| | Radiator Meteorite Bumpers | .200" |
| | Meteorite Bumper for Multi-Wall Structures | .020"-.064" |
| Tungsten | High Temperature Applications Rocket Nozzles | |
| Molybdenum | High Temperature Applications | |
| Columbium | High Temperature Applications | |
| | Nuclear Liquid Metal Ducting & Radiators | .060" |
| Stainless Steel 300 Series | Station Multi-Wall Structures | .020"-.040" |
| | Storage Vessel Multi-Wall Structure | .008"-.032" |
| Magnesium | High Pressure Tubing | .020"-.040" |
| | Storage Vessel Multi-Wall Structures | |
| | Radiation Storm Shelter Composite Walls | |
| | Wall Structures of Vehicle | |

Table 5. Screw Size and Material Thickness Trends



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## Tradeoff Considerations

General.—Perusal of a standard tool catalog[1] or a stroll through a well stocked hardware store or tool crib reveals the ultimate magnitude to which space tool development might expand. It is obvious that the practical limit for a space tool kit is well below this. The Air Force, in developing maintenance capabilities for different missions has defined several technician's tool kits as well as general lists of tools required at higher organization levels which are still fundamentally beyond the desirable limits for space tool kits[2].

Since the primary concern of this program is the demonstration of the feasibility of tools for performing space maintenance tasks, a complete, exhaustive list of tools and accessory equipment is not required in the completed kit. Instead, the emphasis has been placed on defining the major tool using problems and developing tools and equipment which demonstrate a solution to these problems. The major tool functions of torquing, impacting and cutting are discussed to illustrate the logic of the selection of development items.

Torquing.—Torquing is perhaps the primary tool function expected for space maintenance and assembly operations. It is highly unlikely that assembly and disassembly techniques will be developed which do not rely upon some sort of twisting force application by the worker. Screw-type connections whether several turns or fractional turns, are almost universal. Even in electronic chassis where plug-in components are used, twist-type retainers may be found. In general, the range of torquing tasks is divisible into two parts; structural fasteners requiring relatively large torque and component fasteners and adjustments requiring small, but carefully controlled torques. The application of small torques can be accomplished readily by hand tools reacting impulsively against the astronauts mass moment of inertia. Sustained large torques will cause the zero-G worker to accelerate in the opposite direction and some method must be used to overcome this. There are two possibilities. Either the tool can eliminate the reaction or the worker can be restrained. Restraint of the worker is less preferred because it makes adjustment of work attitude and position more complex and difficult. Initial development of space tools has demonstrated the feasibility of zero reaction torquing tools of many different types. Additional considerations can, therefore, be brought into the problem.

In addition to a zero reaction tool, a relatively efficient tool is desired and tools which cover the complete operating range. The range of fasteners indicated as probable for space applications will require torques for tightening and loosening from the low inch-ounce range up to 80 to 90 foot-pounds. The capability to run nuts completely off or on is also necessary. Hand tools can be developed for space use using zero reaction principles, but just as earth maintenance shops use power tools for worker efficiency, a space maintenance worker whose work efficiency is important, will require a power tool. An in-house study of torquing fasteners with standard hand tools under simulated zero-G conditions with and without restraints (see Ref. 3) indicated that task times with hand tools are on the order of one minute. With the zero reaction power wrench, less than 5 seconds are required to run a nut completely on and tighten it to maximum torque. For the higher torques the optimum length of a hand wrench becomes significantly longer than a power tool, increasing the accessibility problem. The capability to adjust the torque output of the zero reaction power wrench

was indicated by analysis of the basic tool mechanism, but tests on small fasteners showed that the lowest torque range would be greater than could be tolerated in the numbered sizes of machine screw threads. It was, therefore, decided to develop both an adjustable power wrench for the upper torque range and a hand-held screwdriver for the lower range. The screwdriver would require special development because manipulation of a standard screwdriver in a pressure suit has been demonstrated in the Human Factors testing to be one of the more difficult tasks. Although it was decided that a screwdriver capability is a necessary function for a space maintenance tool kit, development of the screwdriver was limited to solving the problem of tool manipulation with the gloved hand. Power tool operation in the low torque range is still desirable from the standpoint of work efficiency if a reasonable mechanism can be developed.

Hammering.—Besides the capability of torque application a maintenance worker may need to hammer. The hammering requirement is oriented more toward occasional blows of varying intensity, such as a machinist would use in freeing a tight piece or tapping a component into final alignment, than toward fast repetitive constant amplitude blows such as are required for riveting or chipping. If riveting as a fastening technique is used in space operations it is highly probable that a blind rivet would be used to eliminate the necessity for bucking. Explosive rivets, squeeze rivets or other types of blind fasteners are a much more logical approach. Development of commercial power hammers has been oriented toward riveting and chipping tasks and are, therefore, not suitable for machine shop type tasks. In view of the limited use anticipated for a hammer, it was decided to include only a hand held hammer. Development would be restricted to solving the problem of holding the hammer with the spacesuit glove and restraining the operator.

Drilling.—Drilling holes is expected to be a significant maintenance function where structural repair or modification is required. Holes for fasteners to hold stiffener plates or large patches may be required. To drill a hole with standard tools, two forces must be exerted by the tool on the work, an axial thrust and a torque. Use of hand powered drills is obviously inadequate for space use. Drilling with battery operated tools on earth has already been commercially developed by the Black and Decker Company, so the only major problem is developing a mechanism which will permit zero reaction drilling to eliminate the need for operator restraint. Tests of standard drill motors have shown torques in the range of 30-40 foot-pounds on the handle when the drill is stalled in the work. Use of the impact mechanism used for wrenching would eliminate the handle torque reaction, but the axial force requirements necessitates development of a new mechanism.

In addition to the zero-G requirements, studies of force capabilities under 1/6-G showed that peak forces would be about 20 pounds. Tests of drilling with standard 1/4-inch twist drills showed that a force of about 40 pounds is required for optimum cutting rates. For larger diameter holes even more force is required. Surprisingly, machinist's handbooks and other similar data do not give drill bit pressure. Only feed rates are recommended and chip formation in drilling with hand tools versus machine tools indicates that the machine bit pressures are significantly higher than hand tool pressures. Use of a standard drill motor would require a restraint system and restraint forces upwards of 40 pounds would be required. This is considered near the upper limit of expected restraint systems.

Cutting.—Cutting will be required for repair of damaged spacecraft structures. Meteoroid impact or in-space collision with wayward cargo or spacecraft modules may have jagged edges which would have to be cut away prior to patching and special patch shapes may be required. If modification of spacecraft for rearrangement or addition of new experiments is required, cutting of existing structures to accommodate the new equipment is not wholly unlikely.

Cutting may be accomplished by several methods. Sawing, shearing and burning are the most commonly used cutting approaches. Burning is considered unwise in the high packing density environment of a spacecraft. It is also relatively dangerous and costly in terms of weight penalty of consumed fuel. Therefore, it was not seriously considered.

Shearing has advantages of not creating the debris problem associated with sawing, but the shear mechanism required for thicker, higher strength materials is considerably heavier than a comparable saw. The difference between shears, more precisely nibblers, as opposed to saws is one primarily of chip removal volume per stroke. A nibbler may cut a chip 1/8 x 1/16-inch per stroke, while a saw takes a small chip of .002 inch by .030 inch in each stroke. Because the stroke rates are different and the motor gear-down is different, the ultimate rate of material removal per unit power input may be similar in thinner materials. In thicker materials the force requirements for shearing go up significantly with concomitant increases in structure size and weight even though the cutting rate is reduced to use the same motor. The change for sawing is not significant. Saw cutting rates in thicker materials are reduced, but saw structure size and weight does not necessarily increase. The increase would be required only if cutting rates were not to be reduced.

Sawing also shows a potential for more efficient cutting of structures like honeycomb or sandwich type skins because the blade can cut both faces and the core simultaneously even though the stroke is less than the thickness of the material being cut. A shear's stroke must be the full thickness of the material. Since this type of structure is highly probable, sawing as a cutting technique was selected.

There are three basic types of saw mechanisms, reciprocating, rotary, and continuous rectilinear as in a band saw. The band saw approach was rejected because the idler pulley would require major supporting structure on both sides of the work.

The rotary saw was felt to be limited because of the depth limitation to a fraction of the saw radius and the inability to make radius cuts.

A reciprocating saw could be of two types, rectilinear or rotary reciprocating. An example of a rotary reciprocating saw was tested to determine its feasibility for use because of its particularly attractive safety feature. Tests were conducted of a Zimmer orthopedic cast saw. The saw uses a circular blade which oscillates approximately $7^{O}$ at 32 K to 36 K strokes per minute. The saw will cut rigid, hard materials but because of the rapid short stroke it will not cut soft, yielding materials such as flesh or fabrics. This characteristic made it attractive as a potential space tool. The standard Zimmer saw blade is merely a serrated disc which would not stand up under metal cutting loads so a special blade with four different tooth shapes was fabricated for the test. The tooth shapes were based upon designs of power hacksaw, band saw and circular saw blades. The tool was tested in cutting .028 inch Ti - 6 Al - 4 V STA and .063 inch 7075 - T6 aluminum. The tool was difficult to control in the work and the cutting rate was only 1 inch/minute in aluminum as

compared to 6 inches per minute for a jig saw.

The handling difficulty and the existence of the same drawbacks as the rotary (circular) saw dictated the selection of the reciprocating jig saw as the basic sawing approach. This type of saw has the advantages of reasonable variation in depth of cut (large models handle 4-inch thick wood), ability to cut a small radius and reasonable safety in that only the front of the blade is really dangerous during operation, although at no time is contact with the blade advisable.

Other Tool Functions.—The other tool functions listed in the maintenance analysis section were considered not to present significant development problems or to be of relatively less significance as a separate function. Joining techniques such as bonding, welding or soldering were considered properly the subject for other development programs where the emphasis would be on fabrication techniques, not tool development. This is not to say that tools may not be required to accomplish them, but selection of the optimum joining techniques must precede development of the associated tools as part of an integrated tool kit.

Tool Function Integration.—Multipurpose tools as a means of saving weight and volume offer attractive advantages for a space tool kit. Consideration of multipurpose tools involves a more complex systems evaluation of the maintenance function than can profitably be undertaken in an initial program such as this. More precise definition of tool function and performance range is required before detailed multipurpose tool tradeoffs can be made. For instance, it makes sense to consider a screwdriver integrated into the handle on a pair of pliers. But if the pliers must be "10-inch slipjoint" and the screwdriver required is a "1-inch No. 1 Phillips" obviously the two would not be integrated. For power tools, if the requirement is for a 200 foot-pound impact wrench and a saw for only .016 SO aluminum, the motor requirements would not match.

However, it is possible to consider some major aspects of the problem and to define criteria which will be applicable. The advantage of multipurpose tools is one of weight and volume economy. In contrast, the disadvantage is that the tool may not be suited to the task because compromises have to be made in its design. Within this writer's experience, for instance, most multipurpose hand tools have been extremely awkward to use because what is the screwdriver handle cannot be adequately grasped.

For power tools a different set of relationships is valid. The major integration of power tool functions revolves around interchangeability of the drive mechanism with separate functional attachments. The tradeoff considerations require compromises in motor, handle and working mechanism physical and power relationships. The relations vary according to the function and an optimum physical relationship between motor and handle for drilling is different than for sawing. The choice facing the designer then is how to arrange the motor, handle and attachment interface to optimize all functions without sacrifice of performance capability. The applicable criteria are:

(1) The relationship between the working end of the attachments and the tool handle for a multipurpose tool must be as similar as possible to the single purpose tools;

(2) The force requirements for the proper operation of the multipurpose tool must be similar to the single purpose tool;

(3) The installation and removal of attachments must require a minimum of operational dexterity and time;

(4) Those elements which are common to all functions should be located in the basic unit;

(5) Loss of overall working efficiency (ease of use and accessibility + power efficiency + operator efficiency + operating range + operating precision) must not outweigh the weight and volume advantages.

With these criteria in mind an analysis of the power tool functions was made to evaluate the advantages or disadvantages of using a multipurpose power tool approach. There are actually two ways to achieve multipurpose capability, addition of accessories such as a grinding wheel for a drill motor or taps for an impact wrench and the prime mover/attachment combination. The former is desirable inasmuch as it is generally simpler to accomplish. Major functional differences make the latter approach more realistic when the considerations of efficiency, control and accessibility are important. The tool functions considered were torquing, drilling and sawing. If separate tools are developed, the complete tool kit will require three separate motors of roughly equal power. The motor and handle of a power tool constitute a major portion of the overall tool weight and since it would be impossible for the astronaut to use more than one at a time, there is no advantage apparent for separate tools. Examination of the control and output forces (Figure 19) shows that the basic force relationships are similar and the possibility of a multipurpose tool using a basic motor and handle and separate attachments is good. It would be desirable to minimize the number of attachments by using simple accessories where possible. The comparison of drilling and torquing control forces shows that they are very similar and it was decided to attach the drill mechanism to the front of the impactor. This would provide the zero reaction torque feature. Only the drill bit pressure mechanism would have to be developed separately. Additionally the impactor provides an over-torque safety feature when the drill bit stalls in the work. With the impactor, the stall merely initiates impacting action and the usual violent twist is not felt. The saw mechanism for commercial tools requires no such safety feature and the consideration of overall length, control forces and efficiency indicated that a separate attachment rather than attachment to the impactor would be desirable.

Since the tools would be battery operated, consideration was given to either including the battery in the motor housing or providing a battery in the tool box. Since there were accessory functions requiring electrical power the tradeoff was one primarily of efficient use of battery power. Three power drains would be encountered during a maintenance operation; the power tools, the work light and the restraint attachment. The work lights would be a low current continuous drain. The adhesive restraint would be a high current short drain at the beginning of the maintenance operation and the power tools would require high current intermittent operation throughout the period. Since battery weight increases significantly with peak current capacity, it would require more weight to provide separate batteries for the adhesive restraints and the power tool. The adhesive restraint design was capable of matching the motor voltage requirement so one battery was feasible. In addition, when working on the light side of the craft, the work light requirement could be traded off for tool performance time if they had a common source.

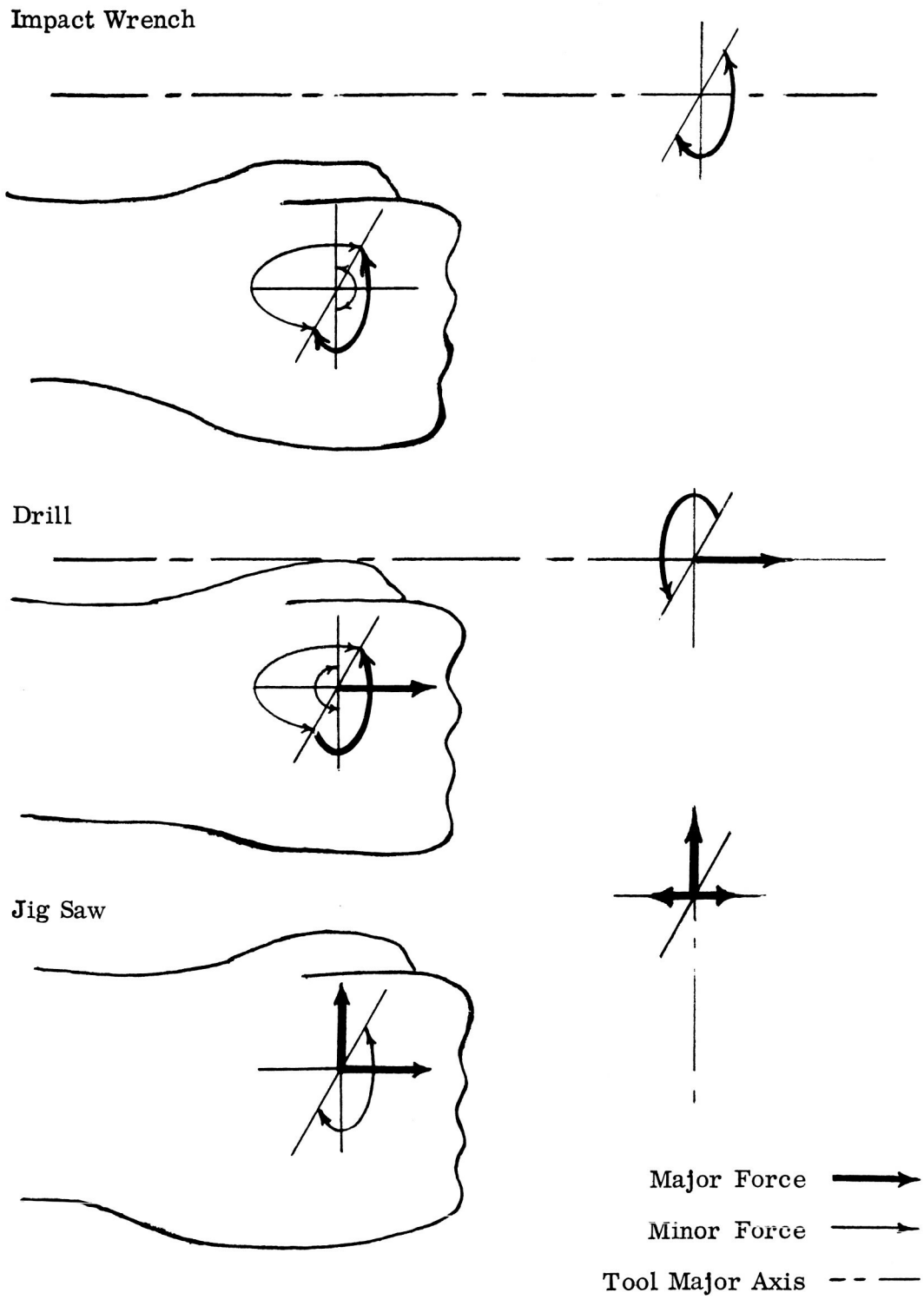



FIGURE 19. POWER TOOL FORCE RELATIONSHIPS

An additional factor is the tool envelope and its effect upon accessibility. Experience with earlier space tool development had shown the volume of a tool handle is insufficient to effectively hold the battery so that the basic envelope would have to be larger to include the battery in the tool and, therefore, a large work space would be required to use the tool. The power cable running between the tool box and the tool was not considered a hindrance because it would also serve as a lanyard. Taking power/weight as the major factor it was decided that a common battery should be included in the tool box rather than separate batteries supplying each electrical function.

Accessory Equipment.—The space environment will require accessory equipment for the worker in addition to his basic tools. Besides the problem of packaging the tools for efficient transportation and use, the worker will require as a minimum, a work light, a tether or restraint system and means of manipulating small parts.

The work light is required for both dark side illumination and shadow fill light as defined in the Human Factors Studies

The restraint system requirement is somewhat more complex. Although it is a goal to develop tools with zero reaction on the worker, there are periods when the worker will be using hand tools or when he will be handling components directly. During these periods he will be exerting forces and developing reactions which create the same problems as tools, but of lesser magnitude. Therefore, he will require at least a minimal restraint system. The restraint system capabilities must be matched with the tool capabilities and requirements to form an integrated system. The basic considerations for a restraint system can be defined as follows:

(1) Restraint attachment must be quick and easy;

(2) Attachment must be made wherever maintenance work is expected to be performed;

(3) Adjustment of attitude or position must require the minimum time and effort;

(4) Maximum advantage must be taken of unique environmental characteristics such as weightlessness;

(5) Restraint system force capabilities must exceed the expected reaction force loads in all significant directions;

(6) Interface requirements with the space suit must be minimized.

There are two restraint approaches which can be taken. On the one hand, we can attempt to approach as closely as possible the unrestrained condition providing only a minimal tether, while on the other hand we can attempt to reproduce the mechanical situation used on earth where the back, shoulder and leg muscles can be used in the accustomed way.

Restraint attachment for either concept would be expected to be essentially the same with the exception that a simple tether will require only one attachment point where a complete restraint would require at least three. Attachment at the work site would be the same in either case. Adjustment of attitude or position is far

easier with a simple tether than with a complete restraint system since the complete system requires adjustment of three legs for major shifts of position. The attempt to reproduce the earthside mechanics fails to utilize to the maximum the weightless condition. Anyone who has ever crawled under a car hood or into other awkward situations can attest to the desirability of being able to stand on your head at times. The high density packaging of spacecraft will create accessibility problems which in conjunction with the problem of limited space suit mobility will make the maximum flexibility of working position highly desirable. Since in orbit there is no inherent up and down at the work site, restraint attachment requirements should not impose an up and down orientation. Restraint force capabilities would be spread over at least three fasteners in the complete restraint system. This means that for properly positioned attachment points, the yield strength can be less than for a single point tether. However, implied in this tradeoff consideration is a total system approach of complete restraint plus standard tools versus a simple tether and zero reaction tools. If zero reaction tools are used with either restraint concept the three point concept has a definite strength advantage and safety advantage through redundancy. The interface with the spacesuit will be more complex with three restraint connections than with one, not only because of the number, but also because the structural loads will need to be transferred across the suit from one member to the other.

It would also be worthwhile to examine the fundamental concept of three-point restraint to compare the mechanics with what is found in a work situation on earth. On earth the forces applied to a tool are essentially generated by developing a moment around the contact point of one foot with the floor. The weight of the subject acting around his foot creates a force on the tool through the, for this purpose, rigid body structure. The skeletal muscles must develop enough force to transmit the moments developed around the foot to the point of application at the tool. In a zero-G restraint situation the forces are essentially applied at similar points because the human center of gravity is reasonably well aligned with the hips. But the direction of forces is different. On earth the leg extensor muscles support the weight and develop the tool force. Under the zero-G condition the flexor muscles are required to develop the tool force by creating a bending moment around the hip restraint point. This is an unnatural condition and therefore potentially troublesome.

To evaluate the restraint concepts further before making a final choice for development, the brief study described below was conducted.

The purpose of this study was to investigate several different restraint concepts, and to select an optimum configuration that will enable an operator to adequately perform extravehicular space maintenance.

Whether or not extravehicular maintenance can be performed in zero-G depends largely on how well the operator is restrained at the work area. The study concentrates on a zero-G environment, over a 1/6th and 2/5ths G, since it appears to represent the most difficult of the three working environments. While it may be necessary for an operator to be restrained in performing certain tasks on the lunar and martian surfaces, such a restraint system is not likely to be as sophisticated as that required under zero-G.

A space maintenance worker, restrained in a work position equivalent to that on earth for a given task, should encounter no increase in task difficulty as the result of

the zero-G environment. Present day "state-of-the-art" pressure suits, which are not designed for space maintenance, will increase task difficulty by reducing operator mobility and dexterity. A review of the literature indicates that various types of space maintenance restraint systems have been investigated. The NASA Human Engineering Criteria (Ref. 4) makes a comparison of tethering devices and finds toe hooks and adjustable waist straps the most satisfactory. Encapsulation devices such as space tugs (Ref. 5) have been studied indicating present day thinking on the topic of operator restraint.

Four restraint concepts were tested and evaluated. They include: (1) Two point hip and one point toe, (2) Two point shoulder and one point toe, (3) One point navel and one point toe, and (4) One point navel.

The current investigation utilizes an empirical approach. Two trained maintenance operators participated as subjects. The Lunar Gravity simulator was modified for this evaluation. The work panel, normally mounted at the rear of the standing platform, was mounted at the front of the platform (see Figure 20 and 21). The work panel and platform were balanced in the vertical plane. Since the 5 degree-of-freedom simulator does not permit vertical (up and down) movement, the work panel served as the apparent 6th degree (vertical) of motion. No pressure suits were available for this study.

A two point hip and one point toe concept was studied (see Figure 20). The subject was restrained from two points at the waist, at a slight downward angle, to two points located at the lower corners of the work panel. The left foot (in the case of a right handed operator) was secured to an 11-inch strip mounted below the simulator platform. The simulator was started, and the subject readily performed drilling, sawing and torquing tasks. These tasks were selected because they involve a representative sample of directional forces on the part of an operator. The restraint was evaluated as adequate for all two-handed tasks. It provided the operator with stability in all axes, and maximized body mobility and flexibility from the waist upward. If an operator desired to rotate around the work area for better accessibility, it would be necessary for him to free himself from three points to relocate the restraint system. Thus, a restraint utilizing fewer tie-down points would be easier to set up and relocate.

A two point shoulder and one point toe restraint was investigated (see Figure 21). The same toe tie-down was utilized as for the waist restraint concept. Here, two points at the shoulder were secured at an upward angle to two points at the upper corners of the work panel. It was necessary to tether the shoulder points to the back of the simulator rather than on the subject's body. The subject proceeded to perform the selected tasks with the simulator in operation. Evaluation of this restraint concept indicated it to be less desirable than the two point waist and one point toe approach. The limiting factor with the shoulder concept was that it significantly reduced body mobility from the waist upward.

A two point restraint concept was next explored. The two feet together constituted one point, while the second point extended from the subject's navel to the working surface. The zero-G cradle was used instead of the 1/6th-G cradle because it restrained the subject's feet. The subject was secured in the simulator, and the feet were restrained to the platform of the lunar gravity simulator. A nylon tether was fastened to the center of the waist strap at the subject's navel and secured to the

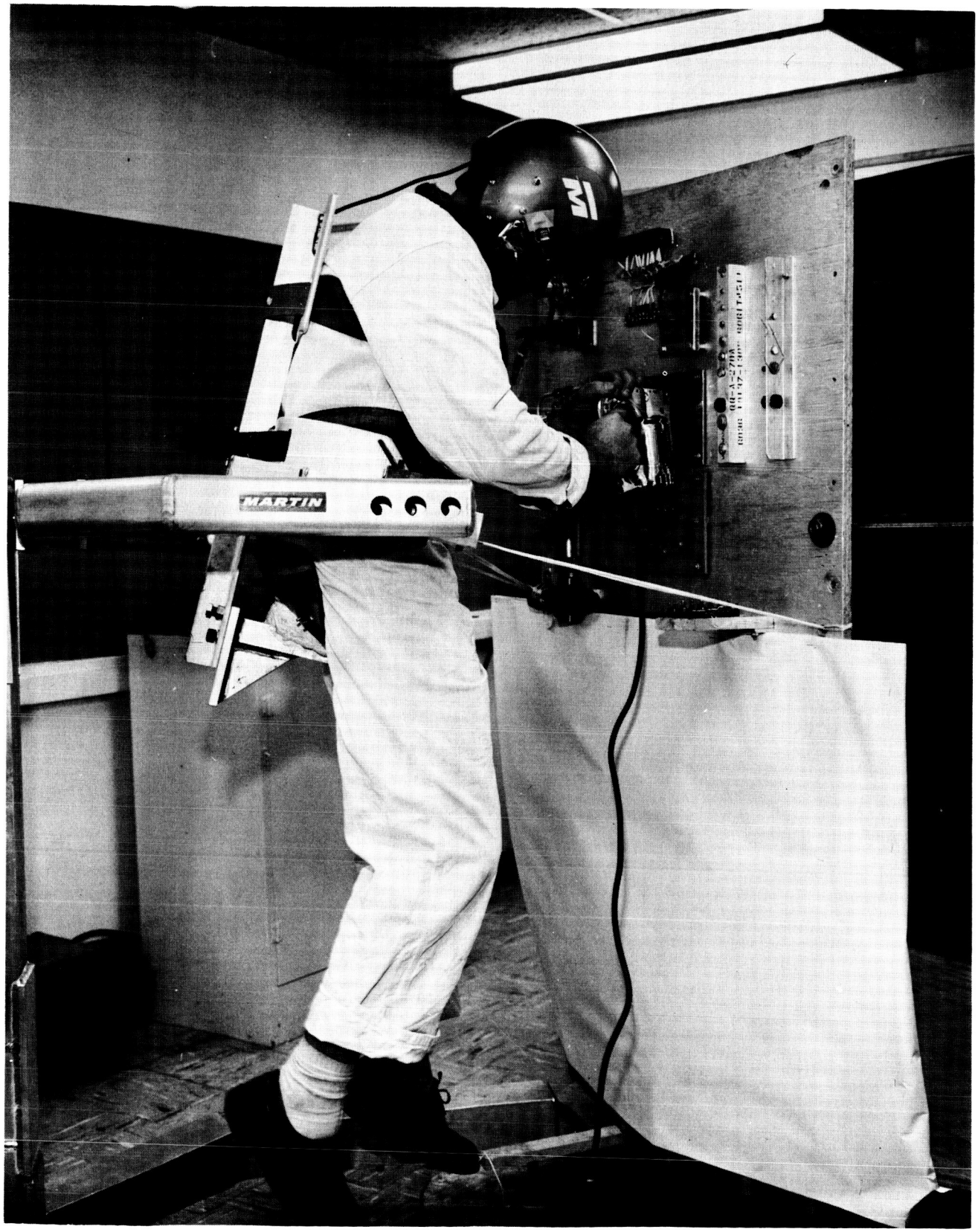



Figure 20. Hip-Toe Restraint

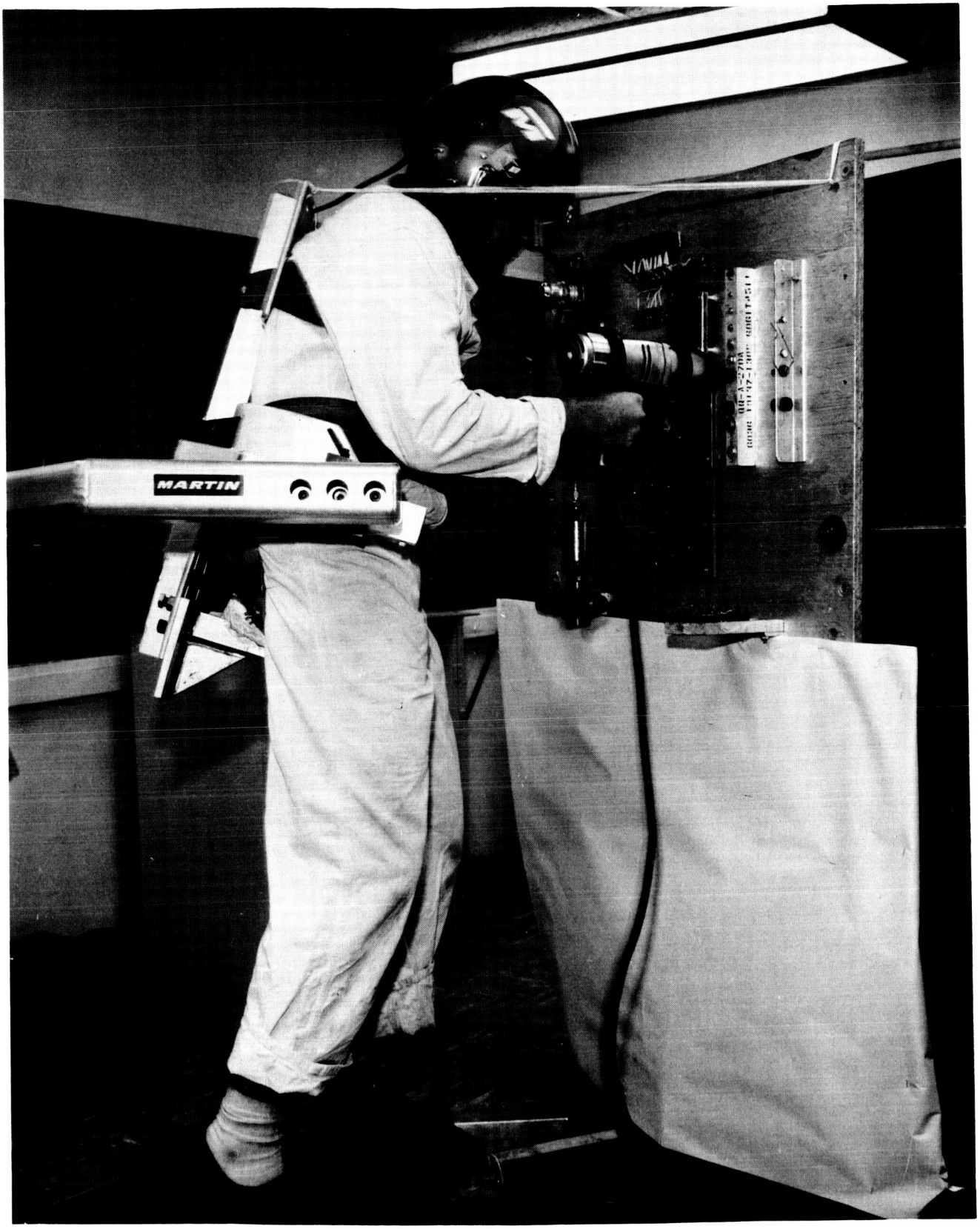



Figure 21. Shoulder-Toe Restraint

lower portion of the work panel. The subject had little difficulty drilling, but had the tendency to yaw while sawing and torquing. The subject reported that having the feet together was unnatural and uncomfortable.

The final single point concept investigated was similar to the last, except the feet were not restrained. Also, the subject was allowed to hold onto the panel for stabilization with one hand while working with the other. The same apparatus was used as for the last concept studied. A single point restraint was extended from the subject's navel to a point at the bottom of the panel. In reality, the tie-down would be located just below an access opening. By placing the feet against the lower panel (side of a space vehicle) of the simulator the subject was able to drill and saw, but considerable more time was necessary. Actually, the subject was able to utilize his hands and feet as additional restraint points. While such a restraint system would be inadequate for two-handed maintenance, it is conceivable that such a system may have merit for such maintenance tasks as visual inspection, and one-handed tasks which are not time consuming. One advantage to such a restraint is that it allows the operator to rotate about the work area for better accessibility. It is also quicker and easier to install and relocate one attachment point than two or more.

As a result of the study and with consideration of the 6 points discussed above it was decided that a single point tether would provide the optimum restraint system particularly in view of the fact that present spacesuit designs do not incorporate flexible waist joints so that hip restraint will cause a considerable reduction in torso mobility while working.

Small Parts Manipulation.—Manipulation of small parts with a pressurized glove was indicated as a problem during the human factors studies. Consideration was given to the relative importance of this problem in the overall maintenance situations.

It is probable that maintenance situations will require component removal and replacement and that many of the components will be attached with relatively small fasteners. The danger of accidental dropping is twofold. The part dropped will in most cases be irreplaceable because logistic considerations will minimize stockage of these types of items as on-board spares and the dropped part, since it does not "fall down," may find its way into areas like electrical terminals or reaction control jets where its presence could become catastrophic. It is not unreasonable to assume a high degree of caution while manipulating parts, but the consequences of an accident make it desirable to provide the best possible assurances that parts will be retained.

When the task sequence is examined, the problem resolves itself into two parts, manipulation and temporary storage. A worker must loosen the part, remove it from the tool or the last portion of its travel, transport it to a storage area and place it out of the way while other tasks are being performed. For replacement the opposite sequence is used. While it is highly desirable that components use captive fasteners wherever possible it is felt that in all cases this will not be feasible and the problem could at best be minimized, but not eliminated.

The critical periods appear to be at the points where the part is being picked up or released. Therefore, the following criteria can be defined:

(1) The minimum number of grasping or releasing actions is required;

(2) Storage and manipulation must be accessible to both hands;

(3) Natural grasping actions are preferred;

(4) Positive control of the part is required at all times while it is being transported or positioned;

(5) Modification of the part for the purpose of handling or storage is undesirable.

The temporary storage problem can be approached in three basic modes. In one mode each part removed is immediately grasped and stored in an individual device, in the second, parts are stored in a closed container, preferably with a self-closing feature and third, the parts are positively held on a storage rack.

The second is the least desirable because a part can be lost while the lid is open and another part is being inserted. In the third mode the positive holding must make use of forces other than gravity. The forces must be capable of restraining motion in various directions which could occur from accidental contact with one part while handling another. Since the materials from which parts will be made may be nonferrous metals or non-metals, magnetic holding cannot be used. Use of a holding technique like "Velcro" which requires a two-part system is impractical because it would require the attachment of one part of the system to every part which would be expected to be removed. Contact adhesives have been shown to be impractical because they deteriorate through pickup of dust, debris and lint, which may not be a problem in orbital extravehicular work, but which could be on the lunar surface and inside the cabin, and because they lose strength at temperatures below 200° F so that hot parts would not be retained. Mechanical retention devices show the greatest advantage. Choosing between a separate mechanical holding device for each part or a mechanical retention storage rack is dependent upon the evaluation of the complete activity sequence for whatever devices could be developed. It was decided to study and develop two mechanical concepts and make a final selection after evaluation of both. This effort is reported in the section on small parts manipulation and restraint development.

In summary, the following tool and accessory functions were selected for development:

| Tool Function | Development Problem |
|---|---|
| Torquing | |
| Power Wrench | Provide a zero reaction torque capability up to ≈ 75 foot-pounds for use by a pressure suited astronaut. |
| Hand Screwdriver | Adapt a screwdriver capable of use by a pressurized gloved hand. |
| Drilling | Develop a zero reaction power operated drill capable of use by a pressure suited astronaut for drilling up to 1/2 inch diameter holes. |

| Tool Function | Development Problem |
|---|---|
| Sawing | Develop a zero reaction jig saw capable of use by a pressure suited astronaut for cutting typical spacecraft skin materials. |
| Hammering | Develop a hand hammer optimized for use with a pressurized gloved hand. |
| Integrated Functions | Develop a basic motor/handle with separate attachments for impact wrenching and sawing. Drilling requires an accessory for the impact wrench. |
| | Provide a common battery supply for tool and accessory power. |
| **Accessory Function** | |
| Work Light | Develop a work light which provides intensity and direction suitable for space maintenance work. |
| Restraint System | |
| Attachment Fitting | Develop an attachment fitting capable of simple emplacement and strong enough to withstand expected work forces. |
| Restraint Link | Provide a simple single point tether between the worker and the work site. |
| Small Parts Manipulation and Restraint System | Develop a complete system for positive mechanical control of small parts during assembly and disassembly operations. |

Tool Kit Tradeoffs.—Tool and accessory development alone provide only basic functions which must be integrated into a complete useable system for effective space maintenance work. It is to no avail to provide an astronaut with tools and accessory equipment if he is confronted with unnecessary difficulties in using each piece of equipment to accomplish his tasks. Therefore, consideration was given to defining the criteria for development of the tool kit package for the tools and accessories to be developed. Tool kit development must approach the problem from the point of overall worker efficiency at the work site. Whatever will reduce total maintenance time and possible error is preferred. This thinking was applied to the selection of tool and accessory concepts and it is equally valid for solution of the work site problem.

Criteria for development of the worksite layout can be defined as follows:

(1) Equipment should be placed immediately adjacent to the work to minimize waste motion;

(2) Equipment placement must not interfere with task performance;

(3) Equipment should be accessible, in the stored position, to the hand which will use it;

(4) Most used tools or accessories should be placed in the most easily reached areas;

(5) The storage position should present the tool or accessory for grasping in the using position;

(6) Discrimination between locations of major use items should be possible by touch.

Besides the general criteria of work site layout the space environment imposes additional criteria on the tool kit packaging:

(1) The tool kit must positively retain the tools to prevent drifting away when not in use;

(2) The heat loads from space craft skins and incident solar radiation should be minimized;

(3) The tool kit must be restrained at the work site to minimize problems inherent in removing and replacing equipment;

(4) The tool kit must be transportable without loss or disturbance to its contents.

## References


1. Snap-On Industrial Catalog, 65-P, Snap-On Tools, Kenosha, Wisconsin, 1965.

2. Sets, Kits and Outfits of Hand Tools, Federal Supply Catalog, C6-7-SL-CB5, FSC Class, DOD, 1965.

3. Malzahn, D. F.: A Study of Zero-G Effects on Space Maintenance Problems and Techniques, a Simulated Approach. Martin Report ER 13165, Martin Company, Baltimore, Maryland, October 1963.

4. Ling-Tempco-Vought, Inc.: A Preliminary Design Guide of Human Engineering Criteria for Maintenance and Repair of Advanced Space Systems (HECMAR). Report No. 00.474, Dallas, Texas, 19 June 1964.

5. Adams, C. R. and Hanff, G.: Some Human Factors Considerations for Orbital Maintenance and Materials Transfer. Douglas Paper No. 3017, Douglas Aircraft Company, Inc., Santa Monica, California, 1964.

## TOOL DESIGN DEVELOPMENT & EVALUATION

General.—Design of specific tools to fulfill the functions selected for development required further tradeoff considerations between the desired functional characteristics and the current state-of-the-art. In general the approach taken was to evaluate specific concepts which were promising so that one could be selected for actual fabrication and evaluation. The concepts listed in Table 6 were felt to be reasonably capable of development. The table is not necessarily comprehensive in covering all possibilities.

TABLE 6

ZERO REACTION TOOL CONCEPTS EVALUATED

| FUNCTION | CONCEPTS EVALUATED |
|---|---|
| Torquing | |
| Power Wrench | Battery operated Zero Reaction Wrench |
| | 1. Torque adjusted by Controlled Internal Restraint clamp |
| | 2. Torque adjusted by spring preload |
| Screw Driver | 1. Special handle with ratchet adaptor |
| | 2. T-Bar with ratchet adaptor |
| Drilling | 1. Standard twist drill inside sleeve type adhesive holder |
| | 2. Trepanning (Hole saw) with internal needle restraint |
| Sawing | 1. Blind side Hold down - top side feed |
| | 2. Kerf hold and feed - continuous |
| | 3. Kerf hold and feed - reciprocating |
| | 4. Spring hold down and feed |

In order to evaluate the concepts certain basic data on tool performance was required. A series of studies were made to define;

Drilling Parameters
Saw Parameters
Tool handle/glove interface
Needle penetration Parameters

These studies along with the Maintenance Analysis, Human Factors and Environmental Constraints were used to define the Performance Goals for the tools listed in Table 7.

TABLE 7

NASA TOOL PERFORMANCE GOALS

| | | |
|---|---|---|
| 1. Impacting | | |
| | Performance | 1. 75 foot-pound maximum adjustable to 6 foot-pound in at least 3 steps |
| | | 2. Remove and install 30 1/2-20 bolts or equivalent nuts in a 3 hour period |
| | Reaction on Handle | 1. Torque - 2 inch-ounce |
| | | 2. Push-away restraint provided on spindle for sockets. Socket will provide restraint to fastener. |
| 2. Sawing | | |
| | Performance | Cut a total of 50 inches in a 3 hour period |
| | Materials | 1. Al - 7075, 6061, 2024 |
| | | 2. Stainless |
| | | 3. Ti - 6 Al, 4V |
| | Thickness | 1. 1/4-inch maximum |
| | | 2. Nominal .025 - .050 |
| | | 3. Cut both faces of 1-inch honeycomb simultaneously |
| | | 4. All cuts normal to surface |
| | Reaction on Handle | Push-away 0 |
| | | Parallel to work 0 |
| | | Torque 0 |
| 3. Drilling | | |
| | Performance | 18 holes in a 3-hour period |
| | Size | 1/8-inch to 1/2-inch OD |

TABLE 7 (Cont)

NASA TOOL PERFORMANCE GOALS

| | | | |
|---|---|---|---|
| | Thickness | 1. 1/4-inch maximum | |
| | | 2. 020 nominal | |
| | | 3. Drill both faces of 1-inch honeycomb from same side | |
| | Materials | 1. Al - 70 5, 6061, 2024 | |
| | | 2. Stainless | |
| | | 3. Ti - 6 Al, 4V | |
| | Reaction on Handle | 1. Torque | 2 inch-ounces |
| | | 2. Push-away | 0 |
| 4. | Hammering | Deleted - no power tool requirement. | |

Note 1: BATTERY CRITERIA

Tool use - performance per tool criteria worst case

Combined operations in one 3-hour period:

| | |
|---|---|
| 6 holes | Ti .028 |
| 12 bolts | 5/16 NF |
| 10 inch total cut | Ti .028 |

Illumination - 12 vdc 1A continuous for 4 hours

Restraints - 12 vdc 8A for 15 seconds

Repeat 4 times during 4-hour period.

Recharge - to full capacity at maximum possible rate 30-50 cycles.

Environment - meet requirements of Figure 2.

Note 2: HUMAN FACTORS CRITERIA

Pressurized glove operation of all switches, adjustments and attachments.

All corners rounded to maximum possible radius.

Switch operation protected from accidental throw.

Blind operation of all attachments in minimum possible time and number of steps.

Discussion of the concepts for achieving zero reaction for each function is contained in the section of that particular development. The final prototype tool design evolved from work with preliminary feasibility models which were operated and tested to determine the optimum design features to be incorporated in the final models. Figure 22 shows the final configuration power tool motor/handle and attachment. During development, initial feasibility models of the motor/handle, impactor and saw were made and tested (Figure 23). Because development of the drill mechanism was slowed by technical problems only one configuration of it was constructed.

## Motor and Handle Development

Design Approach. — The design concept arrived at after the trade off studies required a separate motor and handle to which the attachments could be connected. The motor concept employed was essentially the same as that employed in the early Air Force tool development (1) and in Black and Decker commercial practice. The 12 vdc motor has a permanent magnet field. The silver graphite brushes used were developed by Stackpole for high altitude operation. In earlier tool development they had been tested at a vacuum of $10^{-6}$ torr with negligible wear in a 6-volt motor of the same design. The motor was supported in two ball bearings to isolate torques developed on the field housing from the handle for zero reaction operation. Power connections from the handle trigger switch to the brush assembly is made through a pair of slip rings at the back end of the motor unit. The handle contains the trigger and reversing switch and the power connection. The handle was located under the center of gravity of the combined motor and impactor attachment.

Three major development problems required solution. The handle design required optimizing with the space suit glove, the attachment interface required a simple reliable mechanical connection and the motor performance in a pure oxygen environment had to be investigated to ascertain whether or not any special safety features were required because of potential explosive or contamination hazard.

Glove/Handle Interface Study. — A brief study of the interface between the pressurized space suit glove and the tool handle was conducted for the purpose of determining if a special handle design was required. The following Black & Decker tools (Figure 24) were evaluated for significant features of handles and switches.

Black & Decker Catalog No. (Counterclockwise from the bottom of Figure 24)

| | |
|---|---|
| 681 | Screw drill |
| 624 | Impact Wrench |
| 668 | 1/4 inch Shorty Drill |
| 980 | Polisher with rocker type switch |
| 687 | 1/2 inch Reversing Drill |

The 1/2 inch reversing drill was evaluated with the trigger switch guard both in place and cut off. The method of evaluation was to hold the tools in the normal work position and move them through a range equivalent to task performance. Subjective evaluation by two pressure suited subjects in the Gemini and ILC suits was based upon the feel of control and the ease with which the switches could be operated. The pistol grip handles were clearly preferred because of the good control of switching without a high degree of tactile sensitivity and the relative safety of a trigger switch as compared to the paddle switch on this 1/2 inch shorty drill or the rocker switch on the polisher.

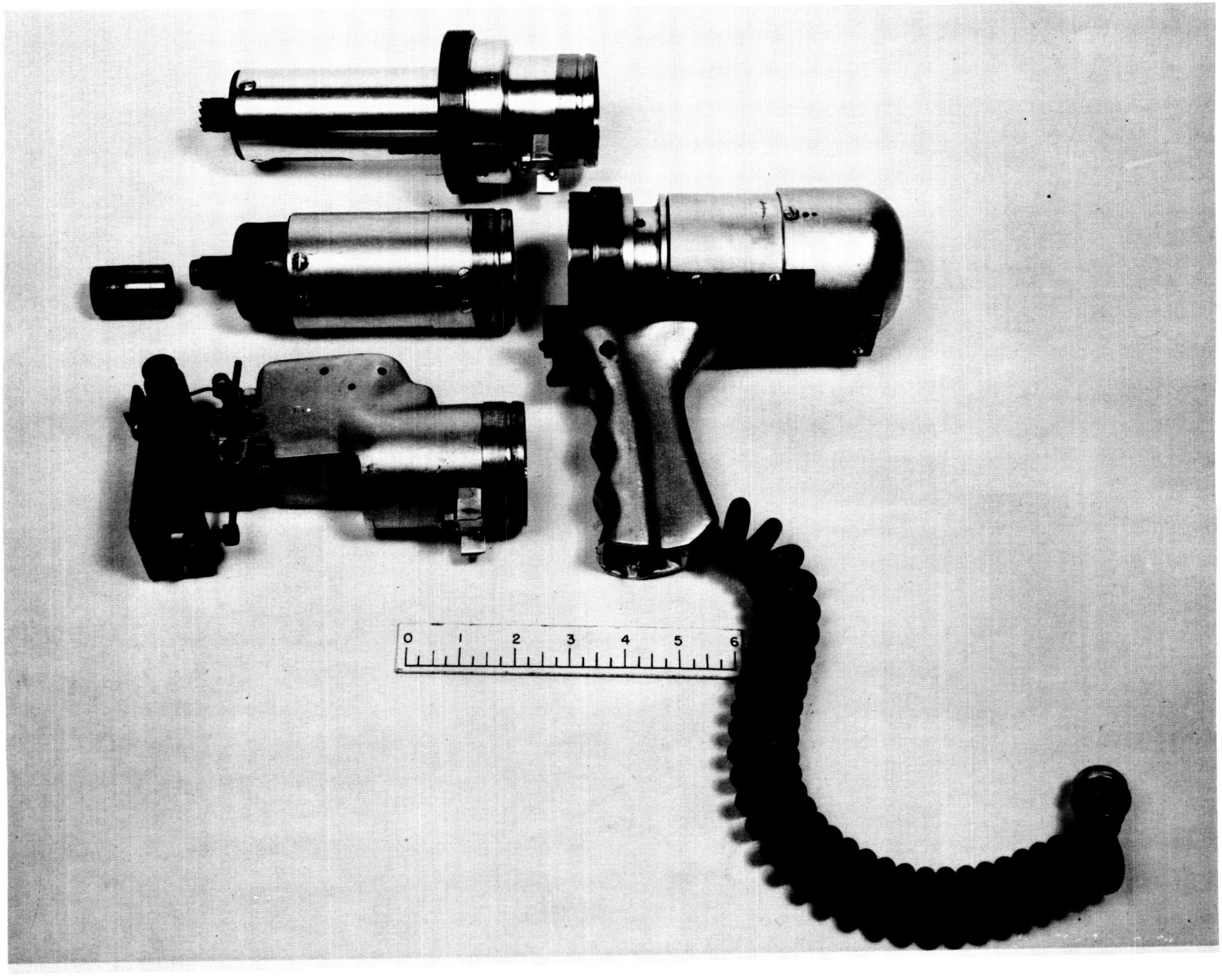



Figure 22. Final Model Power Tool and Attachments

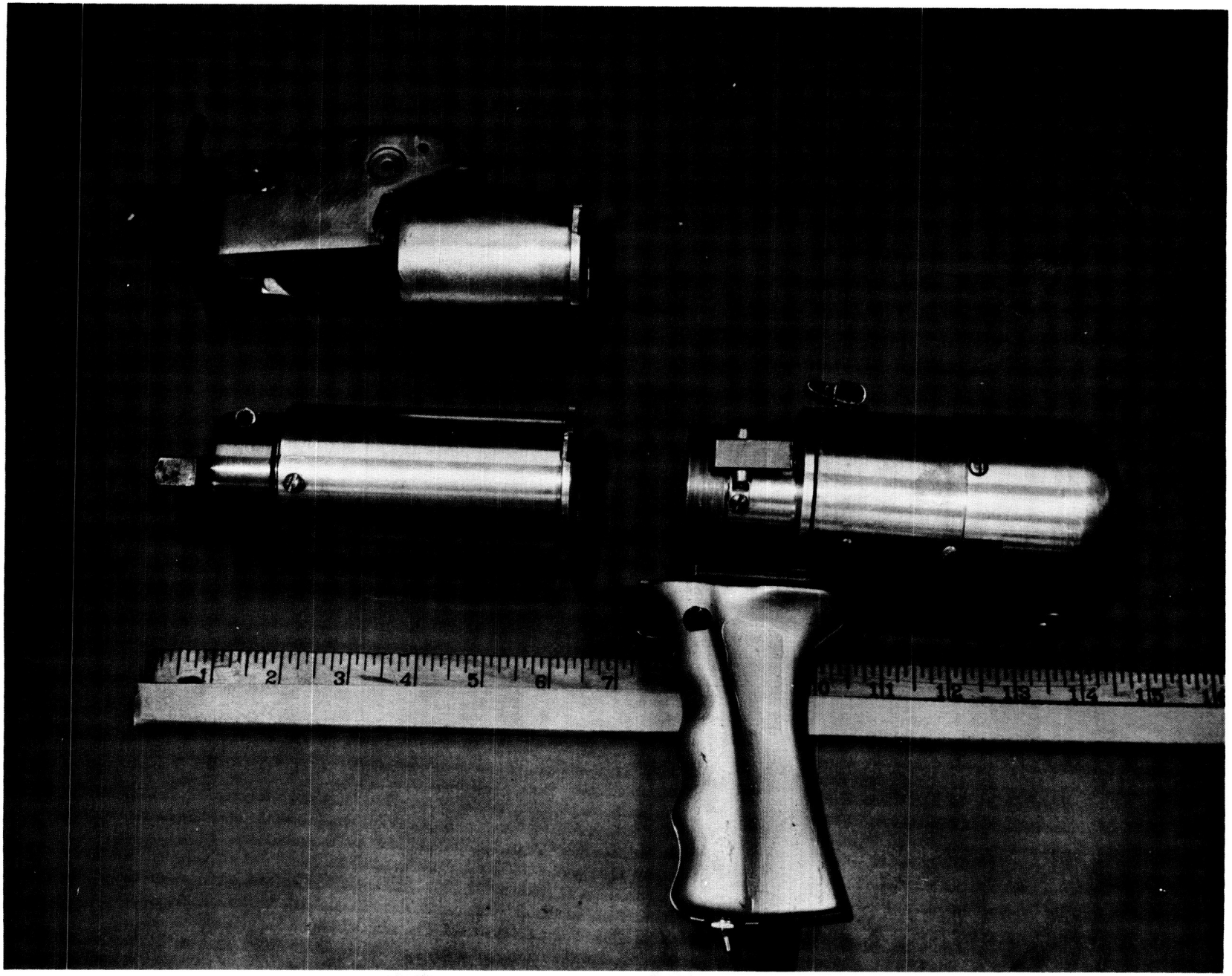


Figure 23. Power Tool Feasibility Model

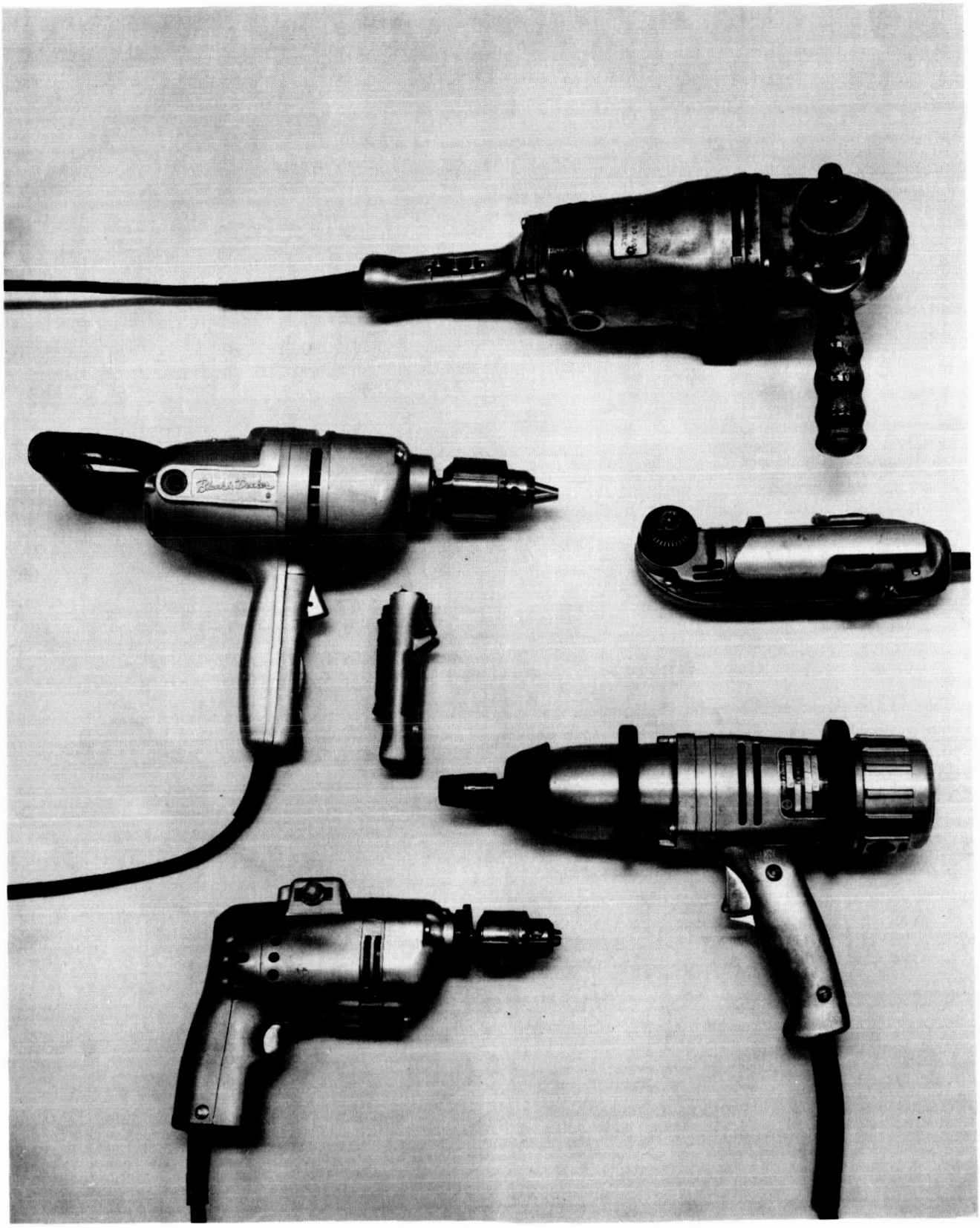
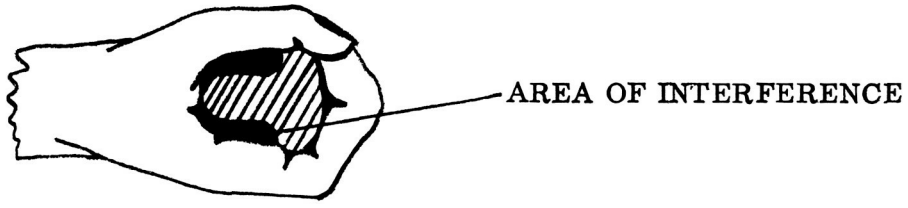


Figure 24. Glove Handle Interface Tools

Difficulty was experienced by both pressurized subjects and by three other subjects wearing only the gloves in grasping the tool with a feel of control. A comparison of the ILC state of the art glove grasp (Figure 25) and the bare hand grasp (Figure 26) shows that the essential elements of a good firm grasp are absent in the gloved situation. Notice in Figure 26 that the fingers come well around the handle to nestle the tool firmly against the palm and the heel of the hand. The thumb is securely holding the handle in the V between the lower joint and the knuckle of the index finger. In figure 25 these characteristics are absent. The glove construction prevents the snug grasp against the palm and as a result the fingers and thumb are not able to close securely and the tool is more free to slide and twist in the grasp. Since a poor grasp would contribute to operator fatigue and reduced precision, a new handle shape was developed to mate with the glove. Use by either hand was considered necessary. The essential feature of the new handle shape was a T cross section so that the glove palm structure did not interfere with the grasping action as shown below.




The narrow end of the T can also fit into the V of the thumb more easily. A pattern of the handle shape was originally developed in clay and after fitting it was cast in plastic to serve as a pattern for the tool handle fabrication.

As a part of the handle evaluation, consideration was given to the location of the reversing switch. Location on the front under the trigger switch was rejected in favor of a switch on the bottom of the handle. The bottom location was easier to reach for two handed use. In the initial handle design the switch was exposed. The human factors design evaluation recommended recessing the switch to prevent accidental tool reversal and this feature was incorporated in the final handle design.

Attachment Interface Development.—The efficiency of the multipurpose concept of one motor and attachments is dependent upon the ease with which the attachment can be installed and removed. Although this concept has been commercially developed, commercial attachments do not usually include an impact wrench with its vibratory forces. The design approach taken was to provide a clutch for power transmission from the motor unit to the attachment so that a simple mating action would connect both the housings and the power transmissions.

The motor speed of approximately 14 K rpm is reduced by a 7.1 planetary gear to 1.8 K rpm at the clutch. The clutch is a tooth type which mates when the attachment housing is fitted to the motor housing. Three forces are encountered by the interface during tool operation: drive torque in the clutch, reaction torque in the housing, and axial thrust. The rotary drive torque is transmitted through the clutch teeth. In the original clutch design 6 equally spaced teeth, each 20 degrees wide, were provided. Because different attachments would be used in sequence, the relative position of motor

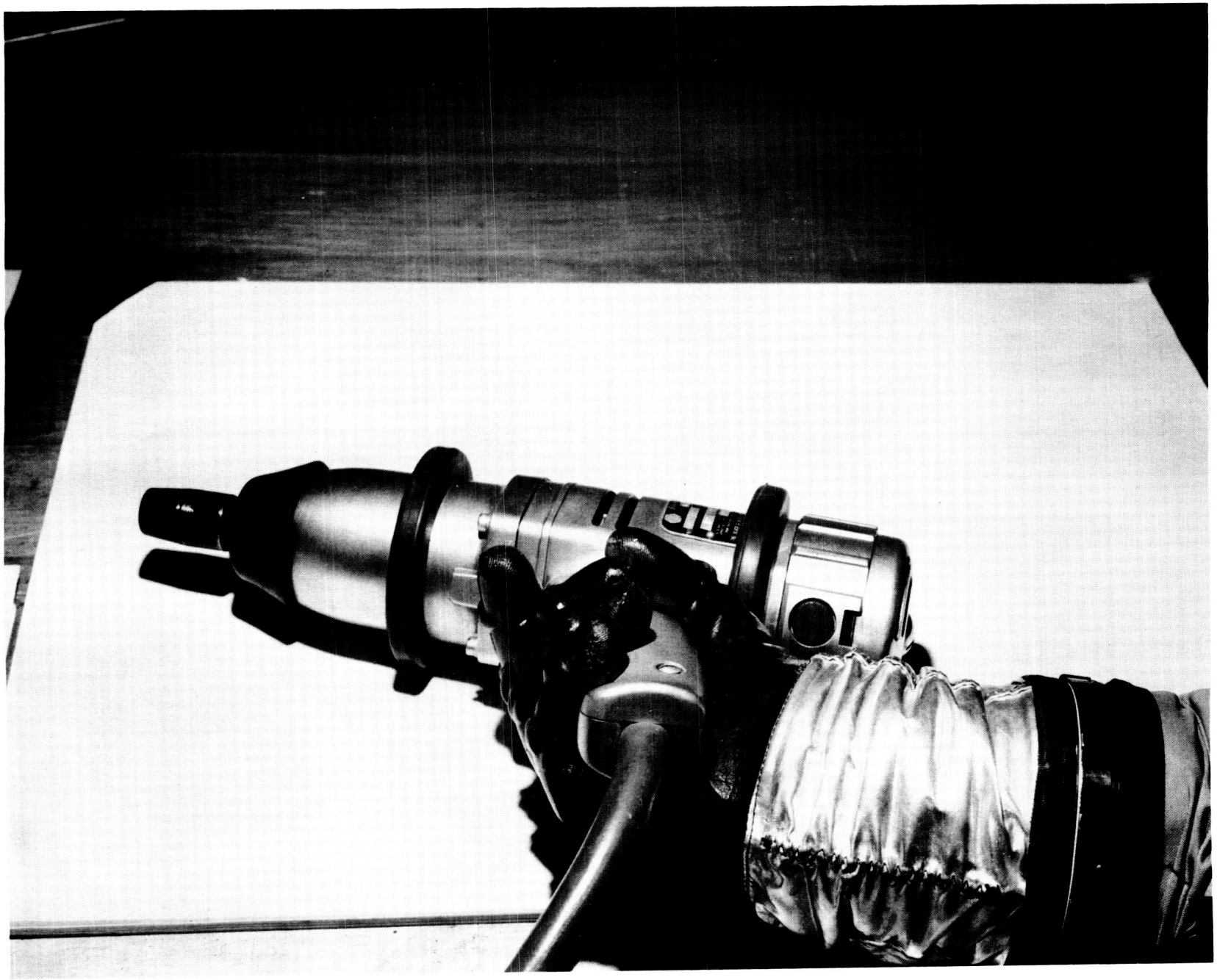


Figure 25. Gloved Hand Grasp

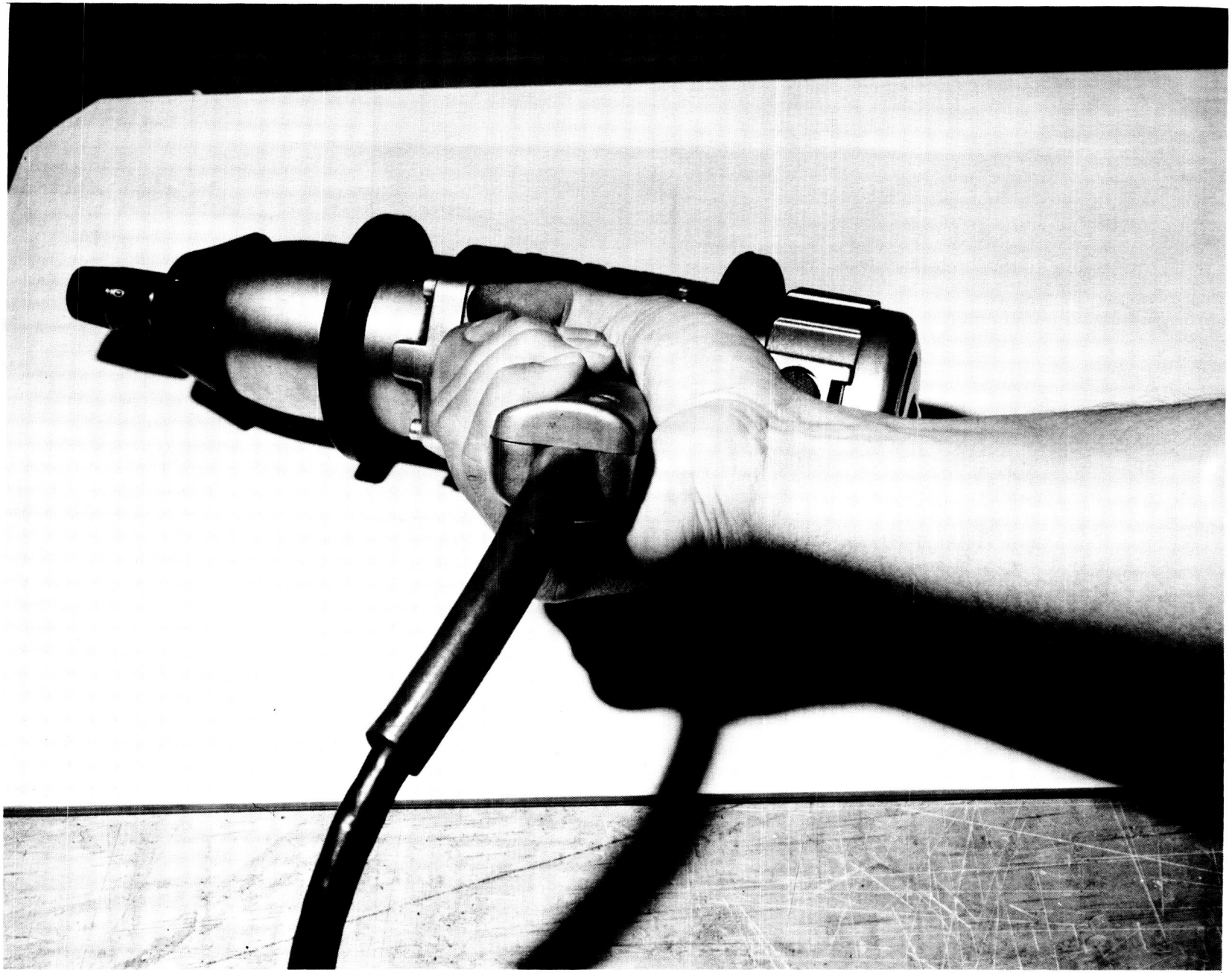
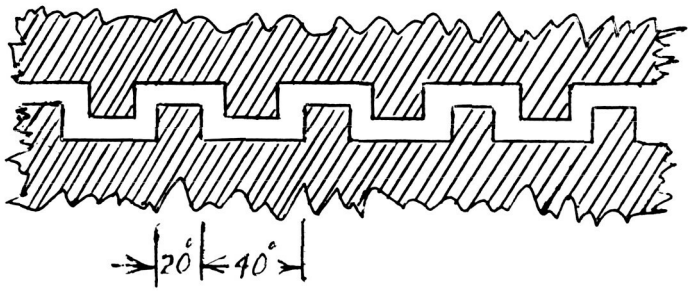
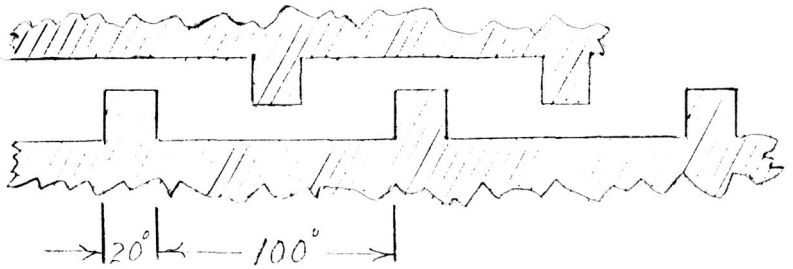


Figure 26. Bare Hand Grasp

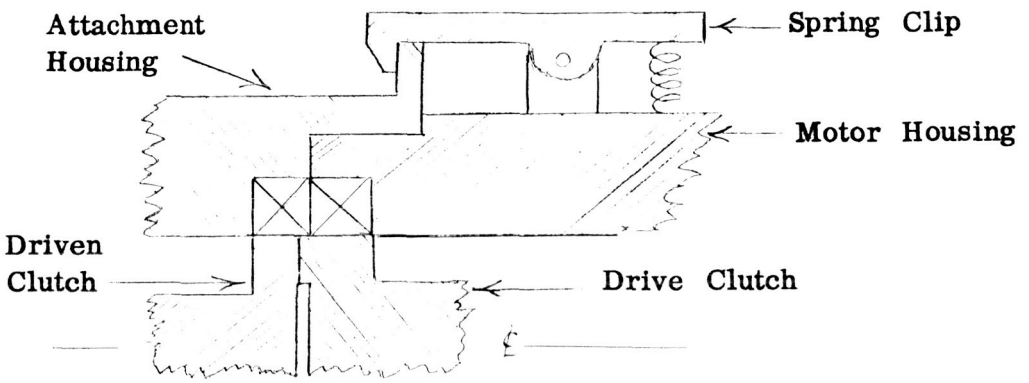
and attachment clutch teeth is randomized and the probability of mating without rotating to clear teeth was .5.




In use this was found to be low so the clutch was changed to 3 teeth, each 20 degrees wide and 120 degrees apart. This increased the probability of mating to .8 which was found satisfactory.



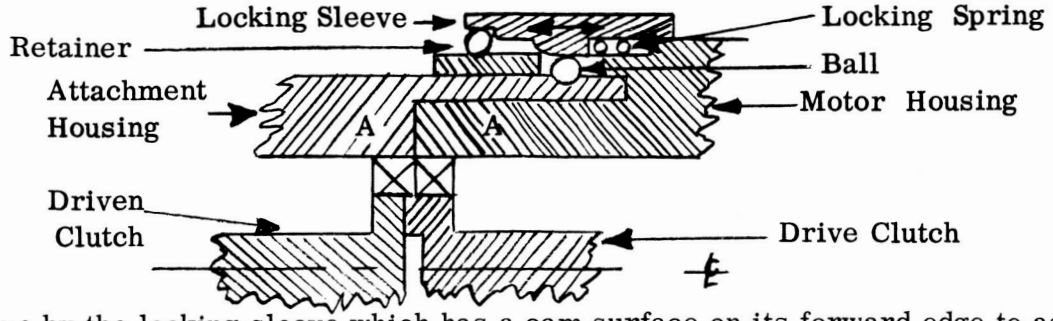

The reaction torque and axial thrust were originally transmitted through the housings by two diametrically opposed spring clips as shown below. The clips fit into notches




in the attachment housing flange so that torques were reacted against the side of the notch. In use this proved unsatisfactory. The spring clips and attachment were aluminum and the vibratory torque of the impactor distorted the notch shape from radial to wedge shaped sides allowing the attachment to force the clips outward and to rotate relative to the motor housing.

In addition the protruding angles and corners of the clips were dangerous to the gloved hand and awkward to handle. Attachment and removal required pre-positioning of the housing so that both clips could be opened simultaneously.

For the final configuration, the interface was redesigned to provide a ball detent which would be operable in any position and which would have no dangerous corners. The ball detent mechanism shown below utilizes two diametrically opposed balls in the motor housing which fit into a groove on the attachment housing. The ball is held in the




groove by the locking sleeve which has a cam surface on its forward edge to assure seating of the ball. The locking spring keeps the detent engaged unless the sleeve is held in the rear position by the operator. During operation it was discovered that the attachment was rotating relative to the motor although it had been anticipated that the friction in the slip fit would be sufficient. Screws were inserted in the housings just outside the OD of the clutch (Point A in the sketch). The screws acted as clutch teeth to transmit the reactive force. Brinelling of the groove in the attachment caused binding which made removal and attachment difficult. The Brinelling was caused by axial vibration of the impactor aggravated by a wedging action of the housing screws. A hardened steel sleeve replaced the aluminum groove and the pan-head screws were replaced with cap screws to provide radial contact surfaces. This was satisfactory for operation for a limited time. After torquing approximately 50 bolts, the screws sheared off. A final fix was incorporated consisting of hardened steel dog teeth set into the motor housing and the impactor housing. The dogs are 30 degree sectors inset 1/8 inch into the basic structure to distribute the impact loads. This fix has proved satisfactory in several hours of running time during oxygen and torque performance testing.

Oxygen Compatability Tests.—As discussed under the Environmental Constraints the possibility of explosion and contamination of cabin atmosphere required definition before a final motor design could be developed. The purpose of the tests was to resolve the need for a sealed or explosion proof motor.

The power tool motor operates on 12 V dc drawing approximately 30 amperes under full load with the impactor attachment. Four test sequences were run, one with a prototype tool and three with the final tool. In addition, a preliminary test was made with a dynamotor with deliberately damaged brushes to maximize arcing.

All tests were conducted in a similar manner to evaluate the explosive hazard. At the ordnance revetment at Martin Company airport, the tool was set up in a bell jar and pumped down to 0.1 psia. Then pure oxygen gas was introduced until the pressure reached 5.0 psia. The tool was run in a short duty cycle which allowed the motor to reach speed or stop and therefore assured maximum starting surge current. After approximately one hour operation the test was stopped and the bell jar set up moved to the chemical lab for gas analysis.

Initially analysis was concentrated upon detection of ozone and a gross indication of the presence of other contaminants by potassium iodide reaction. On the final tool tests an infrared absorption spectrophotometer was used to identify the presence of bonds which could be compared against a table of allowable contaminants, (Table 8), to determine the presence of various contaminants.

TABLE 8

RECOMMENDED MAXIMUM CONCENTRATIONS OF ATMOSPHERIC CONTAMINANTS[10]

| COMPOUND | PPM | $mg/M^3$ (Approx.) | Test Results | | |
|---|---|---|---|---|---|
| | | | Definitely < PPM | Probably < PPM | Possibly > PPM |
| ACETIC ACID | 10 | 25 | | X | |
| ACETONE | 1000 | 2400 | | X | |
| ACROLEIN | 0.5 | 1.2 | | | X |
| AMMONIA | 100 | 70 | | X | |
| AMYLACETATE | 200 | 1050 | X | | |
| AMYL ALCOHOL | 100 | 360 | | X | |
| BENZENE | 25 | 80 | | X | |
| BUTYL CELLOSOLVE | 50 | 240 | | X | |
| CARBON DISULFIDE | 20 | 60 | | X | |
| CARBON MONOXIDE | 100 | 110 | | X | |
| CARBON TETRACHLORIDE | 25 | 160 | | X | |
| CRESOL | 5 | 22 | | X | |
| CYCLOHEXANE | 400 | 1400 | | X | |
| DIOXANE | 100 | 360 | | X | |
| ETHYL ACETATE | 400 | 1400 | | X | |
| ETHYLENE DIAMINE | 10 | 30 | | X | |
| FLUORINE | 0.1 | 0.2 | | | X |
| FORMALDEHYDE | 5 | 6 | X | | |
| HYDRAZINE | 1 | 1.3 | | X | |
| HYDROGEN CHLORIDE | 5 | 7 | | X | |
| HYDROGEN FLUORIDE | 3 | 2 | | X | |
| HYDROGEN PEROXIDE, 90% | 1 | 1.4 | | X | |
| HYDROGEN SULFIDE | 20 | 30 | | X | |
| LITHIUM HYDRIDE | | 0.025 | | | X |
| METHYL ALCOHOL | 200 | 260 | X | | |
| METHYL CELLOSOLVE | 25 | 80 | | X | |
| NITROGEN DIOXIDE | 5 | 9 | | X | |
| METHYL ETHYL KETONE | 200 | 590 | | X | |
| OZONE | 0.1 | 0.2 | | | X |
| PERCHLOROETHYLENE | 100 | 670 | | X | |
| PHENOL | 5 | 19 | | X | |
| PHOSGENE | 1 | 4 | | X | |
| PHOSPHINE | 0.05 | 0.07 | | | X |
| SODIUM HYDROXIDE | | 2 | | | X |
| SULFUR DIOXIDE | 5 | 13 | | X | |
| SULFURIC ACID | | 1 | | | X |
| TEFLON DECOMPOSITION PROD. (AS FLUORINE) | | 0.05 | | | X |
| TOLUENE | 200 | 750 | | X | |
| TRICHLOROETHYLENE | 100 | 520 | | X | |
| XYLENE | 200 | 870 | | X | |

The initial test was run with a dynamotor in the 5 psia $O_2$ environment to evaluate only the explosive hazard. The dynamotor was operated remotely from a 28 V dc source through five start-stop cycles. No explosions occurred, although the motor brushes had been deliberately "hacked up" with a knife to assure arcing.

Following the dynamotor test a prototype tool was operated in a bell jar with a gas volume of approximately 900 cubic inches. Prior to the test the tool was disassembled, cleaned with methyl ethyl keytone, lubricated with GE G-300 silicone grease and reassembled. A taut rubber band was placed in the bell jar to detect the presence of $O_3$. The tool operated at 12 V dc 20 amperes for 57 cycles of 30 seconds on 30 seconds off in the impacting mode. This is equivalent to tightening or loosening fifty-seven 1/2-inch bolts. No explosions occurred, although the commutator brushes were open to the $O_2$ atmosphere. At a concentration greater than .01 ppm cracks will develop in taut rubber[9]. The process is time dependent and in approximately 1 hour no cracks were detected, indicating a concentration of at least less than 1 ppm. Additionally, the gas was sniffed by an experienced chemist.

The chemistry of $O_3$ indicates that little would be detected because, although it may be generated by arcing at the commutator brushes, its reactivity is high and it would immediately react with the hot copper commutator segments.

For detection of other contaminants the gaseous vapors were passed through a scrubbing material which removed any $CO_2$ present in the gas. The resulting mixture was then passed through a platinum catalyst which converted all organic material to $CO_2$. The $CO_2$ was collected and absorbed in the catalyst. The amount of deposit was then determined by the change in weight of the catalyst which occurred during the test. This change in weight was then used to determine an equivalent amount of $CO_2$.

Although this method does not identify the organic materials present, it does provide a basis of determining the probable levels of organic contaminant generated. The specification for breathing "quality" oxygen allows up to 25 ppm of methane ($CH_4$), the most probable organic contaminant in air. Our chemical analysis shows the presence of 60 ppm $\pm$ 20 ppm of contaminant. This figure is probably optimistic due to the possibility of condensation of contaminants and air leakage during the 30 minutes which elapsed between the end of the operational tests and the start of the chemical analysis. If bell jar volume (0.85 $ft^3$) was expanded to a minimum capsule size (approximately 100 $ft^3$) the concentration of contaminants can be reduced to less than 0.6 ppm which is well below the allowable methane level. It is also less than the allowable level for most other organic gases.

For the final tool testing the test set up was the same, (Figure 27). The tool was also cleaned and relubricated with silicone grease. The tool was run for 48 minutes at a duty cycle of 3 seconds on 3 seconds off with 12 V dc 25 amperes power input from the tool kit battery. This is equivalent to a total of 480 bolts tightened in 48 minutes, far in excess of what would be expected in a maintenance situation. During the operating period the motor case temperature rose from ambient to 156° F. No explosions occurred.

The test set up was moved to the chemistry lab for gas analysis, but during the set up, human error destroyed the integrity of the sample and the gas analysis was not made.

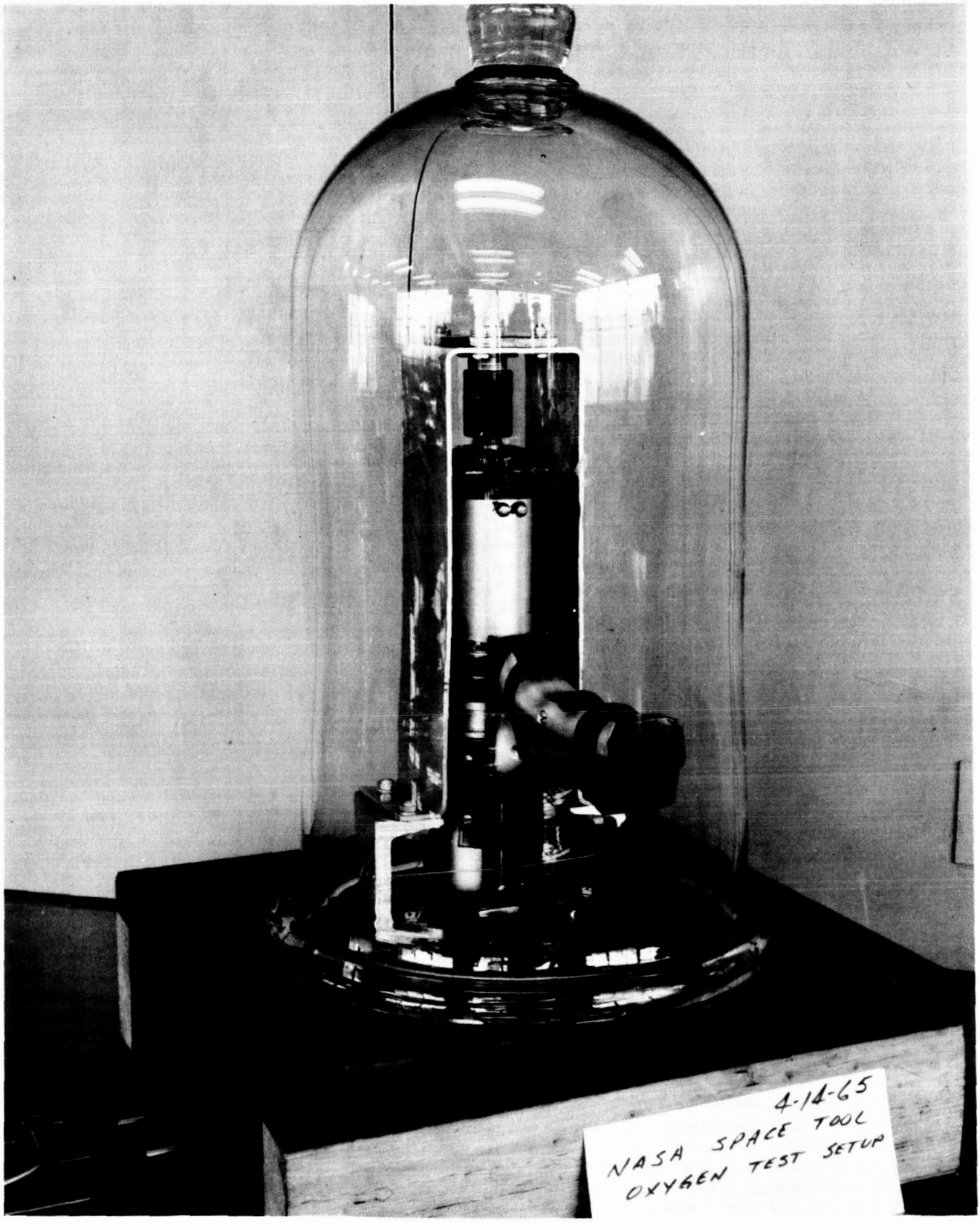



Figure 27. Oxygen Test Set-Up

Examination of the tool indicated no mechanical problems except for an expected high rate of wear of the bronze bushing in the CIR collar. Brush length was measured before and after the test with only .002 in. wear during 24 minutes actual elapsed running time. Since the brushes are .400 long this is considered insignificant.

A second tool test was conducted with the same set up and operating procedure. During this test the tool was operated for 600 cycles and it reached a case temperature of 177° F. After transfer to the chemistry lab the bell jar gas was passed through a 10 meter cell in a Perkin Elmer model 21 infrared spectrophotometer. During the analysis the spectrophotometer calibration was lost and the data was considered invalid. A third attempt was made successfully. The tool was operated for 600 cycles and reached a case temperature of 172°. Again no explosions occurred. For the final gas analysis the curves in Figure 28 show the calibration of background radiation (A) and the tool sample (B). The third curve (C) is the response for 5 cc of ethane injected into the gas cell. This represents a concentration of 1,250 ppm or a detection capability of 25 ppm. Comparison of curves A and B shows significant responses in the 728, 2050-2220, 2850-3200 A° bands. The other indicated responses are attributed to water vapor. Comparison of the responses observed with the characteristic responses of known bonds indicates the definite presence of the water and C-H bonds, namely $CH_2$, $C-NO_2$ or COOH. Possibly present are silicones ($S_1H$) amine ($-NH_2$) amides ($-C^{O}_{\parallel}NH_2$), amine substituted aromatics ($\emptyset-NH_x$) and mono substituted aromatics. Presence of amides and amine substituted aromatics is considered highly probable due to the anticipated presence of ozone. The presence of alcohol (p, s) can be discounted at this point due to the lack of necessary O-H bonds in any discernible quantity.

In applying the existance of these bonds to the compounds listed in Table 8, a number of basic assumptions have necessarily been made in view of the cursory nature of the analysis. It should be noted at this point that it was not the intent to perform a complete quantitative gas analysis at this time, but to perform only those tests necessary to uncover and identify gross problem areas. The detection sensitivity for example has been established, through insertion of 5 cc of ethane, to be 25 ppm. Strictly speaking, this sensitivity is applicable to C-H bonds only. For this quick look data, however, 25 ppm will be assumed to be the detection sensitivity for all bond structures. It is also assumed that ozone is present. This will virtually preclude the presence of the relatively unstable compound, formaldehyde. Lastly, it can be assumed that the volume ratio between the test jar and a typical cabin volume is in the order of 1/250. This has the effect of increasing the detection sensitivity to .1 ppm, thereby discounting the presence of all Table 8 compounds in excessive amounts with the exception of acrolein, fluorine, ozone, phosphine, and traces of lithium hydride, sodium hydroxide, sulfuric acid, and teflon decomposition products. No opinion can be formed regarding the presence or concentration of these compounds without further and more exhaustive testing, calibration and analysis. Such testing is considered beyond the scope of this program, however, techniques are available to perform the analysis and could include the following:

1. Higher resolution infrared analysis through use of a Beckman IR12 spectrophotometer. This instrument is also capable of reaching 50u as opposed to the 14u attainable on the Perkin Elmer Model 21.

2. Use of concentrating cold traps to increase the sensitivity three or more orders of magnitude.

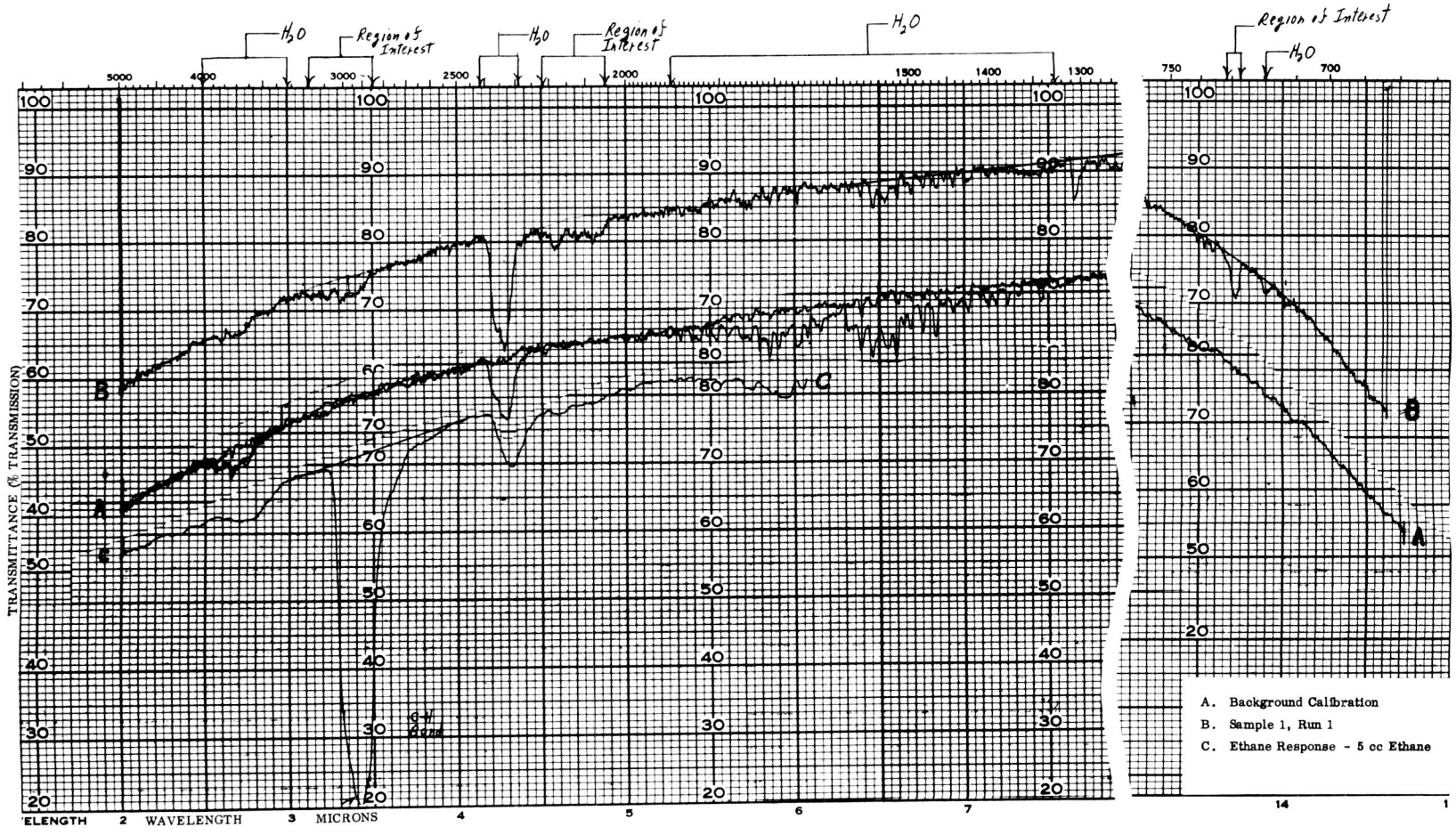



Figure 28. $O_2$ Test IR Response Curves

3. Use alternate analytical methods such as gas chromotography and mass spectrometry.

Generally speaking, the following conclusions may be drawn from the preliminary analysis:

1. Some compounds on the reference list are not present above the specified limit.

2. Some compounds on the list may be a threat. Further testing is required.

3. No compounds on the list have been shown conclusively to be a threat even though the tool was operated much more extensively (600 cycles - 30 minute run time) than any anticipated requirement for lunar or orbital missions.

4. The explosion hazard from the 120 V dc motor is minimal. 1,737 on-off cycles in 5 psia $O_2$ at starting surge currents of approximately 60 amperes is far in excess of what would be expected for normal use.

## Impactor Development

The impactor mechanism used in the power tool is an adaptation of the Black and Decker model 624 impact wrench. The initial development for zero-reaction capability was accomplished under USAF auspices and is reported in reference 1. A detailed analysis of the operation of the mechanism is contained in reference 1. For clarity and continuity this report will describe the basic operating principles and the significant new development areas which were a part of this program.

Theory of Operation.—Figure 29 is an engineering assembly drawing of the impact mechanism and the motor and handle assembly. The separation plane is just above the trigger switch $^{-37}$. The clutches $^{-2}$ transmit the motor torque to the impactor assembly. The basic principle of operation is a repeating cycle of compressing and releasing a spring $^{-117}$ which accelerates the impact mass$^{-115}$. The mass impacts at the end of its travel against the output spindle$^{-102}$ delivering a sharp torque impulse. After impact the spring is again cocked and released and the impact repeats at about 3 K impacts per minute.

The cocking action occurs when the impact barrel $^{-115}$ cannot rotate because the hammer dogs in the front end are engaged with the anvil dogs on the spindle $^{-102}$ and the anvil is held by the controlled internal restraint collar $^{-106}$ (CIR). Although the impactor $^{-115}$ cannot rotate, it is free to slide axially and the cocking cam $^{-122}$ and balls $^{-116}$, which are rotated by the clutch torque, move the impact mass $^{-115}$ to the rear, storing energy in the spring $^{-117}$. At the end of rearward travel, the dogs disengage and the impact mass $^{-115}$ is free. The spring force causes the impact mass to rotate forward following the cam surface for 180 degrees, at which time the hammer dogs impact against the spindle anvil. The momentum of the impacting mass is transferred to the output spindle by the blow, through the spindle to the socket and to the fastener.

The zero-reaction capability is achieved by providing a closed force path between the motor and impactor cam so that no handle torque is developed. Drive torque is developed in the motor by the magnetic circuit of the field and armature. Torques in opposite directions are developed on the armature$^{-36}$ and the field housing$^{-20}$. This

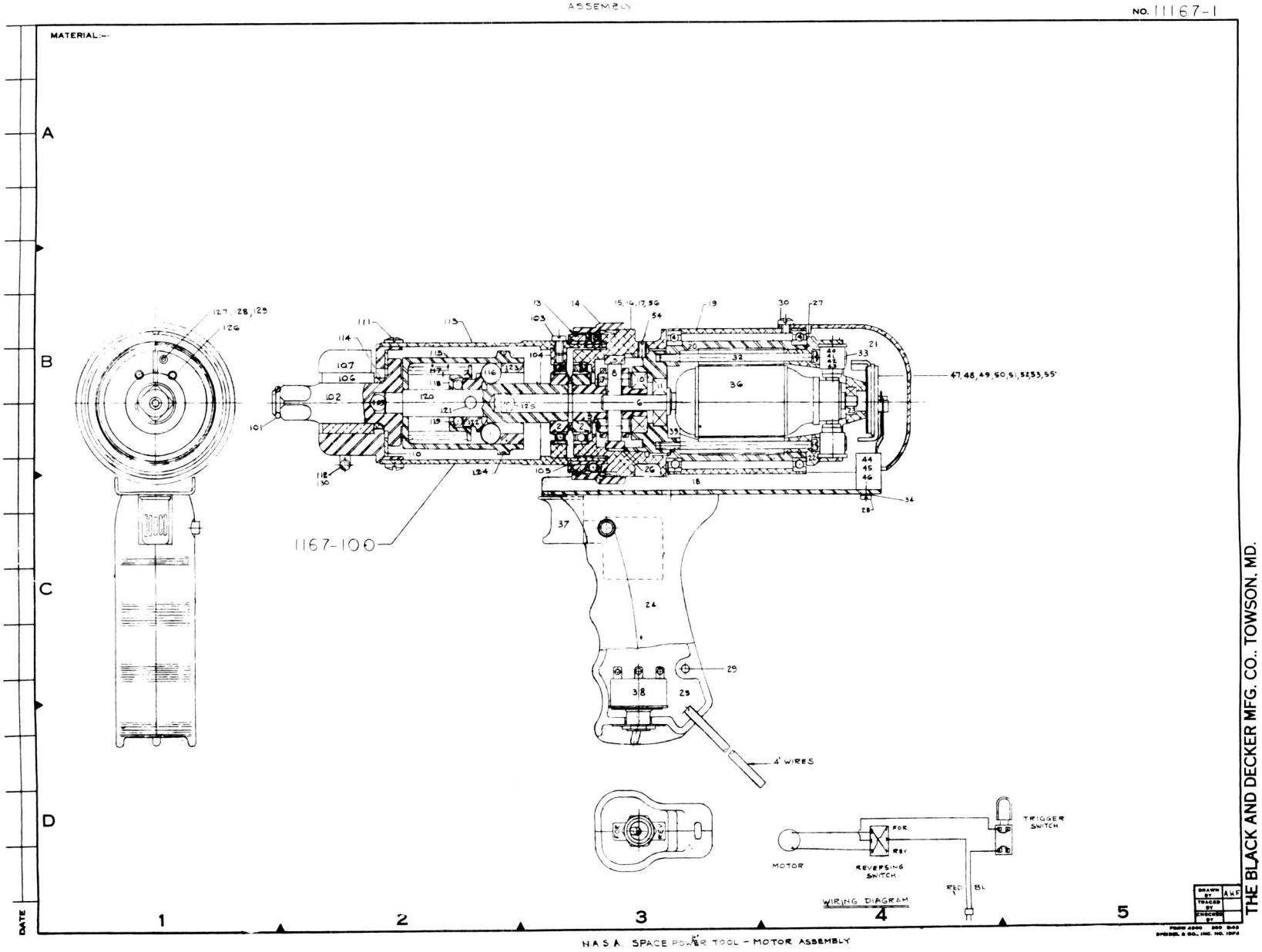
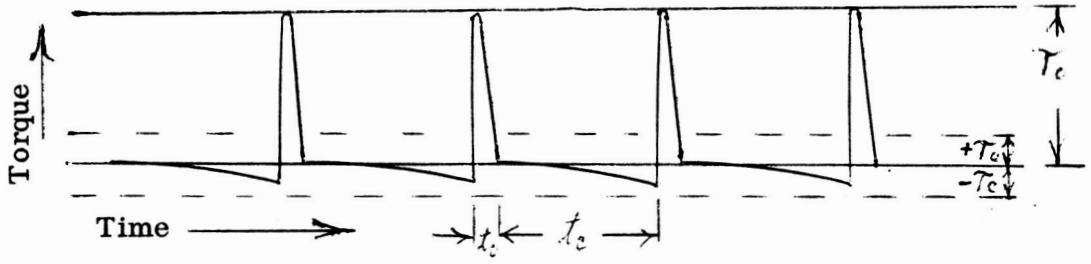



Figure 29. Impactor Mechanism

armature torque is carried through the reduction gears -6, 7, 8, 9 to the clutch-2 and through the clutch to the cam shaft-120. The field torque is carried through the motor housing assembly -20, 15, 16, 17 to the impactor housing -104, 113 so that the camshaft-120 and the barrel-113 are torqued in opposite directions. No torque except bearing drag is felt on the handle.

To cock the spring, it is necessary to connect the barrel-113 to the spindle-102 so that the cocking action will be as discribed above. However, to achieve a torque output, it is necessary for the spindle to rotate relative to the barrel. The frictional characteristics of the contact between the steel spindle-102 and the brass collar-106 are utilized. In cocking the mechanism the torque forces are a relatively low 30 inch-pound while impact of the impactor on the spindle anvil creates torques of about 85 foot-pounds. The collar pressure is adjusted by the clamp-107 which is attached to the barrel-113 so that the force required to cause relative motion of the spindle-102 in the collar-106 is slightly more than 30 inch-pounds. Under these conditions the cocking force is transmitted through the CIR collar to the spindle causing the cam-122 to move the impactor-115 axially compressing the spring-117. When the impactor dogs release and the impactor rotates and impacts against the spindle anvil, the energy of the blow is sufficient to create a sharp torque pulse of 85-foot-pounds. Since this is greater than the frictional force of the collar-106 on the spindle-102 the impact causes rotation. A time base diagram of forces acting on the spindle is shown below.




$T_o$ is the peak amplitude of the output pulse and is consequently the maximum possible torque. $T_o$ is the torque required to cause relative motion of the collar and spindle. The positive peaks occur when the impactor strikes the anvil. The negative torque between pulses is the torque transmitted by the spindle to cock the spring. The energy stored in the spring during time $t_c$ is released during $t_o$. Since the times are unequal the force amplitudes are also different. The useable output is that portion of the output pulse above the level of $T_c$.

Development Areas.—Since the basic zero-reaction impactor was already developed, the areas of interest in this program were development of (1) a higher output of at least 75 foot-pounds, (2) an attachment interface permitting use of the motor for other purposes and (3) investigation of a method for controlling output torque.

The output torque was increased primarily as a function of motor power. With a 12 V dc motor, more power is available to compress the spring so that more impact energy is available per blow. During development testing of the impact mechanism three different springs were tested with spring constants of 100, 65 and 34 pounds per inch. The output torque on a 1/2-inch bolt and the input power were measured. Output torques of 80 to 100 foot-pounds were recorded with all springs; however, the power consumption with the 100 pound-inch

spring was about 75% higher than the 34 pound-per-inch spring for the same output. Because the power efficiency was considered significant for battery selection, the low rate spring was selected.

The attachment interface was developed along with the motor and handle and has been described in that section. The major developmental problem encountered was the transmission of torque impulses reflected from the impactor mechanism to the motor. The impact attachment controlled the design of the other attachments in this area.

Investigation of control of output torque was made during the testing of the initial feasibility models of the impact mechanism. The CIR torque forces were varied by adjusting the clamping force. The clamping force was adjusted by a cap screw in piece 107 which spreads the collar to release the clamping force. The screw was adjusted to set the torque required to turn the spindle, as measured by a torque wrench, to a given value. After the CIR torque was set, 1/2 inch bolts were tightened and the torques measured

Theoretically the variations in the value of $T_c$ should have two effects. One is to increase the torque level at which the output becomes useable and the other is to increase the width of the output pulse, thereby decreasing the peak amplitude since the area under the curve is constant. The tests indicated that it was feasible to vary CIR torque to obtain a difference in torque output. Problems were encountered in setting the torque value of the CIR and in maintaining the value during use. The adjustment was modified by adding a locking screw and nylon plug to the CIR collar to keep the tool vibration from causing the screw to back out and thereby increase the CIR torque. To set the torque, a lever was added to the CIR collar to turn the adjustment screw (Figure 30). The pitch of a 1/4-28 thread on the screw moves the collar jaws .036 inch per revolution or .0001 inch per degree. A total swing of 15 degrees would give the CIR collar a travel of .0015 which initial work had indicated was the maximum required for adjustment from full CIR clamp to no CIR clamp. Setting of the CIR adjustment would require adjustment of the lever to either forward, middle or back position for an approximate torque setting to high middle or low torque values.

Test. —The impactor mechanism for both feasibility and final models was the same with the exception of the interface attachment and the CIR adjustment. The impactor was tested for torque output on a series of bolts tightened into a hardened steel plate. Cadmium plated bolts and dry holes were used to minimize the disturbing effect of lubrication. The results are shown in Table 9. The first set of figures show the torque values averaged over 10 bolts of the size listed. The second set shows the variation in torque with time for a 1/2 inch bolt. The shortest time is about the minimum possible. The CIR setting for these tests was held constant at 30 inch-pounds.

Handle reaction torque was measured in two ways. In the first the tool was clamped vertically by the output spindle and power was applied through flexible leads to the motor slip ring brushes. Measurements were made of the force necessary to move the handle relative to the barrel with the motor off and of the force necessary to restrain the handle with the motor on. The second method was to hold the handle in the vise and with a fine thread wrapped around the barrel and a 0 - 8 ounce spring scale to measure the force required to restrain the barrel rotation. The average of all values was 5.6 inch-ounces. Measurements made with the tool impacting are more sifnificant because the vibration significantly reduces the effective friction of the housing bearings.

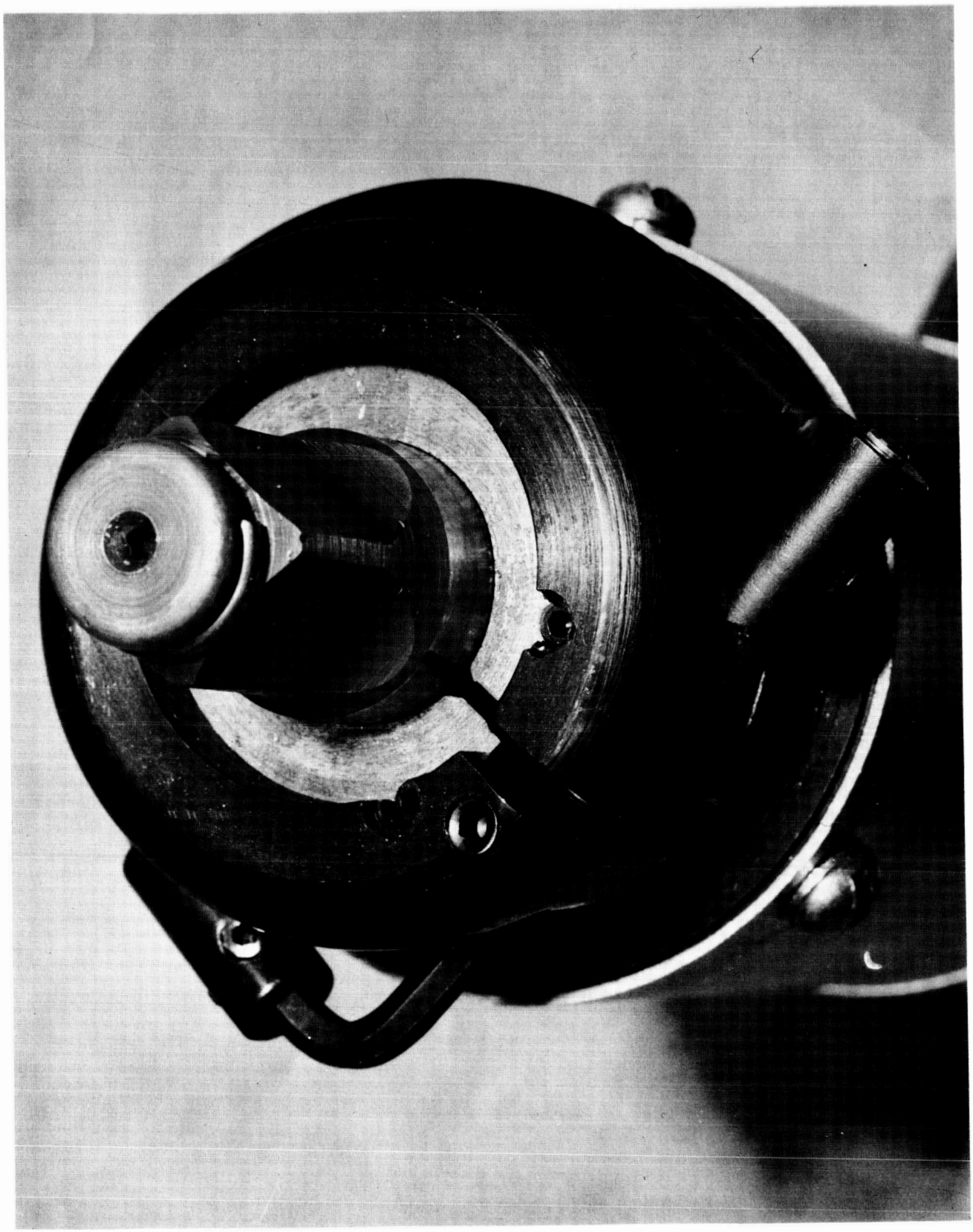


Figure 30. CIR Adjustment Lever

TABLE 9

IMPACTOR PERFORMANCE

| Fastener Size | Torque | Power Input | Time |
|---|---|---|---|
| 1/2-inch | 85.0 ft/lb (10) | 12V 30A | 3 seconds |
| 3/8-inch | 49.0 ft/lb (10) | 12V 25A | 3 seconds |
| 5/16-inch | 34.2 ft/lb (10) | 12V 25A | 3 seconds |
| 1/2-inch | 85.0 ft/lb | 12V 25A | 8 seconds |
| | 82.0 ft/lb | 12V 25A | 2 seconds |
| | 75.0 ft/lb | 12V 25A | 1 second |
| | 62.0 ft/lb | 12V 25A | 0.5 second |

Handle Reaction Torque - 5.6 in/oz.

The CIR adjustment was tested to determine the adequacy of the adjustment technique. The variations in CIR clamp forces were highly erratic and difficult to control. Repeatability of the setting was almost impossible to accomplish without a torque wrench measurement on the spindle. An attempt to plot the curve of CIR torque versus locking lever angle was abandoned because the settings were not repeatable. Analysis of the problem indicates that the difficulty lies in the contact area of the brass collar and the fact that a few thousandths of an inch difference in collar spread changes the concentricity and eccentricity of the spindle shaft and collar. It is now felt that this approach has inherent difficulties in repeatability and a more fruitful approach will be to vary the energy stored in the spring and imparted to the impact mass. The locking lever was removed from the final model.

## Saw Development

In order to develop the saw attachment certain basic data were required on saw forces and power requirements. A series of tests was run to measure the basic sawing relationships. The test set up consisted of a Black and Decker drill stand modified to hold a Model U-351 jig saw. A large diameter pully was mounted on the top and a cable attached to the holding fixture was run over the pully to a counter weight. The counter weight was varied to provide a net downward force against the work of 3 to 8 pounds. Table 10 shows the test results for a series of blades cutting .028 inch thick titanium and Aluminum .062 and .135 inch thick. The purpose of the tests was to determine an

## TABLE 10

## SUMMARY DATA - SAW CUTTING

SUMMARY DATA - JIG SAW CUTTING - TITANIUM GRADE C120A .028" THK & ALUMINUM 7075-T6 (.062 & .135" THK)

| TEMPER-ST | CUTTING RATE IN/MIN | POWER INPUT WATTS | BLADE SPEED FT/MIN | TOOL PRESSURE LBS | | TEMPER STA | BLADE NO. | CUTTING RATE IN/MIN | POWER INPUT WATTS | BLADE SPEED FT/MIN | TOOL PRESS. LBS |
|---|---|---|---|---|---|---|---|---|---|---|---|
| BLADE NO. 2 | 1.71 | 67.6 | 174.0 | 3 | | | 30 | 3.02 | 93.1 | 211.5 | 5 |
| 3 | 2.50 | 72.4 | 164.0 | 4 | | | 31 | 3.31 | 100.0 | 206 | 6 |
| 4 | 2.67 | 77.6 | 153.0 | 5 | | | 40 | 3.23 | 104.9 | 200 | 7 |
| 5 | .39 | — | | 2 | | | 33 | 3.02 | 115.3 | 248 | 5 |
| 6 | .91 | 84.0 | 232.0 | 3 | | | 34 | 3.59 | 121.5 | 232 | 6 |
| 7 | 2.78 | 90.5 | 218.0 | 4 | | | 36 | 3.24 | 118.3 | 234 | 7 |
| 8 | 4.36 | 97.5 | 210.0 | 5 | | | 44 | 4.90 | 136.2 | 214 | 8 |
| 9 | .44 | 40.0 | — | 2 | | | 37 | 2.49 | 83.7 | 177 | 5 |
| 10 | 1.62 | 100.5 | — | 3 | | | 38 | 3.52 | 89.3 | 172 | 6 |
| 11 | 1.56 | 98.5 | — | 4 | | | 39 | 3.52 | 92.4 | 164 | 7 |
| 13 | 3.38 | 109.7 | 251 | 4 | | | 41 | 2.84 | 133 | 238 | 5 |
| 14 | 4.49 | 127.2 | 226 | 5 | | | 42 | 3.97 | 146 | 231 | 6 |
| 15 | 5.76 | 128.7 | 223 | 6 | | | 43 | 6.05 | 161 | 218 | 7 |
| 16 | 8.94 | 139.5 | 211 | 7 | | — JIG SAW | CUTTING ALUMINUM | | | | |
| 17 | 3.51 | 137.1 | 290 | 4 | | THK. INCHES | BLADE TYPE-No. | | | | |
| 18 | 3.77 | 145.0 | 290 | 5 | | .062 | 51 | 2.26 | 93.6 | 220 | 3 |
| 19 | 5.32 | 147.5 | 282 | 6 | | .062 | 52 | 2.74 | 100.0 | 208 | 4 |
| 20 | 6.41 | 161.0 | 268 | 7 | | .062 | 53 | 4.59 | 105.0 | 200 | 5 |
| 21 | 3.59 | 151.2 | 324 | 4 | | .062 | 54 | 5.20 | 110.0 | 194 | 6 |
| 22 | 4.19 | 159.2 | 317 | 5 | | .135 | 55 | .59 | 92.0 | 224 | 3 |
| 23 | 4.52 | 165.0 | 313 | 6 | | .135 | 56 | .98 | 88.0 | 218 | 4 |
| 24 | 7.50 | 181.0 | 288 | 7 | | .135 | 57 | 1.35 | 102.0 | 200 | 5 |
| 27 | 9.38 | 116.4 | 188 | 7 | | .135 | 58 | 1.24 | 102.0 | 198 | 6 |
| 28 | 5.21 | 104.8 | 202 | 6 | | .062 | 59 | 6.87 | 105.0 | 200 | 4 |
| 48 | 5.14 | 118 | 178 | 8 | | .062 | 60 | 9.90 | 114.0 | 188 | 5 |
| 49 | 5.24 | 130 | 218 | 8 | | .062 | 61 | 11.1 | 120.0 | 167 | 6 |
| | | | | | | .062 | 62 | 9.48 | 127 | 174 | 7 |
| | | | | | | .135 | 63 | 1.37 | 102 | 202 | 4 |
| | | | | | | .135 | 64 | 1.85 | 109 | 197 | 5 |
| | | | | | | .135 | 65 | 2.62 | 115 | 190 | 6 |
| | | | | | | .135 | 66 | 3.86 | 122 | 169 | 7 |

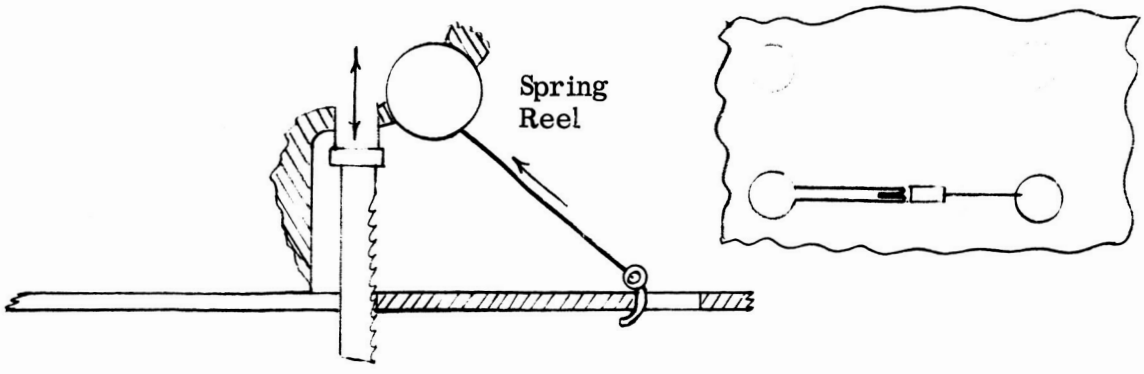
optimum cutting pressure for the saw. The materials selected for test were felt to span the range of sawing problems. The summary data in Table 10 shows that the cutting rate is a function of power input, blade speed and cutting pressure.

A design value of five to seven pounds was selected as optimum. Eight pounds produced rapid dulling of the blade and less than five pounds was too low a cutting rate. The possible concepts for zero-reaction sawing were evaluated against their capability to provide the tool pressures, and the mechanization in the simplest manner. The spring hold down and feed concept consists of a short cable on a spring loaded reel attached to the saw mechanism just ahead of the blade (Figure 31, A). Holes would be drilled at each end of the desired cut and the saw inserted in one. The cable would be stretched out and hooked in the other and when the saw was turned on, then it would be pulled toward the other hole. Difficulties anticipated with this approach were that cutting out areas in a skin would require drilling several holes and the hold down force would vary as a function of the angle of the cable to the surface, being low at the start. This would cause trouble with saw vibration. The major advantage would be that it would be a very simple mechanism. A simple test was made with a standard jig saw pulled with a cable and the difficulty of holding the saw on the work was confirmed. In one instance the blade caught on the edge of the kerf and bounced the saw out of the work. This approach was abandoned.

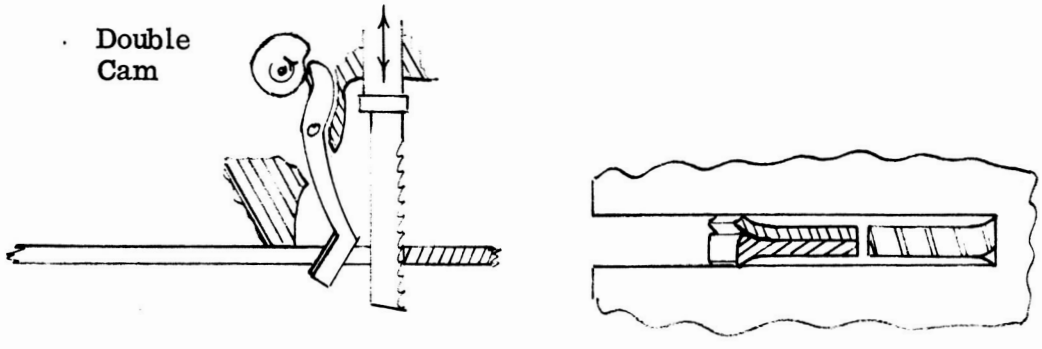
A mechanism which would hold the saw by gripping the sides of the saw kerf was considered to offer advantages of (1) not limiting the material thickness being cut, (2) accepting variable thickness is one cut and (3) being simple to start in the work. The disadvantages were that the problem of holding the saw in the kerf was considered risky because the kerf is normally rough on the edges and the saw would be difficult to start in the work because a minimum cut would be necessary before the zero-reaction feature would take hold. Two possible mechanisms were considered (Figure 31, B and C). One required a reciprocating action of a set of pusher blades which would fit behind the saw blade and the other required a single thread screw in the kerf behind the saw blade.

The pusher blade concept had a set of two blades constructed so that a relative motion between the blades along the line of the cut caused them to wedge into the kerf. Once wedged they could then react the forces necessary to provide hold down and feed pressure. The blades would be actuated by a cam arrangement to control the timing of each action. The difficulties inherent in this approach were two fold. The holding strength possible with this mechanism was dependent upon the stiffness of the material being cut and the smoothness of the walls of the saw kerf and the mechanism would be released and not holding during part of each advance cycle. The first would be a problem in thin material, which could buckle and the grip would be lost and the second could be a problem because the grip would have to be maintained on a cutting up stroke and released on a down stroke with the possibility that the saw blade could catch on the material and bounce the saw out of the kerf.

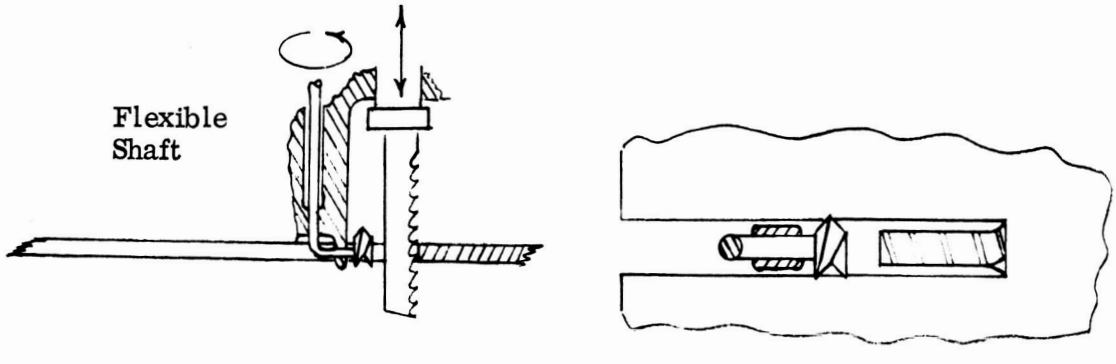
The screw behind the blade suffers from the same problems with the additional one of requiring watchmaking precision to fit the screw into an .030 inch wide kerf. The approach chosen was holding on the blind side with a top side feed. With this approach the work would be gripped continuously and feed pressure applied without destroying the saw kerf or releasing the grip on each stroke. The concept is limited to a maximum thickness of material to be gripped and to a maximum rate of change of thickness per inch of travel but it is inherently better than the other approaches. The concept went through two stages of hardware refinement.




A




B




C

FIGURE 31. ZERO REACTION SAW CONCEPTS

Theory of Operation. —The basic mechanism contained in the housing consists of a mating clutch and a bevel gear to transmit power to the blade drive crank. The drive crank provides a simple harmonic motion to the saw blade shaft causing the blade to move over a 5/8 inch stroke. As it moves a saddle and guide pins give the blade a slight rotation so that an epicyclic path is followed by the saw teeth moving them into the work as they cut. This portion of the mechanism is adapted from the standard Black and Decker model U-231 jig saw. Emphasis in the program was placed upon the development of the zero-reaction feature. Figure 32 shows the original zero-reaction configuration.

The standard saw shoe plate has been shortened to accomodate the hold down and drive mechanism. The drive mechanism utilizes two Torrington needle clutches with knurled steel tires pressed on the outside. These clutches are mounted on a shaft in front of the saw blade (Figure 33). Connected between the shaft and the saw blade holder are two "S" shapped springs (Figure 32). The clutches are mounted so that torque on the shaft, in the counterclockwise direction as you look at (Figure 32), engages the clutches and drives the mechanism to the left. Torque in the opposite direction disengages the clutches and allows the shaft to rotate freely clockwise. The spring is set so that its free position occurs at the bottom of the blade stroke. On each stroke the spring is stretched upward applying a counterclockwise torque to the shaft and through the clutches to the drive tires. With the drive wheels held against the work surface this causes a driving force which keeps the blade in contact with the work. The drive wheels are held against the work surface by the structure which is pivoted on the shoe support. At the rear is a pressure cam and spring. A quarter turn on the cam provides a downward force on the drive wheels. In the photograph, the shoe is lifted at the front end, just behind the drive wheels, because the hold down pressure is not applied by the hold down foot. The hold down foot fits under the work and encloses the blade. The cross section immediately behind the blade is .030 in thick so that it fits through the saw kerf just behind the blade. In operation the hold down cam is released so that the shoe fits flush on the work surface. The hold down foot is pulled up snug under the work and locked with the lever just behind the spring and then the cam pressure is applied so that the work is squeezed between the hold down foot and the drive wheels. Because of the clutches in the drive wheels only the spring torque which will drive the mechanism forward is seen at the wheels and feed pressure is applied while the mechanism is held on the work. Although the angular displacement of the shaft over a full stroke is only about 5 degrees, the clutches have effectively no backlash and feed pressure is maintained. Maximum feed rate would be 1800 strokes per minute at 5 degrees per stroke or 25 wheel revolutions per minute. With a .75 inch diameter wheel, this is (2.35 x 25) 58.75 inch per minute.

Test. —The original saw configuration was tested in .020 Aluminum. Sawing rates of approximately 12 inch per minute were achieved. The saw was inserted in a 5/8 inch diameter hole and the cut was made in a straight line to a hole approximately 6 inches away. Because the hold down pressure was applied at a point approximately 1/4 inch ahead of the blade, the last 1/4 inch of cut was not completed. A change in the hold down shoe was defined for the final configuration. All cuts were made with the work horizontal and the saw resting on the work. Power was applied and the saw allowed to operate hands off. Hold down and feed were accomplished solely by the saw mechanism. For the final design configuration delivered to Martin by Black and Decker a modified hold down foot, shown in Figure 32, was used. Tests were made of saw operation in a range of metal thicknesses.

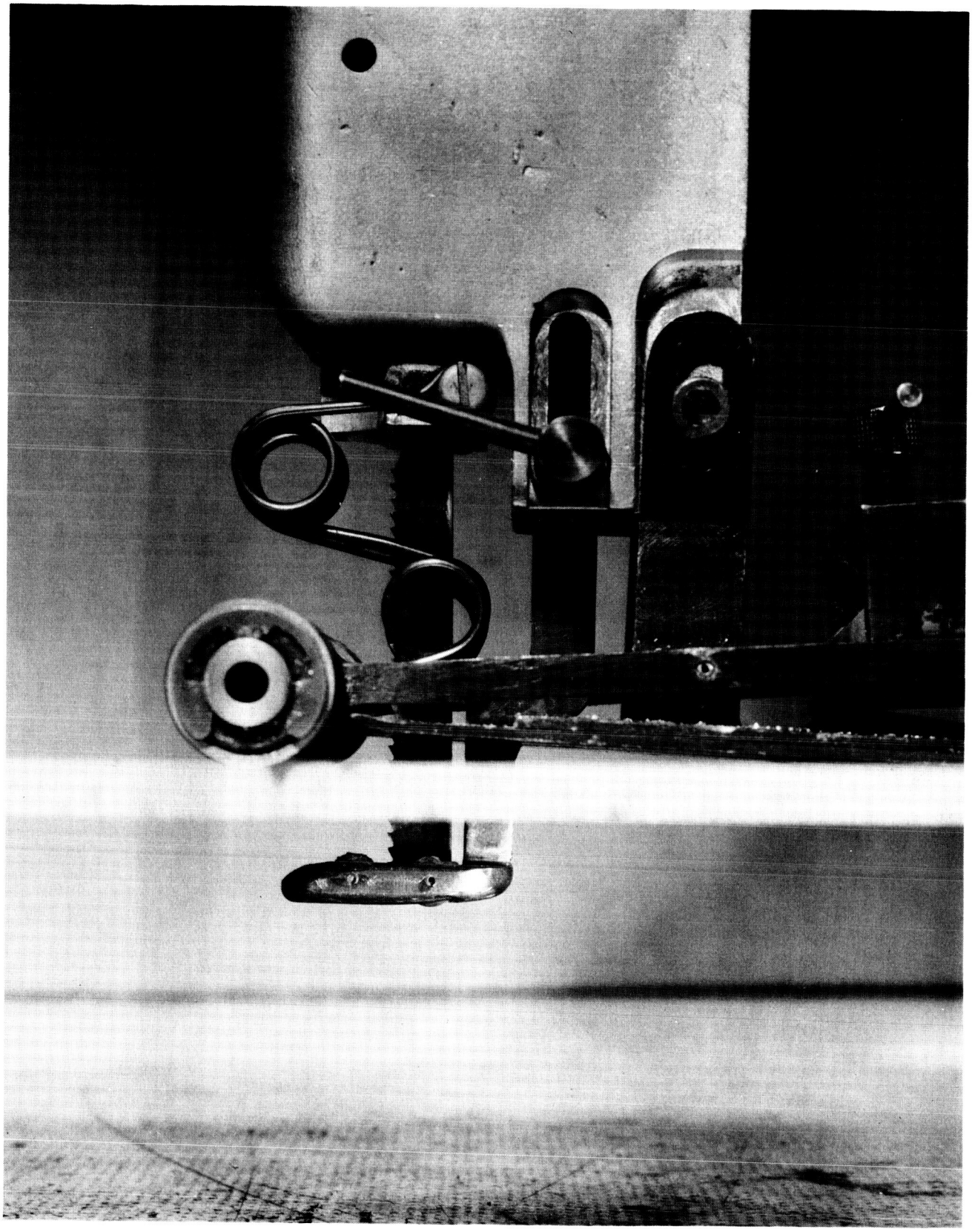


Figure 32. Saw Attachment - Side View

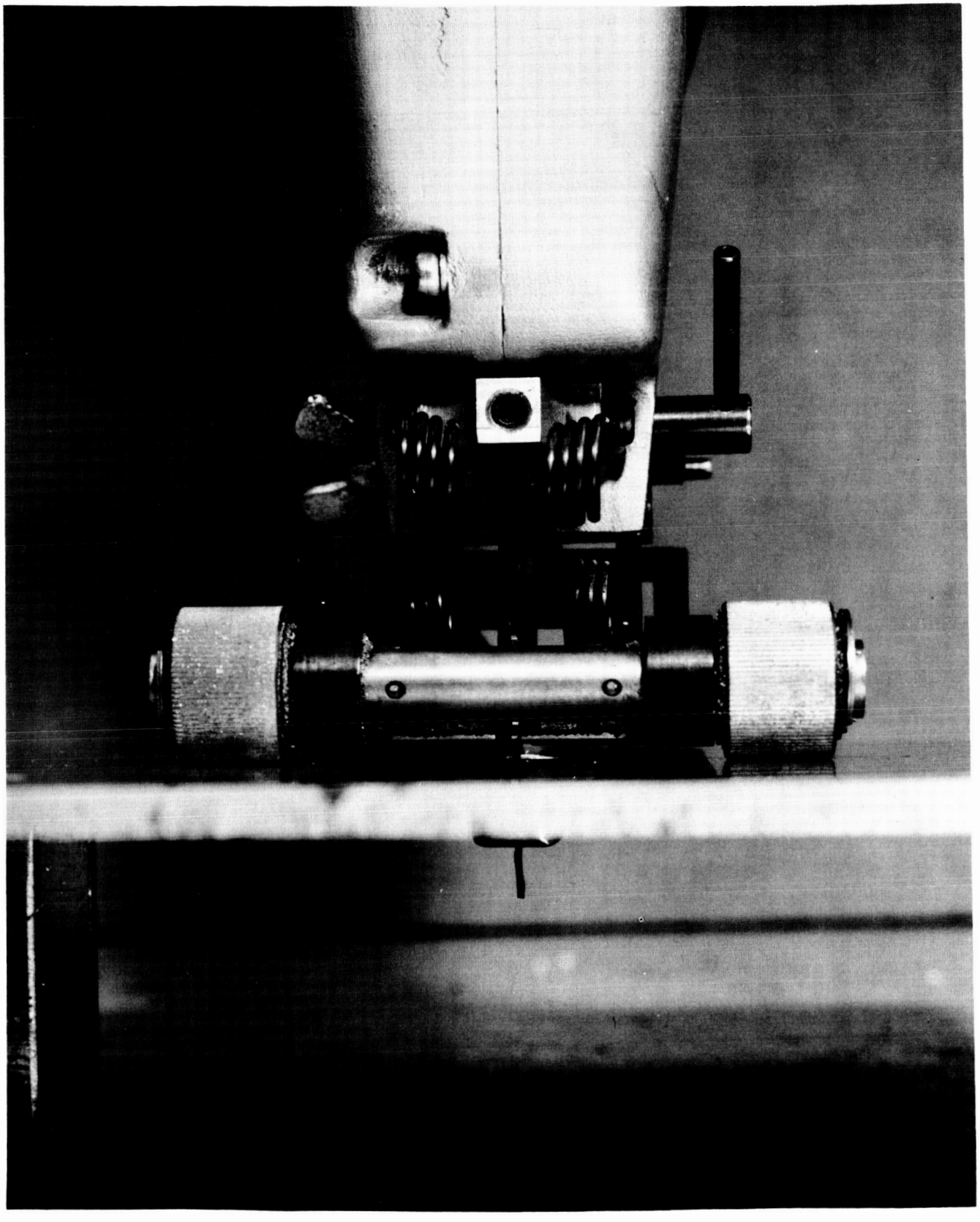


Figure 33. Saw Attachment - Front View

Slow feed rates in thick ( $>.100$) and thin ( $<.040$) material were noted. The test method varied from the original tests of the prototype in a significant manner. Originally, the work was supported in a wooden frame work, similar to a carpenter's mitre box, which allowed clearance below the work for the blade and hold down foot. The full weight of the tool was supported on the work and the tool moved along the work-piece. In the new test set-up the tool was inverted and held with the blade extending vertically upward. The work was stabilized by two supports in front of and behind the saw and the strip of metal being sawed was allowed to move. This latter approach provided a more realistic simulation of zero-G mechanics and showed that the feed rate problem consisted of two separate parts. In the thicker materials the feed rate is a function of the maximum forward force which can be exerted against the saw blade. This force is limited by the tractive force of the drive wheels. The knurled drive wheels slip on the aluminum surface when exerting a 5 pound forward force. Consideration was given to methods of increasing the tractive force through changes in knurl design or other wheel surface treatment, but it was decided that resolution of this particular problem was beyond the present scope of the program.

The second part of the problem, low feed rates in thin material, was discovered to be the result of "oil canning" of the thin material by the saw feed mechanism. The hold down and feed forces are applied at three points. A force is exerted upward on the bottom of the work just ahead of the saw blade by the hold down foot. The downward force exerted by the drive wheels is applied by the wheels at points about 1 inch to either side of the cutting line. Therefore, the 25 pound hold down force is reacted at two points with moment arms of 1 inch and bending moments in the workpiece cause it to lift upward at the cutting point leaving only the corner of the drive wheels in contact. With the contact area of the drive wheels reduced to a point, tractive force was lost. Analysis of the mechanism showed that with a redesign of the drive mechanism it would be possible to move the drive wheels closer to the cutting line to eliminate this problem.

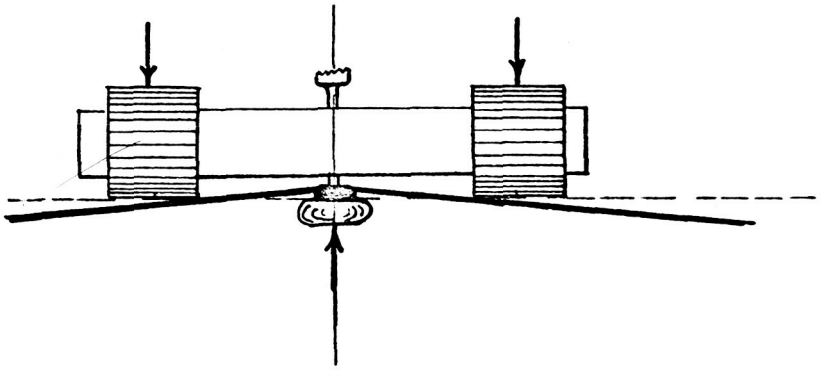


Figure 34 shows the final configuration. The drive wheels have been moved close to the blade and the springs replaced by a single straight spring actuated by a linkage from the saw blade holder. In addition the pivot for the wheel lever has been changed to move the hold down point to 1/8 inch in front of the saw teeth. The final configuration was retested with the results as shown in Table 11. The feed rates for thicker material will require an improvement in tractive force to achieve a significant increase in sawing rate.

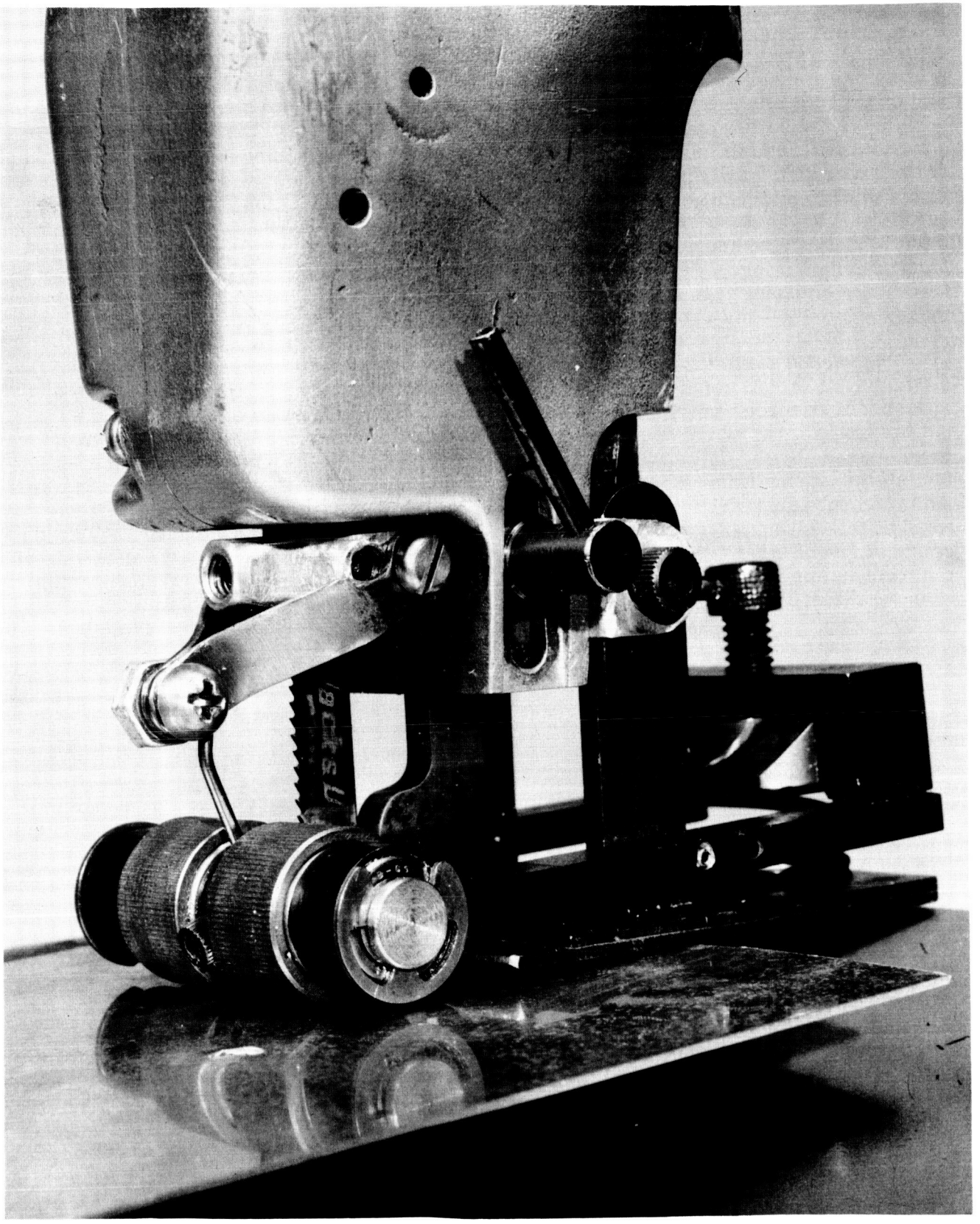


Figure 34. Saw Attachment - Final Configuration

TABLE 11

SAW PERFORMANCE

| Material | Distance (Inches) | Time (Seconds) | Rate (In/Min) | Hold-Down Force (Pounds) |
|---|---|---|---|---|
| .016 7075-T6 | 12 | 16 | 45 | 25 |
| | 18 | 20 | 54 | 25 |
| | 24 | 31 | 46 | 25 |
| .040 7075-T6 | 6 | 20 | 18 | 25 |
| | 12 | 45 | 16 | 25 |
| | 14 | 55 | 16 | 25 |
| .050 2014-T6 | 4 | 57 | 4.2 | 32 |
| | 6 | 75 | 4.8 | 32 |
| | 7.5 | 100 | 4.5 | 32 |
| .063 6061-T4 | 6 | 25 | 14 | 30 |
| | 12 | 55 | 13 | 30 |
| | 17 | 77 | 13 | 30 |
| .100 7075-T6 | 1 | 15 | 4 | 20 |
| | 4 | 45 | 5.3 | 20 |
| | 10 | 120 | 5 | 20 |

| | |
|---|---|
| Cutting Radius | 3 inches minimum |
| Blind Starting Hole | 5/8-inch diameter |
| Operator Reaction | Steering forces only |
| Power Input | 12.5 VDC, 5 A, 62.5 Watts |

## Drill Development

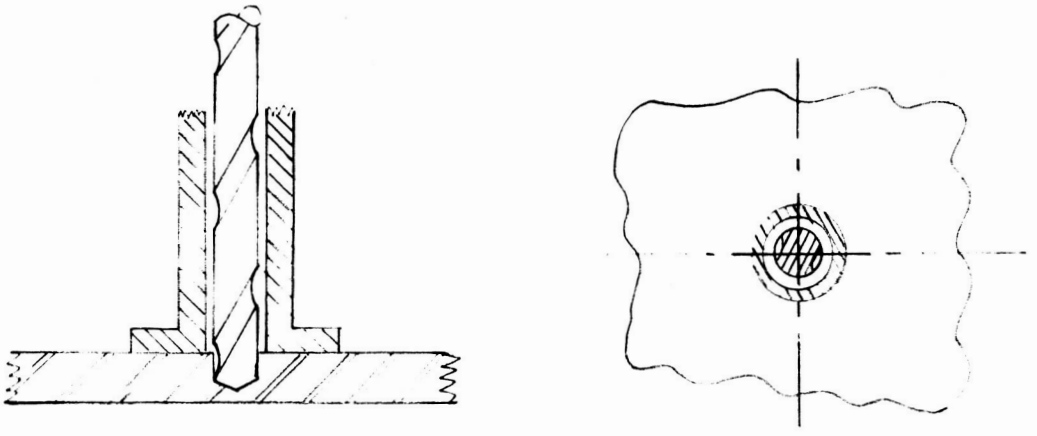
Drilling requires the application of two forces to the drill bit, an axial thrust and a cutting torque. Prior to developing a zero reaction drill mechanism it was necessary to define the magnitude of the forces involved and to evaluate mechanisms for applying these forces without undesirable reaction on the operator. A search of machinists handbooks revealed that standard shop practices define the feed rates for twist drills in terms of RPM and linear feed in inches per minute. No sources were found which gave optimum cutting pressures for different twist drill sizes and different work material. Experienced workmen using a hand drill motor learn to gauge pressure as a function of cutting rate using such criteria as motor speed (sound), chip formation, heat and known bit sharpness. None of these provide clues to the actual forces required on the drill bit to drill a hole. Tests were made in the laboratory utilizing a torque wrench with an adaptor modified so that drill bit pressure was applied at a radius of 1 inch. An approximate maximum force of 40 pounds was applied under 1 G shirtsleeve conditions while drilling a 1/4-inch diameter hole in a 1/8-inch thick 6061 -T6 Al plate. This force was considered as the maximum which a zero reaction tool would have to react. Torque requirements were not measured because it was anticipated that the torque could be developed by the impactor mechanism and this would be zero reaction. Tests on previous programs had shown the feasibility of drilling with an impact mechanism although the drill rate was relatively slow.

Two concepts were evaluated for achieving the zero reaction drill capability (see Fig. 35). One required a drill turning inside the holding fixture while the other required the holding fixture inside and the drill turning around it. Both concepts require a method of attaching the tool to the work to react the necessary axial cutting pressure. It was also apparent that if the axial force of 40 pounds could be reacted, a minimal torque reaction would also be possible.

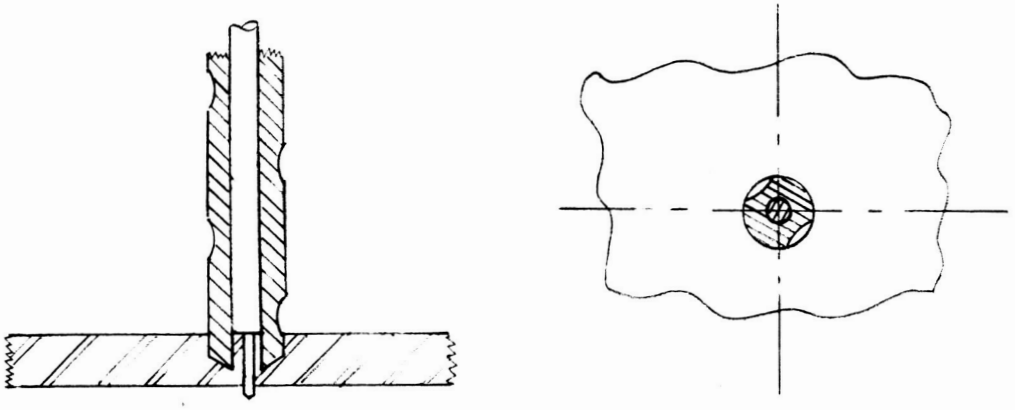
The possible means of attachment considered were (1) adhesive bonding, (2) welding, (3) nailing or stapling (interfrictional fastening). Adhesive bonding offered potential and an adhesive fastening attachment was already planned for a restraint defice. The adhesive would require approximately one square inch of contact area. This meant that for drilling holes the adhesive would be on the outside of the hole area and consequently would require removal after the hole was complete. Welding was considered attractive in its speed and ease of attaching, but it would require surface preparation and the equipment for application would be more complex. Welding could be used for either inside or outside holding concepts. Interfrictional fastening (nailing or stapling) is quick, simple and the mechanization appeared to be a relatively simple problem. This concept is better suited for inside holding, but it is adaptable for either concept.

Because of the relative complexity of welding equipment, the use of welded holding devices was discarded. The adhesive and interfrictional fastening approaches were studied further. The adhesive development is discussed in the section on the Restraint System Development. Its development for drill use would be essentially the same as for personnel restraint.

The use of an interfrictional fastening device was explored in a series of tests. Early in the program, as these tests progressed, an attractive element of the complete tradeoff came to light. This will be discussed first along with the selection of



A. Holding Outside



B. Holding Inside

FIGURE 35. DRILL CONCEPTS

drilling concept. Then the test and development of the holding mechanism and the drill mechanism will be covered.

One of the prime considerations was the efficiency with which a hole can be drilled. The basic drill concepts included standard twist drills working concentrically inside a holding fixture (Fig. 35, A) and a modified drill working concentrically around the outside of the holding fixture (Fig. 35, B). Looking at B, it becomes apparent that a portion of the metal is not removed when the hole is cut. In effect, the action is exactly like a two tooth hole saw. In fact a hole sawing (trepanning) approach has other advantages. Using a hole saw requires removal of a thin area of metal around the circumference of the hole only. Therefore, the total amount of metal removed on larger diameter holes is considerably less. Where R is the basic hole radius and t is the thickness of the saw kerf, the amount of metal removed is $\pi (Rt-t^2)$ for a unit thickness. The power required to remove the metal is proportional to the total amount removed and the chip size, both of which are smaller with a hole saw than with a standard twist drill. In addition, a hole saw can be designed to throw the chips inward instead of outward so that they can be trapped by the saw blade and the debris problem minimized. For these reasons it was decided to investigate the use of a hole saw operating concentrically around a holding device as a primary approach to the drill problem. The hole saw would be driven around a small diameter needle which has been driven into the work surface at the center of the hole. With a target of 1/8-inch to 1/2-inch diameter holes, the needle diameter would have to be small to be effective, probably on the order of 1/16-inch.

Attention was turned to the problem of penetrating a typical skin with a small diameter needle and achieving a pull-out force adequate to apply pressure to the hole saw. Two series of tests were conducted. One defined the axial force required on the hole saw and the other defined the parameters of achieving needle penetration.

To determine the hole saw characteristics, a set of 3/8-inch saws was fabricated of three different materials; M34, M4 and T-15 Tool Steels. The saw blades were tested in the same fixture used for sawing tests modified to hold a Black and Decker No. 457A 1/2-inch Holgun. The fixture was counterbalanced to provide cutting forces of 5 to 11 pounds. A series of holes was drilled in .028-inch Ti-6A1-4V with saws of different materials run at different pressures and speeds. A range of 5 - 11 pounds was covered in 1-pound increments with variations in speed from 1200 RPM up to 2300 RPM. The time for cutting a hole was recorded along with power input, RPM and cutting force. The optimum point was selected at 10 pounds cutting force at 1300 RPM. The M34 hole saws performed better than the others and this material was selected.

Following the optimization study, a series of holes were cut to determine probable blade life and expected average cutting time. In cutting 85 holes an average of 10 seconds/hole was experienced. The initial time for cutting with a new blade was 2.4 seconds and the average time dropped to a fairly steady level of 10 seconds by the 25th hole. From the 25th to the 85th hole, the rate declined to about 12 seconds/hole. The T-15 and M5 saws drilled rapidly. In 25 holes their rates were down to 18 and 16 seconds/hole respectively.

A series of tests on needle penetration was conducted to determine the feasibility of this approach for restraining the drill. The purpose was to define the feasibility of utilizing a needle driven into the spacecraft skin as a restraint capable of greater

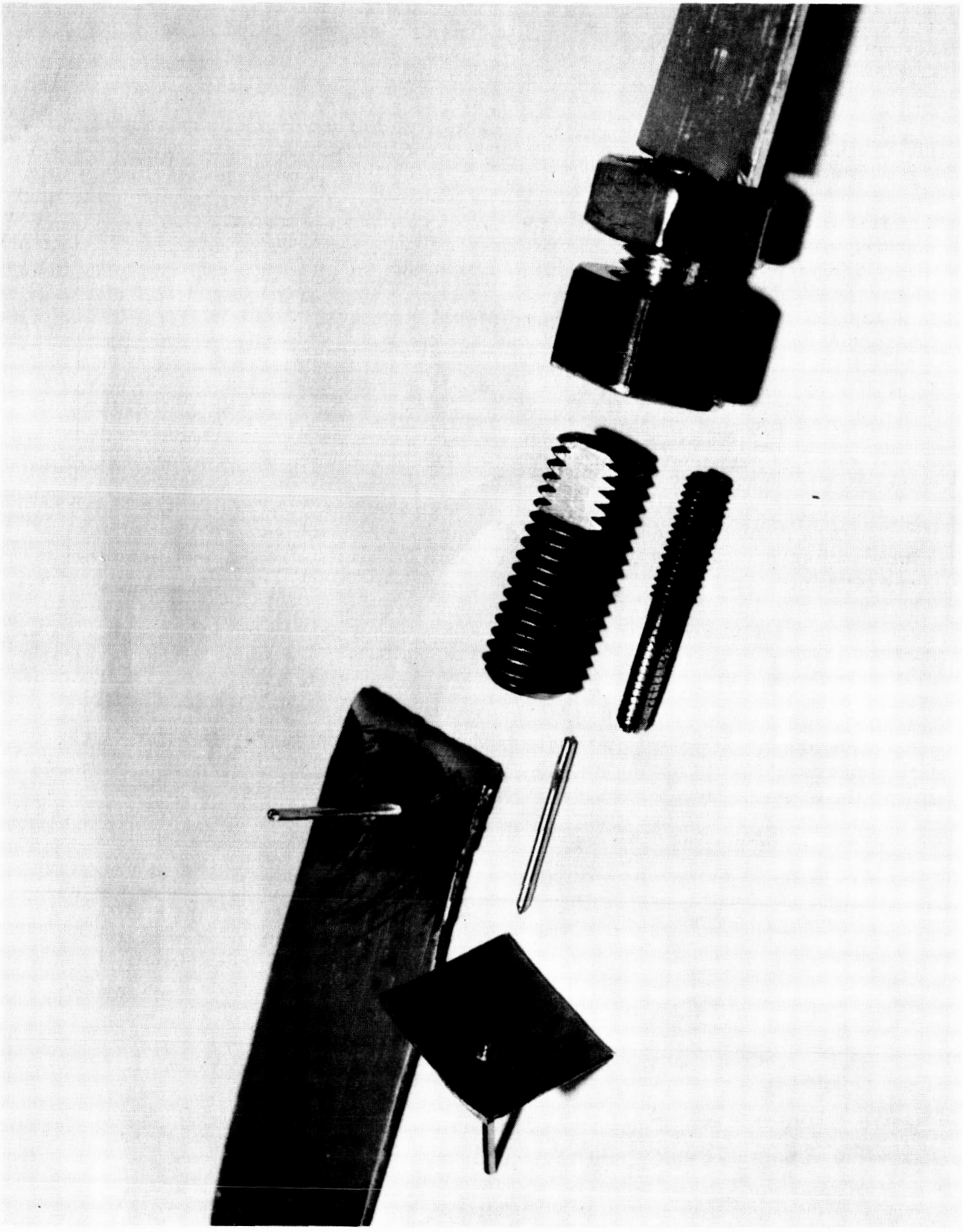


Figure 36. Needle Penetration Test Collet

than 10 pounds axial load. Initial tests were made by imbedding a Singer Sewing Machine needle, No. 11GY, .030-inch in diameter in a wood block holding fixture with the point in contact with an .050 inch thick piece of 6061.T6 Aluminum backed up by a steel block with a 3/16-inch diameter hole under the needle.

A 100 gram weight was dropped 24 inches onto the needle which made a partial penetration. The needle did not pass through the metal sample; however, it did take 9 pounds to pull the needle out. A one pound weight was then dropped 24 inches with the same needle and the needle passed .062 inches through the same sheet of aluminum. The test was repeated twice and the pull-out load was measured at 17 pounds and 19 pounds. A one-pound weight was then dropped 12 inches and passed through the same sheet 1/32-inch. The test was repeated twice and the pull-out force was found to be 21 pounds and 23 pounds. A new needle was then placed in the fixture and another test was run with the one-pound weight dropped 24 inches. 1/16-inch penetration of the needle resulted and the pull-out force was measured 20 pounds. The apparent disparity of the heavier impact blow producing less pull-out force was due to the configuration of the needle. The deeper penetration of the needle resulting from the 2 foot-pounds energy blow resulted in reduction of the contact area of the needle in the hole. There is a relief groove for the lead-in to the eyelet of the needle. The needle did not penetrate sufficiently in the 1 foot-pound blow to enter this region of relief in the needle.

Tests were also made of the penetration into .028 titanium. With energies of 1 and 1.5 foot-pounds, the needle points broke. In a third test, a one pound weight was dropped two feet (2 foot-pounds of energy) and made to impact the needle holder. The needle with holder was held approximately 1/16-inch off the workpiece prior to impact by a piece of modeling clay. The needle (Simanco #16 Sewing Machine) passed through the .028-inch thick titanium workpiece. The tip of the needle broke at a point forward to the eyelet. The point remained in the workpiece. Microscopic inspection of the imbedded needle point on the back side of the workpiece revealed that the modeling clay had also passed through the metal. The clay may have provided lubrication for the needle as it passed through the workpiece. This was the first successful test of the many attempts to pass a needle through titanium. Since the needle broke, additional work was required.

The three factors which contributed to this successful test are:

(1) The workpiece and related hardware were rigidly clamped together. The workpiece could not shift as the needle struck it.

(2) The needle had velocity as it struck the work. Prior tests, the needle was in contact with the workpiece as the dropped weight struck the needle holder (zero needle velocity).

(3) The modeling clay followed the needle through the workpiece and provided lubrication.

As all three changes were made at one time, it is not possible to ascertain the relative value of the three above mentioned factors.

Subsequent to these tests a fixture which could be driven by a Duo Fast CT859 Staple Tacker was fabricated to hold small needles. Figure 36 shows the fixture

along with a sample of .050-inch diameter needles which successfully penetrated samples of titanium. Emphasis was placed upon titanium penetration rather than aluminum because of the relative hardness. The blade at the right fits into the slot of the stapler, as shown in figure 37, with a 1-inch travel prior to contact with the metal. The needles were fabricated from 1/2-inch long segments of #6 sewing needles with a 90-degree included angle in the point. Successful penetrations were made with pull-out strengths of greater than 15 pounds. At 15 pounds the needle collet slipped off the needle. A high speed motion picture was made of the penetration action because the penetration was erratic. The film showed that the stapler actually slowed the needle from 34.4 feet/second peak velocity to 22.8 feet/second at impact. Also, the material was deflecting at impact and the collet deviated from the impact axis on rebound which might cause the needle to be dislodged after an initially successful penetration. An improved test fixture was fabricated (Figure 38) with the needle supported in a loose fitting hole and the impacting mass constrained to move axially by the fixture barrel. With this fixture a series of tests was made of different needle configurations. Figure 39 shows a test sample of titanium with varied needle penetration attempts. Table 12 lists the results of the tests. This series was concerned with achieving successful penetration and determining maximum torque capability. The configuration of test no. 12 was chosen for development. A 10X photo of this needle (Figure 40) shows the point geometry and holding action. The wedge action provided by the point assures a firm grip while the flats resist torque. With a basic definition of the saw force requirements and needle capabilities established it was possible to begin definition of the actual drill attachment. The objective was to make the attachment as automatic as possible to minimize the time required in its use and to simplify the operation.

A basic sequence was defined as follows:

1. Install proper saw and needle;
2. Cock the mechanism;
3. Set at work point and penetrate;
4. Cut hole;
5. Remove from work;
6. Remove slug and chips.

The design problems encountered were 1) to mechanize the needle penetration inside the hole saw drive mechanism, 2) to use motor power to cock the mechanism, 3) to trigger the release and 4) to reset the mechanism. In so far as possible these would be automatic operations requiring simple positive motions to initiate. As the design evolved, it became apparent that the zero reaction feature of the impactor was not required and that the mechanism would be fairly long, so it was decided to make it a separate attachment rather than something to work on the front of the impactor.

The solutions to the design problems and the operational sequence refinement were straightforward and resulted in the mechanism whose operation is described below.

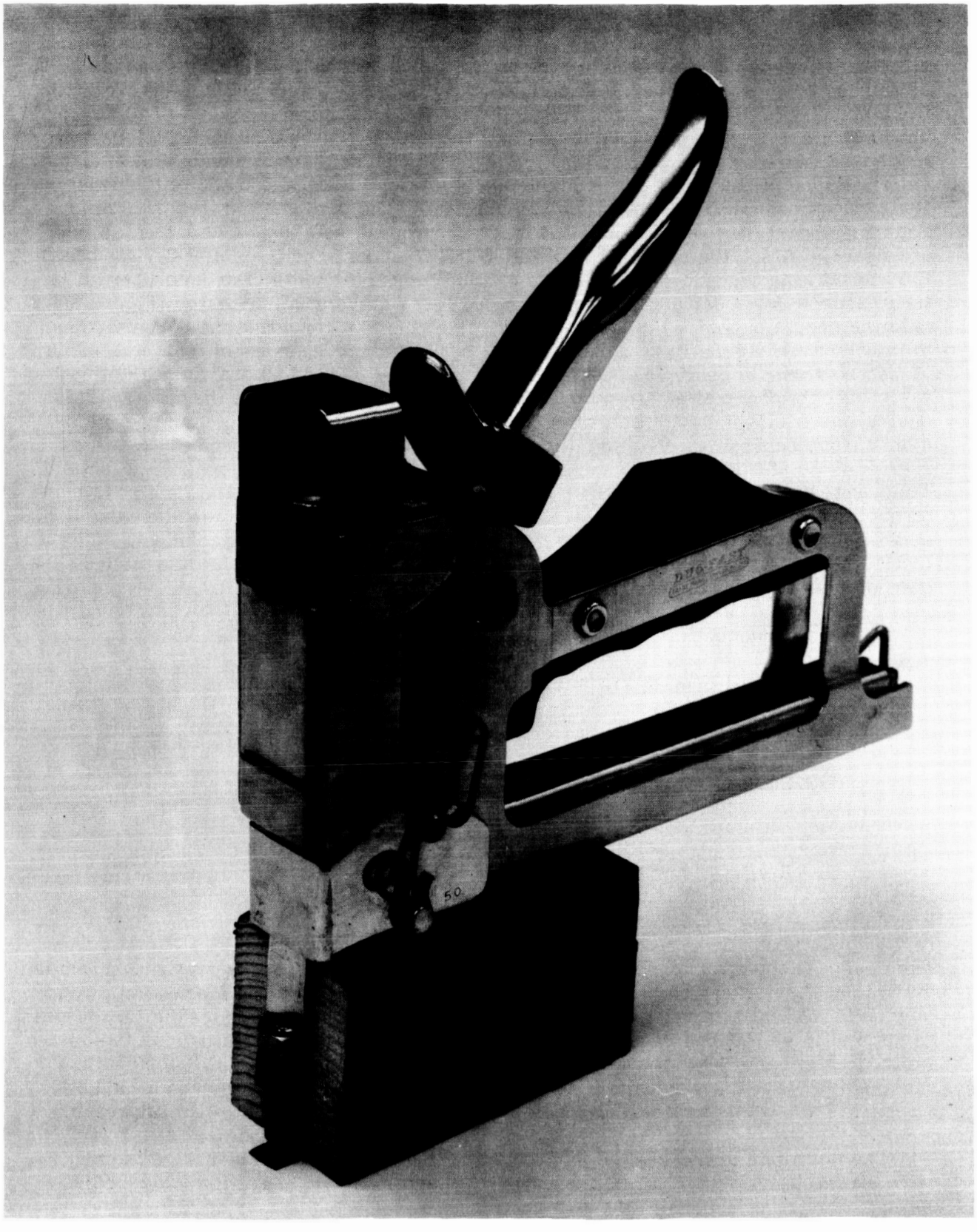


Figure 37. Needle Penetration Test Set-Up

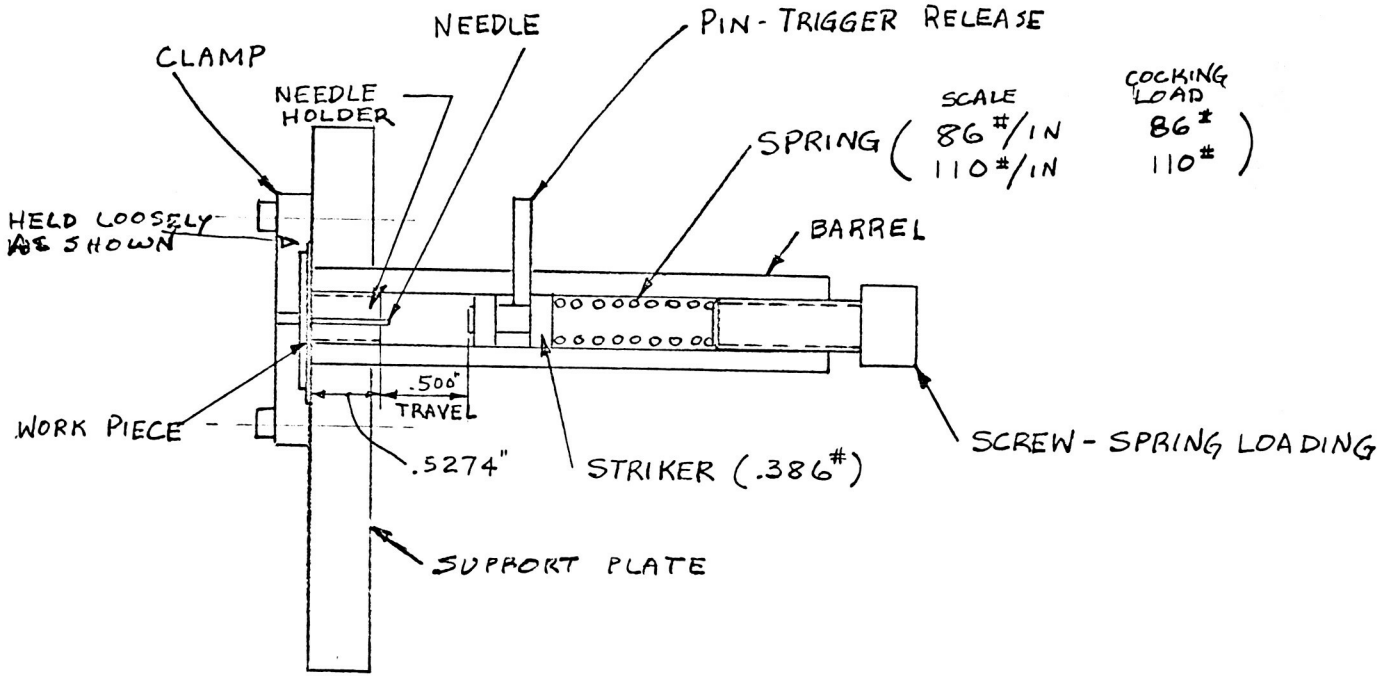



Figure 38. Needle Penetration Fixture

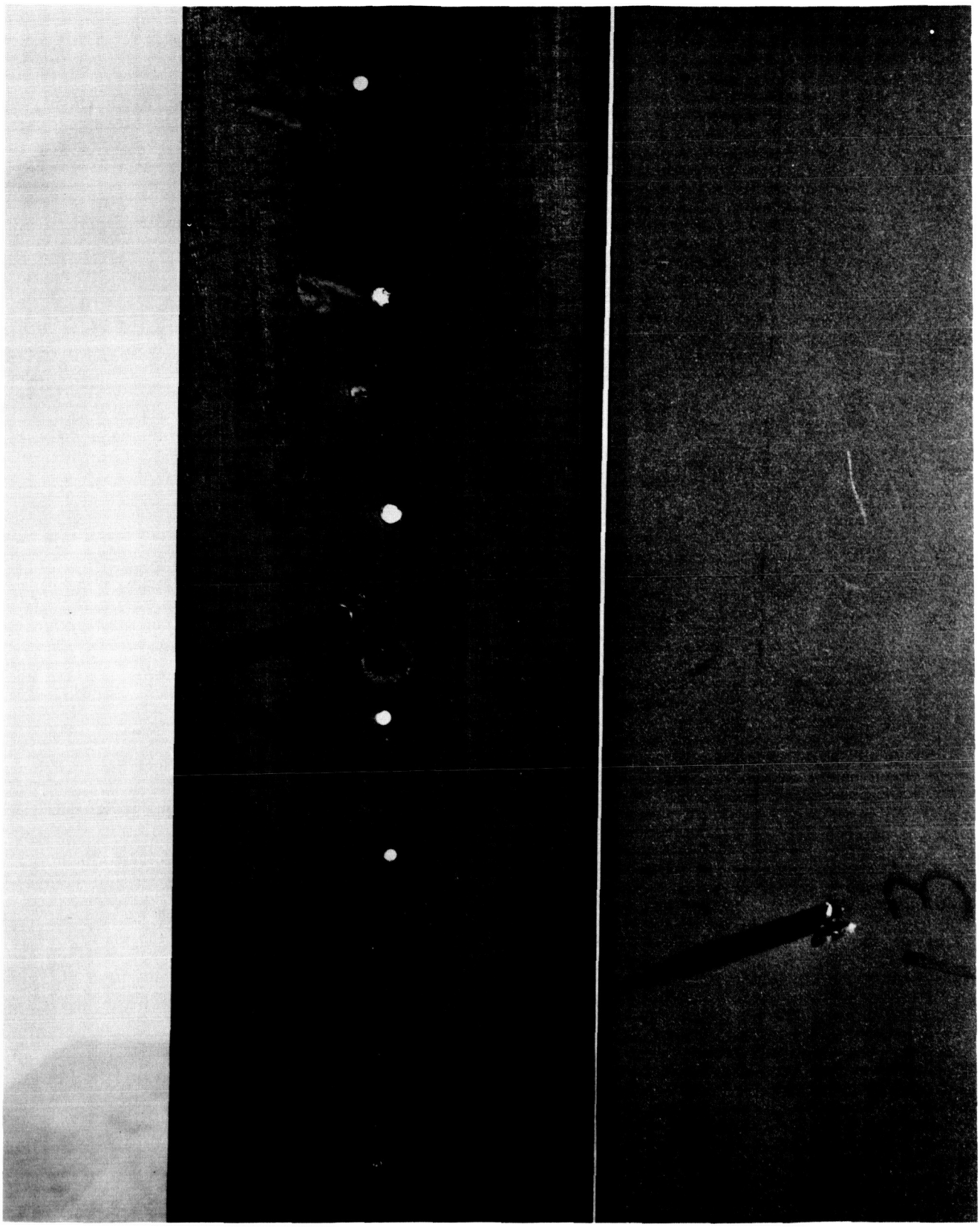


Figure 39. Needle Penetration Test Samples

TABLE 12

NEEDLE PENETRATION RESULTS

| Hole | Diameter | Needle Type Point Geometry | Results |
|---|---|---|---|
| 1 | .034 | 60° included angle | Point dulled - no penetration |
| 2 | .034 | 120° | No penetration - point slightly off center |
| 3 | .030 | 120° precision ground | no penetration |
| 4 | .036 | 120° | no penetration |
| 5 | .036 | 120° | 1/32-inch penetration with solid backup |
| 6 | .046 | 120° | 1/32-inch penetration - torque 4 inch-ounces |
| 7 | .062 hex* | 120° | Partial penetration |
| 8 | .062 hex | 120° | 1/32-inch penetration - torque 3.45 inch-pounds ragged hole on exit side |
| 9 | .062 6 spline* | 120° | Torque 3.5 inch-pounds |
| 10 | .046 | 120° | 1/32-inch penetration - torque too low to measure |
| 11 | .062 hex* | 120° + Flats | 1/16-inch penetration - torque 6.75 inch-pounds needle broke just behind flats. |
| 12 | .078 | 120° + Flats ground to 1/16-inch width and 10° included angle | 1/8-inch penetration - torque 12.75 inch-pounds |
| 13 | .083 | Roll Pin | 3/32-inch penetration - torque 3.0 inch-pounds |

Note: Tests 1 - 7 were conducted with 86 pound spring
8 - 13 were conducted with 110 pound spring
5 - 13 were conducted with piece solidly backed up.

*Hex needles were modified Allen wrenches
6 spline needle was modified Bristol wrench

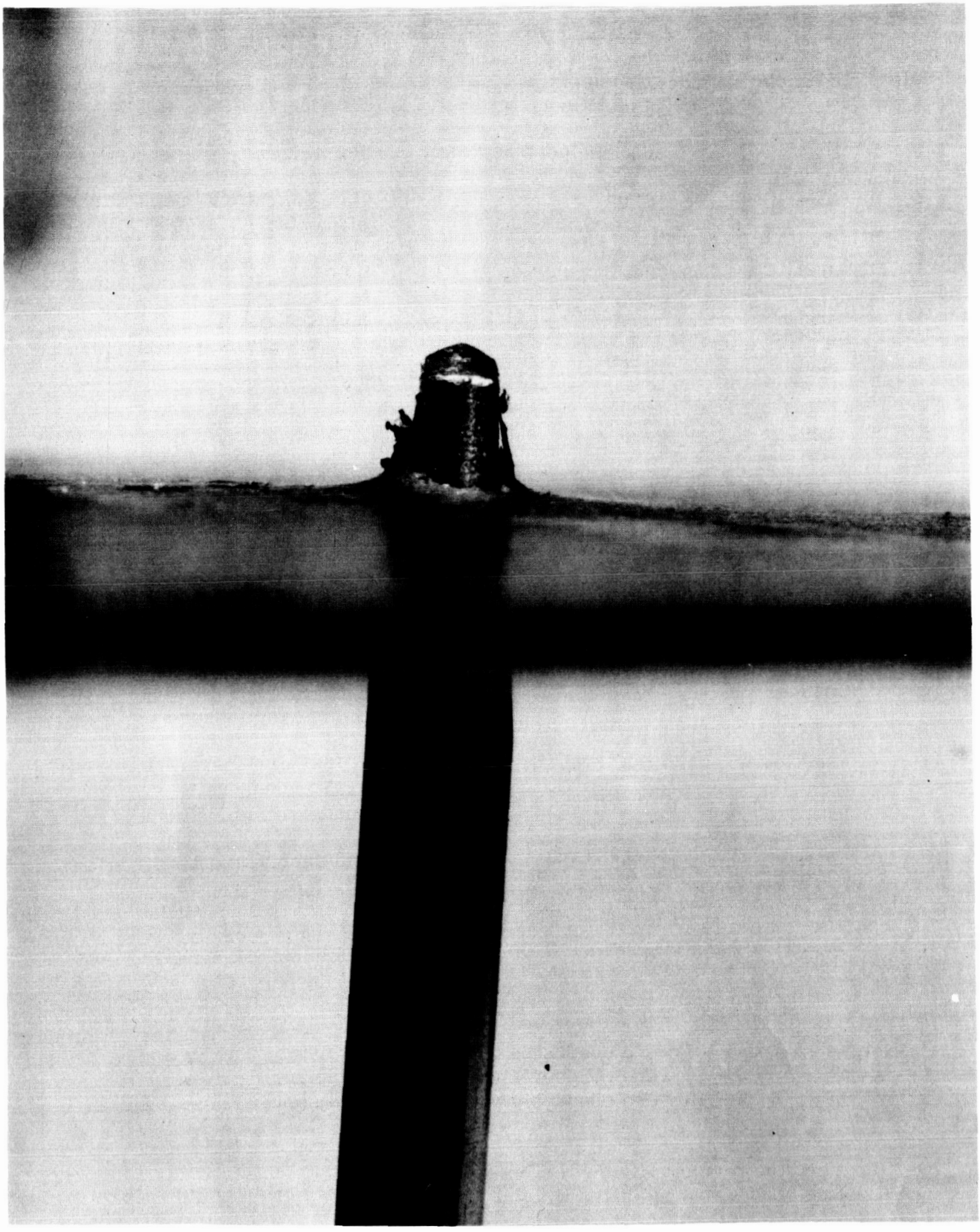
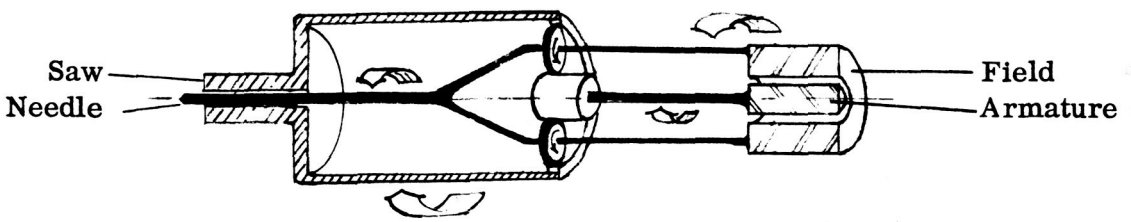


Figure 40. Needle Penetration in Titanium Sample

Theory of Operation.—Figure 41 is an assembly drawing of the drill mechanism. The right end mates with the motor section of Figure 29 in the same manner as the impactor and power is applied through the clutch and housing. The drill/motor assembly is free to rotate in the housing bearings so that effectively no torque is felt on the tool handle.

Unlike the other attachments the center mechanism of the drill stays stationary and the outer barrel rotates around it. The motor housing normally stays still and the armature rotates so it is necessary, in effect, to turn the drive mechanism inside out. This is accomplished by the reduction gears which transmit clutch-2 rotation through a 2.05:1 reduction gear train -317, 316, 5. The Ring gear -5 is held rigidly to the barrel -310 so that the barrel rotates driving the saw holder -302 and saw -320. The supporting structure -314 for the planetary gears -316 is connected to the motor housing through the attachment interface -105, 104. This structure is also connected through the needle mechanism to the needle so that ultimately the torque developed between the motor armature and field is felt between the needle and hole saw. The sketch below shows the simplified force paths.




Since the needle resists torque forces the field does not rotate. The arrows show relative rotation.

This portion of the mechanism accomplishes the hole saw drive. It is also necessary to drive the needle into the work and to pull on it as the hole saw is rotating. The exploded view of the mechanism (Figure 42) shows the method of accomplishing this. The central column which transmits the torque and pull forces to the needle consists of the lower group of parts. Reading from left to right these are the needle -301, (numbers are keyed to Figure 41), needle holder -303, striker latch -304, detent ball -330, stationary shaft -305, striker 312 and tension spring -306, reset nut -309 and striker spring -308. The needle is screwed into the needle holder which is held on the stationary shaft by a pin through the striker latch. Figure 43 shows the relationships of these parts. The pin at the left end of the striker latch, which is extended past the end of the stationary shaft, engages the slots in the needle holder. When a torque is applied to the striker latch this pin transmits the torque to the needle holder. In Figure 42 the pin -327 extending out of the stationary shaft nor-

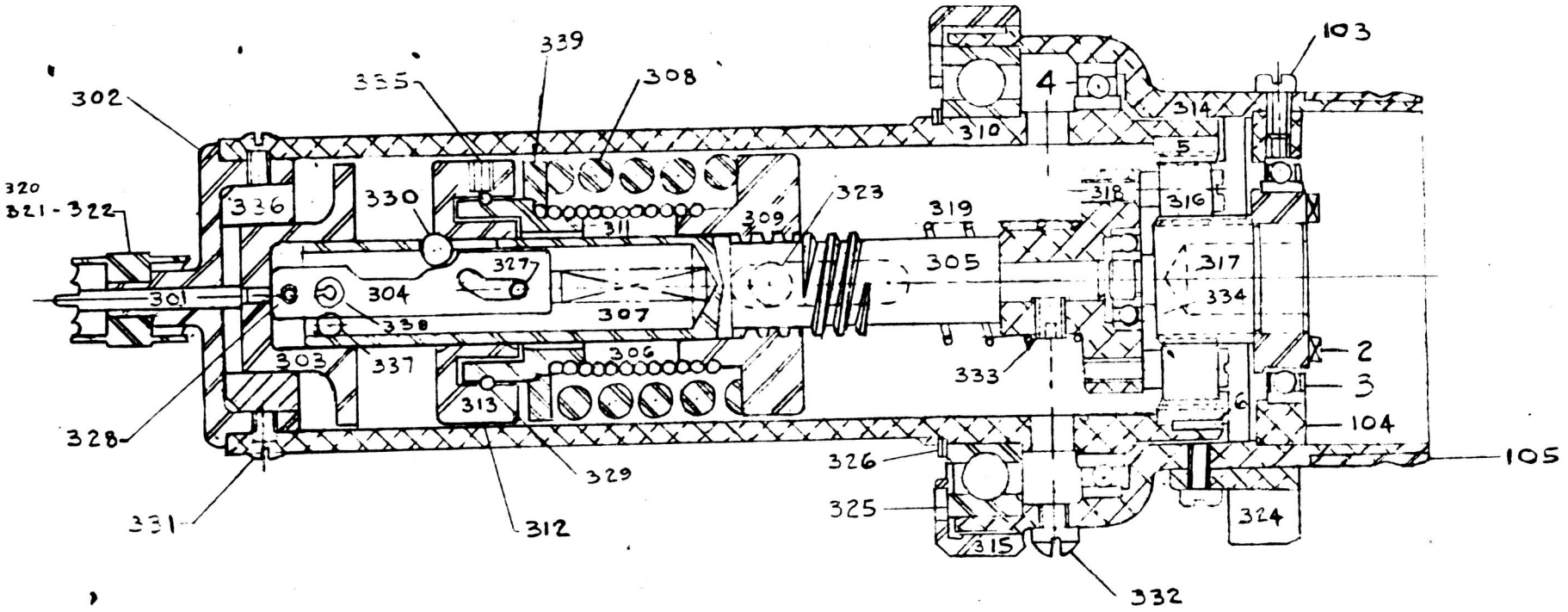



Figure 41. Drill Mechanism Assembly Drawing

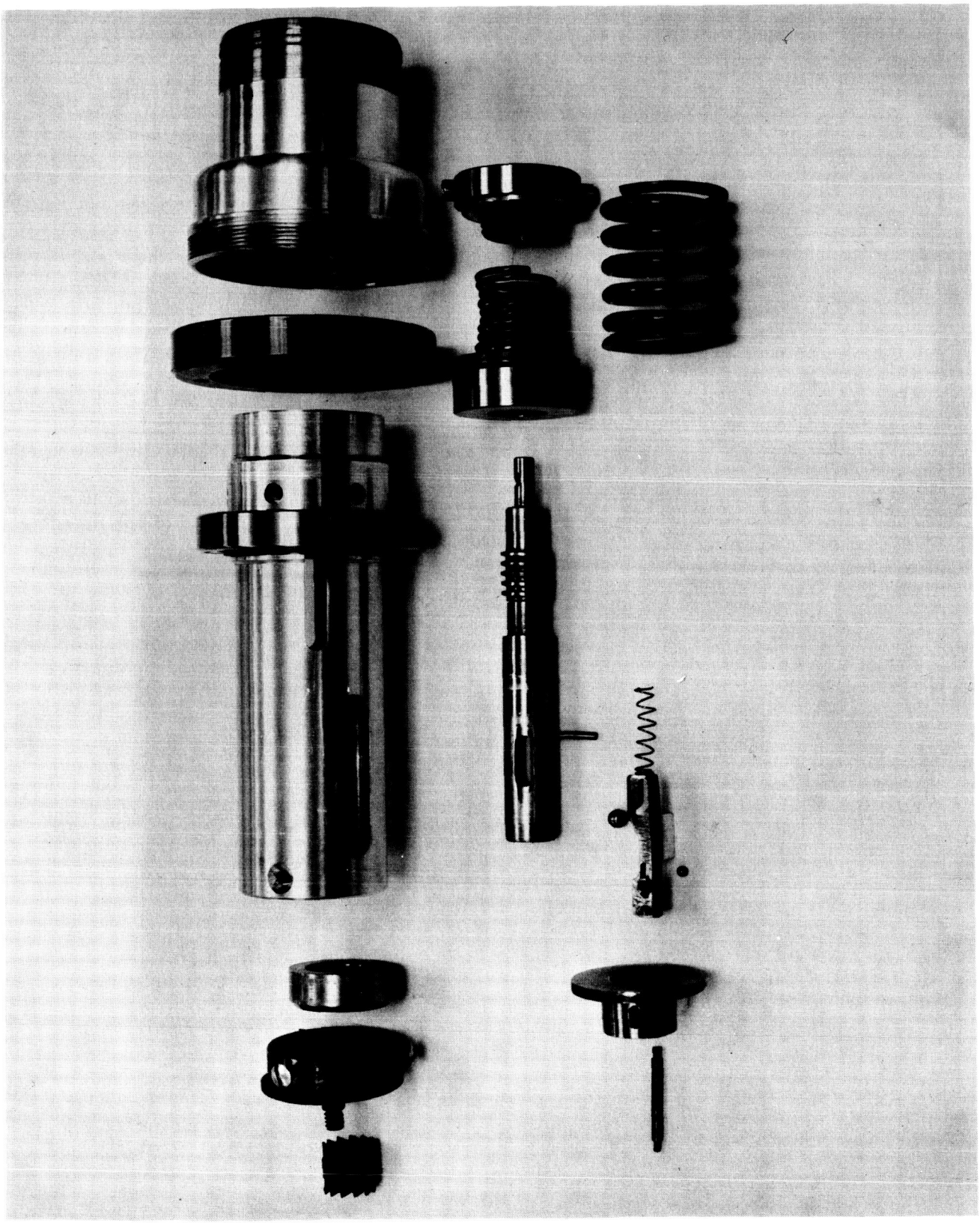


Figure 42. Drill Mechanism Exploded View

mally engages the striker latch through the dog leg slot so that torques on the shaft are transmitted through that pin -327 to the striker latch -304, through the striker latch pin -328 to the needle holder -303 and to the needle. At the right end the stationary shaft is clamped rigidly to the attachment interface structure. The torque required to cut with the saw blades is clockwise looking at the work. Therefore, the torque on the needle is counterclockwise. It was necessary to provide a left hand thread on the needle to prevent its unscrewing during the drilling operation.

In addition to torque forces it is necessary to exert axial forces on the needle to both penetrate and to apply cutting pressure. The penetration will be discussed first. In Figure 43 the spring assembly is shown above the stationary shaft. This assembly fits on the shaft loosely so that the striker can move forward and strike the needle holder. Figure 44 shows the spring assembly on the shaft in the cocked position. Notice the detent ball, which is also visible in Figure 43. The ball rests on the surface of the striker latch so that it projects up through a slot in the shaft and obstructs the forward motion of the striker. The reset nut engages the Acme thread at the rear of the shaft and compresses the spring as it rotates. Rotation of the reset nut is achieved by slots in the barrel engaging the shoulder screws -323 on the nut so that, as the barrel rotates in reverse, the nut travels forward (left). The thread length is short enough to allow the nut to run off the end of the thread to avoid jamming the mechanism. The spring force is felt between the shaft and the detent ball in contact with the striker. Since the shaft slot restrains the forward movement of the ball the only possible motion is down. The striker latch prevents downward motion of the ball to release the striker until it moves back approximately 1/32-inch. The configuration of the release surface on the striker latch was selected so that the downward force on the ball is reacted rearward assisting the movement of the striker latch. This was done to minimize the friction forces caused by the 110-pound spring force acting downward against the striker latch. The surface is canted downward 5 degrees so that the downward force of the striker spring against the ball creates a rearward thrust on the striker just slightly less than the force required to slide it rearward under load. In this manner the release force can be minimized. The striker latch is moved back by motion of the needle and needle holder. A rearward travel of 1/32-inch of the needle moves the striker so that the ball moves onto the 30-degree slope. The spring pressure now pushes the ball completely below the surface and the striker moves forward, impacting against the needle holder. The striker energy is transmitted to the needle holder and needle, driving the needlepoint into the work and driving the needle holder forward against the saw holder if the needle penetrates the work completely. This limits the maximum needle travel so that the needle cannot act as a projectile if accidentally released.

Once the needle has penetrated, the drill is turned on to run in the cutting direction. The reset nut runs to the rear (right) in the first few turns and the mechanism applies the saw force by means of the tension spring which is stretched between the striker and the reset nut. The striker is prevented from moving by the detent ball. Once it has impacted it is forward (left) of the slot in the shaft and the return spring -307 pushes the striker latch forward raising the ball above the shaft surface. The spring force is transmitted from the striker through the ball to the striker latch shoulder pulling the striker latch to the rear (right). However, the striker latch pin is engaged in the needle holder slot and the striker latch cannot move relative to the stationary shaft unless the needle also moves.

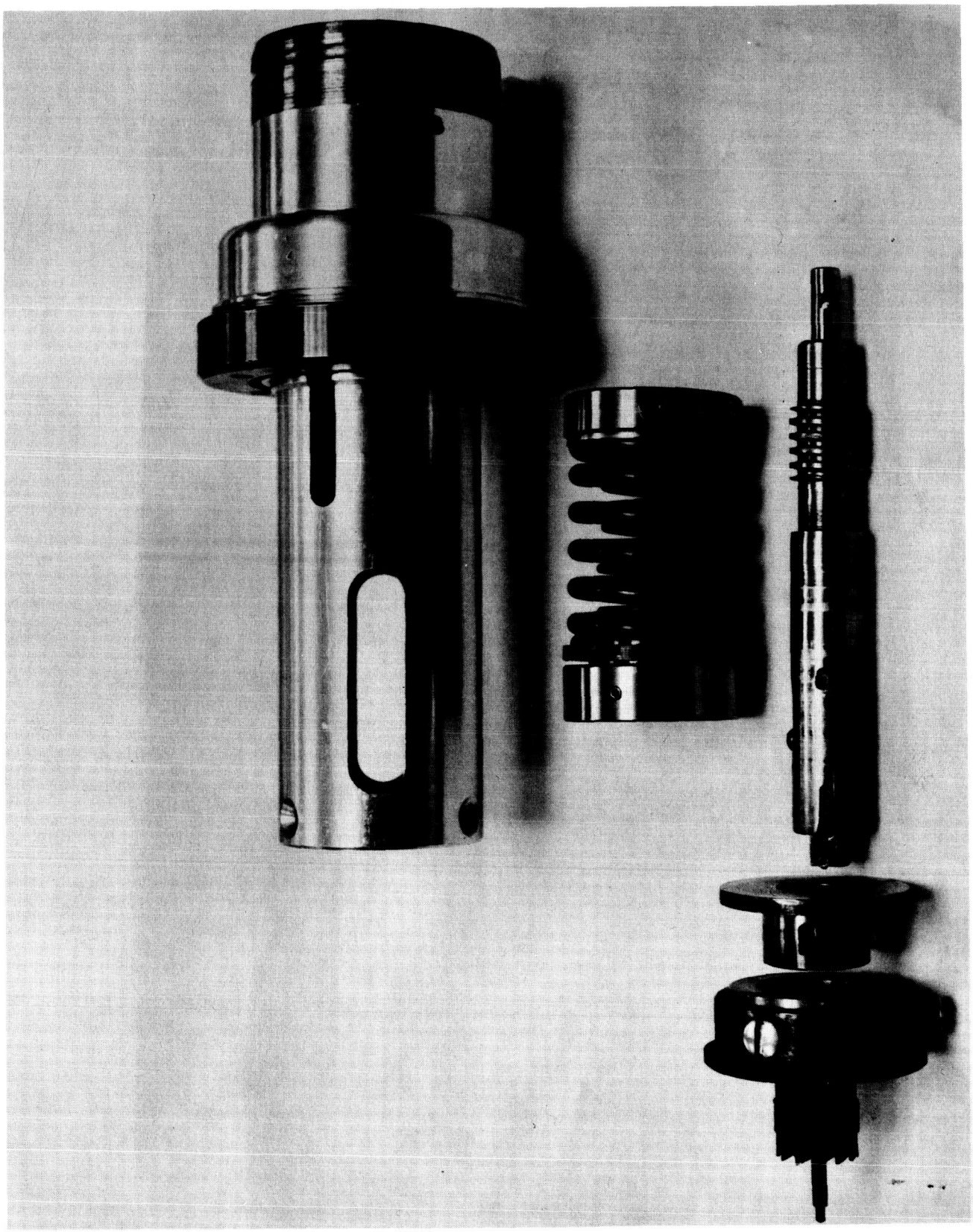


Figure 43. Drill - Spring Mechanism Disassembled

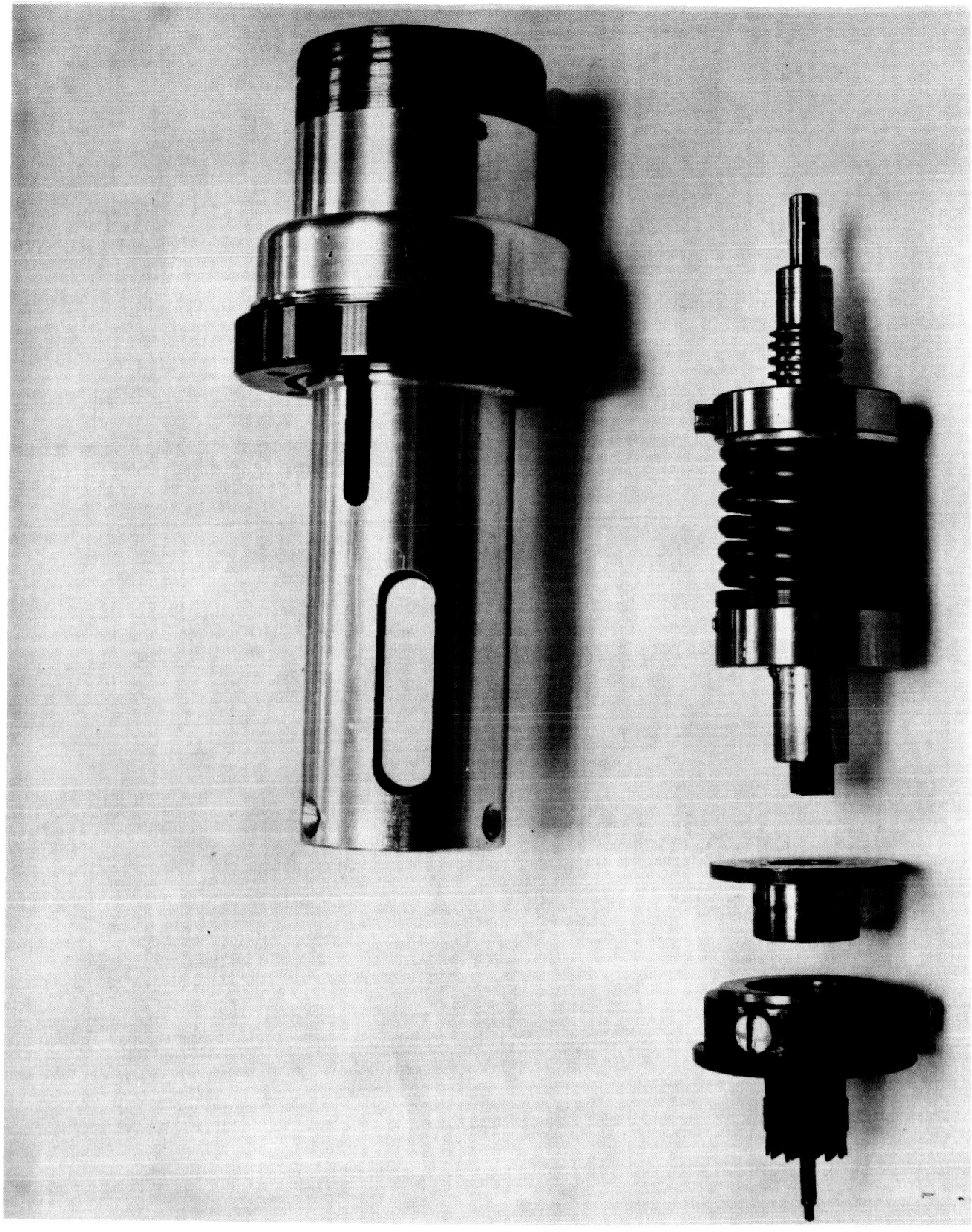


Figure 44. Drill - Spring Mechanism Assembled

The spring tension is felt through two paths. At one end of the spring it is through the striker, ball, latch, pin, needle holder and needle to the work piece. At the other it is through the reset nut, shaft support structure, bearings, barrel, saw holder and saw to the work. Therefore, the spring tension results in pressure of the saw teeth against the work.

As the saw cuts through the work it moves forward relative to the needle or in other words, the needle moves backward relative to the saw. The needle movement results in striker latch movement relative to the shaft. For the purpose of saw feed there are two moving assemblies connected by the tension spring. One is the striker latch force path and the other is the shaft and barrel path as described above. As these move relative to each other the pin -327 moves in the dog leg slot in the striker latch. After the striker latch has moved about 3/16-inch rearward, i.e. the saw has penetrated 3/16-inch, the striker latch moves onto the dog leg portion of the slot, allowing the ball to drop below the shaft surface and the striker to be pulled to the rear. This removes the spring tension from the needle, stopping the cutting action and resetting the striker.

The tool is run in reverse to unscrew the needle which is now held in place by the slug cut out of the workpiece. The needle, slug and chips can be disposed of, although the needle is reuseable.

Replacement of the needle allows the operational sequence to be repeated.

The sequence of actions is summarized below. Starting with the needle and saw in place and the mechanism uncocked, the following happens:

(1) The motor is run in reverse to cause counterclockwise rotation of the barrel.

(2) Barrel rotation causes the reset nut to run forward compressing the striker spring.

(3) The tool is placed against the hole center.

(4) A sharp push is applied causing the needle, needle holder and striker latch to move rearward relative to the shaft.

(5) The ball rides onto the $30^{\circ}$ latch surface, drops below the shaft surface and releases the striker.

(6) The striker is accelerated forward by the spring impacting against the needle holder, driving the needle into the work.

(7) The motor is run forward to cause clockwise rotation of the barrel.

(8) The barrel rotation causes the reset nut to move rearward applying spring tension through the striker, ball, latch, pin, needle holder and needle.

(9) The tension on the needle pulls the mechanism forward bringing the saw into firm contact with the work.

(10) The saw cuts through the work moving slowly relative to the needle. Torque is reacted through the stationary shaft, pin, striker latch, pin, needle holder and needle.

(11) The striker latch moves relative to the shaft until the dog leg portion of the slot is reached. This occurs in two stages. Initially the saw is penetrating the work. After penetration the slug is pulled into the hole saw below the plane of the teeth. This total movement limits the maximum thickness which can be cut assuming that the needle has penetrated the work fully.

(12) The latch drops the ball below the shaft surface releasing the striker, allowing it to move rearward, and thereby removing cutting pressure.

(13) The tool is stopped and removed from the work.

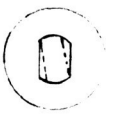
(14) The tool is run in reverse to free the needle so that the slug and chips can be removed.

(15) As a safety precaution the tool is run forward to assure that the spring is not cocked until needed.

Test.—After fabrication the drill was assembled and tested. The mechanism required extensive debugging and fine tuning to make it work properly. Several of the ultimate design features were developed during this period. These were the striker latch configuration, the needle point geometry and the compression thread length. The problems will be discussed in the order mentioned.

The striker latch was originally designed with the ball surface horizontal (parallel to the major axis). With the spring pressure of 110 pounds acting on the upper portion of the ball, a force of about 100 pounds is exerted sideways against the striker latch supports. This required an excessive axial force to allow sliding contact between the ball and striker latch and the striker latch and its supports. To reduce this force the surface was machined at a $5^{o}$ angle to provide an axial component which assists the release force. It was then discovered that the pressure on the ball was brinelling the surface as the ball moved over the corner onto the $30^{o}$ slope with the effect that the ball rested on a sharper slope in a lower position. This caused the striker contact to be higher on the ball and the release force to be less. This aggravated the situation to the point where the ball would not hold the striker. A hardened steel pin was inserted and ground to the proper angles to eliminate the problem. During drilling attempts it was discovered that feed pressure was low because the striker latch was rubbing against the shaft and the slot was rough. The slot was polished and a stabilizing pin -338 inserted to reduce this frictional force. Also, a slight misalignment of the needle holder slot allowed the loads on the detent ball to be transmitted through the latch and pin to the needle holder causing it to drag. This was corrected by enlarging the slot width so that only tension or torque is transmitted and all side loads are reacted by the support ball -337 and the shaft pin -327.

The needle point geometry was a problem because the needle was working loose under the vibratory loads imposed by the saw and, when penetrating the higher temper aluminums, was breaking loose a section rather than deflecting it so that the grip force was lost. The final needle geometry is shown in figure 45. The needle diameter is 1/8-inch with an .070 point with a 120-degree included angle and a 10-degree included angle on the flats. The flats have been undercut as shown below to provide



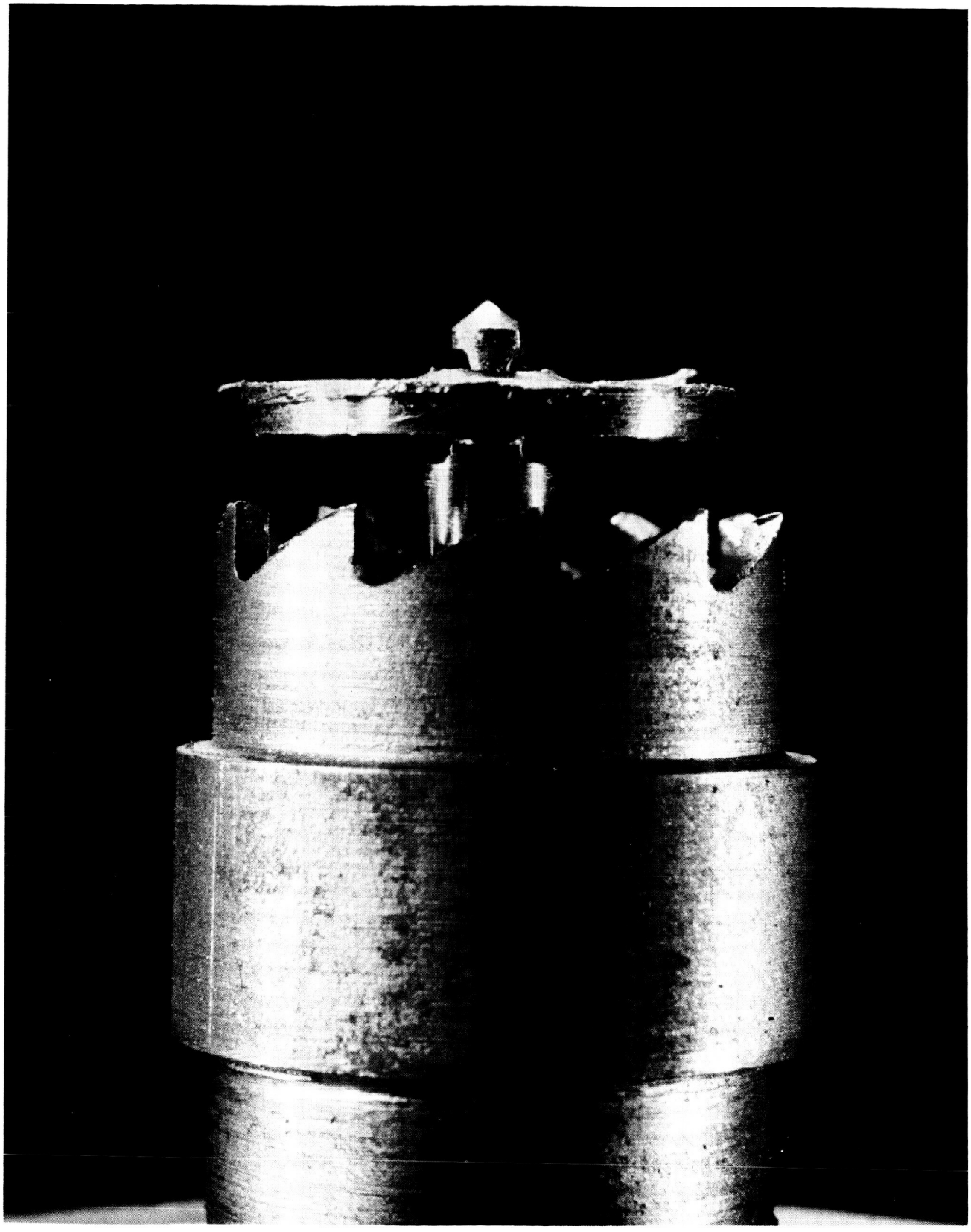


Figure 45. Final Needle Geometry

a cam locking action. After the needle has penetrated the metal the torque reaction as the saw bites in turns the needle so that the lips overlap the material and resist pulling out. Additional concepts were considered such as threads similar to an Easy out, but fabrication and test of different needle point geometries was considered beyond the scope of the present program.

The free length of the compression spring was selected so that the spring would move the striker into position to pick up the detent ball when the reset nut was at its rearmost point. The tension spring pulls the assembly up to the shortest position allowed by the compression spring. With the original thread length this would position the striker over the shaft slot requiring the ball and latch to move to the forward end of the slot to reset. It was discovered that the ball was becoming trapped under the striker instead of moving forward and the mechanism could not be recocked. During investigation of this problem, two access slots were cut in the forward portion of the barrel. It was finally determined that the best fix was to extend the thread to the rear 1/4-inch to move the striker initial position back to the rear edge of the slot. This clears the ball and as the mechanism is cocked the striker merely moves forward until the detent position is reached before the spring starts to compress.

After the fine tuning on the mechanism the performance was improved to a point where performance testing could be meaningful. A series of test holes was attempted in the same materials used for saw testing. Hole saw sizes of 1/4-inch, 3/8-inch, 1/2-inch and 21/32-inch were tested. The 21/32-inch hole saw was included because this is the minimum size hole required for starting a blind saw cut.

The drill was used to cut a series of holes of varying sizes. Initial attempts were made to cut holes in .016-inch 7075-T6 aluminum. As the saw started cutting the torque on the needle increased to the point where the needle twisted out the hole, like a reamer, and pulled out, stopping the cut. No holes were completed because of this.

In the next size of material .040 -7075-T6, a series of holes was cut with the four different size hole saws. The cutting rates are tabulated along with other basic data in Table 13. The discrepancy for the 3/8-inch diameter hole is attributed to faulty fabrication of the hole saws. The teeth were not properly sharpened and hardened. Previous feasibility tests indicate that the expected time would be consistent with the trend shown by the other size hole saws. The reason for a decrease in cutting time with an increase in diameter is that the tooth speed relative to the work increases with radius. All other factors are constant, RPM drill pressure, material, tooth shape (except 3/8). This indicates that the cutting RPM is below the optimum. Further study will be required to determine the relationship between cutting rate and tooth speed to determine the optimum. Speed of the drill is about 1000 RPM. It is possible that the internal friction in the mechanism has slowed the motor more than anticipated since the input power of 210 watts is higher than expected.

During the refinement of the mechanism and the performance tests it was discovered that the material which can be penetrated by the needle is limited. Attempts were made to penetrate varying thicknesses of aluminum from .016-inch up. The penetration success is dependent upon material thickness, material deflection, material stiffness and impact velocity. No problem was encountered in successfully penetrating up to .040 T6 aluminum. Only .028 titanium was tested. The samples were supported on a backing plate with holes up to 1/2-inch in diameter. With .050

aluminum the needle made only partial penetrations, not sufficient to hold securely enough to cut a hole. It is probable that the energy transfer mechanics from the striker to the needle will require a more thorough dynamic analysis to reevaluate the effects of material characteristics and internal friction in the mechanism on penetration. The existing design would require major rework to properly evaluate this empirically. Therefore, this line of investigation is beyond the scope of this program.

Additional tests were made to determine other drill characteristics. The drill torque, needle release impulse and travel needle pull-out and saw force were made and are also tabulated in Table 13.

A vacuum test of the drill was made to evaluate possible heating effects on the drill and motor combination. Investigation of the interaction of saw teeth and metal in a vacuum were not studied as a part of this program although such a study will be necessary for future development work. This type of study will require more sophisticated test fixtures to permit control of saw rates and work pressures in vacuums greater than $10^{-5}$ torr.

The tool setup consisted of the motor and drill unit held in a fixture which supported the drill but allowed rotation of the outer drill barrel and counter-rotation of the motor field housing. In front of the drill, a solenoid was mounted with a plunger for applying the release force. The tool and release solenoid were remotely operated. No saw was attached, although a needle was emplaced. The front end of the drill was supported in an aluminum alignment bushing with a teflon thrust washer. The chamber was pumped down to $2 \times 10^{-5}$ torr.

The drill operation consisted of cycling the drill through its complete operation at four cycles per minute for 45 minutes at $2 \times 10^{-5}$ torr. This is equivalent to drilling 180 holes in rapid succession. The cycle consisted of the following:

1. Drill motor run to cock mechanism for 2 seconds.

2. Solenoid release of impact mechanism.

3. Drill run in cutting direction for 12 seconds.

Drill power and motor case temperature were recorded. The motor temperature rose in the 45 minutes to $170^{\circ}$ F. Drill power inputs varied for each segment of the cycle. Cocking runs the motor almost to stall with 45 amperes at 9.5 V dc peak power in. Running draws approximately 9 amperes at 12 V dc. During the test the chamber pressure increased from $1.8 \times 10^{-5}$ to $4.6 \times 10^{-5}$ torr.

Examination of the motor revealed the reason for the increase in chamber pressure. The motor unit used for this test was the early design and it had mistakenly had standard copper graphite brushes rather than the high altitude silver graphite brushes. Previous tool developments had proved the capability of the silver graphite brush to perform so this is not considered a problem.

Hand Tools.—The tradeoff studies showed that screwdriving and hammering were necessary functions best performed by hand tools. In the evaluation of tools by pressure suited operators it was apparent that the standard screwdriver was inadequate

| Hole Size | Drilling Time | |
|---|---|---|
| 1/4 in. | 21 sec. | .040 7075-T6 Al Power In 210 watts |
| 3/8 in. | 48 sec.[1] | |
| 1/2 in. | 15 sec. | |
| 21/32 in. | 9 sec. | |
| 1/2 in. | 35 sec. | .028 Ti-6Al-4V - ST |

| | |
|---|---|
| Max. Cutting Travel[2] | 3/16 in. |
| Cutting Force | 7 lb. |
| Max. Cutting Torque | 65 in-lb. |
| Impact Release Impulse | .5 lb. ft. |
| Impact Release Travel | .015 in. |

1. Discrepancy in time is from improperly sharpened and heat treated hole saws.

2. Maximum cutting depth is dependent upon hole saw length and needle length, e.g., to cut a hole in the back face of 1-inch honeycomb would require a needle and hole saw 1-inch longer than normal and a second cut.

TABLE 13. DRILL PERFORMANCE

and that a ratchet screwdriver with a pistol or T-grip is preferable. The handle developed for the power tool was adapted to hold a standard Snap On Tool Co. model TM-67A midget ratchet adapter with a 1/4-inch drive shank (Figure 46). The ratchet is reversible and capable of accepting any standard 1/4-inch drive attachment such as sockets, screwdriver blades and extension bars.

The hand hammer was adapted by casting a plastic handle of the same shape as the power tool handle around the normal wooden hammer handle to assure that a good grasp of the hammer is possible with the gloved hand (Figure 47). The hammer proper is a Lixie soft face dead blow hammer. The head contains a cavity which is 75% filled with small cast iron shot. As the hammer impacts the shot controls the energy distribution to minimize rebound and effectively deliver more energy to the work. Comparison of rebound between the dead blow and standard machinists hammer confirms this feature.

In both the screwdriver and hammer portions of the program the emphasis was limited to providing a minimum functional completeness to the tool kit. Hand tool development, as contrasted to power tool development, was not stressed.

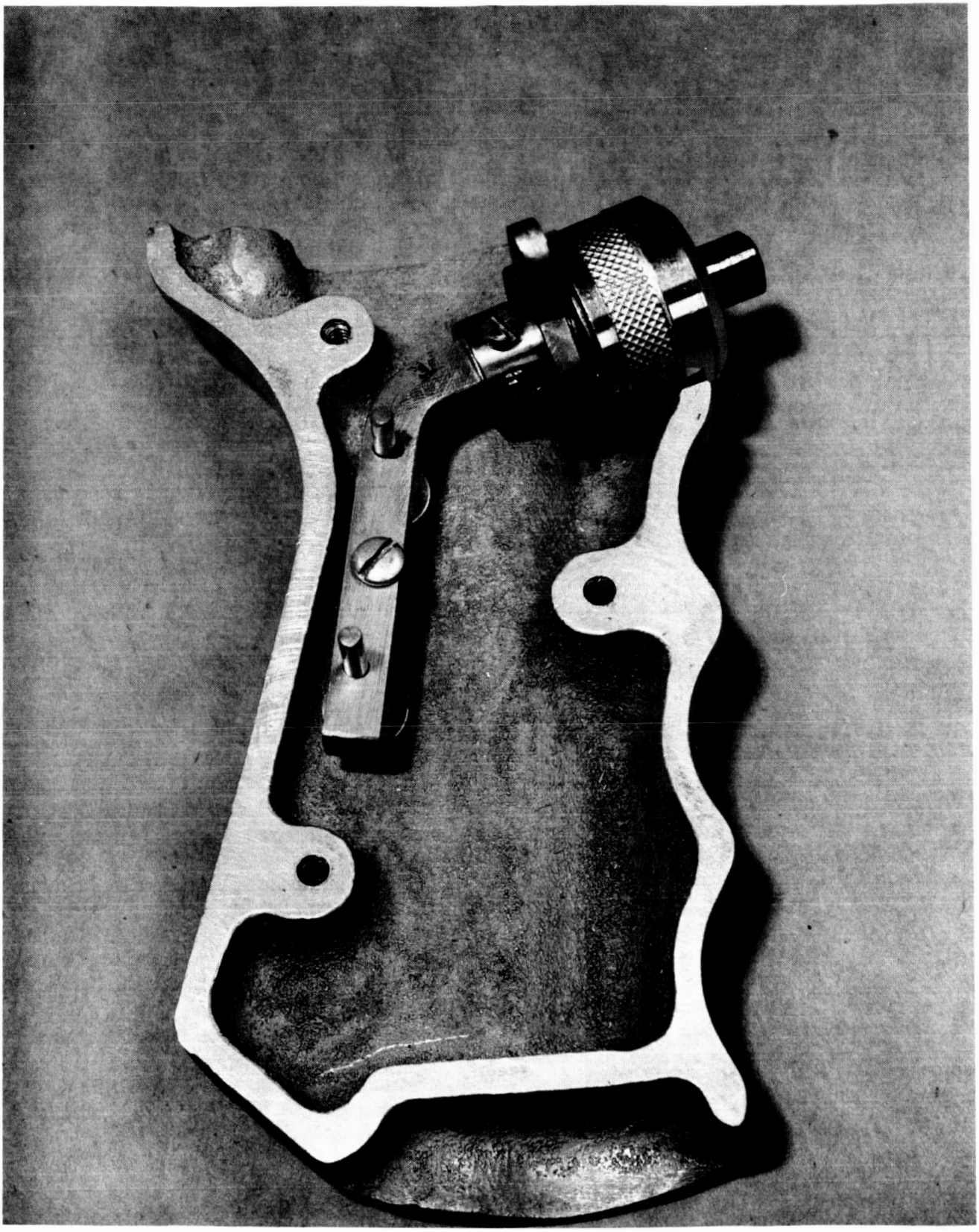


Figure 46. Screwdriver Handle and Ratchet Assembly

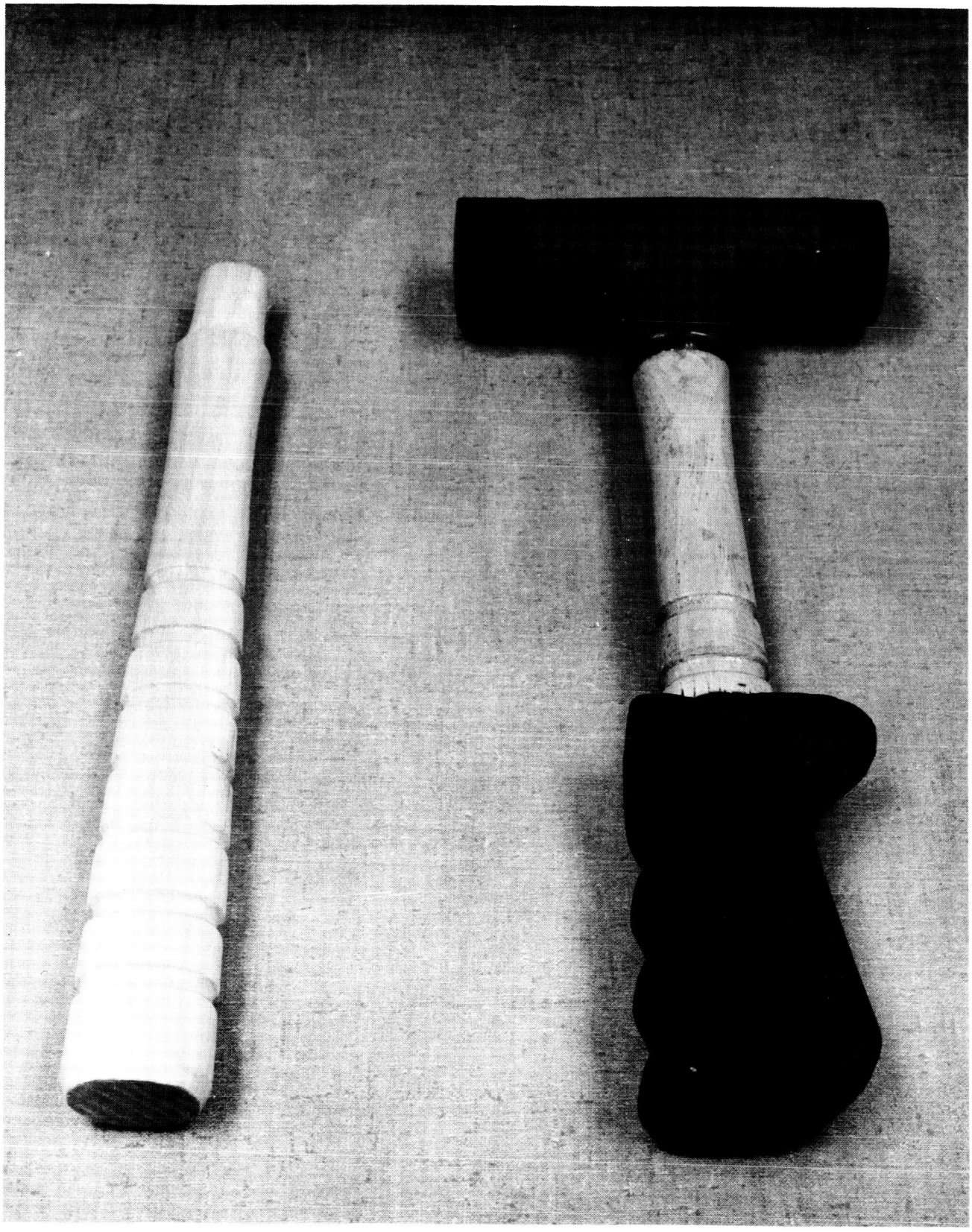


Figure 47. Dead Blow Hammer

## ACCESSORY DESIGN DEVELOPMENT AND TEST

### General

The development of accessories required the solution of three problems, one major and two minor. The minor problems were the development of the work light and small parts manipulation and restraint system. These are minor only in so far as the solutions were fairly well defined by the tradeoff studies and they did not present difficult technological development problems. The major problem was the development of the personnel restraint system. The development of these items concentrated upon design and fabrication. Testing was conducted to determine some performance parameters, but because the subtle interactions of these accessories with the basic tools and the actual work site will be highly significant, it is felt that a full and complete evaluation will require performance of actual maintenance tasks under zero-G and other environmental constraints.

### Restraint System

The restraint system concepts evaluated for development are based upon a basic approach which divides the problem into two parts. On one hand, any restraint system must be attached to the spacecraft , and on the other, any attachment method could be used for any of the various restraint systems. The major developmental effort was concentrated on the method of attachment. Efforts were made in the tradeoff studies to define the basic restraint method. A single point tether is optimum and the definition of the tether was considered no problem at present. The one end of the tether would be a matter of spacesuit interfacing with some sort of clip or hook attachment while the other would be an interface with the attachment to the spacecraft and could be developed as a part of the attachment development. The rigid rope discussed later, represents another more sophisticated tether approach.

The adhesive restraint attachment.—The means of attachment to the spacecraft is dependent upon several considerations. One of the most important is the tradeoff between methods of attaching. It is possible that an attachment could be made mechanically, as with a screw or bayonet type fastener or some type of expandable clamp in a mating hole, but this approach would require location of the mating part at all possible positions on the spacecraft structure where maintenance might eventually be required. Because this would lead to increased weight and complexity of design, it was felt that a simpler system was highly desirable. The goal was set that the system have the following characteristics:

| | Mandatory | Desirable |
|---|---|---|
| Lightweight | X | |
| Install at any point as required | X | |
| Attach in minimum time | X | |
| Reuseable | | X |

| | Mandatory | Desirable |
|---|---|---|
| Removable | | X |
| No preparation of surface required | X | |
| Minimum complexity of installation | | X |

The installation at any point as required, on closer examination, resolves into installation in three major areas which can ultimately be defined only after an extensive maintenance analysis of a particular spacecraft. There will be 1) high usage areas such as equipment compartments which require the continual return to the same point or nearby points to accomplish maintenance tasks, 2) medium usage areas where the frequency of activities is low, but some activity is definitely probable, and 3) the areas where no maintenance activity is planned and the probability of even doing any is low. In the first type of area, it is probable that some type of mechanical attachment fitting could be profitably built in to the structure. In the second and third, it is better to reduce the structural complexity by attaching as needed. The necessity for attachment in the third type of area would be predicated upon other factors affecting the decision to do maintenance such as availability of repair materials and tool capabilities. However, it is wise to seek, at this point in development, the broadest possible capabilities so the restraint attachment should be oriented toward the widest possible usage.

The requirement for reuseability and removability are optional and dependent upon the mix of "in place" and "as needed" attachment points. Basically, the attachment would be most efficient if it did not need to be removed and did not degrade in place to preclude its reuse. The need for removal is again dependent upon other factors which define three basic areas. These are 1) surfaces where the re-entry aerodynamic characteristics are critical, 2) surfaces where the re-entry temperatures would destroy the attachment but without disturbance to the spacecraft re-entry characteristics and 3) surfaces on spacecraft which do not re-enter such as LEM. The present state-of-the-art of re-entry aerodynamics restricts the disturbances allowable on aerodynamic surface profiles. It is likely that this restriction would minimize, if not eliminate completely, the need to perform maintenance in that area because equipment compartment accesses would not penetrate a re-entry surface. On the second type of surface the attachment should present a minimum profile and minimum heat sink to minimize the local disturbance to spacecraft heat flow during re-entry. A device which burns off quickly is preferred. Surfaces which do not experience re-entry present no problems other than that the attachment should not introduce excessive stresses or upset the long term thermal equilibrium by significantly changing adsorbtion or emittance characteristics.

Preparation of the surface should not be necessary because it requires tool use in the work area and the first task to be accomplished on arriving at the work site will be attachment.

The possible techniques to select from are limited. A welding technique which would give almost instantaneous attachment requires surface preparation. The needle penetration technique developed for the drill mechanism is promising, but since the restraint requirements are more rigorous it was decided that the adhesive

attachment fitting would be pursued. By pursuing the needle attachment for the drill and the adhesive attachment for personnel restraint, both concepts could be brought to the point where a knowledgeable decision between them could be made for subsequent refinement.

The adhesive restraint attachment development goals were defined as follows:

1. Strength - 50 psi tension load applied at 100 lb/sec.;
2. Cure time - approximately 5 sec. - 30 sec. required;
3. Cure temp. - $<600^{o}F$
4. Apply to typical spacecraft materials;
5. Maintain strength through $-150^{o}$ F to $300^{o}$ F range
6. Surface temp. at application, $-150^{o}F$ to $300^{o}F$;
7. Applicable in vacuum of $10^{-6}$ to $10^{-12}$ Torr;
8. Store at $85^{o}$ for 6 months;
9. May be applied to one side of bond prior to storage.

The basic attachment concept had been investigated by Martin in-house and an initial design concept developed for a small button which could be bonded to the spacecraft skin. A similar concept was studied by Viron Company for the USAF Aero Propulsion Lab

The basic button concept (Figure 48) evolved into a circular disc of 1.13 in diameter .030 2024 aluminum with a swivel eye in the center. Mounted on the disc is a sandwich of 1/32-inch thick of FM-98 epoxy film and a wire grid of no. 32 copper wire on 1/32-inch centers (Figure 49). The aluminum button is hard-coated with an aluminum oxide surface film to provide electrical insulation. Power connection to the wire grid is made through the two wrapped contacts on the outer edge (Figure 48). The hole between the contact and the swivel is for the applicator thermocouple which was found necessary to assure an adequate bond.

Initially it was hoped that a simple "Band Aid" type of adhesive button could be developed. Tests with contact adhesives showed that it was possible to achieve the required 50 psi strength at room temperature, but the contact adhesives will not maintain strength over the temperature range encountered. Another problem which became apparent early was the discrepancy in adhesive strengths when subjected to tensile/shear or peel. Peel strength is several orders of magnitude lower and is therefore the limiting factor. The sketch below shows that under a peel situation, the stress is concentrated on the peel line which has a very small effective area. The adhesive is locally stressed beyond its ultimate strength and when it fails the stress point merely moves back without reduction of the local stress on the adhesive. Actually, as the peel line moves closer to the center of the button the moment arm

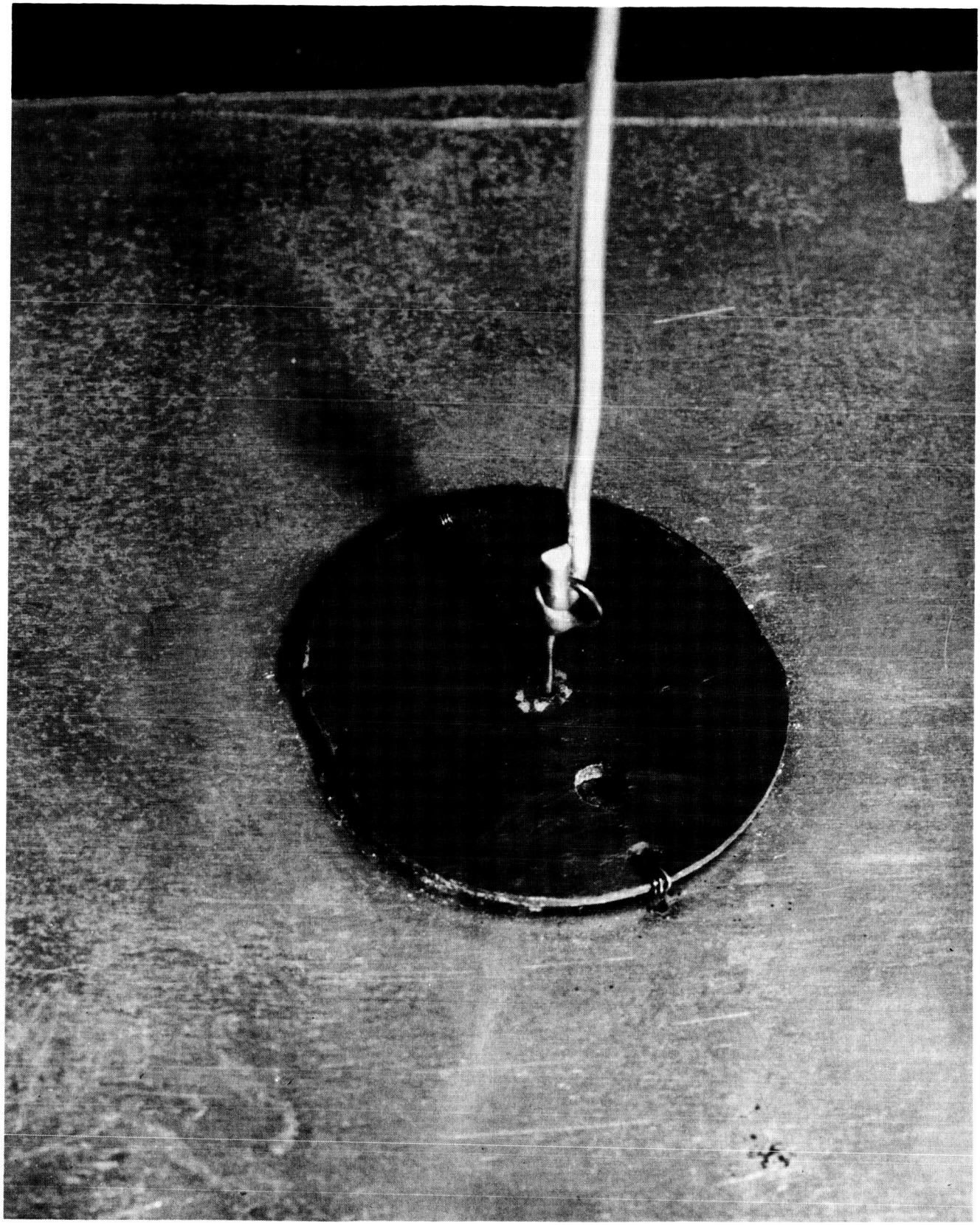


Figure 48. Adhesive Restraint Top View

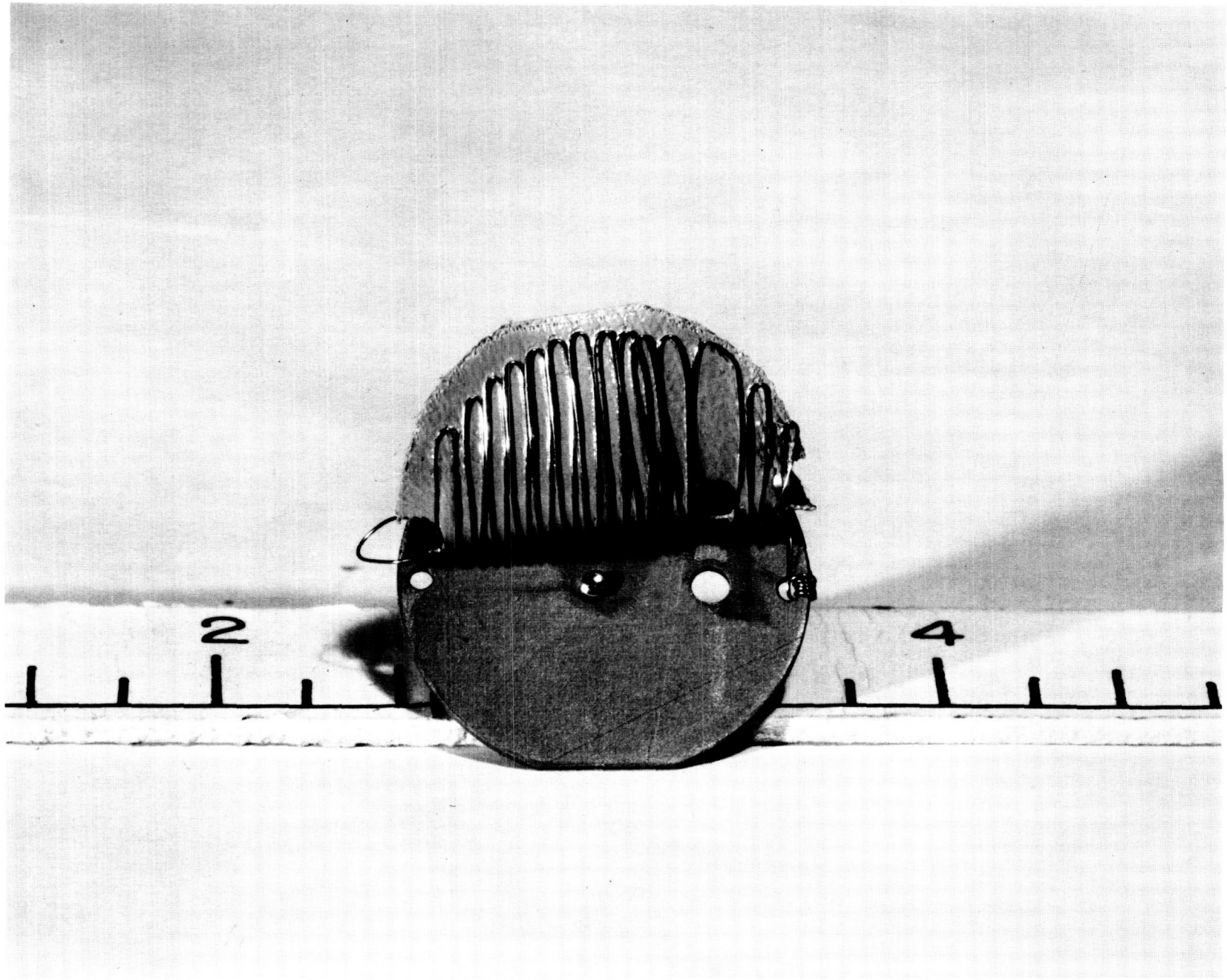
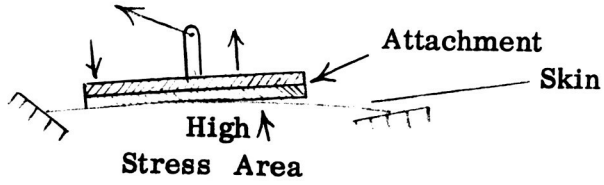
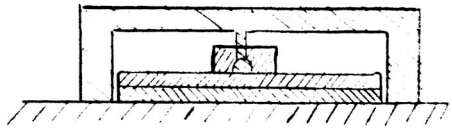



Figure 49. Adhesive Restraint Bottom View

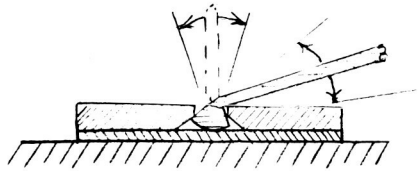



of the peel force is reduced, increasing the local stress and accelerating the process. The solution to the problem is to eliminate the peel force to take advantage of the tensile shear strength of the adhesive. A mechanical system for eliminating the peel force had been developed and tested in house. The essence of the system was a two part button as shown below.



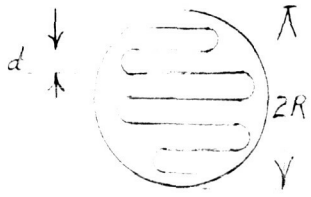
The inner disc has a swivel joint at its center. Around the disc is a ring with a bridge structure to connect the outer and inner pieces through the swivel. Angular loads on the outer piece are resolved into shear and peel forces on the outer structure and pure tension on the inner disc. The swivel allows the inner disc to distribute the tension load over the whole button. Even when the outer ring is completely free it cannot slide sideways and develop a peel force because of the mechanical fit of the two parts prevents it. During the developmental testing of the buttons, where the direction of tensile forces could be controlled, a simple swivel was used without the outer ring. The swivel was fabricated from a pin. The oval head of the pin fits into a countersunk .030 hole; the point is then bent into a circle and soldered. The pin heads have been tested to 70 pounds failure loads. Since the design strength of the button is 50 pounds, this is considered adequate. Refinement of the swivel for greater loads is primarily a matter of material selection.

An analysis of the loads on the restraint fitting during testing showed that most loads could be expected at angles fairly close to the normal. The swivel construction would not develop peel forces because of the short moment arm across the pin head and if the angular limits of the swivel travel were exceeded it would merely bend. Since the excursions of tension forces away from the established position would be low, the bending would be a one time occurrence and fatigue failure of the swivel would not be expected. The distribution of loads with the swivel bent would



still be possible to assure the use of the full tensile shear strength.

The adhesive system was developed in a series of tests based upon an available honeycomb bonding adhesive. During a survey of adhesive manufacturers, it became apparent that the adhesive strengths required for the restraint system fall in the range of failure of a structural adhesive so that data on adhesives which would meet the requirements was not readily available. Discussions with several manufacturers revealed that few adhesives could be expected to meet the temperature, time and strength requirements. The Bloomingdale Department of American Cyanamid Corporation recommended investigation of their FM-96 honeycomb bonding epoxy film which they felt could be used, although they had no data which showed its suitability. The normal cure cycle for the adhesive is 30 minutes at 300 degrees F. The epoxy cure cycle could possibly be accelerated to less than 30 seconds at a temperature of 600 degrees. A series of samples of adhesive sandwiches was bonded, by heating with a torch, to establish the initial feasibility of a quick cure cycle. Temperatures were not monitored but the adhesive was visually observed for a characteristic color change and the time was recorded. When the adhesive had cured, the sandwich was quenched in water and tested in tensile shear. Good bonds were obtained in times varying from 5 to 30 seconds. Button samples were then fabricated from 1.13 in. aluminum discs to give a 1 square inch bonding surface so that data would be directly translated into psi strengths. An adhesive sandwich was fabricated using a wire grid embedded between two layers of adhesive. Initial experiments concentrated on developing a grid pattern which would be uniform over the button surface. Plain copper wire was selected rather than resistance wire because the resistivity is lower and the heat distribution could be better optimized. A grid of parallel wires on 1/32-inch centers was fabricated from 28 inches of No. 30 wire.




$$L = \frac{\pi R^2}{d} + n \frac{d}{2}$$

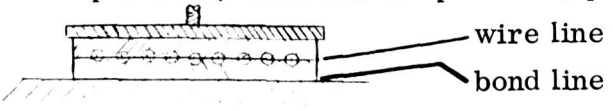
$$\frac{nd}{2} = R$$

$$L = \frac{\pi}{d}(R^2 + R) = 27.6 \text{ in.}$$

L = length of wire
r = Radius of button (.507 in.)
d = spacing (.031 in.)
n = number of segments

Tests were made of wire heating under high currents to establish the basic parameter of wire size. Length and voltage were determined by the basic configuration. No. 30 wire drawing about 10 A was found to be optimum.

A series of buttons were bonded and tests made of their tensile strength. Recordings were made of bond line temperature, wire line temperature, power input and




bonding time. With about 80 watts input, bonds could be made in 30 to 60 seconds with strengths of 50 pounds. The recordings of wire line temperatures showed that a drop of about 250 degrees F was experienced across the 1/32 in. layer of adhesive. Since the adhesive begins to char and lose strength at 700 degrees, this limits the

maximum bond line temperature to 450 degrees. Efforts to accelerate the bonding by introducing more heat would merely char the adhesive at the wire line and weaken the adhesive at that point.

A consultation with Bloomingdale on this problem resulted in use of their FM-98 which is the basic system with a 70% filler of aluminum powder. With this the drop in temperature was reduced to 150 degrees F and the required bond line temperature could be reached in under 30 seconds.

With reasonably reliable bonding performance at room temperatures it was possible to examine the problems of temperature variations. Since the adhesive cures at high temperatures, it was decided that the low temperature performance would probably present the more difficult problem. A cold test chamber was prepared for testing at low temperatures. Figure 50 shows the test equipment. The aluminum plate with the test sample and thermocouples has an aluminum wick which is submerged in the flask of liquid nitrogen. The initial test was conducted on six buttons which were bonded at room temperature (Figure 51). With 12 V dc, 8A input the bond line temperature reaches 400° F in 15-20 seconds which is sufficicient to achieve about 50 pounds tensile strength. At 3 seconds after power is applied, a 1 sec.-1 lb. push was given to seat the button on the surface. During the rest of the cure cycle the only forces holding the button and test surface together were the weight of the button and the wetting strength of the adhesive. The test surface was .030 in. thick aluminum. Six (6) buttons, bonded at ambient temperature, were cooled to -250° F and pulled with the following results.

| Button | Pull Force at Failure | | |
|---|---|---|---|
| #1 | | 2 lbs. | |
| 2 | | 24 | |
| 3 | | 57 | |
| 4 | | 33 | |
| 5 | | 13 | |
| 6 | T + 0 min | 50 | Pulled 6 times |
| | T + 5 | 56 | |
| | T + 30 | 53 | |
| | T + 60 | 35 | Test tool failed |
| | T + 90 | 70 | |
| | T + 150 | 80 | Eye broke on button |

Button #6 was cured for 27 seconds with an average power input of 126 watts.

Button #1 failure may have been caused by a misaligned thermocouple which caused bending in the test surface when it was screwed onto the cold plate, with a consequent peeling action.

This confirmed the inherent strength of the bond and attention was turned to accomplishing a reliable bond on surfaces of varying temperature.

In the course of development testing, it had become apparent that the bonding action was highly dependent upon initial surface temperature and the heat sink characteristics of the surface. It was discovered that exceeding a critical temperature was

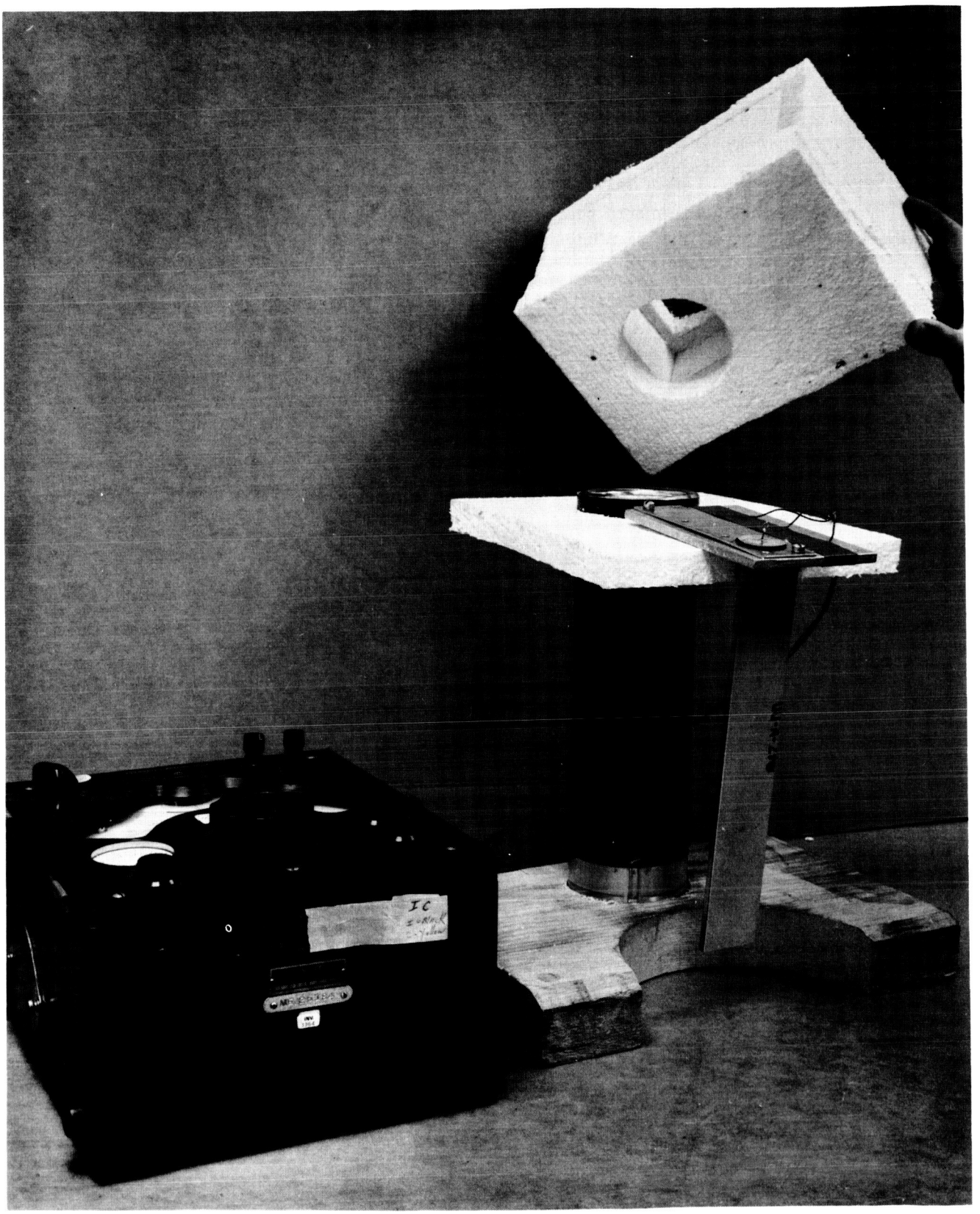


Figure 50. Adhesive Cold Test Equipment

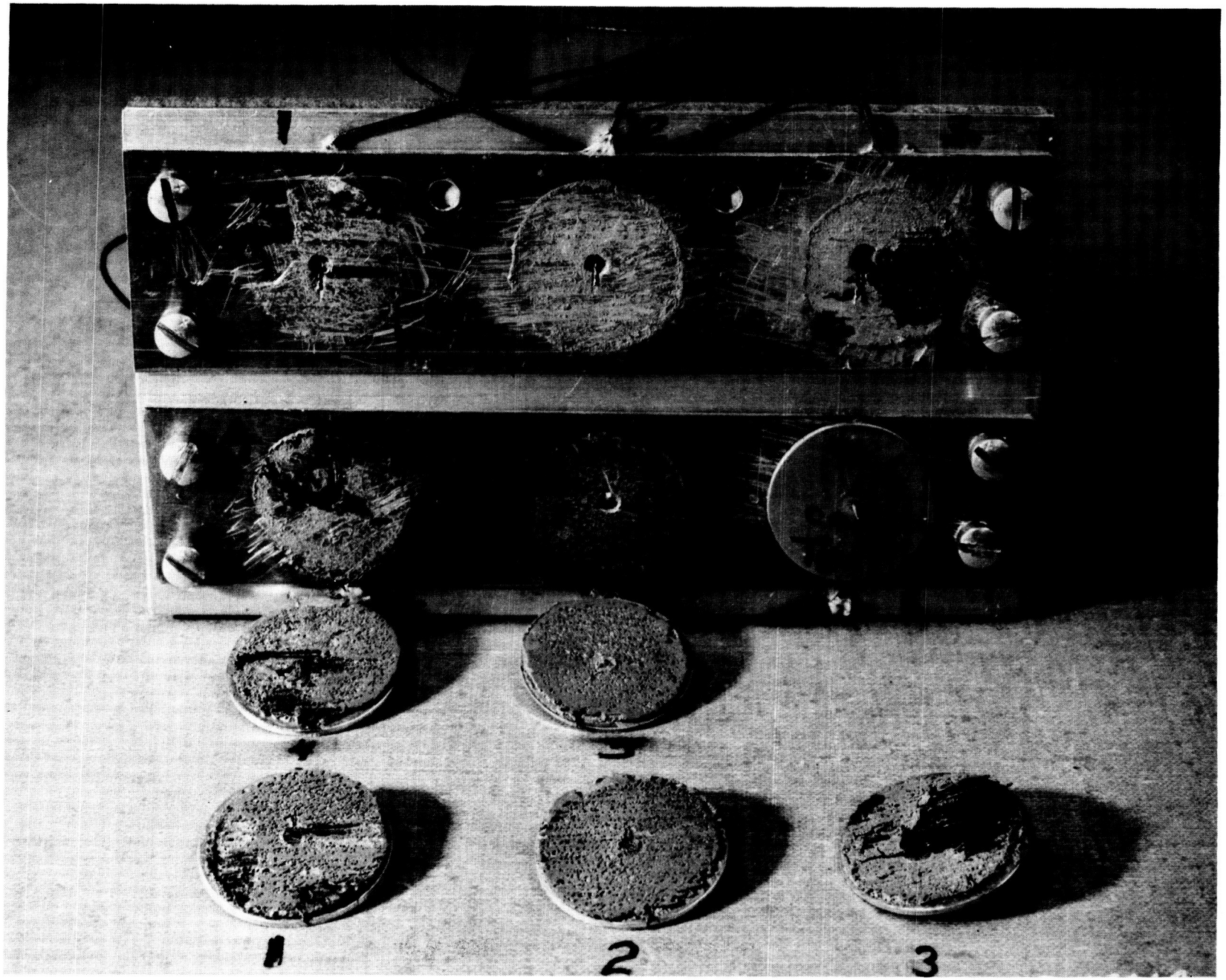



Figure 51. Cold Test Samples

required, but that using time alone as a controlled variable was an unreliable indication of good bonding. A modification to the buttons was made to permit thermocouple monitoring of the bond line temperature and a monitoring circuit developed (see Figure 71 - Tool Kit Electrical Circuit). By this time the basic button configuration was established and development was started on the applicator mechanism. The applicator had to accomplish the following:

1. Apply electrical power;

2. Monitor temperature and give positive indication of proper bond; and

3. Allow limited astronaut drift during cure cycle.

The applicator developed has two basic parts, which are used with the tool kit screwdriver handle (Figure 52) plus the monitor circuit mounted in the tool kit. The button holder on the left holds the button in place and provides the necessary electrical contacts and thermocouple. In figure 53 can be seen the electrical contacts, the thermocouple probe and the slot for the eye of the swivel on the button. The trigger assembly in Figure 52 houses the indicator light, which comes on when the bond line reaches 450°F, the trigger switch which energizes the power and thermocouple circuits and the pressure probe which allows a controlled impulse to be applied to the button to seat it properly as the epoxy softens.

Tests were made to determine the capability of a subject to apply a light pressure to the button without applying enough to push himself away before the button is bonded. Figure 54 shows the relationships between push impulse, drift time and distance for a 250-pound astronaut. This indicates that an impulse of 0.2 pound seconds or less is necessary to allow him to remain within reach during the average bond cycle. If he can drift up to 2 feet an impulse of 0.5 is acceptable. In the final testing of the applicator a pressure suited subject applied seating forces with the applicator to the frictionless simulator which was weighted to provide a 250-pound mass (Figure 55). The relative motion is similar except that the astronaut stood still and the work was accelerated. Control of impulses to limit drift was demonstrated as feasible, although a training period will be required. Impulses resulting in drifts on the order of 10 ft/min have been made in the pressurized condition. Six feet of cable was installed on the applicator to allow for the 30-second bond time.

After the applicator had been developed and the button configuration established performance testing of the adhesive system was started. While the buttons were being bonded erratic temperature indications were noted. The monitoring circuit controls a light which comes on as the thermocouple output reaches 13 milliamps indicating a temperature of 450° F. In some cases the light would fail to come on even though a good bond was achieved. It was discovered that when the button was pushed to seat, the adhesive would flow into the probe hole and lift it off the surface, causing erratic readings. One layer of adhesive was removed to reduce the flow and correct the problem. It was then discovered that erratic indications were occurring because the thermocouple circuit (Figure 71) was seeing the battery voltage on the thermocouple input. The epoxy becomes conductive as it is heated and before it is cured and this was causing a short circuit. This was corrected by isolating the monitor circuit with a separate battery source which is switched on when the bonding power is applied.

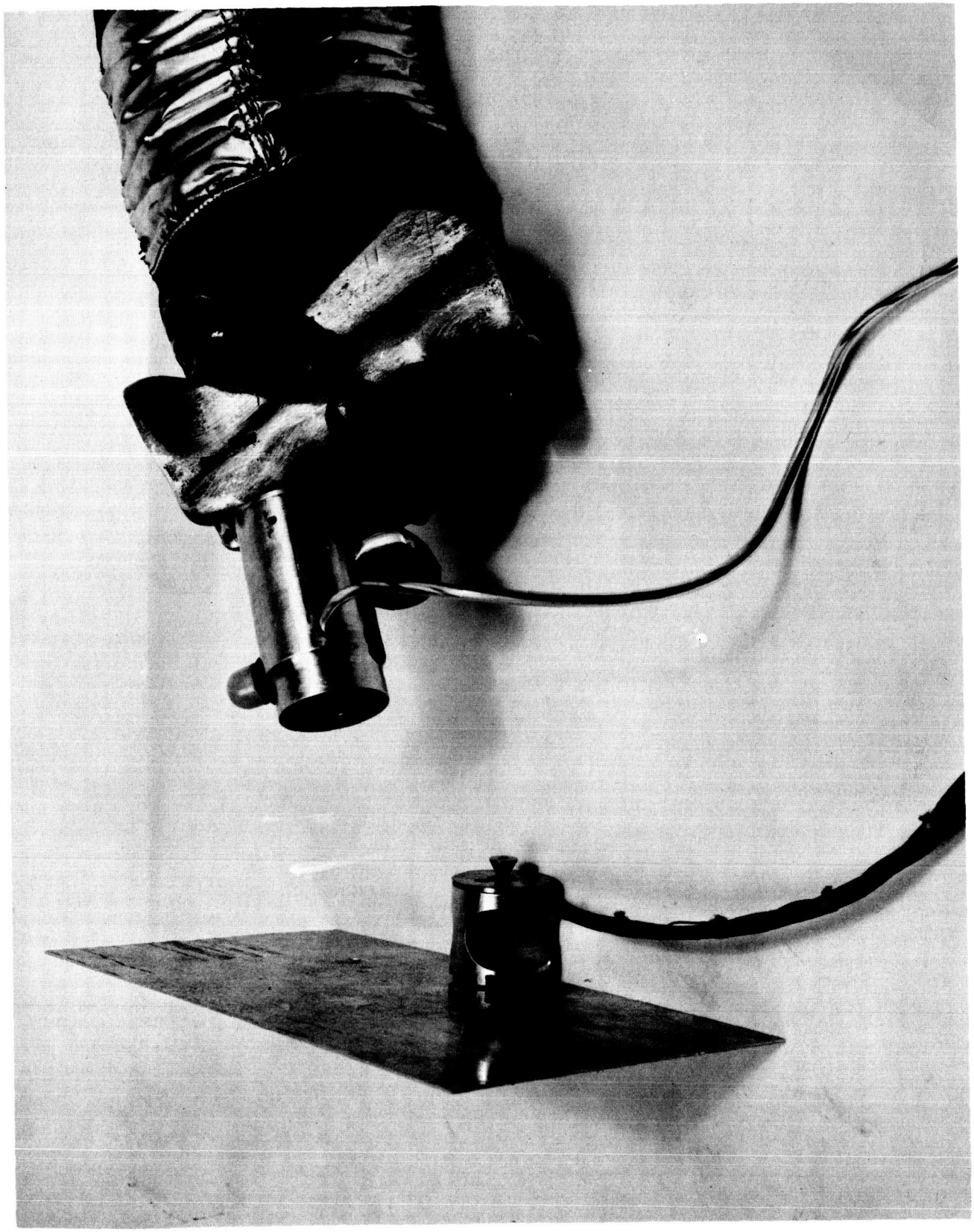


Figure 52. Adhesive Applicator

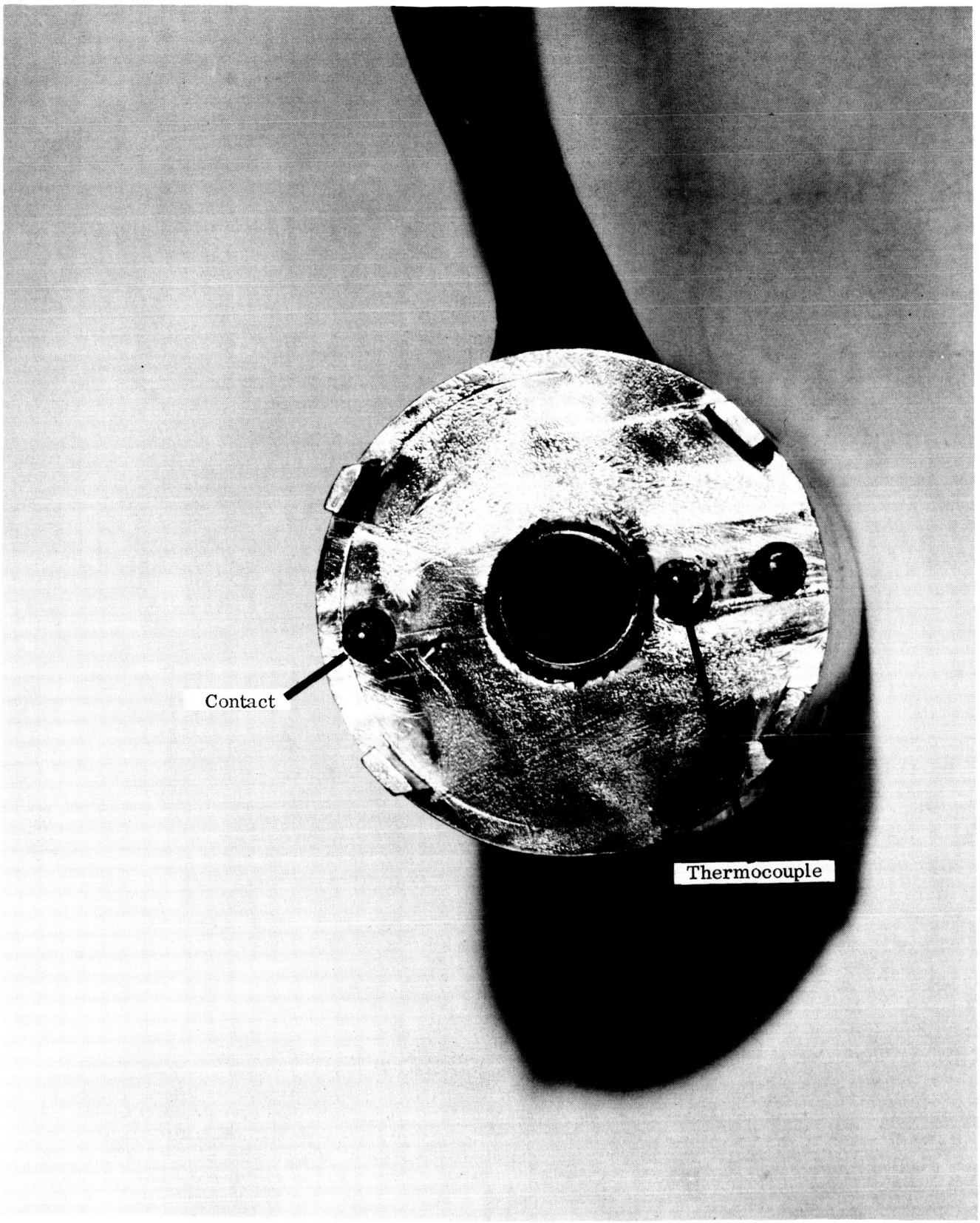



Figure 53. Button Holder

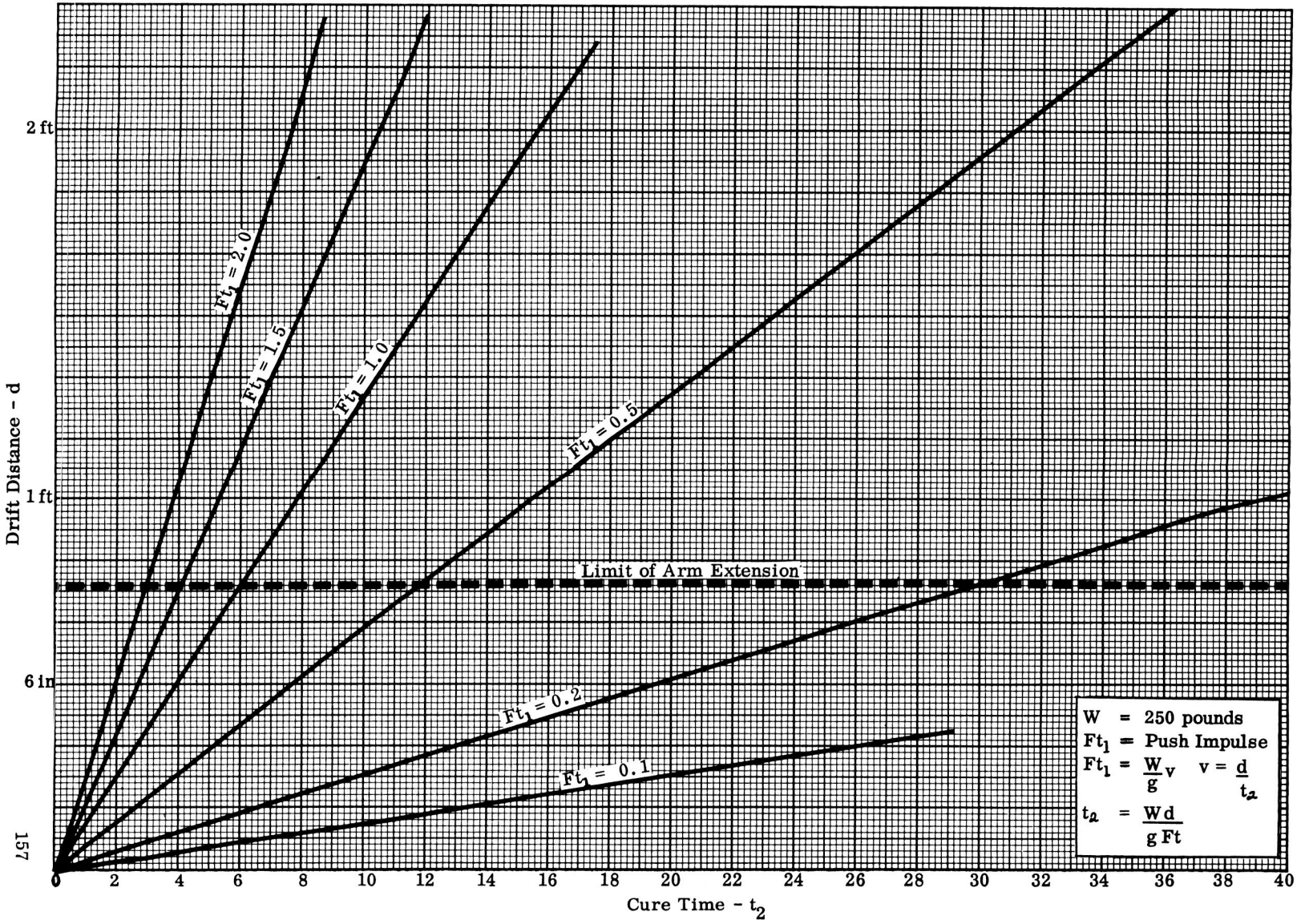



Figure 54. Push Force vs Drift Time and Distance




Figure 55. Push Away Tests

A series of buttons was bonded to varying thicknesses of aluminum at varying initial temperatures and test pulled in tension after bonding. Figure 56 plots the relationship of the bond time versus initial temperature for different thicknesses of material. The curves arising from the left are typical time temperature curves of individual buttons bonded on surfaces at the initial temperature indicated by the origin of the curve. The points plotted show the end points of the bonding cycle for the samples and indicate whether the bond strength was above or below 50 pounds. In most cases the bond strength was either 10 pounds or less, indicating the critical temperature had not been reached, or the button experienced a mechanical failure of the swivel eye indicating a good bond. On the basis of this sample, a critical temperature line was plotted. The extension of the curve into the 450 to 600 degree F region was accomplished by placing samples of adhesive on a hot metal plate and recording the time required to cure to a hard state.

The heat sink capacity of the bond surface plays a significant part in determining the bonding time. Figure 57 is a family of curves illustrating the effect of material thickness and initial temperature on time required to achieve a good bond. Prediction of bonding time is accomplished by entering the chart at the surface temperature and moving up to the appropriate thickness. The bonding time is then read as the ordinate of the intersection.

Although significant progress has been made in developing the adhesive system the performance can be improved in the areas of shortening the bond time by achieving higher temperatures at the bond line, increasing the mechanical strength of the swivel and developing an adjustment, preferably automatic, to compensate for different heat sink capacities.

Rigid Rope.—Part of the problem in restraining the astronaut is that his mobility and dexterity limitations while working in a pressurized spacesuit make it highly desirable that he be able to change position easily, while at the same time, to apply significantly large forces to the work he must be restrained. In a test of restraint adjustment two pressure-suited subjects were placed in the simulator and given the task of removing and replacing a 6-inch square metal panel located 9 inches behind an access panel simulating the side of the spacecraft. The simulator was locked on all axes to simulate a rigid restraint and the subject was instructed to perform the task sequence as far as possible in one position. He would be allowed to change position only when a sub-task was impossible to accomplish in his present position. The new position was then held until it was impossible. A second run was made with different ground rules. In the second run the subject was allowed to change position whenever he desired. Only one subject was run in the second sequence because of a suit failure. In the first sequence 8 and 6 positions were used by the two subjects. In the second run the first subject requested 35 changes of position.

A possible solution to the conflicting requirements of rapid shift of work position and adequate strength is the "Rigid Rope" (Figure 58). This is a prototype device which can be made rigid or flexible at will. The device shown on the photograph is a feasibility model which is quite large and heavy. The mechanism could be refined to accomplish the same task with a much smaller device. In operation the end fastened to the metal block would be attached to the spacecraft and the end with the lever would attach to the spacesuit. Actuation of the lever would unlock all joints simultaneously, allowing the relative position of the worker and work site to be

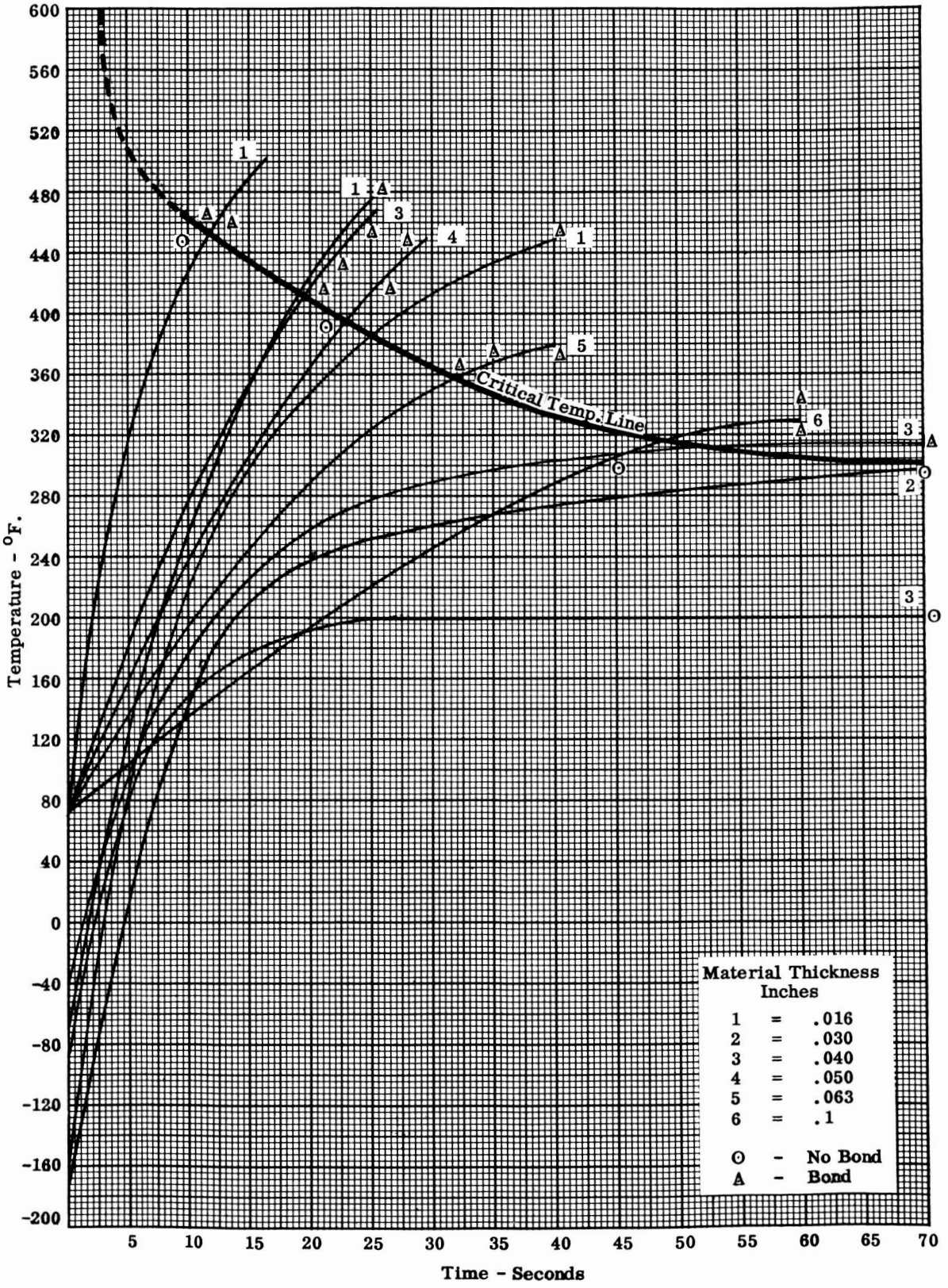



Figure 56. Adhesive Button Performance

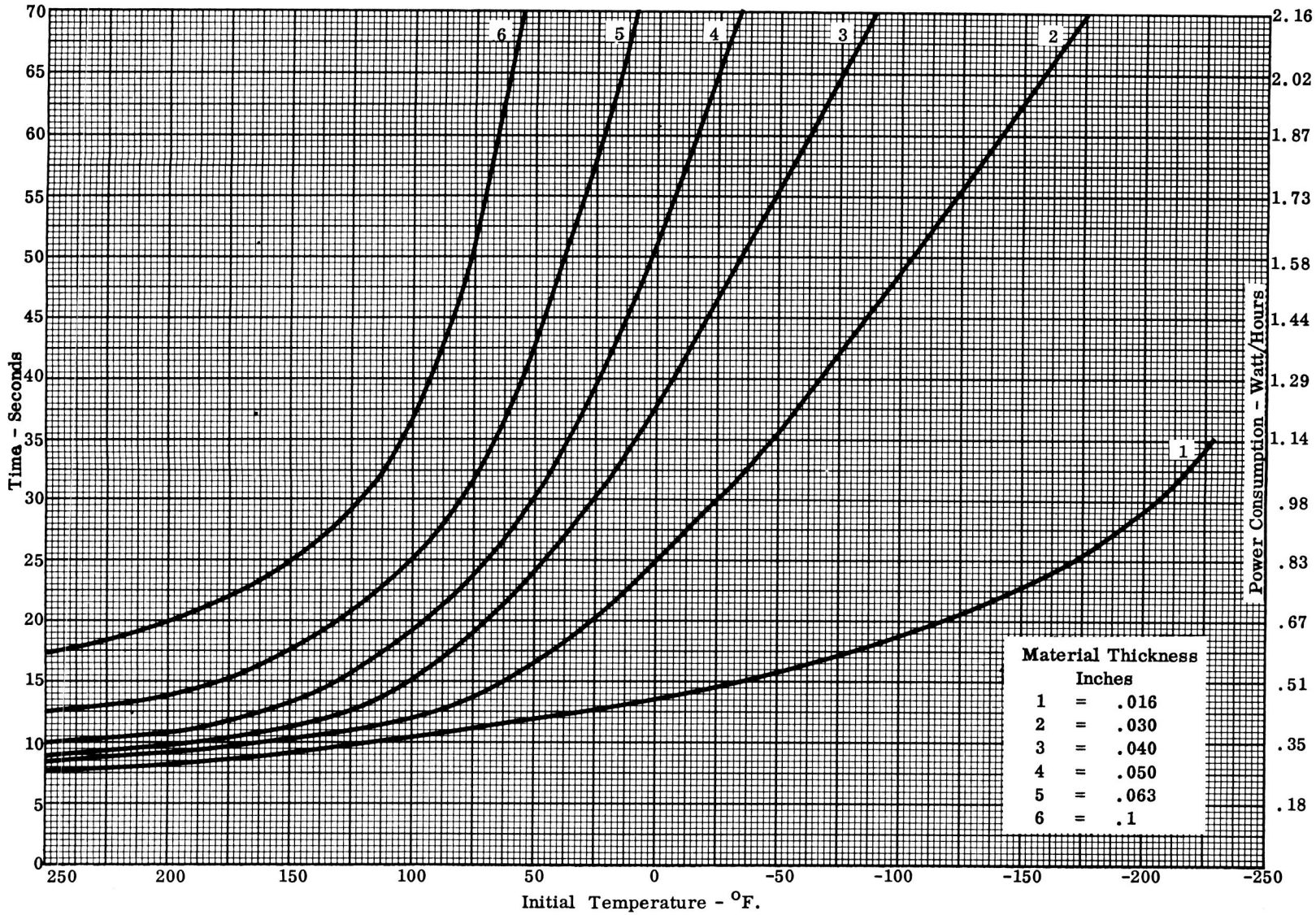



Figure 57. Predicted Bonding Time

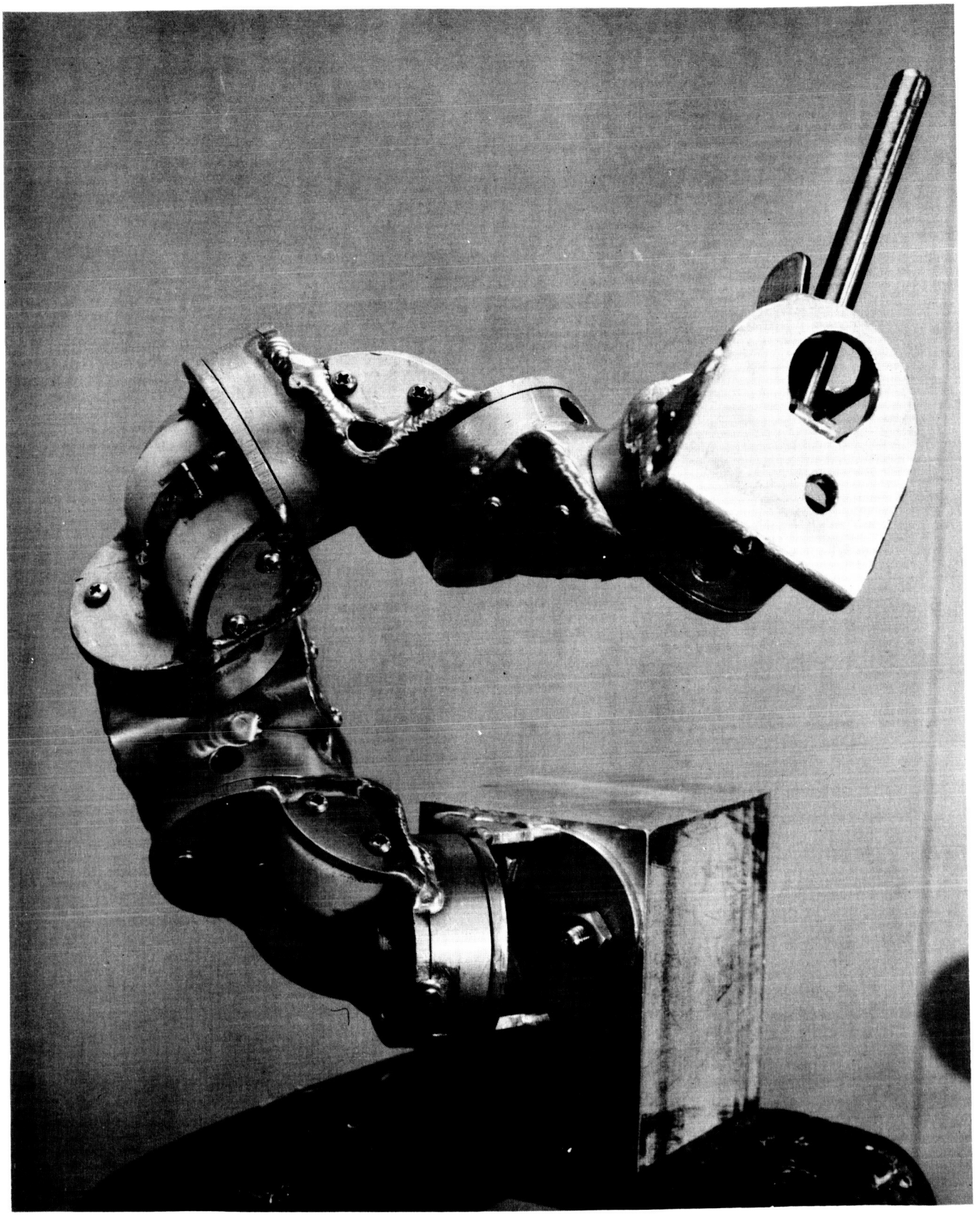


Figure 58. Rigid Rope

changed with little effort. When the new position is achieved all joints are locked simultaneously and the system becomes rigid. The model shown has successfully resisted 100 foot-pound torques.

The joint mechanism (Figure 59) is a set of ring gears attached to each side of the joint structure with a spur gear of the same pitch diameter mating with both. The spur gear is moved in and out of mesh in all joints simultaneously. When it is meshed with both ring gears it acts as a fine-toothed spline and transmits the torque across the joint. With a 64-tooth gear each joint can index a minimum of 5.625 degrees. At a two foot radius this would correspond to a change in position of about 2 inches.

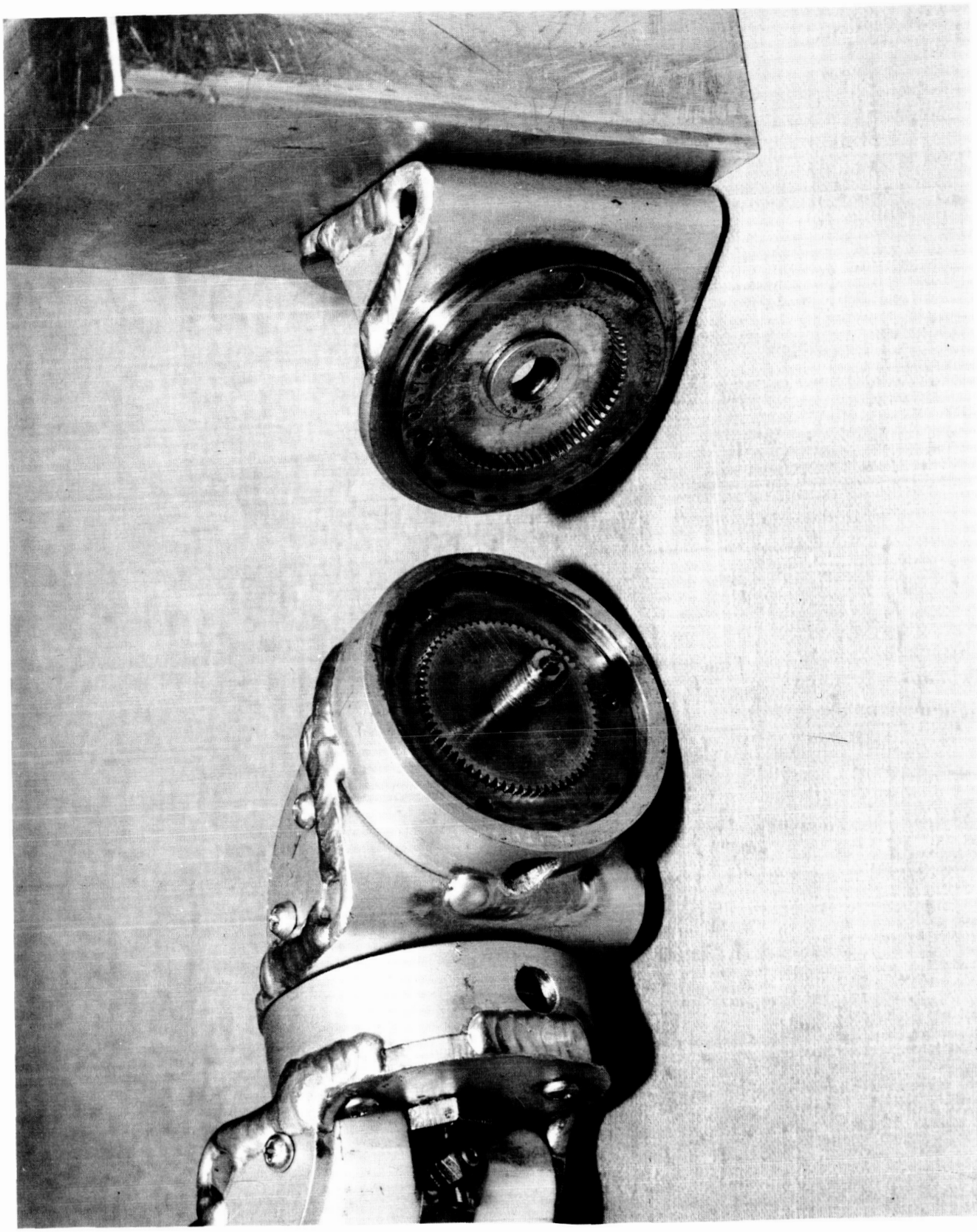


Figure 59. Rigid Rope Joint

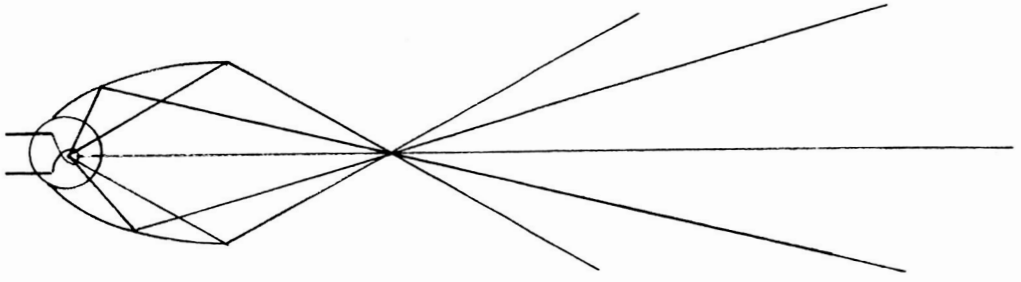
## Small Parts Manipulation and Restraint

The problem of small parts manipulation and restraint requires that the astronaut have a simple secure method of grasping parts and safely place them in a storage device. The storage device developed (Figure 60) consists of a set of small diameter brushes set vertically in a holder so that parts like nuts and washers with holes can be inserted over a single brush while parts like screws which are solid can be inserted between adjacent brushes. The stiffness of the short bristle brush is more than adequate to hold lightweight parts, yet the insertion and removal forces are reasonably low. The final configuration (Figure 61) was used in the toolbox for holding parts and the hole saws used with the drill.

In addition to the storage device a "fingernail" was fabricated which slips on over the space suit glove to make it easier to pick up and handle small items. This type of device should be integrated into the glove design as an optimum measure.

## Work Light

A simple work light was fabricated to provide illumination to meet the requirements established in the human factors studies. The work light (Figure 62) was turned from bar stock and the fins were left on for machining purposes. A refined model would be lighter and smaller if spun. The reflector shape is a $30^{o}$ semiellipsoid of revolution. The 12 VDC 6.9 watt GE No. 68 lamp is mounted so that the filament is located at the focus. The beam then appears to emanate from the conjugate focus and maximum reflector efficiency is achieved.



The beam patterns shown in Figure 63 are taken parallel and perpendicular to the finament plane and represent the 40 foot candle beam contour. The change in brightness as a function of input voltage affects the total light output but not its distribution. One light is mounted on each side of the helmet at the temples in the optimum location. Slight adjustments in beam spread by focusing are possible but an optimum ellipsoid integrated into the helmet would be possible.

## TOOL KIT DESIGN DEVELOPMENT AND TEST

Several tools and accessories must be integrated into a complete package capable of being handled efficiently by the astronaut. The basic shape for the tool kit was established by the limitations of pressure suit performance and work site layout. Within this basic shape it was necessary to package the items listed in Table 14 and shown in figure 1. Because there were three separate items using electrical power it was necessary to

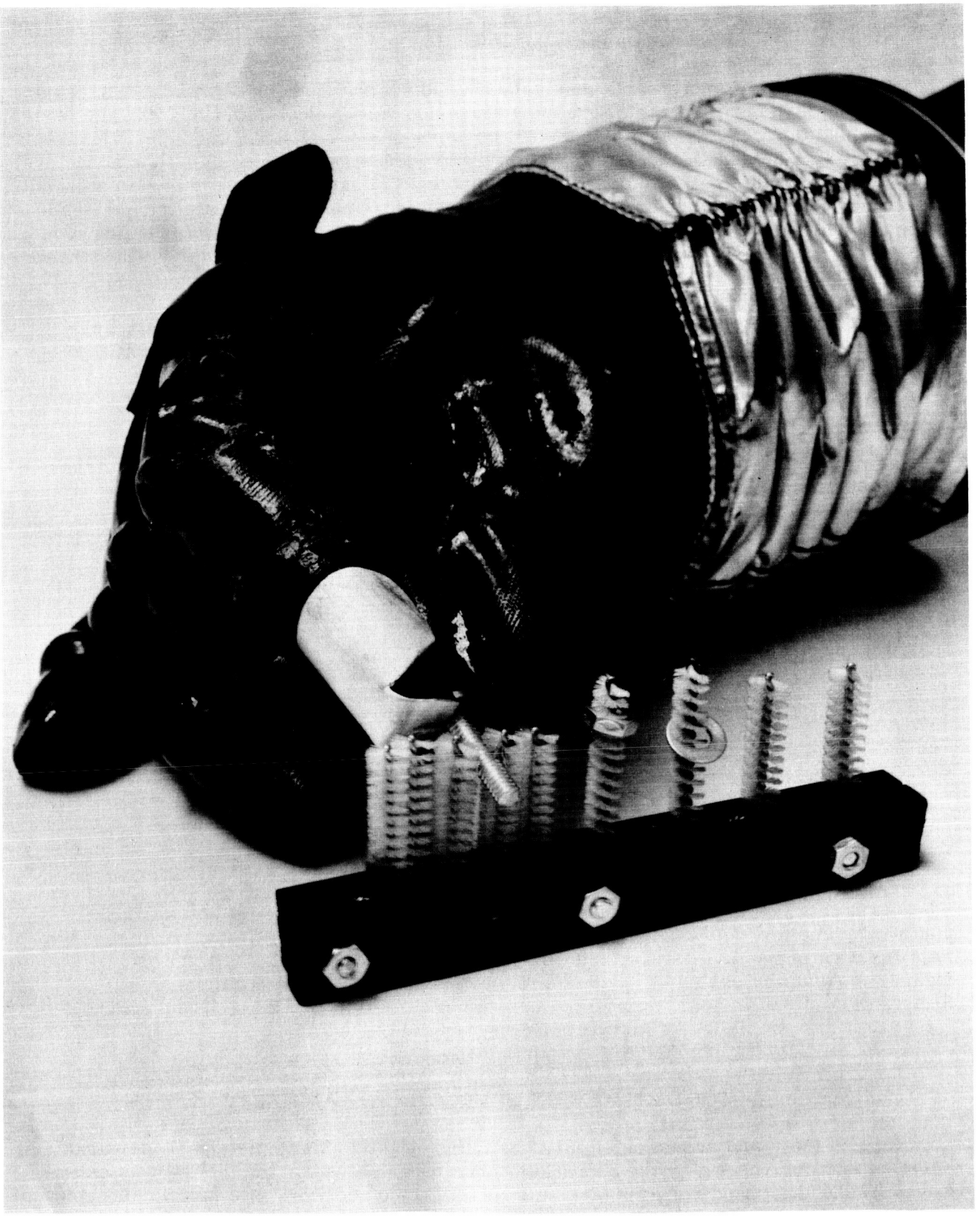


Figure 60. Small Parts Holder

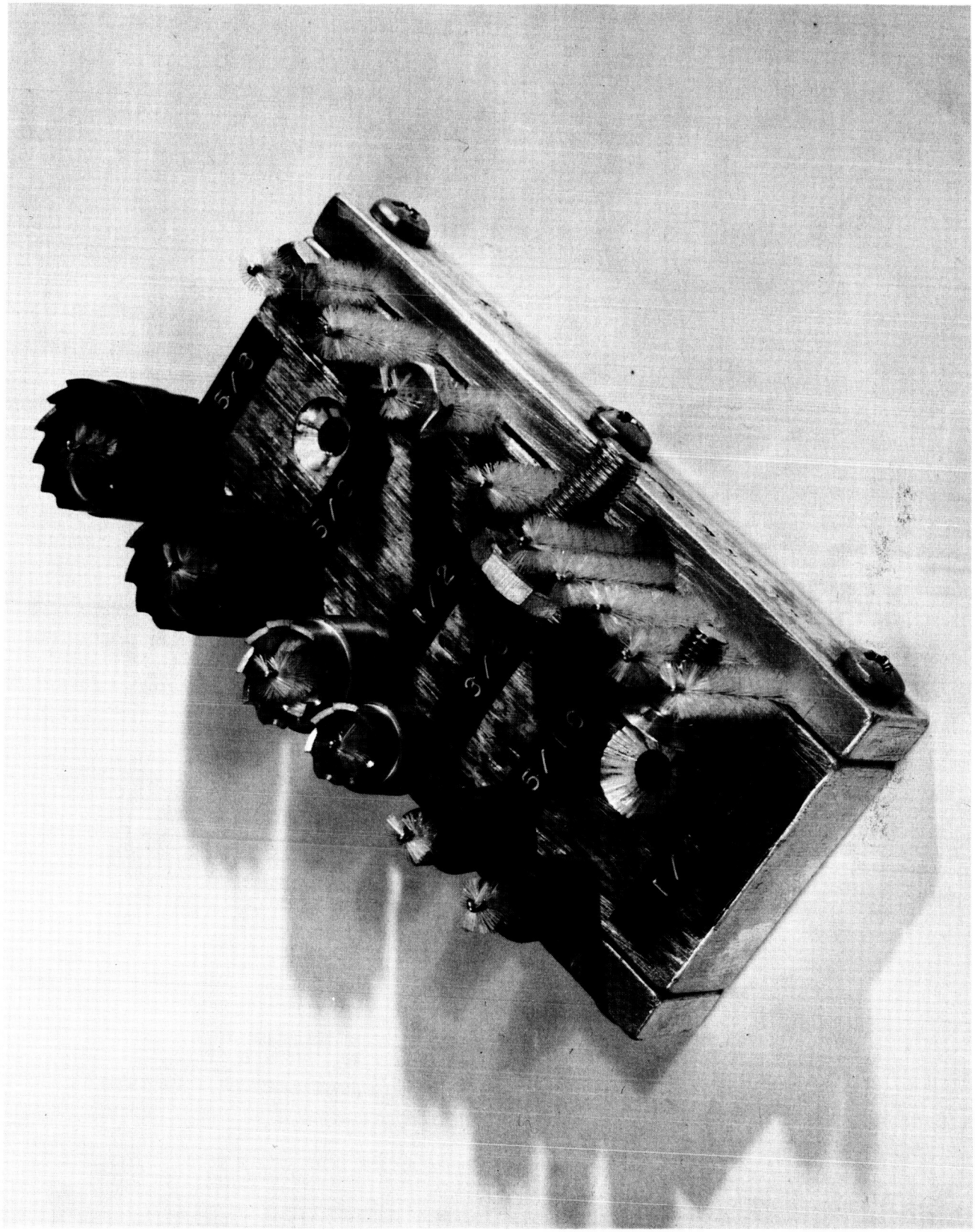


Figure 61. Small Parts Holder - Final Configuration

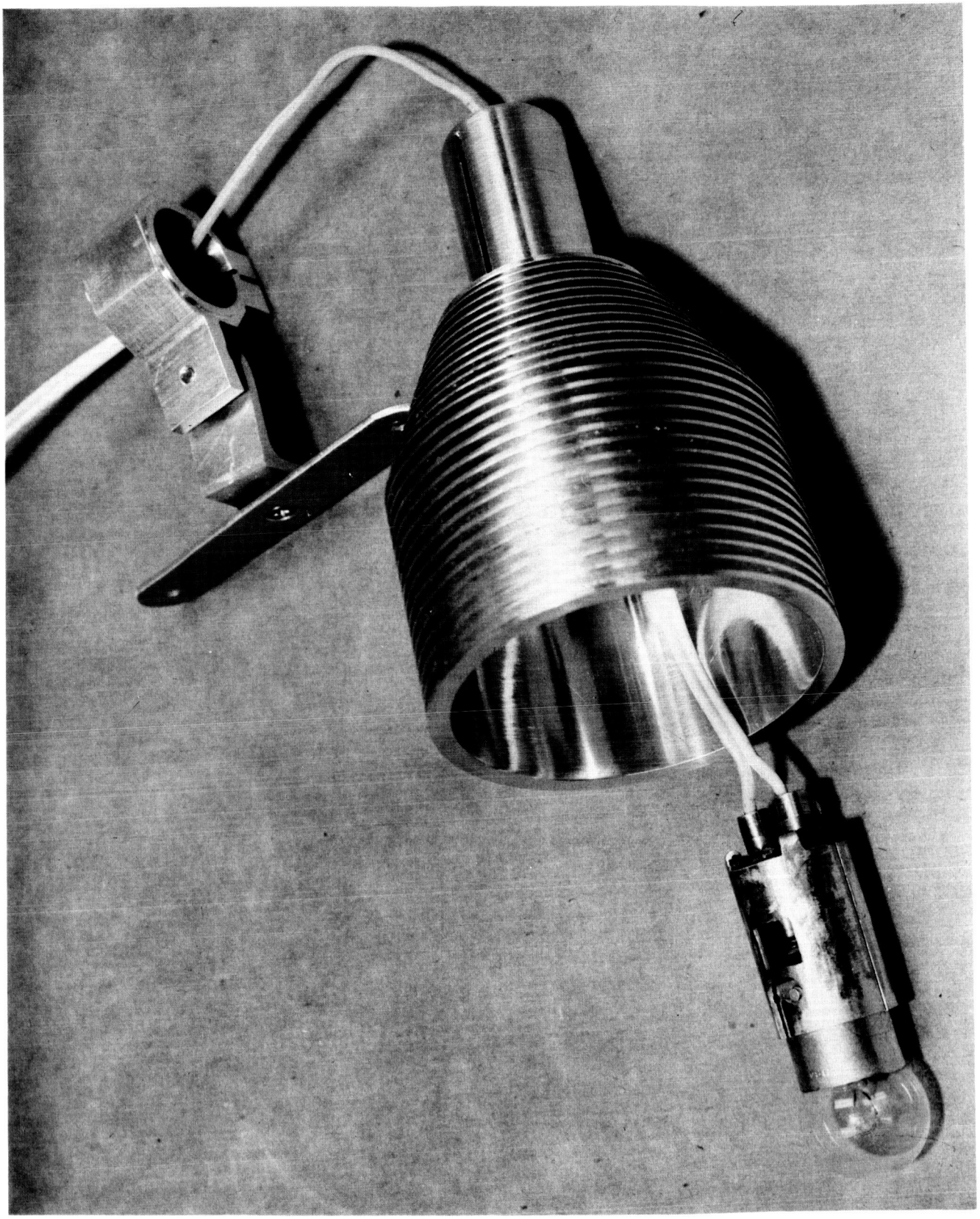


Figure 62. Work Light

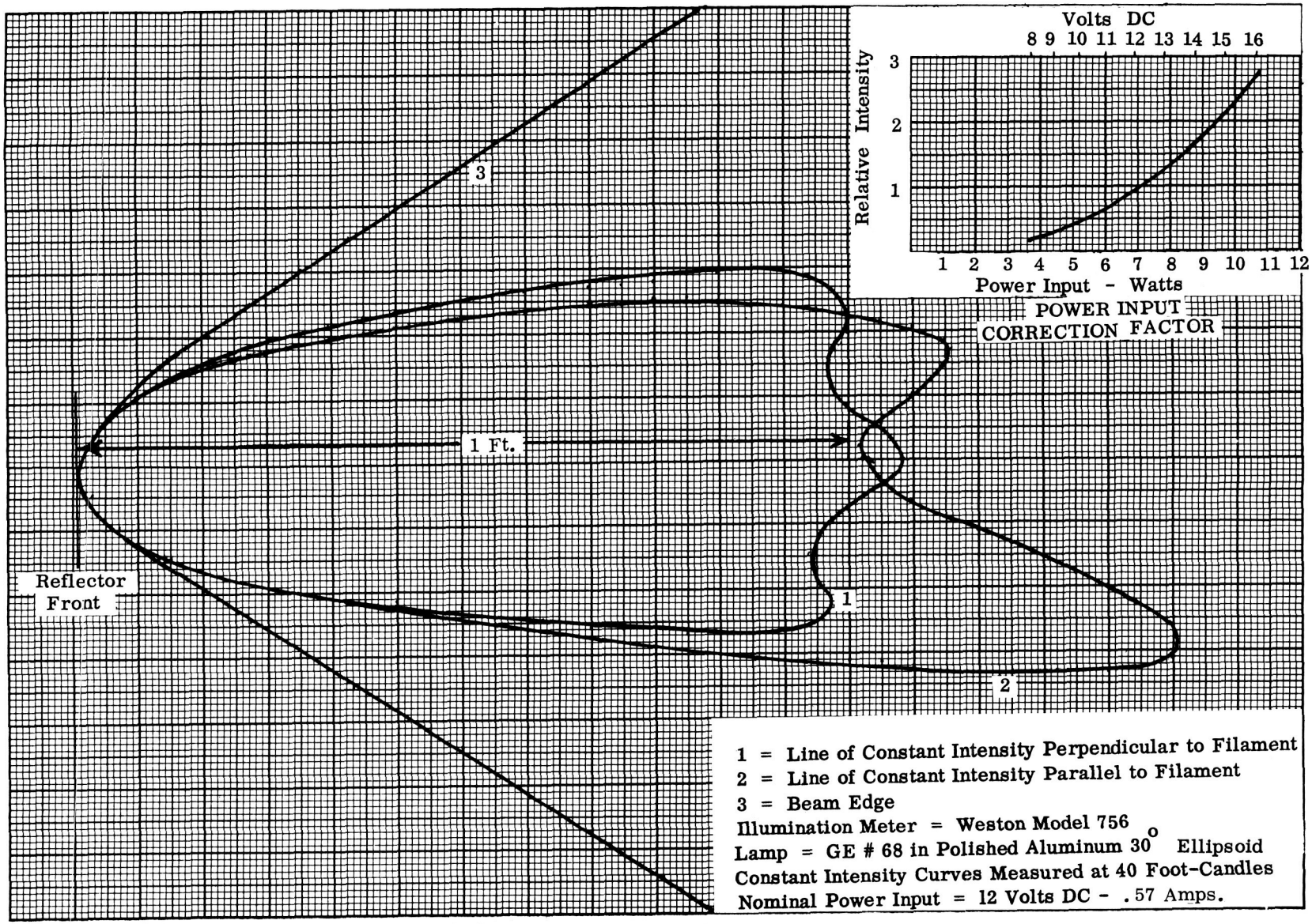



Figure 63. Work Light Beam Profiles

TABLE 14

TOOL KIT CONTENTS

1. Motor Unit
2. Impact Attachment
3. Socket - 1/2-inch drive - 3/4-inch
4. Socket - 1/2-inch drive - 5/8-inch
5. Socket - 1/2-inch drive - 9/16-inch
6. Socket - 1/2-inch drive - 1/2-inch
7. Socket - 1/2-inch drive - 7/16-inch
8. Socket Holder
9. Extension Bar - 5 inches
10. Saw Attachment
11. Spare Saw Blades
12. Saw Blade Holder
13. Allen Wrench - 3/32-inch
14. Allen Wrench - 1/8-inch
15. Drill Attachment (Trepanner)
16. Saw - 5/8-inch diameter (2)
17. Saw - 1/2-inch diameter
18. Saw - 3/8-inch diameter
19. Saw - 5/16-inch diameter
20. Saw - 1/4-inch diameter
21. Needles
22. Hammer 1-1/4 pound dead blow
23. Screwdriver Ratchet Handle - 1/4-inch drive
24. #2 Phillips Head Bit
25. Short Bit - 1/4 x .032
26. Extension Bar - 6 inches
27. Work Lights - 6.9 Watts (2)
28. Adhesive Restraint Buttons
29. Applicator Holder
30. Applicator Control Unit
31. Applicator Temperature Sensor Electronics Package
32. Storage Rack
33. Restraint Button Attachment Cables
34. Small Parts Manipulator
35. Small Parts Holder
36. Battery - 12 Volt - Silver Zinc Cells (8) - 163 Watt Hours
37. Battery Case - Pressurized to + 6 psi differential
38. Astronaut Tether

integrate all requirements when making the battery selection and to place the battery as closely as possible to the center of the tool kit for maximum thermal protection.

Battery selection will be discussed first and then the development and evaluation of the integrated tool kit.

## Battery Selection

Criteria for battery selection were established by analyzing the electrical loads expected in one maintenance period. An exact power profile cannot be plotted but average loads can be estimated with a reasonable degree of confidence. The estimated loads listed in Table 15 are conservative.

TABLE 15

| | Task Time Per Operation | Power Consumed Per Operation | | | Total No. of Operations | Total Power watt hours | |
|---|---|---|---|---|---|---|---|
| | | A | V | W | | | |
| Impacting | 6 sec | 37 | 10 | 370 | 60 | 37 | Peak Load |
| Sawing | 30 sec/inch | 6 | 11.5 | 69 | 50 inches | 28.8 | Peak Load |
| Drilling | 12 sec | 10 | 11.0 | 110 | 18 | 6.6 | Peak Load |
| Work Light | 4 hrs | .55 | 12 | 6.9 | 2 lamps | 56.2 | |
| Adhesive | 30 sec | 8 | 10 | 80 | 6 | 4.0 | |
| Combined Tasks | 12 | | | 110 | 6 holes | 2.2 | |
| | 6 | | | 370 | 12 bolts on and off | 14.80 | |
| | 30 | | | 69 | 10" Sawing | 5.75 | |

Total combined load 82.95 watt hours

Maximum Peak Load impacting and lights and adhesive 97.2

Because the battery performance at extreme temperatures was not known, a 100% safety factor was applied. A Yardney Silvercel Model HR-15 Silver-Zinc Cell was selected. Basic data are listed as follows:

Estimated Capacity - 40 amperes (15 amp hours at 10.9 volts
- 163 watt hours

8 Cells - O.C. Voltage (No Load - 15.0 volts (1.87 volts/cell)
- 10 amperes - 12.0 volts (1.5 volts/cell)
- 40 amperes - 10.9 volts (1.37 volts/cell)

Weight - 5.0#

50 Cycles

Six Months Shelf Life

Size - 2-5/16" x 6-13/32" x 4-15/16" (height)

Watt Hours/Lb = 32.6

Voltage Drop with Temperature - (Estimated) 3 millivolts 1°F to 32°F

Drop is faster below 32°F

Spec. Heat = .30 BTU (#) °F for Yardney cells

The battery is packaged in a sealed case with a pressure relief valve which maintains the internal case pressure 6 psi above the ambient (Figure 64). The case is thermally isolated from the tool kit shell by the solid urethane foam. Figure 65 shows the results of a cold soak test made to measure the variations in battery temperature as the tool kit is exposed to varying ambient temperatures. The complete tool kit was instrumented with thermocouples and placed in a closed box of Expanded Styrene Foam. Liquid Nitrogen was added and temperatures of the tool kit shell, the box ambient, the battery case and battery were monitored. The peaks and valleys of the lower curve represent the changes in tool kit shell temperature as $LN_2$ was added. The middle curve is the change in battery case temperature and the upper curve is battery cell temperature. A drop of about 30°F/hour occurs when no load is drawn from the battery and no heat input is contributed by the power tool attachments cooling in the tool box after use.

## Tool Kit Envelope

The tool kit basic envelope is determined by the requirements for placing the tools within easy reach of the worker without obstructing his access to the work. The basic shape utilizes the triangular space in front of the worker in his normal working position. Prototype tool kit shell was made of drawn .063 aluminum. The limitations of the Martin Marform deep drawing process limit the maximum draw depth to 1.5 times the minimum cross section radius. Layouts were made of the tools and accessories to determine the minimum tool kit depth and therefore, the minimum draw required. The minimum corner radius could then be plotted. The longest piece in the tool kit is the drill attachment of 9 inches. Allowing a small clearance brings the minimum box thickness to 9.5 inches. The tool kit requires a lid to retain components during transport and storage. The lid parting line was established as 2 inches below the top surface

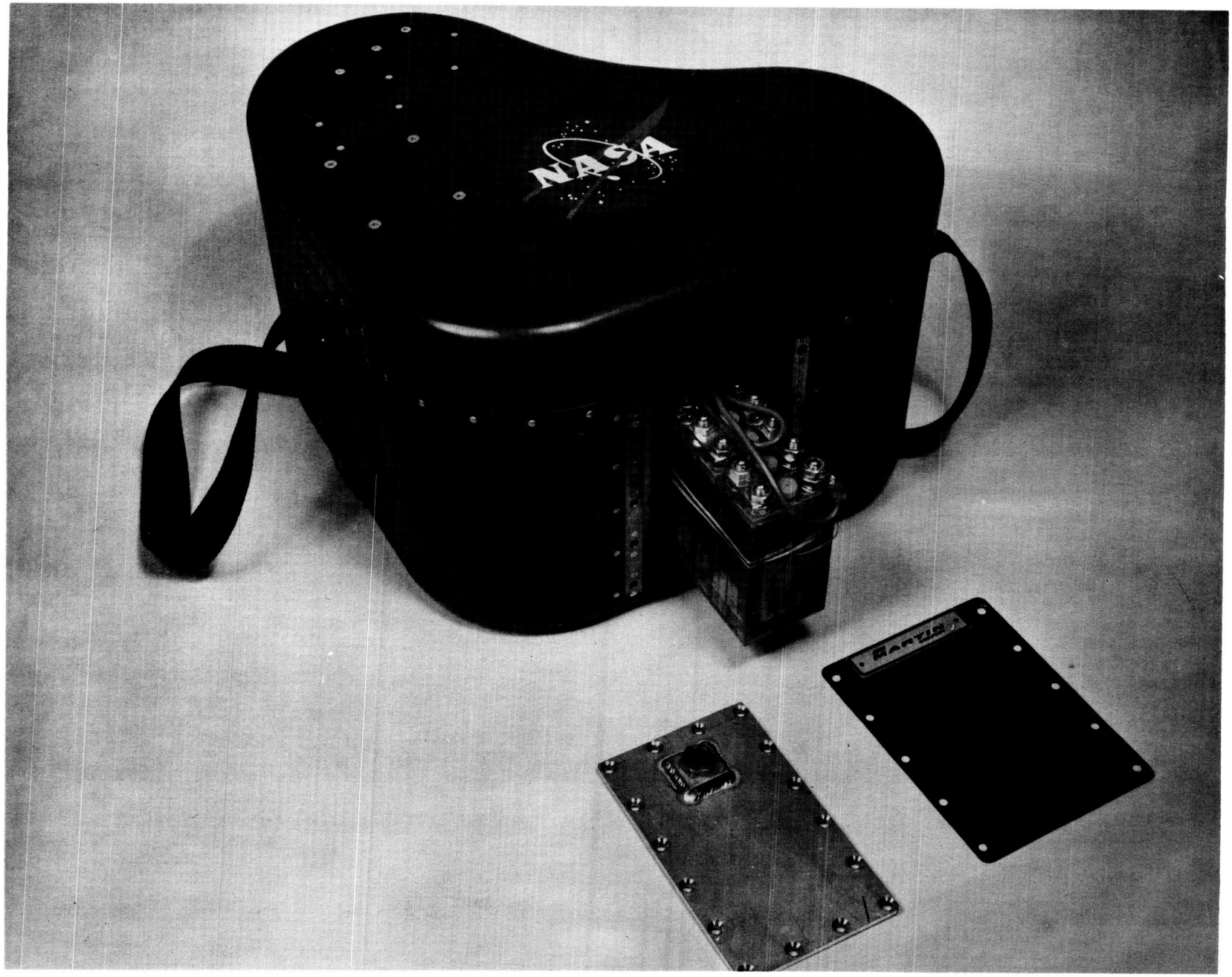



Figure 64. Battery Packaging

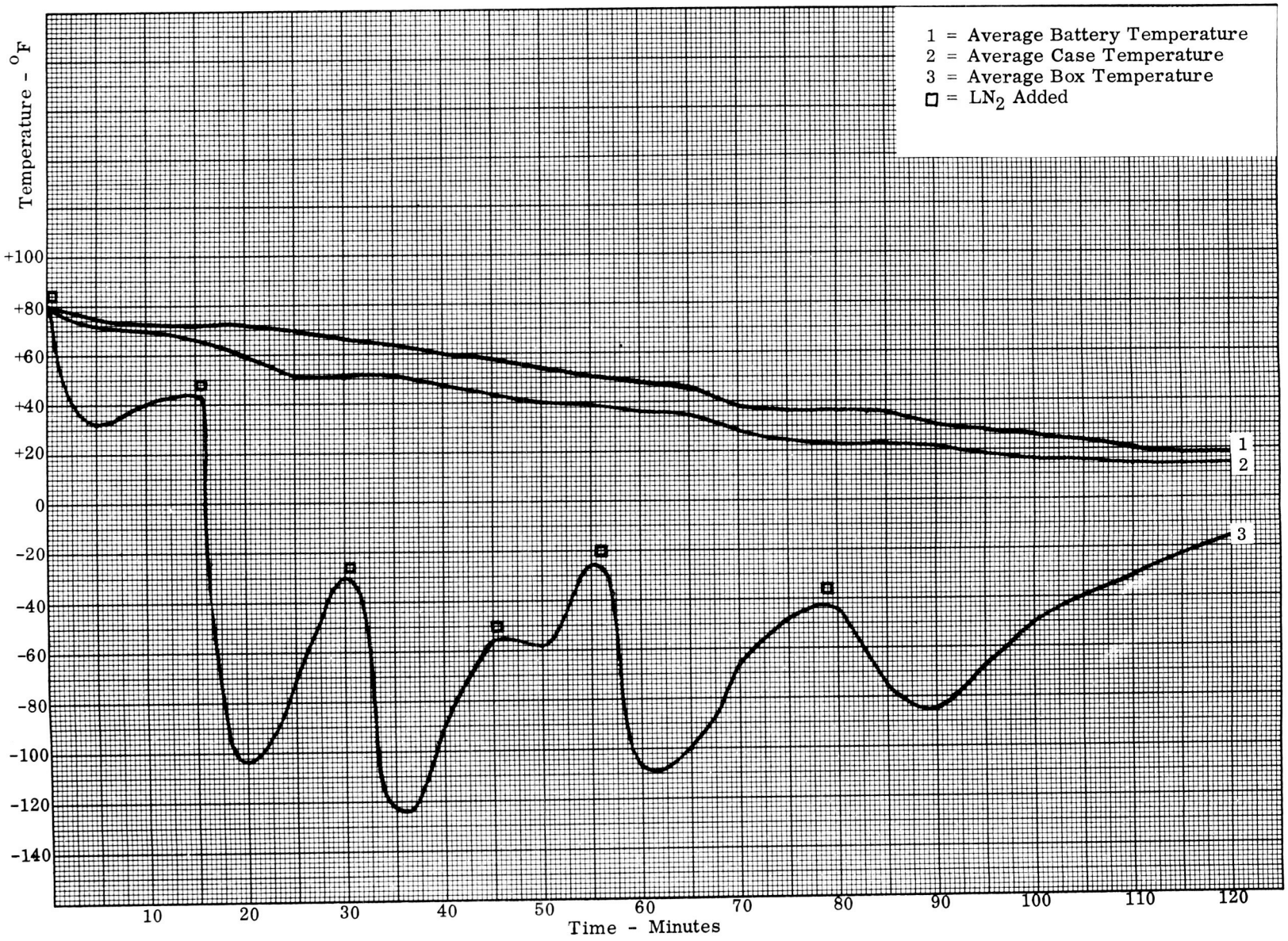
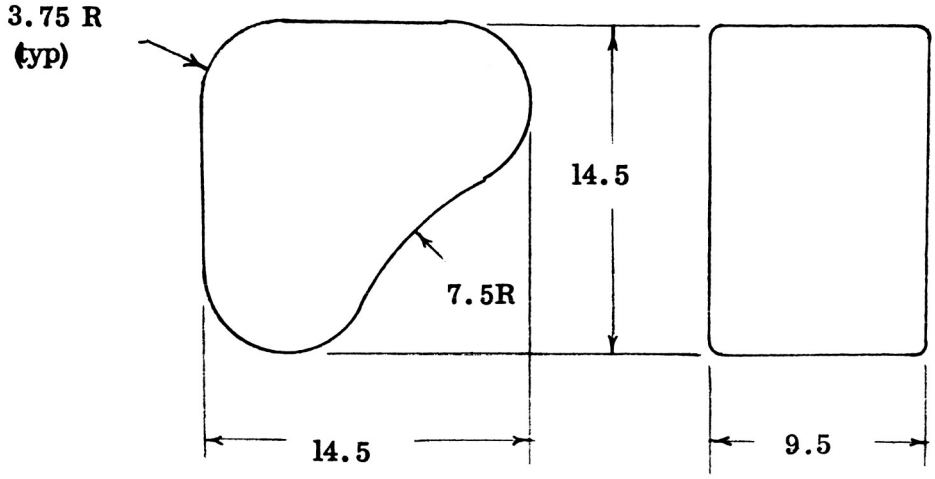



Figure 65. Tool Kit Cold Test

to allow for grasping the items in their storage compartments. The shortest draw is, therefore, 7.5 inches and the minimum cross section radius is 3.75 inches. The basic shape is shown below.




The lid was divided into two segments to allow separate access to the adhesive applicator buttons and hand clearance for grasping the tool handle (Figure 66). The buttons are stored in a rack on the edge of the upper lid. The lids were hinged on the right side to allow approximately 90-degree swings. Bending loads from the upper hinge are supported by a stiffener which also provides support for the motor unit.

Lid opening and closure is accomplished by a "Velcro" strap on each lid which provides a simple, reliable, easy to grasp, latch (Figure 67). A nylon web carrying strap is provided.

Attachment points on the tool kit are provided for attaching the tool kit to the work site and the astronaut to the tool kit. During the human factors evaluation of the prototype tool kit the optimum restraint point was in line with the kit. Since it would be necessary to restrain the kit anyhow in order to effectively insert and remove components, attachment points were added to the tool kit to carry the restraint loads through the box to the spacecraft. A 1/4-turn cam lock fastener was located on the front surface for astronaut tethering and two adhesive attachment points were located on the rear surface at the maximum practical separation of 9-inches (Figure 68). The wire cables retract into the tool kit after attachment to pull it up flush with the spacecraft skin. With the lid open, tool kit stability around all axes and astronaut tethering is achieved with a minimum of two restraint points.

Component Stowage.—The packaging of the contents inside the basic shell was developed on the basis of locating items of highest usage closest to the preferred reach positions (Figure 69). The motor unit and battery dictated the basic layout with other items being fitted around them. The motor unit required access with the right hand. Both-handed access was preferred, but this is impossible to achieve within a small envelope and a compromise was necessary. For thermal protection the battery had to be located with the maximum insulation thickness between it and the shell. In-so-far as possible the major power attachments were located so that they could



Figure 66. Tool Kit In Use




Figure 67. Tool Kit Envelope

Figure 68. Attachment Points

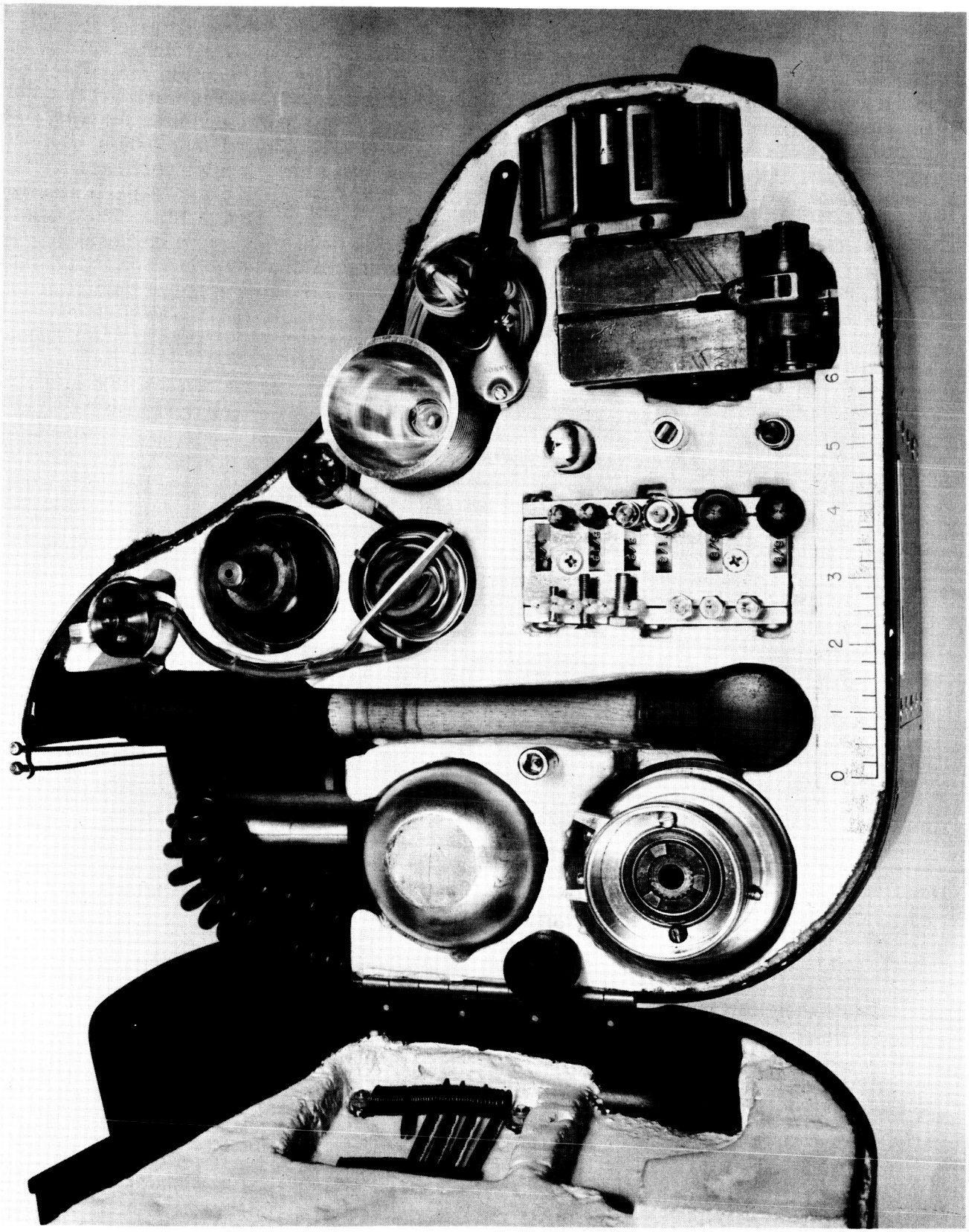


Figure 69: Tool Kit Layout

be grasped in the left hand with the motor unit in the right. Location of the battery and the relatively large diameter of the final drill attachment prevented complete adherence to this. To minimize area on the top surface a turret for storage of the 5 impactor sockets was made to provide positive captive with optimum volume efficiency. The same device was intended for hole saw storage, but use of the brush rack which is located just above the battery case proved more efficient. Below the work lights is the electronics package for the adhesive applicator. This item as well as the battery requires access only for maintenance on the tool kit. The screwdriver handle was stored in the lower lid where it is accessible for use with the adhesive applicator. Screwdriver bits and extension bars were located in available spaces. A preliminary tool kit was fabricated initially of expanded polystyrene foam (Figure 70). The basic dimensions and layout were similar, but a comparison shows several significant changes. The changes resulted from packaging compromises and a more thorough analysis of the accessibility of components by performance of tasks in the simulator with the tool kit. The small parts holder had been expected to utilize the lid area but task performance showed that parts would be removed by either hand and the storage area was relocated to a more central location. It was also anticipated that the hammer, being a low usage item would be stowed below the screwdriver with a pull-out access. It was possible to eliminate the separate access which would have been difficult to reach.

The location of electrical connectors was dictated by the basic lead runs. Three electrical connectors plus a switch and a socket for the electronics package and the battery case are interconnected by the tool kit harness (Figure 71). The battery leads come through the rear of the battery box to a distribution terminal board. The power tool motor connector is located just below and to the left of the handle. Work light connection is through the connector on the right front exterior so that the light wires can be kept out of the workers way. The switch is located just to the right of the stored motor (Figure 66). The work light connector is also used for battery recharging. The adhesive applicator connector is a compromise for efficient packaging. Since the lead lengths are long, a storage cleat was provided. Prior to exiting the spacecraft, the astronaut would dress the leads into the hammer storage area so they can be easily pulled free.

The tool kit stowage compartments are cut from a rigid urethane foam poured in place. The battery case and harness were installed in the tool kit along with the tubes for the adhesive button restraint cables. A dam was installed to form the motor handle area and the lower part of the box was foamed. After the foam had set, compartments for each component were milled in the foam and the top surface was leveled. The lid was foamed and trimmed separately. After the final fitting of parts the foam was given a thin coating of "Renite" epoxy to provide a tough, durable surface.

Special lanyards were not provided for each component because of the total number and the tangle that would result during use. No problem is anticipated with drift forces. All major parts are friction fits and small parts are positively restrained. It is assumed that reasonable care in handling will minimize any requirement for restraint of individual parts.

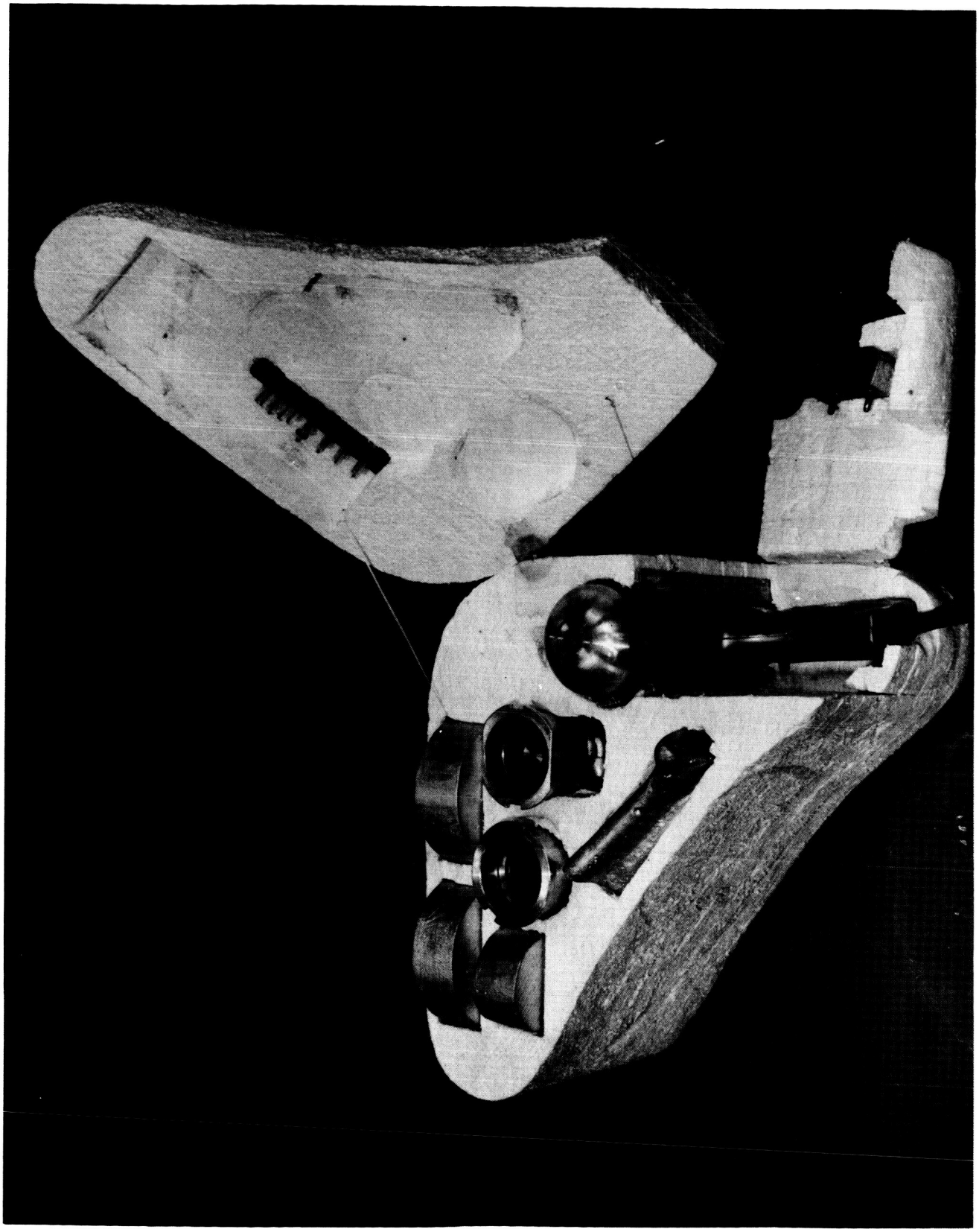


Figure 70. Preliminary Tool Kit

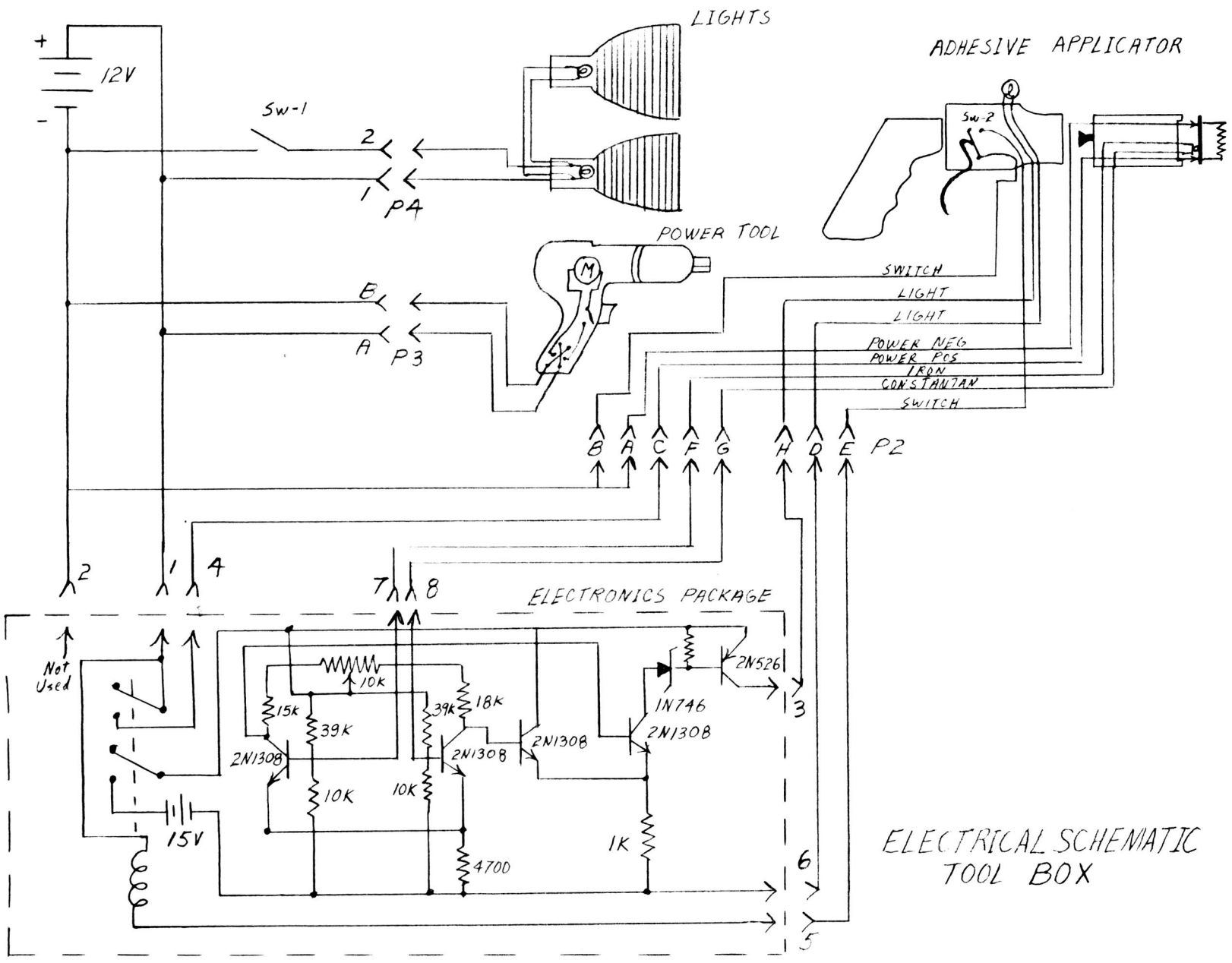



Figure 71. Tool Box Electrical Schematic

Basic Data Summary.—The tool kit design with a completely integrated package has the characteristics listed in Table 16.

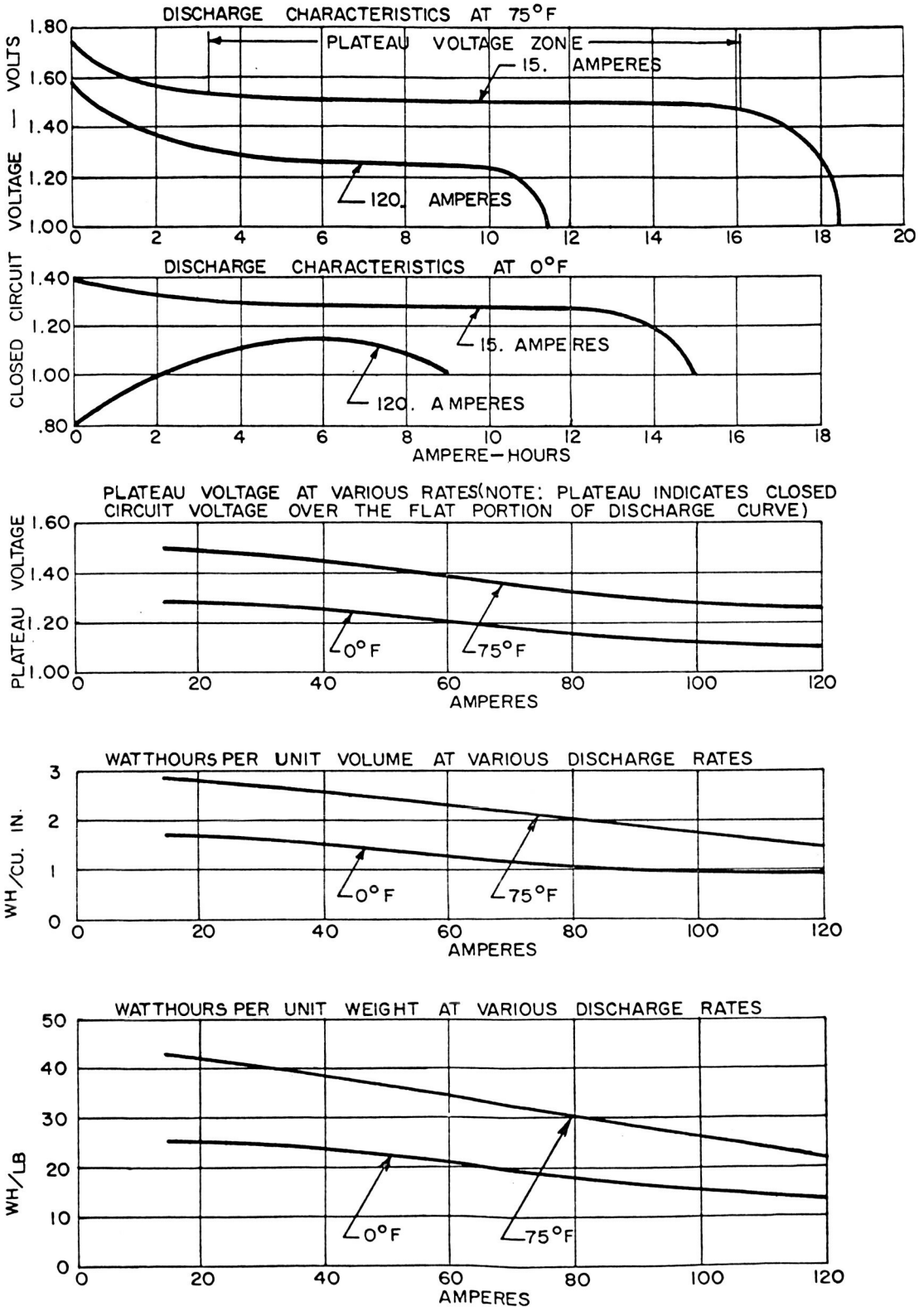
TABLE 16

BASIC DATA

| Item | Wt. | L (in.) | W (in.) | H (in.) | |
|---|---|---|---|---|---|
| Complete Tool Kit | 38 lb | 14.5 | 14.5 | 9.5 | 0.7 cubic ft. |
| Motor | 90 oz | 7.0(1) | 3.0 | 8.0(2) | 12 V dc, PM Field |
| Impactor | 42 oz | 6.4(1) | 2.25 | - | 85 ft.-lb. out<br>6 in.-oz. reaction |
| Saw | 48 oz | 8.0(1) | 2.5 | 4.0(3) | 5/8-in. stroke .030 in Kerf |
| Drill | 56 oz | 8.25(1) | 3.25 | - | 1/4-5/8 holes |
| Hammer | 28 oz | 12.5 | 1.5 | 4.3 | 1-1/4 lb. dead blow |
| Screwdriver Handle | 12 oz | 4.6 | 1.4 | 2.7 | 1/4 in. ratchet drive |
| Work Light (2) | 14 oz | 4.0 | 2.2 | - | 30° Ellipsoidal Reflector 6.9 W |
| Restraint Buttons | 1 oz | - | 1.13 | 0.5 | 8-10 A, 10-12 V, 30 sec. for 50 lbs. |
| Applicator (2 parts) | 12 oz | 3.7 | 1.3 | 2.8(4) | Indicate 450° Bond Temp. |
| Battery | 5 lb | 6.4 | 2.3 | 4.9 | 12 V dc Yardney HR-15 Silver Zinc Cells Pressurized Case - 6 psig above ambient max (See Fig. 72 for power characteristics.) |

| Accessories | |
|---|---|
| Sockets (5) | 1/2 in. Drive, 7/16 - 3/4 in. |
| Hole Saws | 1/4, 3/8, 1/2, 21/32 |
| Saw Blades | Modified Black & Decker U-1354, 14 teeth |
| Screw Drivers | 1/4" Drive #2 Phillips, 1/4 x.032 |
| Extension Bars | 1/4 x 6 in., 1/2 x 5 in. |
| Small Parts Holder | |

1. 1.25 in. overlap when assembled
2. 6.5 in. motor axis to handle tip
3. 3.0 in. motor axis to forward linkage
4. two 3.0 ft. cables for drifting




YARDNEY ELECTRIC CORP.

FIGURE 72. BATTERY PERFORMANCE CURVES

## CONCLUSIONS

The conclusions to be drawn from this program can be summarized in one statement. Tools for performance of maintenance, repair and assembly tasks in extraterrestrial environments will not be the limiting factor in planning such activities.

It is feasible to develop tools which are operable by a worker in a pressurized spacesuit under 0-1 G conditions. The optimization of tools for space use will require detailed analysis of maintenance task requirements and logistic plans to assure a proper matching of tool performance to the work and the other supporting functions such as spare parts and technical data.

Pressure suit performance is perhaps the most significant human factors element affecting tool design and use. Advances in pressure suit development toward shirtsleeve performance will substantially reduce the magnitude of, what are now limiting factors, reach and dexterity. Pressure suit mobility and dexterity not only affect tool use directly, but also indirectly through their influence upon component accessibility and spacecraft equipment packaging. Integration of operator performance with spacecraft design presents problems which will require efficient configuration management during spacecraft design and development. Integration of functions such as restraints, lighting and grasping devices with suit designs is necessary.

Environmental constraints, while not minimal, appear to offer no unsoluble problems in tool development. Zero gravity operation with reactionless tools has been shown feasible although much more development and refinement is desirable. Operation of electric motors in an $O_2$ environment is feasible. Vacuum and temperature problems affecting tool materials will benefit from technological improvements in the basic equipments the tools will be used to repair.

Finally, development of tools will require evaluation and development of techniques under actual extraterrestrial conditions. New approaches, not readily apparent to workers trained to operate under earthside conditions, will be developed by such testing.

## RECOMMENDATIONS

The following items are felt necessary to carry the tool development to the point where maintenance repair and assembly tasks can be realistically planned as integral functions on future space missions.

1. Refine the basic tool concepts to reduce weight and volume while increasing efficiency and output range.

2. Develop experiments for performance in space which will evaluate tool use under actual conditions to uncover subtle interactions.

3. Develop more detailed analyses of tool and accessory functional capabilities to enable matching the tool kit capabilities to specific missions where a significant contribution to the probability of mission success can be made.

4. Expand the investigation of basic functional concepts into other areas such as measurement and alignment, cargo, personnel and equipment transfer and handling, and other functional areas required for extended orbital and lunar stays.

5. Develop techniques for integrating the astronauts tool using capabilities into mission planning and spacecraft design to assure the optimum logistic tradeoffs are achieved.

6. Standardize the fasteners and structural materials used for spacecraft to reduce the range of types and sizes and thereby reduce the complexity of a maintenance tool kit and on-board repair parts stock.

7. Wherever possible utilize captive fasteners on replaceable components.

8. Develop spacesuits which approach as closely as possible the inherent human mobility and dexterity.

9. Develop experiments to define more precisely the human performance capabilities under environmental extremes which are required for space maintenance and assembly.