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FINAL REPORT

DEVELOPMENT OF ADVANCED SOIL SAMPLER TECHNOLOGY

TASK E

PHILCO Ford

Publication No. UG-4410

SPACE AND RE-ENTRY SYSTEMS

FINAL REPORT

DEVELOPMENT OF ADVANCED SOIL SAMPLER TECHNOLOGY

TASK E

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ABSTRACT

This report covers the effort performed under Task E of JPL Contract 951935 and together with Philco-Ford report UG-4289, 31 January 1968, completes the requirements for this contract. The tasks performed in the earlier work covered the laboratory and field testing of eleven sampler breadboard mechanisms. The results of that effort were utilized in the completion of Task E wherever it was applicable.

Design criteria were defined and eight prototype designs were completed for six soil sampling mechanisms and two soil processing mechanisms. These designs were completed in sufficient detail to allow reasonable estimates of the weight, size, and complexity to be made. These designs are based on breadboard models built and tested by the Jet Propulsion Laboratory, Philco-Ford and Hughes Tool Co. The completed prototype designs represent a well defined starting point for more detailed design analysis and development of soil sampling mechanisms to support future surface probes.

It is concluded that these soil sampling mechanisms are of sufficiently light weight and compact size to warrant consideration as part of the payload complement of an early unmanned planetary or lunar payload.

This work was performed for the Jet Propulsion Laboratory, California Institute of Technology, sponsored by the National Aeronautics and Space Administration under Contract NAS7-100.

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SECTION 1

INTRODUCTION

This report summarizes the results of Task E, for the period from 1 January 1968 to 8 May 1968. The intent of Task E was to generate engineering prototype designs of lunar and planetary soil samplers and mechanical sample processors. These designs are based on breadboard mechanisms as identified by the Jet Propulsion Laboratories, Pasadena, California. This task was scheduled to follow the completion of the laboratory testing of two soil samplers, developed at Philco-Ford under a preceeding contract, and the field testing of these samplers along with nine other mechanisms furnished by JPL. It was scheduled in this manner so that the full benefit of the test experience with these soil samplers could be utilized in the completion of Task E.

The sampler and processing mechanisms designated by JPL in the work statement are listed in Table I.

TABLE I

SPECIFIED MECHANISM DESIGNS

Designation	Title
(E-1)	Uncased Rotary Impact Drill Sampler
(E-2)	Cased Rotary Impact Drill Sampler
(E-3)*	Conical Abrading Sieve Cone Sampler
(E-4)*	Helical Conveyor Simple Particulate Sampler
(E-5)*	Backhoe Sampler
(E-6)*	Soil Auger Sampler
(E-7)	Miniature Jaw Crusher
(E-8)	Sample Size Sorter

*Mechanisms tested in the preceeding tasks.

Based on subsequent conversations with JPL, the following interpretations were considered to apply to this design effort.

The purpose of this task is to use the results and demonstration of feasibility from various breadboard sampler models to develop a more advanced engineering design. These designs should emphasize simplicity and reliability. All the mechanisms listed above are identified as geological samplers and any consideration of their use as a biological sampler is strictly secondary. The design should consider the use of the sampler mechanism as a means of enhancing the successful completion of "in-situ" experiments such as was demonstrated for the α-scattering experiment on Surveyor. In this case the soil sampler mechanism was used to aid in dislodging the α-scattering experiment when it failed to deploy properly.

Additional clarification was provided by JPL on the specific design approaches that should be pursued for the simple particulate sampler (E-4) and the particle size sorter (E-8). The simple particulate sampler (E-4) was not to be a linear vertically deployed device, as tested in the field, but rather a curved helical conveyor similar in configuration to a design approach developed by JPL.*

The particle size sorter (E-8) design also was to pick up the basic approach of such a mechanism developed at JPL for the petrographic microscope.** The jaw crusher (E-7) design was also redirected to apply the principles of a rotary rock crusher breadboard which has been built by JPL and undergone some limited testing. Thus, although some preliminary effort was made on a jaw crusher design, the final design of this mechanism is a rotary rock crusher.

The design procedure used in performing this task consists of the following.

(1) Define the design criteria for each mechanism. Utilization of JPL's experience and other sources is made in defining these criteria. JPL assisted in the complete definition of the criteria for each mechanism.

(2) From these criteria and past experience with various applicable breadboard components, preliminary sketches, schematics, layouts, and block diagrams were generated for design review.

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*Reference to Figure 1, SPS 37-46, Volume IV, page 137.

**Reference JPL TM33-353.

(3) From the selected design approaches, a complete layout was generated and supported with the necessary analysis to provide a design in sufficient detail to allow a mechanical evaluation of the mechanism. This evaluation includes estimates of size, volume, and weight and a qualitative estimate of simplicity.

The detailed description of each mechanism design is given in Section 3.0. The basic features of each design are summarized in an operational sequence of events and a detailed weight statement at the end of each mechanism design description. The weight statement identifies the number of different types of parts by item number, the material the part is made of, the total number of parts required, the weight per part, and the total weight per assembly. The weights of major subassemblies are given, where appropriate, to aid in identifying what part of the mechanism is most complex and which items are contributing most to the total weight.

The weights were calculated to four decimal places because of the small size of the parts involved. The final weights should be considered as estimates with a tolerance of 10 to 20 percent. In some cases a more detailed design analysis would probably result in a reduction of the estimated weights.

SECTION 2

DESIGN CRITERIA

At the initiation of this task, it was apparent that in order to minimize confusion in the design goals desired for each sampler or processor mechanism, that a definitive set of design criteria should be drafted. A preliminary set of the design criteria were presented to JPL early in the effort for their review. In general these were acceptable but have since been revised and expanded to conform more nearly with the desired design objectives. The criteria for mechanisms (E-4) helical conveyor simple particulate sampler, (E-7) miniature rotary rock crusher, and (E-8) sample size sorter in particular have been reworked to reflect the specific design approaches requested. These criteria are presented in the following paragraphs.

2.1 UNCASED ROTARY IMPACT DRILL, E-1

This is an uncased rotary impact sampling drill which is based on design components of both the JPL and Hughes Tool Company models. The following initial design criteria are assumed:

(1) This drill must obtain a sample from solid rock, rubble, sand, and cohesive powders. No segregation of sample as a function of depth is required.

(2) The particle size output of the sampler shall be 250μ or less in diameter.

(3) A minimum sample quantity collected per run shall be 2 to 3 grams of soil.

(4) The sampler shall have the capability of repeating a sampling run at least one or more times in several locations, i.e., mounted on a deployment structure.

(5) Continuous sample transport will be used. The operational mode shall be such as to minimize carryover of one sample to the next by clearing the drill and conveyor between sampling runs.

(6) Power required should be 50 watts or less.

(7) The hole diameter shall be as small as possible consistent with acquiring a 2 to 3 gram sample at a reasonable depth in rock; i.e., less than 5 centimeters deep.

(8) Maximum depth of penetration in material other than rock shall be 25 centimeters.

(9) This sampler will start the drilling operation without impact after being deployed to the surface. The sampler shall be capable of sensing feed rate to indicate rock. Impact mode of drilling starts when rock is sensed.

(10) The minimum drill rate in basalt shall be .05 inches per minute.

(11) The axial thrust shall be limited to 20 pounds.

2.2 CASED ROTARY IMPACT DRILL, E-2

This is a cased rotary impact sampling drill in which the casing can be either driven or stationary. The casing is rotated and driven vertically independently with respect to the drill. This design will consider components of breadboard models built by JPL. The following initial design criteria are assumed.

(1) This drill must obtain a sample from solid rock, rubble, sand, and cohesive powder at or near the planetary surface. It shall be capable of obtaining an essentially uncontaminated rock sample from solid rock under an overburden not more than 20 centimeters thick.

(2) The particle size output of the sampler shall be 250μ or less in diameter.

(3) Minimum sample quantity collected shall be 2 to 3 grams of soil. If rock is encountered, two samples will be collected. A sample of overburden and a rock sample.

(4) The sampler shall have the capability of repeating a sampling run at least one or more times in several locations.

(5) Continuous sample transport shall be used.

(6) Power required should be 50 watts or less.

(7) The hole diameter shall be as small as possible consistent with acquiring a 2 to 3 gram sample at a reasonable penetration in solid rock; i.e., less than 5 centimeters deep.

(8) Maximum depth of penetration, if no rock is encountered, shall be 25 centimeters.

(9) This sampler will start drilling without impact. It shall be capable of sensing feed rate to indicate rock. Impact mode of drilling starts when rock is sensed.

2.3 DEEP ABRADING CONE SIEVE, E-3

This design is based on the JPL deep abrading sieve cone tested in the field. The basic concepts of the small half angle cone as a selective acquisition device will be retained in this design. The following initial design criteria are assumed.

(1) The sampler is not required to obtain a sample from solid rock. It is required to obtain a sample from cohesive soils, sand, rubble, and cohesive powder. It should have limited sampling ability on softer rock material such as sandstone and vesicular pumice. Strict segregation of the sample as a function of depth is not required.

(2) The particle size output of the sampler shall be 250μ or less in diameter. The particle size distribution obtained in the field tests for this sampler met this limit with a mean grain diameter of 100μ.

(3) A minimum sample quantity collected shall be 10 grams of soil.

(4) The sampler shall have the capability of repeating a sampling run one or more times in several locations; i.e., mounted on a short boom.

(5) Continuous sample transport using a helical conveyor or batch aerosol pneumatic transport may be used.

(6) Axial thrust during sampling shall be limited to 20 pounds.

(7) A gimbal mount shall be employed to minimize the possibility of defeat due to encountering a surface obstruction (rock).

2.4 HELICAL CONVEYOR SIMPLE PARTICULATE SAMPLER, E-4

This sampler design is based on the JPL concept of a simple particulate helical conveyor sampler utilizing a hard or elastomer lined casing and helical screw. The configuration as shown in Figure 1, SPS 37-46, Volume IV, page 187, will serve as the basis for the design. The basic concepts will be retained; however, design modifications suggested by the results of the field testing will be evaluated. The following initial design criteria are assumed.

(1) This sampler will be capable of sampling only weak cohesive materials, rubble, sand, and cohesive powder. A drill cutter shall be incorporated at the tip of the sampler to enhance its capability in soft rock materials. Segregation of sample as a function of travel is not required.

(2) The particle size output of the sampler shall be 250μ or less in diameter.

(3) A minimum sample quantity collected shall be 2 to 3 grams of soil in traversing a minimum of 5 inches through the soil.

(4) The sampler shall have the capability of repeating a sampling run one or more times by reversing the motor to reposition for the next cycle.

(5) Continuous sample transport using a helical conveyor shall be used.

(6) Deployment rate shall be set to be compatible with the desired feed rate of the sampler.

2.5 BACKHOE SAMPLER, E-5

This is a backhoe type sampler utilizing some form of boom deployment combined with a batch type of sample transport mode, probably a gravity dump. This sampler is considered to be primarily a surface sampler. The following initial design criteria are assumed.

(1) This sampler will be capable of sampling only weak cohesive material, rubble, sand, and cohesive fine powder.

(2) The particle size output of this sampler shall be limited to 5 millimeters or less in diameter.

(3) A minimum sample quantity collected shall be 10 grams of soil per run.

(4) Batch type sample transport using a gravity dump will be considered for this sampler.

(5) The sampler shall have a deployable range up to 5 feet and be able to sample in a sector with a minimum angle of arc of 90 degrees.

(6) Both mechanically extendible telescoping booms and furlable tape closed-section booms (Ryan) shall be considered.

(7) The backhoe scoop design shall be such that it may be closed after clearing the surface without losing excessive sample. Also, the design shall be such that the scoop can be completely closed; i.e., no rocks will be in a position to cause the scoop to be partly wedged open.

2.6 SOIL AUGER SAMPLER, E-6

This sampler is based on the breadboard soil auger developed by JPL and tested in the field. The basic concept of the auger design and sample transfer mode will be retained. This is considered to be a shallow sub-surface sampler. The following initial design criteria are assumed.

(1) The sampler will be capable of sampling only weak cohesive materials, rubble, sand, and cohesive fine powder.

(2) The particle size output of this sampler shall be 5 millimeter or less in diameter.

(3) A minimum sample quantity collected shall be 2 to 3 grams of soil per run.

(4) Batch type sample transport using a spin dump mode of soil transfer will be considered.

(5) The sampler shall consider both vertical deployment and boom deployment to the surface.

(6) If vertical deployment is used, the sampler will be capable of repeating the sampling run one or more times in several locations.

(7) If boom deployment is used, it shall be a simple non-extendable short boom and be capable of being deployed in a sector with a minimum included angle of arc of 90 degrees.

(8) Sample transport from the sampling head will consider a gravity dump mode for the boom mounted sampler.

2.7 MINIATURE ROTARY CRUSHER, E-7

This is a miniature crusher designed to break down larger pebbles to a size useable by geological analytical instruments such as the X-ray diffractometer, the alpha-scattering spectrometer, or the petrographic microscope. The existing JPL breadboard design of a rotary crusher will serve as the basis of this design. The following initial design criteria are assumed.

(1) This crusher shall be capable of accepting and reducing pebbles up to 5 millimeters in diameter contained in the sample delivered to this crusher.

(2) The rock crusher output shall contain particles up to a maximum of 300μ.

(3) The maximum power consumption shall not exceed an average value of 15 watts.

(4) The crusher shall be capable of reducing hard rock such as basalt, quartz, etc., without metallurgical contamination.

(5) The crusher shall include the means of preventing particles larger than 5 millimeters from being ingested.

(6) The crusher shall minimize the probability of ingesting particles larger than 300μ which possess a high magnetic permeability.

2.8 SAMPLE PARTICLE SIZE SORTER, E-8

This processor shall selectively sort by particle size a sample delivered to it by any of the samplers or the rotary rock crusher into particle size cuts suitable for use in geological analytical instruments such as the X-ray diffractometer, alpha-scattering spectrometer, or the petrographic microscope. The JPL particle size sorter developed for the petrographic microscope as described in JPL TM 33-353 shall serve as the basis for this design. The following initial design criteria are assumed.

(1) This size sorting processor shall be capable of separating the sample introduced into it into into three cuts with $d > 300\mu$, $50\mu < d < 300\mu$, and $d < 50\mu$.

(2) The sample introduced into this size sorting processor may contain particles up to 5 millimeters in diameter.

(3) This processor will be capable of repeating the size sorting operation on samples one or more times.

(4) The maximum quantity of sample processed in any given cycle shall not exceed 10 grams.

SECTION 3

DESIGN APPROACH

This section presents the design approaches considered for the sampling and processing mechanisms. The basic considerations for each design were governed by the individual design criteria and functional requirements for each mechanism. The design goals were intended to produce the simplest, most compact, and lightweight mechanisms that satisfy these design criteria and functional requirements. The designs also incorporate the results obtained from breadboard models and any testing that was accomplished with these models. In support of the design effort, JPL made available to Philco-Ford various sketches, drawings, and reports as listed in Table II. This information was supplemented by means of verbal comments and suggestions by JPL during the progress of the design effort.

A complete list of the drawings made during the completion of this task is given in Table III. The first part of the list represents the final engineering prototype designs developed. The remainder of the list identifies the preliminary layouts and sketches made in the course of the evolution of the final design approaches.

The following paragraphs discuss the design of each mechanism in the same sequences as listed in Table I. The alternate design approaches which were considered are also presented.

TABLE II

LOG OF SUPPORTING DATA RECEIVED

Identifying No.	Originating Source	Identifying Title	Date Received
10018381	JPL	Bulk Particulate Sampler, Auger Type	8/0/67
X3024700	Hughes Aircraft Co.	Soil Mechanics/Surface Sampler, Surveyor	3/6/68
6-9376306	JPL	Lunar Drill Assy - Breadboard	3/6/68
Layout	JPL	Cased Drill - Drill Feed Mechanism	3/6/68
SK9079	Ryan Aircraft Co.	Demonstration Model - 12 Ft. Retractable Boom	3/6/68
J10002468-1	JPL	Processor Assy - Petrographic Experiment, Sheet 1 of 30	3/6/68
Final Report JPL Contract 951480	Hughes Tool Co. (Phase I & II)	Development Program of a Lunar & Planetary Geosampler	3/6/68
Layout	JPL	Impact Drill - Hammer Drive Assy	3/21/68
Layout	JPL	Impact Drill - Vibratory Feed Test Extractor	3/21/68
325-491B	JPL	Petroscopic Microscope (Photo)	3/21/68
325-467B	JPL	Petroscopic Microscope Sieve Assy (Photo)	3/21/68
2 Pages of Paper	JPL	Petrographic Microscope Description	3/21/68
29770-10	Ryan Aircraft Co.	12 Pages of Furlable Boom Proposal	3/21/68
Layout 1001	Hughes Tool Co.	B-1 Drill and Transport Assy	3/21/68
Case No. 1384 IR No. 30-1384	JPL	Particulate Auger (New Technology Report)	4/1/68
Job No. 383-20701-2-3220	JPL	Auger Blank Bulk Sampler	4/1/68
Job No. 383-20701-2-3220	JPL	Tip Detail, Saw Tooth Auger	4/4/68

TABLE III

DRAWING LIST

DRAWING NO.	DRAWING SIZE	TITLE	RELATED MECHANISM
PD 37051	D	Rotary/Impact Drill Assy	E-1
PD 37052	J	Rotary/Impact Drill and Deployment Assy	E-1
PD 37070	D	Cased Rotary/Impact Drill Assy	E-2
PD 37053	E	Conical Abrading Sieve Cone	E-3
PD 37054	E	Drive Assy Conical Abrading Sieve Cone	E-3
PD 37055	E	Conical Abrading Sieve Cone Assy	E-3
PD 37074	D	Alternate Design Helical Conveyor Simple Particulate Sampler	E-4
PD 37057	J	Helical Conveyor Particulate Sampler	E-4
PD 37073	C	Drive Assy Details-Conical Abrading Sieve Sampler	E-3
PD 37059	J	Backhoe Sampler	E-5
PD 37060	J	Operational Sequence Backhoe Sampler	E-5
PD 37062	J	Soil Auger Sample	E-6
PD 37063	B	Soil Auger - Lead Screw Feed	E-6
PD 37069	J	Miniature Rotary Rock Crusher	E-7
PD 37061	J	Sample Size Sorter	E-8
PD 37071	D	Extendible Strut Drill Deployment System	E-1
PD 37072	D	Alternate Drive Conical Abrasive Sieve Cone	E-3
PD 37044	D	Simple Particulate Sampler Double Bend Tube	E-4
PD 37045	D	Simple Particulate Sampler Single Bend Tube	E-4
PD 37046	D	Gravity Actuated Helical Conveyor Sampler	E-4
PD 37048	E	Helical Conveyor Double Bend Assy	E-4
PD 37049	E	Helical Conveyor Single Bend Assy	E-4
PD 37041	D	Boom Deployment Soil Auger Concept I	E-6
PD 37043	E	Boom Deployment Soil Auger Concept II	E-6
PD 37047	D	Soil Auger Parallel Linkage Deployment	E-6
PD 37050	D	Simple Boom Deployment Soil Auger	E-6
PD 37042	D 2 Sheets	Sample Size Sorter	E-8

3.1 ROTARY/IMPACT DRILL SAMPLER, E-1

Since one of the most critical parts of this sampler is the impact hammer and rotary drive mechanism, this part of the design was considered first. The basic approach was to determine whether or not a more compact mechanization could be achieved that would deliver the same impact characteristics of the cam actuated hammer tested by Hughes Tool Company for JPL. The basic assumptions were that a helical conveyor would be used to transport soil particles away from the bit, that the helical conveyor would have to pass through the hammer drive assembly to some point above it, and that one drive motor should operate both the impact hammer and the rotary drive of the drill. If possible, it should also provide the drive for the helical conveyor. To satisfy these assumptions, a free wheeling crank system to compress the hammer spring was considered. The crank in this system is connected to the driving source through an over-running clutch so that as top dead center is passed the spring is free to accelerate the hammer carrying the connecting rod and crank with it. After impact the over-running clutch engages the drive shaft which lifts the hammer and compresses the drive spring. The velocity and stroke characteristics for this hammer mechanism were calculated using the parameters listed in Table IV.

TABLE IV

IMPACT HAMMER PARAMETERS

Item	Value
Crank Speed	240 rpm
Hammer Stroke	.5 inch
Hammer Weight	1.75 lbs
Spring Rate	40 lb/inch
Preload Force	30 lbs
Compressed Force	50 lbs
Impact Energy	1.5 ft-lbs

The characteristics for three types of hammers as shown in Figure 1 were compared in order to assess what differences might exist in the operation of the hammer. The type A hammer is one that has been built and tested by JPL but not necessarily with the same parameters listed in Table IV. The type B hammer is the approach using the overrunning clutch, and the type C hammer is a cam actuated device built and tested by both JPL and Hughes Tool Company. The velocity and stroke characteristics are shown in Figure 2 for these hammer types. The most noticeable difference is the number of impact strokes delivered per revolution of the crank drive shaft. The same rotational speed is assumed for each crank drive shaft input.

The type B hammer with the overrunning clutch delivers more strokes per revolution. For this configuration the ratio is 1.7 impacts to one impact

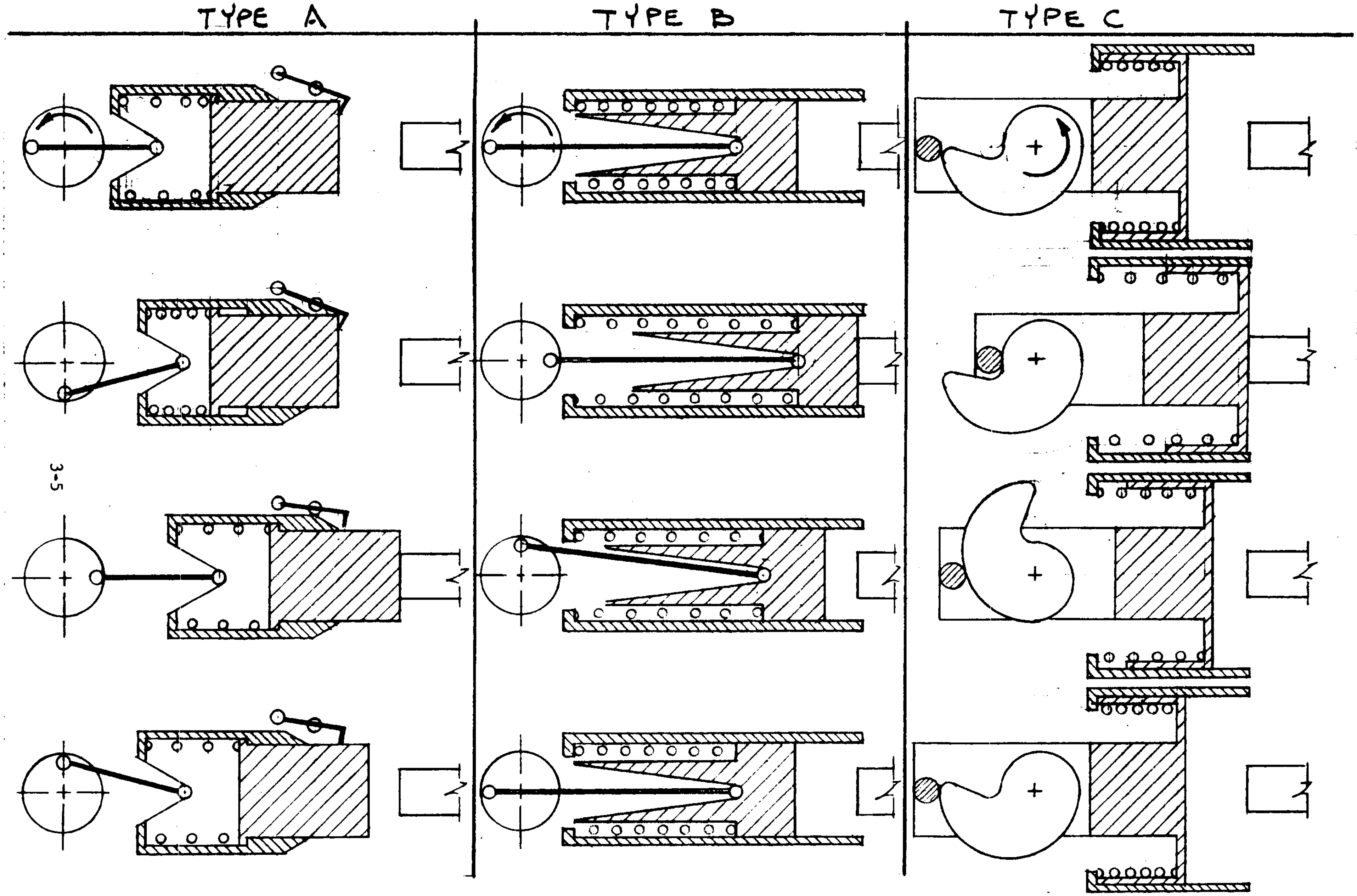


FIGURE 1 SCHEMATIC - IMPACT HAMMER TYPES

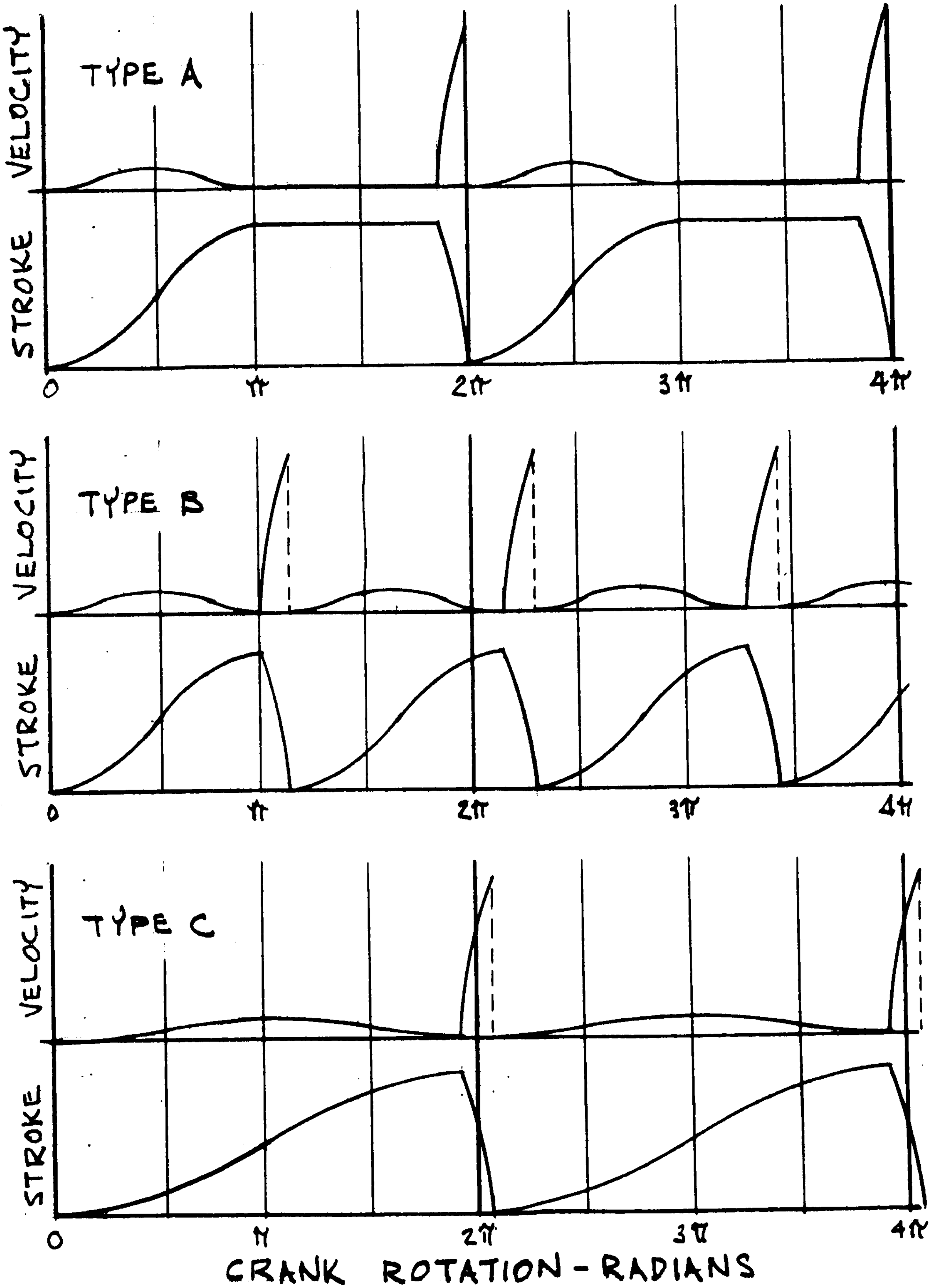


FIGURE 2 IMPACT HAMMER CHARACTERISTICS

for each of the two other types. It should be pointed out that the calculations are made neglecting friction in the system and making no allowances for clearances resulting in lost motion. Thus, the hammer velocity at impact is the same for all three types. In this case it is 8 fps. The peak hammer velocity during the retraction part of the stroke is the same for type A and B and is slightly less than 1 fps. Since type C uses a full rotation of the drive shaft to accomplish the hammer retraction, the peak retraction velocity is half that for types A and B. If the drive shaft velocity for type B is reduced to produce the same impact rate as type A and C, then the peak retraction velocity is also reduced to a value more nearly like that for the type C hammer. Thus, it would appear that the type B hammer is potentially more efficient than type A or C; however, the primary influence on hammer efficiency is the reduction or elimination of sliding friction between the hammer and the guides. This suggests the use of some sort of rolling guide for the hammer which was examined more fully as the design progressed.

Since a lighter hammer might be desired, it is necessary to examine the effect of varying this parameter. The velocity of the hammer while it is being accelerated by the spring is given by the relation

$$v^2 = \frac{kg}{W}(x_2^2 - x^2)$$

where W is the hammer weight, k is the spring constant, x_2 is the maximum compression of the spring, and x is some point in the stroke. Substituting this velocity into the expression for kinetic energy yields

$$KE = \frac{1}{2}\frac{W}{g}v^2 = \frac{W}{2g}\cdot\frac{Kg}{W}(x_2^2 - x^2) = \frac{K}{2}(x_2^2 - x^2).$$

From this it is seen that the mass of the hammer can be reduced without changing the kinetic energy in the hammer at impact; however, the velocity characteristic of the hammer is altered, as is the duration of the impact stroke. Since the spring driving force varies between 30 and 50 pounds, a good estimate of the time variation of the impact stroke can be achieved by assuming an average value for the acceleration of the hammer. This acceleration is given by

$$a = \frac{F}{W}g$$

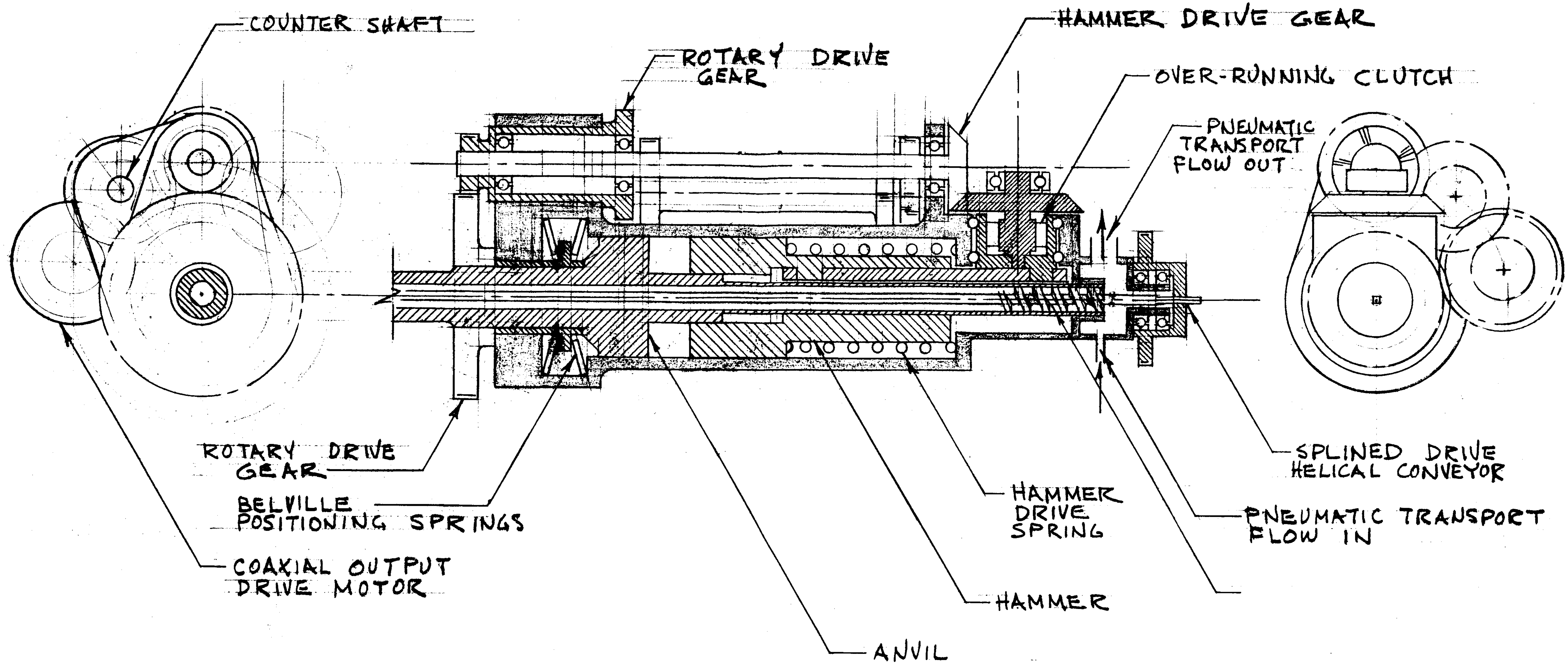
where F is the spring driving force. The time required to travel a distance, s, is given by

$$t^2 = \frac{2s}{a} = \frac{2sW}{Fg} \quad \text{and} \quad t = \left[\frac{2sW}{Fg}\right]^{\frac{1}{2}}.$$

Thus, it is seen that the duration of the impact stroke is reduced in proportion to the square root of the hammer weight. A reduction in hammer weight by a factor of four reduces the duration of the impact stroke by a factor of two. For this reduction in hammer weight, the type B hammer now produces impacts at the rate of 1.84 to one, which is approaching twice the rate of the type C hammer for the same input driving speed, or conversely, almost half the input drive shaft speed for the same impact rate. This weight reduction will yield a hammer weighing slightly less than one-half pound which is more consistent with a light weight drill design than the 1.75 pound hammer tested by Hughes Tool Company.

A potential advantage of the type B hammer is that the over-running clutch will allow the hammer to rebound from the anvil partially compressing the spring. By virtue of the one way or ratchet type action of the clutch, this compression can be retained in the spring. Thus, some energy recovery is possible with this system. Since the rebound velocity of the hammer is determined by the relative mass of the hammer and drill, this feature of energy recovery could be important in a light hammer system in improving the efficiency of the system. A preliminary mechanization of the type B hammer is shown in Figure 3. This approach allows one motor to be used to drive the drill in rotation, operate the hammer, and to drive the helical conveyor. The over-running clutch in the impact hammer drive will also provide the capability of operation in two modes. One is a combined rotary/impact mode of drilling which is the normal mode for hard rock. By reversing the drive motor rotation, it is possible to run the drill in a simple rotary drilling mode since the over-running clutch will not now engage the hammer actuation crank. For the configuration as shown in Figure 3, reversing the drive motor will also reverse the helical conveyor rotation. Thus, if the dual mode of operation is desired, some means of providing the proper rotation of the helical conveyor is necessary. This can be accomplished by using a dual gear train drive to the helical conveyor having outputs of opposite rotational sense. The proper output is then engaged to the helical conveyor through a pair of simple over-running clutches.

As shown in the configuration of Figure 3, the sample is transported up the helical conveyor and empties into a closed chamber. A jet of air or compressed gas is introduced into the chamber which picks up and carries the sample pneumatically by the flow out of the chamber. The sample is carried through a tube to a cyclone collector located in the payload. While this approach is simple to mechanize, it suffers from the fact that a stored gas supply must be available. If the drill is to be used in a large number of repeated or prolonged operations, the amount of gas required could become excessively large. More efficient use of the pneumatic gas supply can be achieved if a valving system is incorporated with the chamber located at the top of the helical conveyor so that the flow is intermittent; i.e., flow occurs for only a small part of each



FOLDOUT FRAME 1

FOLDOUT FRAME 2

FIGURE 3. ROTARY/IMPACT HAMMER DRIVE CONCEPT

rotation of the drill. A completely mechanical sample transport system, while more complex, is probably more desirable for this sampler. Such a system was pursued in the final design of this sampler mechanism.

Two methods were considered for deploying this drill. The first method is a sliding bar arrangement as shown schematically in Figure 4. In this arrangement the intermediate sliding bar is extended at the same time that the drill is traversing along it. The initial intent was to utilize a single drive for all the elements of the deployment mechanism. A defect of this scheme is that if the surface of the soil is encountered at some distance less than 12 inches below the stowed position, the depth of penetration is limited to a value equal to the distance traveled in reaching the surface. This is caused by the intermediate sliding bar reaching the surface which will then preclude further advance of the drill. This can be avoided by moving the drill and then the intermediate bar sequentially at the expense of some additional complexity in the drive mechanism. The maximum extension for this approach is 24 inches. Initial attempts to mechanize this approach did not result in as simple and compact a structure as appeared to be probable.

An alternate method for deploying the drill is shown schematically in Figure 5. This mechanism consists of two parts, a parallel bar linkage to deploy the drill to the surface and a longitudinal feed screw to advance the drill into the surface. The diagonal strut attached to the lower parallel bar link is an extensible member such as a lead screw running in a threaded sleeve. In operation the diagonal strut extends causing the parallel bar linkage to deploy the drill to the surface. When contact with a substantial surface is made, the deployment load builds up which can be sensed to terminate the deployment operation. The diagonal strut and lower parallel bar link then forms a rigid truss in conjunction with the support structure. The axial feed screw is then activated to begin the drilling operation. This mechanism can be stowed in fundamentally the same volume as the first approach; however, it provides more versatility in emplacing the drill both radially and vertically. Eight inches more vertical reach is achieved than with the first approach. One possible disadvantage is that this deployment mechanism could be less rigid; however, since the impact energy is low (approximately 1.5 ft-lbs) and the axial thrust is low (approximately 20 lbs), the requisite rigidity should be achievable. This approach was used in the final design effort for this sampler.

The details of the final prototype design of the rotary/impact drill mechanism is shown in Figure 6. This drill is a combination hard rock drill and deep cone abrading sieve. This combination is used because tests have indicated that the hard rock drill configuration does not transport loose particulate material effectively since it tends to flow away from the helical conveyor entrance ports. Thus, after the drill has penetrated loose material for a few inches, the deep cone abrading sieve encounters the

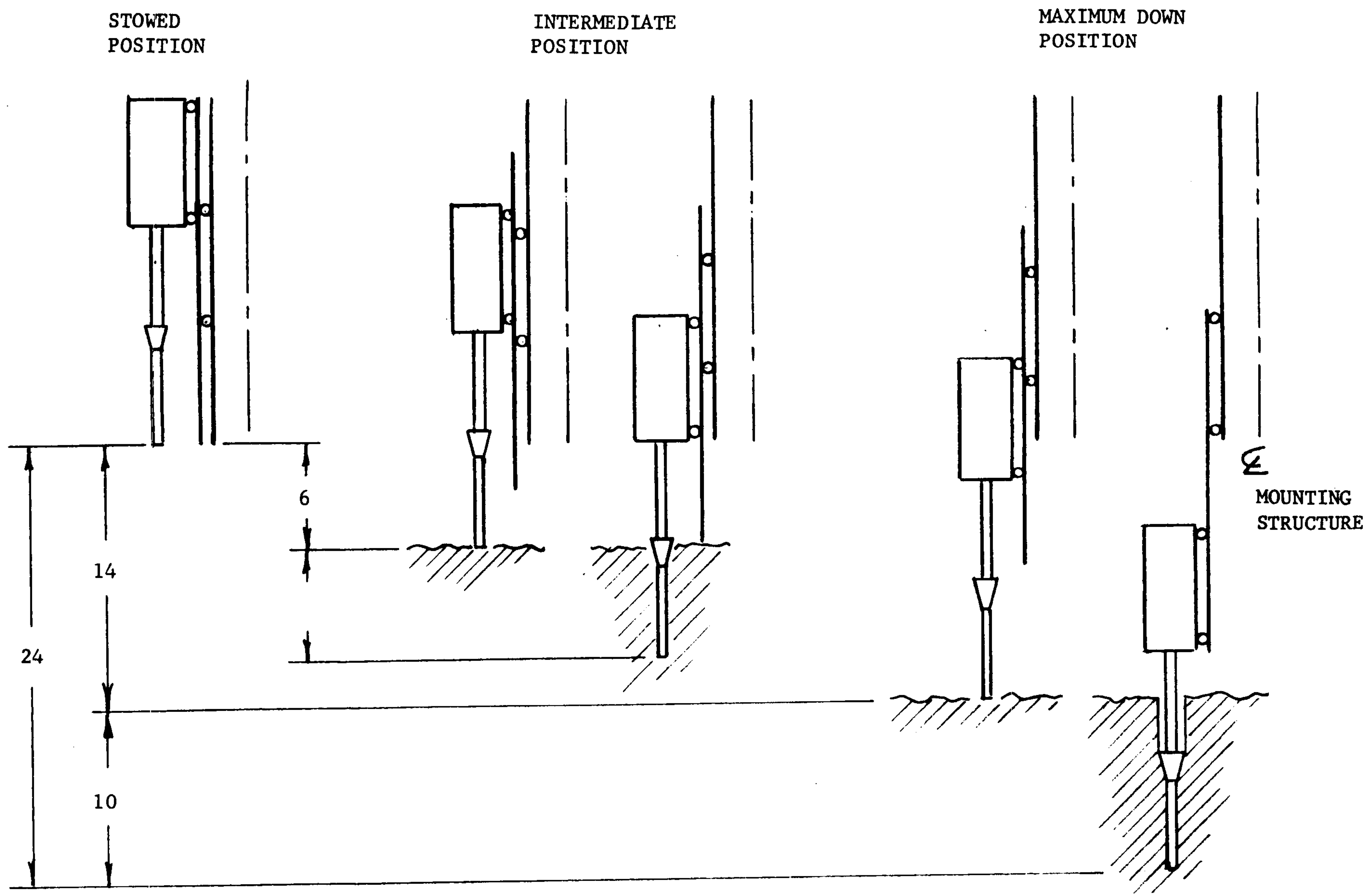


FIGURE 4. SLIDING BAR DEPLOYMENT MECHANISM

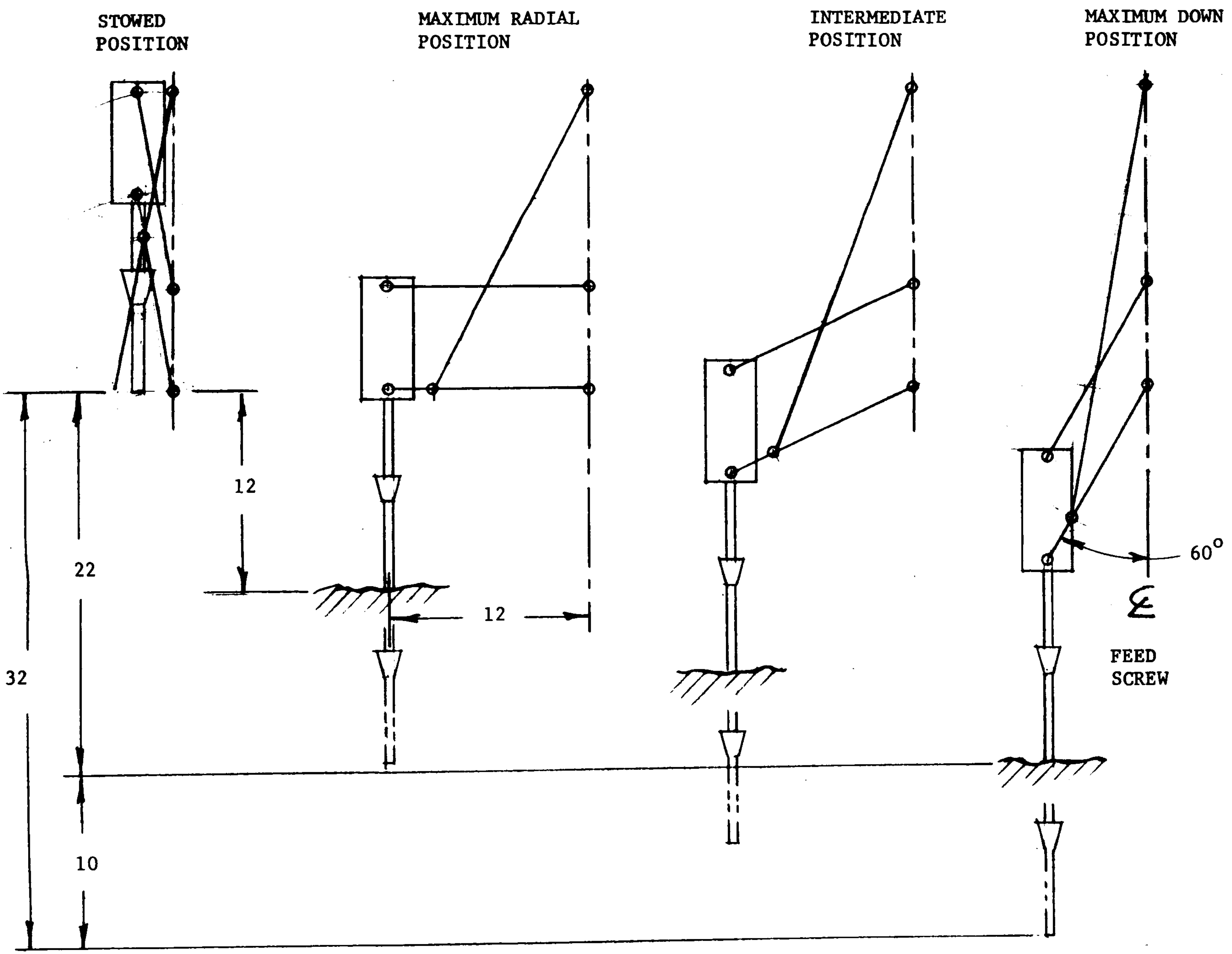


FIGURE 5. PARALLEL BAR DEPLOYMENT MECHANISM

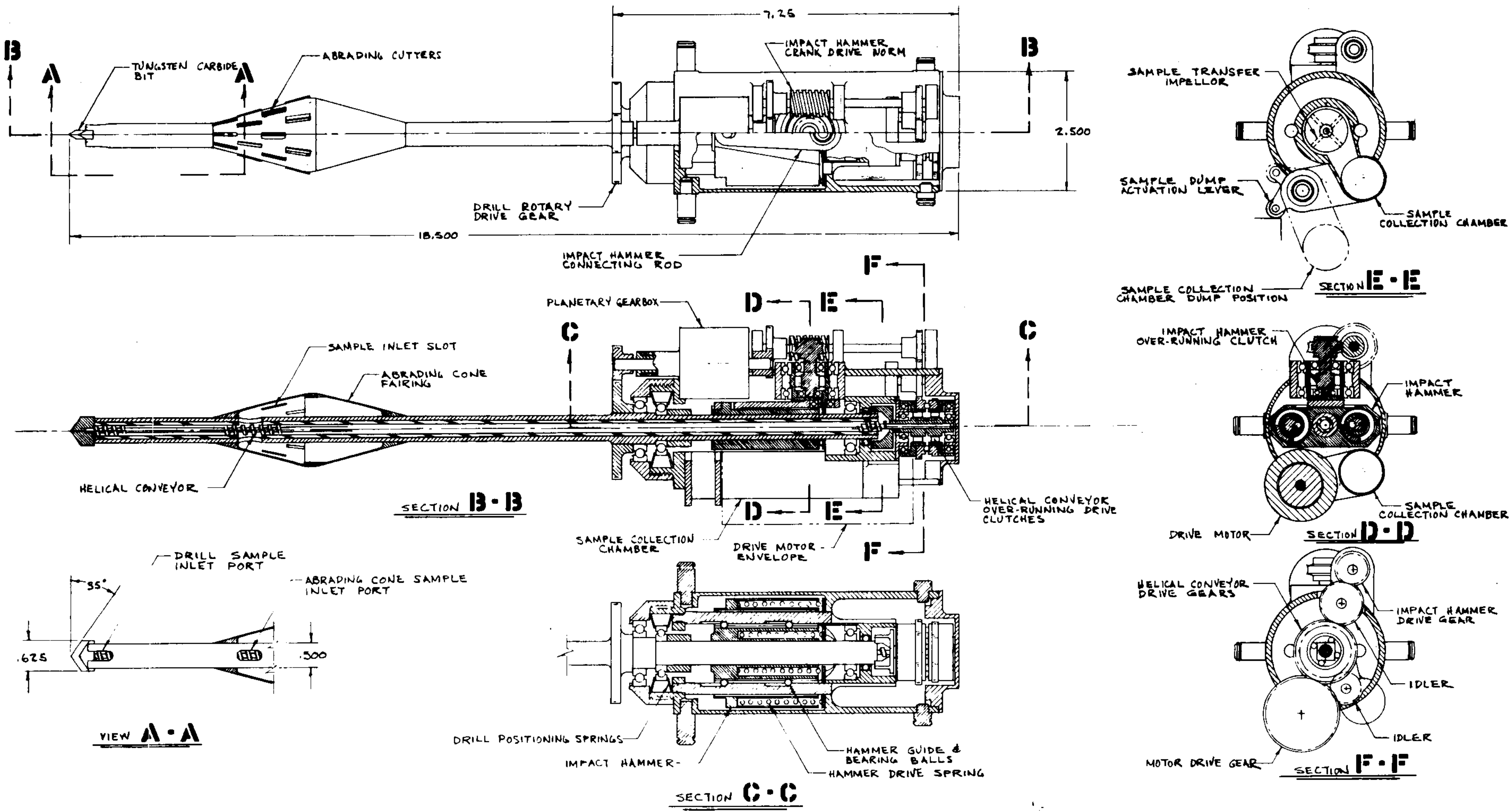
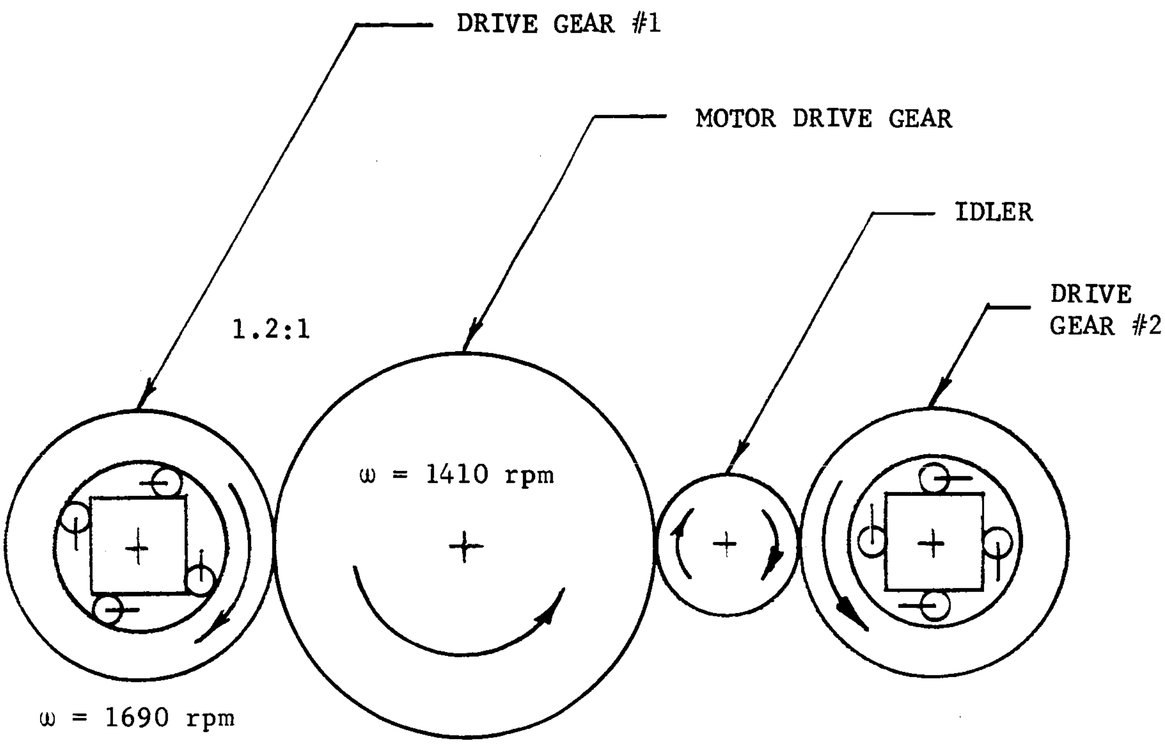


FIGURE 6. ROTARY/IMPACT DRILL ASSEMBLY, E-1

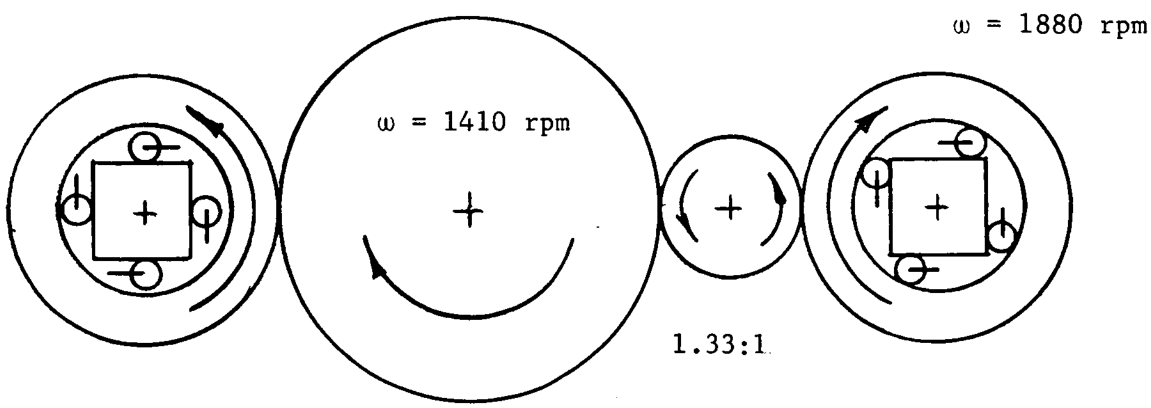
surface. Soil that enters the abrading cone collects at the base of the cone covering the ports to the helical conveyor. Since the outer walls of the cone confine the soil it is more readily picked up by the helical conveyor. Both the drill tip cutter and abrading cone configuration are based on the breadboard models built by JPL. This design represents a compromise that should operate under a larger variety of conditions than either a drill only or abrading cone only. One drawback of this design is that soil being transported up the helical conveyor will spill out of the entry ports at the abrading cone until they are covered, thus delaying the delivery of soil to the sample collection chamber. The amount of soil required to cover these ports can be minimized by shaping the interior of the cone in the area of the entry ports to reduce the free volume. This can be accomplished by filling in the free volume except in the immediate vicinity of the helical conveyor entry ports.

Either a simple rotary drilling mode or a rotary/impact drilling mode can be employed with this sampler mechanism. Since the configuration of the drill tip cutter operates equally well for either direction of rotation, this can be accomplished with one drive motor simply by reversing the direction of rotation. By so doing, the over-running clutch driving the type B impact hammer does not pick up the hammer crank, thus preventing operation of the impact hammer. Since the helical conveyor rotation cannot be reversed, proper rotation can be maintained by using two gear trains with opposite rotational outputs to drive the helical conveyor. The appropriate output is automatically connected to the helical conveyor through over-running clutches as shown schematically in Figure 7. Additional gearing connected to geartrain number 1 is used to transmit power to the impact hammer and to the rotary drive. The complete power train for this sampler is shown in the block diagram of Figure 8. Both drilling modes are shown. Those elements which are inactive or idling are connected with dashed lines. The higher rotational velocity of the output of gear train number 2 is a result of the required location of the gears. Both output gears are mounted coaxially with the helical conveyor shaft. Since a single motor drive gear is used, the output gear of geartrain number 2 must be smaller in diameter so that it will not engage the drive gear directly.

The lower speed helical conveyor output is used when the impact hammer is operating. Less material is transported when drilling rock and running the helical conveyor at a lower speed reduces the power consumed by it. For this configuration, the power is reduced by about 10 percent. All the breadboard drills used separate motors to drive each function such as drill rotation, impact hammer, and helical conveyor. Thus, it was necessary to estimate the size motor required to drive this design. The helical conveyor power was estimated from the power input measured for the breadboard models. The impact hammer power varies throughout the impact cycle and can be calculated from the impact rate and spring forces required to drive



A. GEARTRAIN #1 ENGAGED



B. GEARTRAIN #2 ENGAGED

FIGURE 7. HELICAL CONVEYOR GEAR TRAIN DRIVE

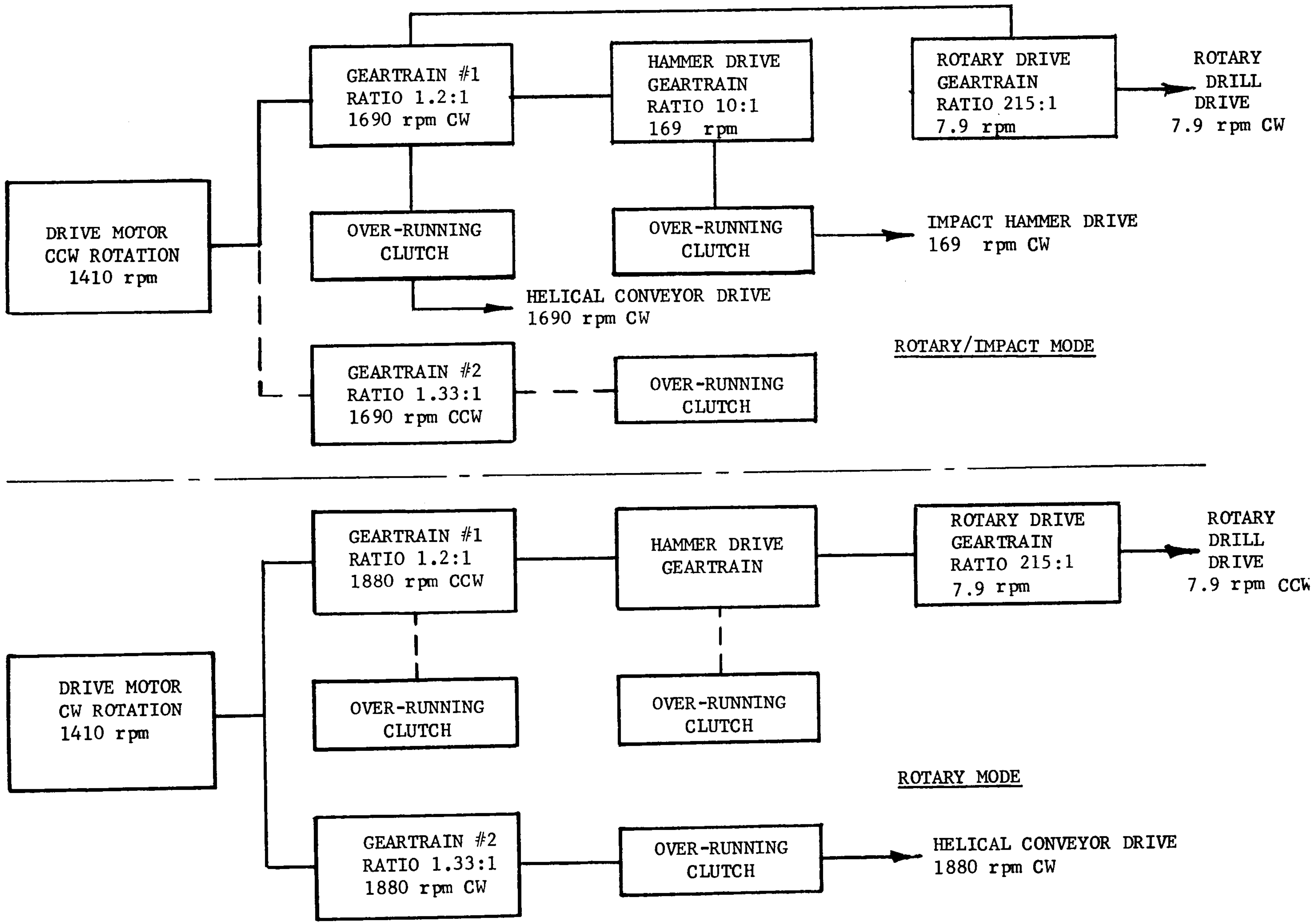


FIGURE 8. BLOCK DIAGRAM - ROTARY/IMPACT DRILL POWER TRAIN

the hammer. The drill rotational power was estimated by assuming the drill tip shears a material of 2500 psi shear strength over a circular area .625 inches in diameter. This is equivalent to the most probable value of shear strength for hard rocks such as granite and basalt. In actual operation, the area in shear would be something less than the full diameter of the hole being drilled. Thus, this is a very conservative estimate of the peak torque required to rotate the drill. These power requirements are summarized in Table V in terms of the torque required at the motor drive gear. The available power at the drive gear is given for a globe BD gear-motor (102A153-11). The planetary gear reducer for this motor has all the gears mounted on ball bearings resulting in a gear train efficiency of 93 percent. The characteristics for this motor are given in Table VI.

TABLE V

ESTIMATED TORQUE REQUIREMENTS FOR THE ROTARY/IMPACT DRILL

		Torque in oz.	% of Rated Torque	% of Stall Torque	% of Cycle
Impact Mode					
Hammer	Peak Load	20.6	111	28	30
	Average Load	10.3	55	14	60
	Minimum Load	0	0	0	10
Helical Conveyor		10.8	58	15	100
Rotary Drive	Peak Load	7.3	39	10	100
	Total Peak Load	38.7	209	52	300
Rotary Drill Mode					
Hammer		0	0	0	0
Helical Conveyor		12.0	65	16	100
Rotary Drive	Peak Load	7.3	8	2	100
	Total Peak Load	19.3	104	26	100

TABLE VI

BD GEARMOTOR CHARACTERISTICS

Characteristics	Value
Motor Only -	
Rated torque, in-oz	3.6
Stall torque, in-oz	14.4
Rated current, amperes	1.0
Rated voltage, volts	27.0
Rated speed, rpm	7800
Planetary Gear Reducer -	
Gear ratio	5.54:1
Gearbox efficiency, %	93
Rated torque, in-oz	18.5
Stall torque, in-oz	74.1
Output speed, rpm	1410

Thus, it is seen that this motor must be over driven in the impact mode; however, it fits the requirements closely enough to be a reasonable estimate for the required size of motor. A more accurate assessment of the power requirements can only be made by testing a prototype model.

In an attempt to minimize power losses in this sampler mechanism, extensive use of ball bearings has been made in this design. The only sliding contact occurs between the drill stem and the drill stem support bearings. Sliding at these points occurs only during an impact from the hammer. This is essentially the manner in which the impact drill stem, tested by Hughes Tool Company for JPL, was mounted. The impact hammer is supported between two guide rods on four ball bearings as shown in Section C-C of Figure 6. Thus, as the hammer oscillates, the ball bearings roll back and forth between the hammer and the guide rods for a distance equal to one-half the stroke of the hammer. In order to ensure the rolling action of these balls, some preload should exist between the guide rods, the ball bearings, and the hammer. In order to prevent the balls from dropping out of their races under vibration and impact, the races are machined with a finite length slightly longer than the travel of the ball. In order to minimize the storage stresses on the hammer drive springs, the hammer is parked in the full down position. This is accomplished by driving the sampler in the rotary drill mode. In so doing the over-running clutch eases the hammer down to the impact position and leaves it there from any position it may have wound up in on the spring compression stroke. This feature will also provide automatic repositioning of the hammer ball bearings should they become loose enough to fall out of position between runs. This occurs during the first spring compression cycle so that any sliding of the balls that is required occurs at the low velocity existing during this part of the cycle. When the hammer reaches top dead center, the balls will then be in the proper position to roll properly as the hammer oscillates. It is not likely that the balls will drift out of their position during hammer oscillation because of the rapid action and bearing preload. At any event, if they do move slightly they will always be repositioned during the slow spring compression part of the cycle. This could increase the electrical power demand to withdraw the hammer but should not result in any reduction in the hammer's impacting efficiency.

The hammer weight for the configuration shown in Figure 6 is .85 pounds which is almost half the weight of the hammer used by Hughes Tool Company in their tests. The ratio of hammer weight to the impacted drill stem assembly is one to one if a steel drill stem is used. This could be raised to 1.5 to one if a beryllium drill stem were used. Also, the higher modulus of elasticity of the beryllium would result in a more efficient transfer of impact energy to the rock being drilled.

The hammer configuration used in this design utilizes two drive springs rather than one. This results in a flat hammer which allows other components to be mounted close to the center line of the drill assembly. It is also possible to pass the drill stem through the hammer thereby obtaining maximum spacing between the drill stem support bearings without increasing the length of the total assembly. The two drive springs operating in parallel combine to produce a spring rate of 40 pounds per inch. The driving force produced by these springs varies between 30 pounds at maximum extension and 50 pounds at maximum compression which is the same as the stiffer hammer spring used by Hughes Tool Company in their testing. Each spring is steel and has a .750 outside diameter, a wire diameter of .091, and a free length of 3 inches with 20 coils.

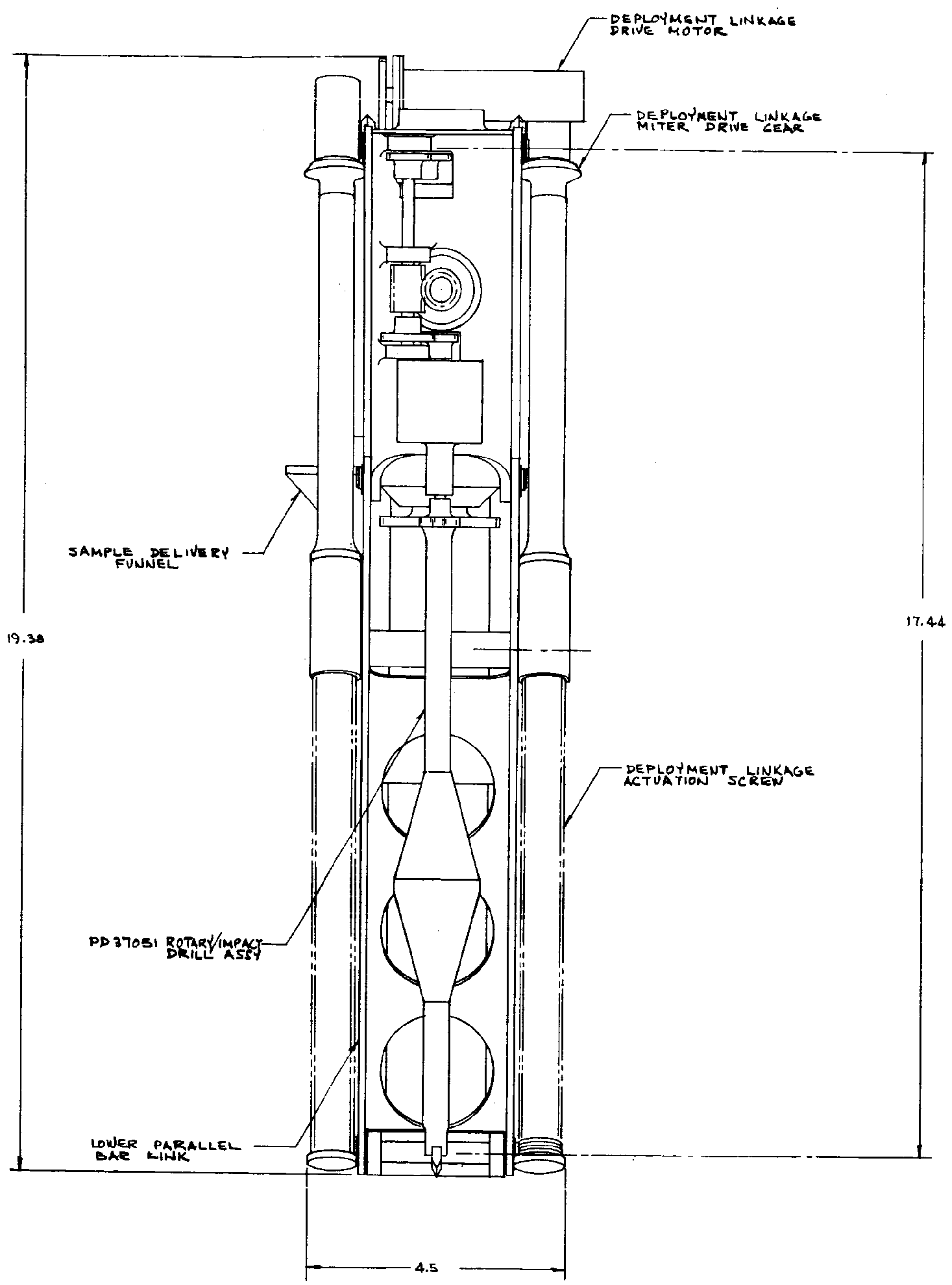
The remaining feature of this sampler assembly, as shown in Figure 6, is that the helical conveyor housing or the drill stem is lined with a silicone foam rubber approximately .06 inches thick. This is based on tests run by JPL with breadboard rubber lined conveyors. The fact that the rubber is resilient and fits snugly around the helix of the conveyor essentially eliminates the grinding characteristics associated with this type of conveyor run in a hard metal housing. In order to eliminate relative motion between the helical conveyor and the drill stem during impact, the end of the conveyor shaft is splined to fit into the driving shaft. This allows axial movement of the conveyor shaft relative to the drive shaft thereby allowing the helical conveyor to move with the drill stem during an impact. An impellor mounted on the end of the helical conveyor drive shaft accelerates the material fed out at the top of the conveyor housing to one side where it falls into a sample collection chamber. This chamber has a volume of 40 cubic centimeters which will hold about 50 grams of sample at a packing density of 50 percent. Assuming no lost sample, the maximum amount of material removed from a 25 centimeter deep hole is about 80 grams. Since it is impossible to retrieve the complete theoretical sample displaced, the collection chamber volume should be adequately large for any given sampling run. If 25 percent porosity is achieved, which is likely when the impact mode is being used, the chamber will hold 80 grams.

The complete drill assembly and its deployment mechanism is shown in Figure 9. The deployment mechanism used is the parallel bar linkage driven by a lead screw extensible strut described earlier in this section. The assembly shown in Figure 9 is shown in the stowed configuration. The support or attach points for the parallel bar links and extensible strut are on a single tube which is free to slide inside the support structure fixed to the spacecraft. An inner tube, also fixed to the support structure, provides additional support for the sliding truss support tube.

The axial feed drive motor and lead screw are housed inside these tubes and is free to float axially. The axial travel is limited by a spring which is preloaded against the base of the motor. This spring is sized to maintain a 20 pound preload minimum which increases to 25 pounds for a deflection of .062 inches. This is a steel spring with an outside diameter of 1.0 inch, a wire diameter of .105 inches, has 5 coils, and has a free length of 1.25 inches.

At the maximum deflection, a switch is actuated to turn off the feed drive motor. Thus, the axial thrust on the drill is maintained between 20 and 25 pounds by the on-off action of the feed control switch. The axial feed provides a total travel of 25 centimeters which is equal to the length of the drill stem free to penetrate the surface. The feed rate was chosen to produce a nominal uninterrupted feed of 2 inches per minute. This is a compromise to prevent an unduly slow feed in weaker materials and excessive actuation of the on-off switch in hard materials.

The support tube housing the feed screw has longitudinal splines or gear teeth machined on a portion of its outer surface. These splines mesh with a motor driven gear which rotates the sliding support tube in its support structured providing azimuth control of the sampler's position. A section through the support structure showing the longitudinal splines is shown in Section B-B of Figure 9. The total displacement in azimuth is plus or minus 60 degrees from the nominal position shown. A nominal traverse rate in azimuth of one degree per second was assumed as a reasonable value. The block diagram in Figure 10 summarizes the general characteristics of the power train drive for the drill feed; drill deployment, and azimuth drive. The motor designations used are for Globe motors since these were readily available for reference.



FOLDOUT FRAME 1

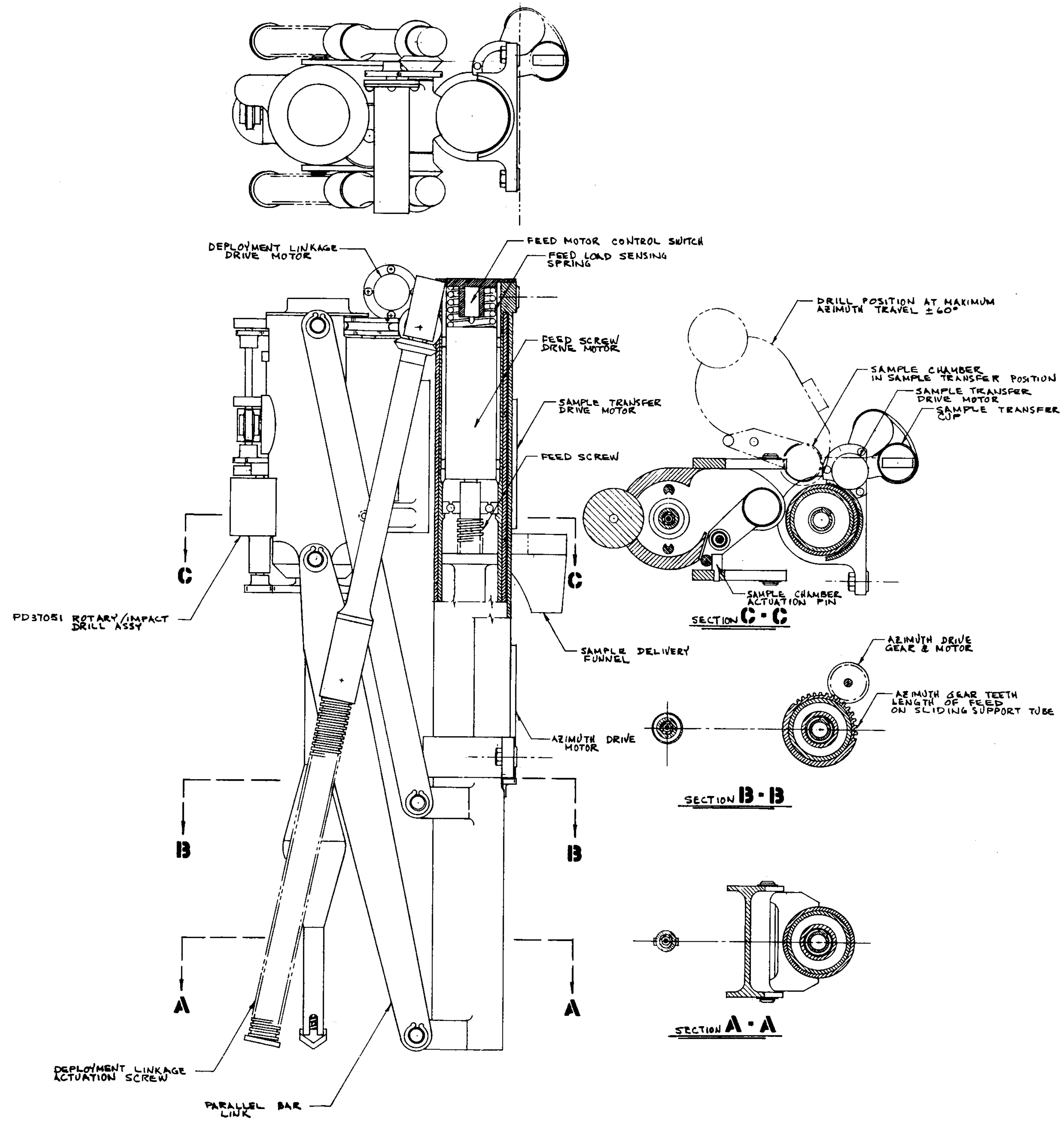


FIGURE 9. ROTARY/IMPACT DRILL AND DEPLOYMENT ASSEMBLY

FOLDOUT FRAME 2

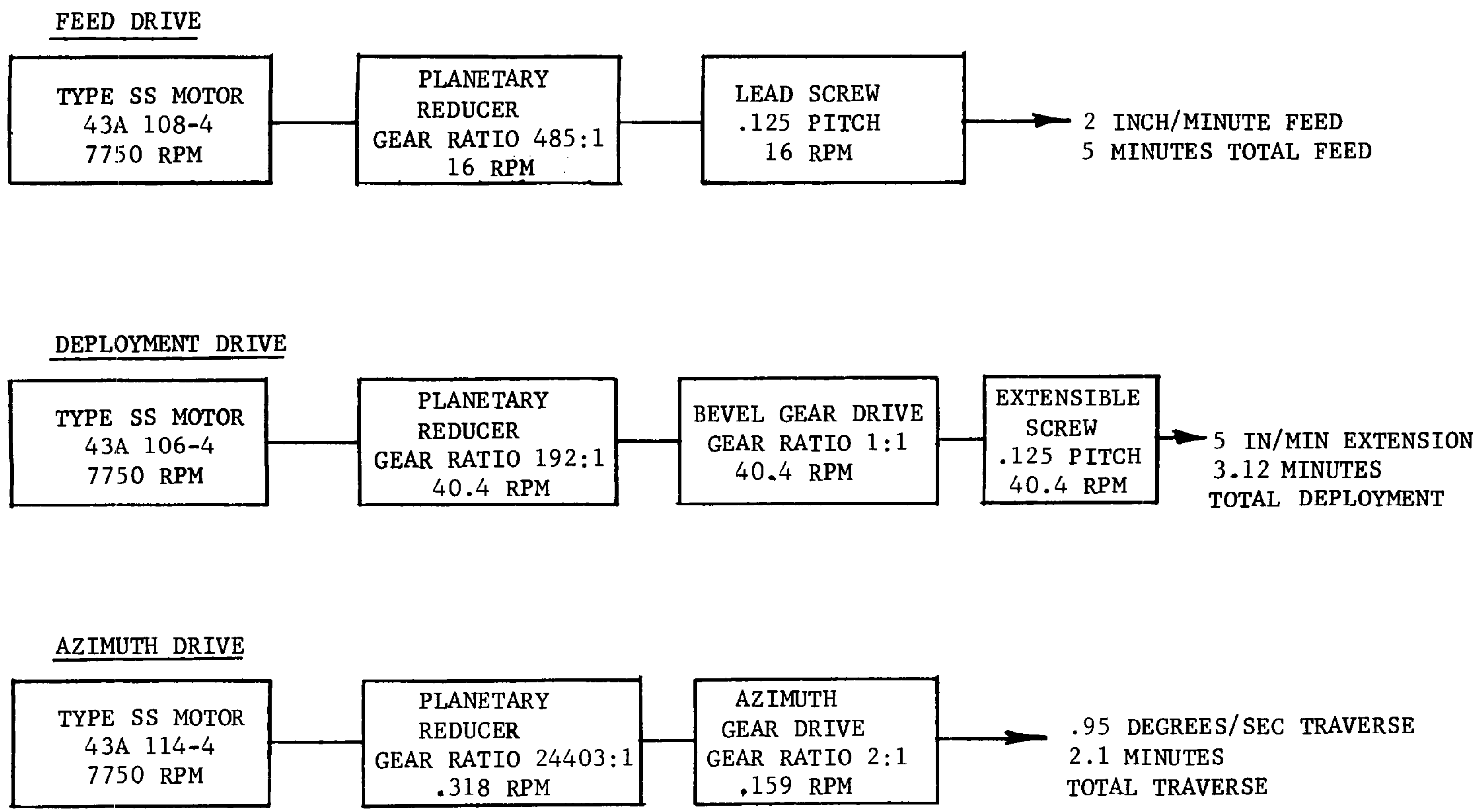


FIGURE 10. BLOCK DIAGRAM - DRILL DEPLOYMENT POWER TRAINS

To effect a transfer of the collected sample, the aximuth motor must drive the sampler to the extreme right hand azimuth position. At this position the sample collection chamber is positioned over a sample transfer cup as shown in phantom in Section C-C of Figure 9. The sample collection chamber is spring loaded to its normal position in the drill assembly housing. During approximately the last half inch of radial retraction, a pin on the upper parallel bar engages a lever connected to the sample collection chamber causing it to swing out to the transfer position. The sampler mechanism must be returned to its extreme right hand azimuth position before the final retraction is made to prevent loss of sample. The bottom of the sample collection chamber is closed by a flat plate mounted on the drill assembly housing. A hole is located in this plate to correspond with the sample transfer cup's position. Thus, as the sample collection chamber slides over the plate it carries the sample with it and drops it through the hole into the transfer cup which also functions as a measuring cup. Since this cup holds only a small quantity of the total sample it is necessary to drive the sampler assembly out radially in order to reposition the sample collection chamber over the plate thereby closing the lower opening. This is to prevent loss of sample when the transfer cup is moved to its dump position. The bottom of the transfer cup is closed with a hinged door that slides to one side as the transfer cup comes into position over the sample delivery funnel, thereby dumping the sample.

Purging the drill between runs is effected by running the drill in the impact mode of operation while the sampler is partially deployed. This will transport the residual soil in the helical conveyor into the sample collection chamber. After running for a predetermined time, the sampler assembly is retracted completely to move the sample collection chamber to the dump position. Running in the impact mode at this position will ensure that all loose material is shaken out of the sample collection chamber. It is anticipated that some residual material will be left in the sampler; however, the total amount should be small since considerable agitation is incurred when impacting the drill in the suspended position.

Figure 11 shows the sampler deployed in the maximum radial position and the maximum down position. The radial position of the drill stem varies between 3.5 inches from the support centerline in either the extreme up or down position to 13.5 inches in the extreme radial displacement. The maximum vertical travel of the drill tip possible, using both the deployment mechanism and the 25 centimeters of feed, is 32 inches. The operational sequence of events is summarized in Table VII.

A detailed weight statement for this sampler is given in Table VIII.

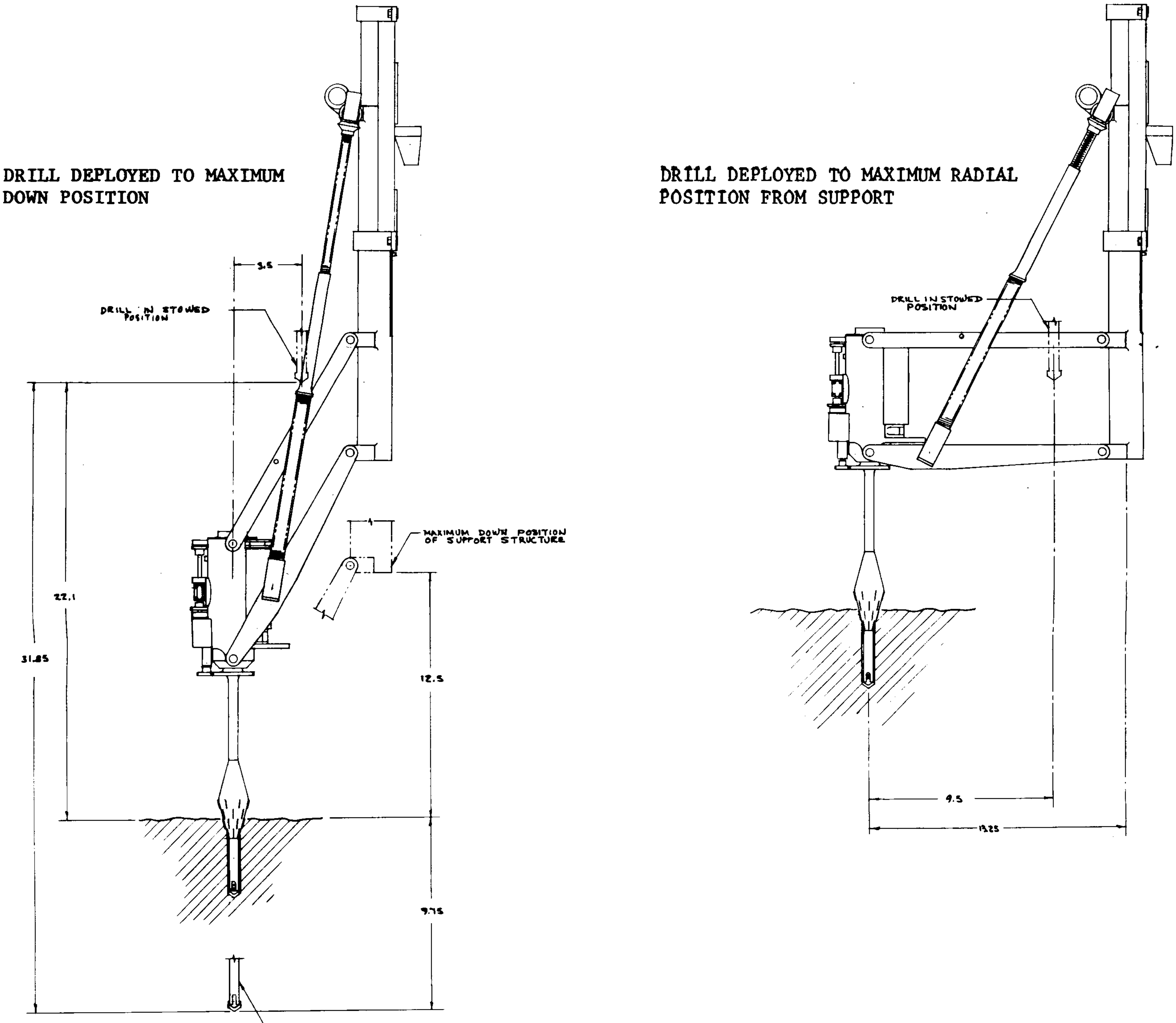


FIGURE 11. ROTARY/IMPACT DRILL DEPLOYED

TABLE VII

ROTARY/IMPACT DRILL OPERATIONAL SEQUENCE

1. Drive drill assembly to desired azimuth.
2. Deploy drill assembly until contact with the surface is sensed. This terminates power to the deployment drive motor.
3. Activate drill in simple rotary drilling mode.
4. Activate axial feed.
5. Monitor feed rate. If the rate drops to a very low value, indicating solid rock, initiate the rotary/impact drilling mode.
6. Drilling is terminated when either some predetermined time has elapsed or when maximum feed has been utilized.
7. Withdraw drill from hole by reversing feed drive.
8. When feed is completely retracted, activate deployment drive in reverse.
9. Stop retraction at rest position before sample collection chamber lever is engaged.
10. Drive the sampler assembly to extreme right hand azimuth position.
11. Position transfer cup to receive sample.
12. Complete retraction of drill assembly to actuate sample collection chamber and position it over the sample transfer cup.
13. Activate impact hammer for a few cycles to assist in sample transfer.
14. Deploy drill assembly to rest position to close sample collection chamber.
15. Drive sample transfer cup to dump position over delivery funnel.
16. Repeat steps 11 through 15 until sample is depleted or it is desired to discard residual sample.
17. Run sampler assembly for a predetermined time at the rest position in the impact mode. This transfers residual soil in the helical conveyor to the sample collection chamber.
18. Retract sampler assembly to stowed position to dump sample collection chamber. This may be done with the sample transfer cup in position if desired.
19. Run impact hammer for a few cycles to shake out collection chamber.
20. Return sampler assembly to stowed position pending the next run.

TABLE VIII

WEIGHT STATEMENT ROTARY/IMPACT DRILL, E-1

Item No.	Item	Mtl	Qty	Weight /Part	Weight /Assy	Total
	Drill Assy					
1	Drive Motor	-	1	1.4400	1.4400	
2	Gear 1.500 PD	STL	2	.0335	.0670	
3	Gear .625 PD	STL	3	.0170	.0510	
4	Gear .750 PD	STL	2	.0185	.0370	
5	Gear .875 PD	STL	1	.0200	.0200	
6	Gear 1.312 PD	STL	2	.0290	.0580	
7	Gear 2.000 PD	STL	1	.0612	.0612	
8	Worm .500 PD	STL	1	.0387	.0387	
9	Worm Gear .750 PD	STL	1	.0231	.0231	
10	Bearing .1875 Bore	STL	7	.0109	.0763	
11	Bearing .250 Bore	STL	3	.0168	.0504	
12	Bearing .500 Bore	STL	2	.0989	.1978	
13	Bearing .625 Bore	STL	1	.1429	.1429	
14	Bearing .750 Bore	STL	4	.1409	.5636	
15	Overrunning Clutch	STL	3	.0203	.0609	
16	.125 O.D. Balls	STL	16	.0022	.0352	
17	.1875 O.D. x 1 Shaft	STL	1	.0138	.0138	
18	.1875 O.D. x 3.75 Shaft	STL	1	.0517	.0517	
19	.250 O.D. x 1.25 Shaft	STL	1	.0173	.0173	
20	Clutch Housing & Crank	STL	1	.0377	.0377	
21	Connecting Rod	STL	1	.0766	.0766	
22	Clutch Housing	STL	2	.0306	.0612	
23	Sample Impellor	STL	1	.0351	.0351	
24	Hammer Guide	STL	2	.0459	.0918	
25	Hammer Spring	STL	2	.0606	.1212	
26	Hammer	STL	1	.8503	.8503	
27	Drill Stem	STL	1	.5500	.5500	
28	Drill Tip	WC	1	.0239	.0239	
29	Rubber Liner	FOAM	1	.0045	.0045	
30	Helical Conveyor	STL	1	.0638	.0638	
31	Anvil	STL	1	.0347	.0347	
32	Abrading Cone	STL	1	.1793	.1793	
33	Abrading Cutters	WC	24	.0010	.0240	
34	Belleville Spring	STL	2	.0145	.0290	
35	Support Pins, Lower	STL	4	.0034	.0136	
36	Hammer Crank Housing	Mg	1	.0304	.0304	
37	Outer Housing	Mg	1	.4200	.4200	
38	FWD Bearing Retainer	Mg	1	.0716	.0716	
39	AFT Closure	Mg	1	.0741	.0741	
40	Sample Chamber	Mg	1	.0393	.0393	
41	.1875 O.D. x .75 Shaft	STL	1	.0058	.0058	
	Subtotal		105			5.8438
	Feed Screw Assy					
42	Feed Screw Motor	-	1	.5000	.5000	
43	Load Sensing Spring	STL	1	.0386	.0386	
44	Feed Screw	Al	1	.1926	.1926	
45	Feed Nut and Tube	Mg	1	.1524	.1524	
46	Feed Screw Bearing	STL	1	.0989	.0989	
47	Outer Sliding Tube	Mg	1	1.0202	1.0202	
48	Inner Sliding Tube	Mg	1	.6565	.6565	
49	Tube Support Structure	Mg	1	.3687	.3687	
	Subtotal		8			3.0279
	Deployment Assy					
50	Lower Parallel Bar	Mg	1	.2847	.2847	
51	Upper Parallel Bar	Mg	1	.2296	.2296	
52	Parallel Bar Shaft	STL	2	.0520	.1040	
53	Extension Nut	Al	2	.0624	.1248	
54	Nut Connecting Tube	Al	1	.0490	.0490	
55	Bevel Gears	STL	4	.0312	.1248	
56	Bevel Gear Shaft	STL	1	.0233	.0233	
57	Deployment Motor	-	1	.4375	.4375	
58	Deployment Screw	Al	2	.2152	.4304	
59	Floating Screw	Al	2	.2815	.5630	
60	Deployment Screw BRG	STL	4	.0168	.0672	
61	Azimuth Drive Motor	-	1	.4375	.4375	
62	Azimuth Drive Gear	STL	1	.0200	.0200	
63	Sample Transfer Motor	-	1	.4375	.4375	
64	Sample Transfer Cup	Mg	1	.0259	.0259	
65	Sample Transfer Funnel	Mg	1	.1319	.1319	
	Subtotal		26			3.4911
	Total Sampler		139			12.3628

3.2 CASED ROTARY/IMPACT DRILL, E-2

A minimal design effort was pursued on this sampler. The additional mechanical complexity of this device was felt to be difficult to justify on the basis of the probability of encountering solid rock within the first 25 centimeters of the surface and the inherent reduction of reliability associated with a more complex mechanism. The intent of this sampler concept is to provide the capability of penetrating a loose overburden until solid rock is reached. At this point the casing is to effect a seal where the hole enters the rock. This is to prevent contamination of the sample collected from the rock by material contained in the overburden.

It is noted that most of the design accomplished on the uncased rotary/impact drill (E-1) is directly applicable to the design of this device. The same impact hammer drive, helical conveyor sample transport, and deployment system can be used. In addition to the drill rotary drive, the casing must be capable of being driven independently in rotation as well as provide a means for the drill to advance with respect to the casing. This sampler design is shown in Figure 12. In order to obtain an independent drive for the casing, it was necessary to add another drive motor to this assembly. It was possible to do this without increasing the configurational envelope by moving the planetary gear reducer in the rotary drive gear train to one side. This then provided sufficient space on the other side in which to install the casing drive motor. Since this drive motor overhangs the housing structure, it was necessary to reduce the diameter of the drill rotary drive gear from 2.0 pitch diameter to a 1.625 pitch diameter. This change does not materially affect the speed of the drill since most of the gear reduction takes place ahead of this point.

The fundamental approach used in this design was based on effecting a seal where the hole enters the solid rock by reaming a tapered opening at the top of the hole with shallow tungsten carbide cutters mounted at the end of the casing. Chips reamed from the rock will tend to pack between the casing tip and the reamed taper in the rock, thereby preventing overburden from falling into the hole. Helical flutes are provided on the side of the casing which can be driven in either direction. Thus, in loose material the casing can be driven to feed material down toward the entrance parts of the helical conveyor which should enhance acquisition and transport of a sample. If solid rock is encountered, the rotation can be reversed so that the flutes tend to feed material upward thereby tending to carry overburden away from the hole entrance in the rock. The casing design in Figure 12 consists of a short section free to move axially along the drill stem. This movement is resisted by a spring which is sized to produce an axial thrust on the casing of 10 pounds when fully extended and 20 pounds when fully retracted. This is a steel spring of 40 coils with a free length of 9 inches. The outside diameter is .750 inches and the wire diameter is .072 inches. The stacked length of this spring is 2.88 inches which will allow the full 3 inch travel of the casing.

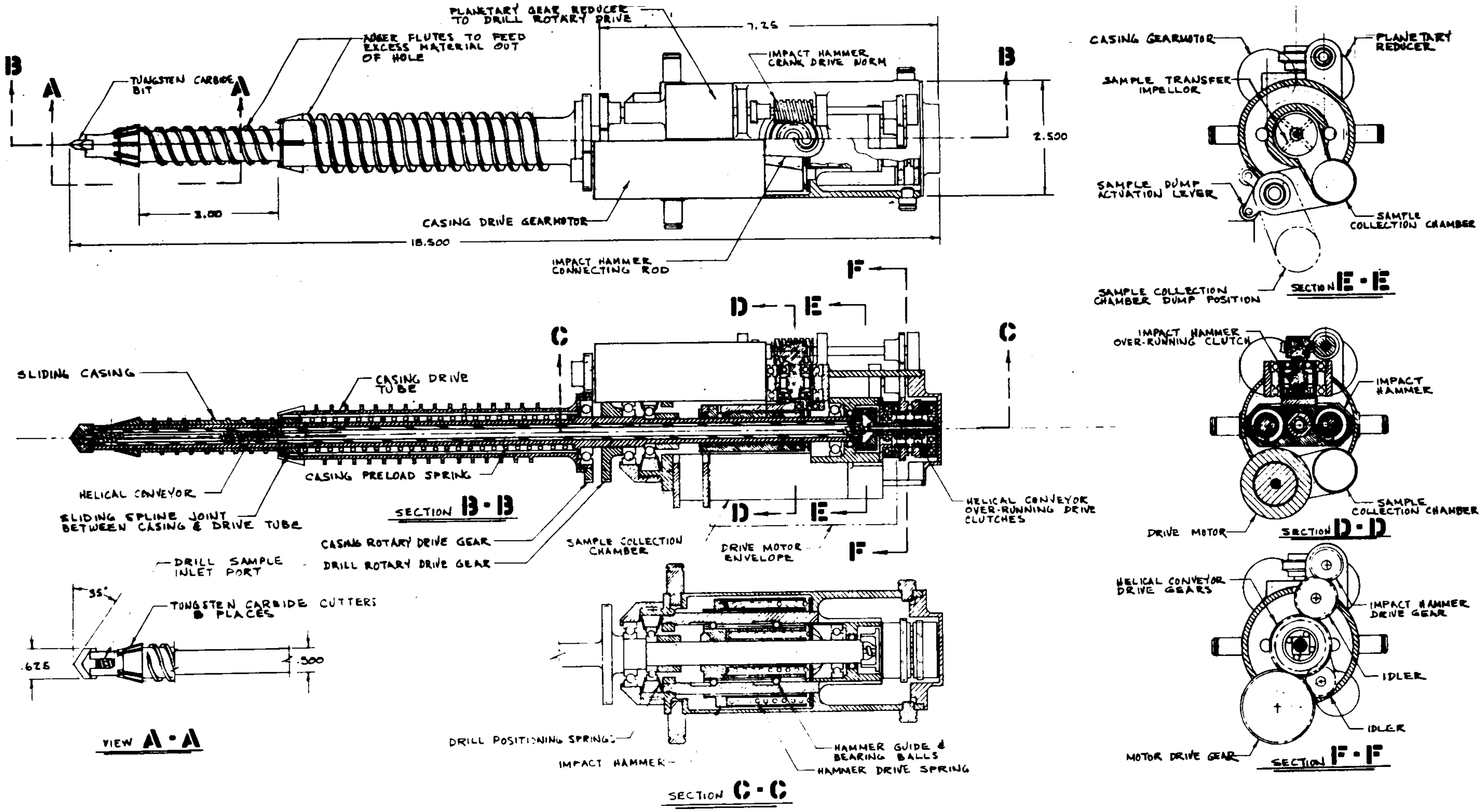


FIGURE 12. CASED ROTARY/IMPACT DRILL ASSEMBLY, E-2

This short section of casing is splined to a longer section which houses the spring and provides the rotary power. A total axial movement of 3 inches can be accommodated for the short sliding casing. A hole of this depth in solid rock would produce a theoretical yield of rock sample of 40 grams which is an adequate size sample.

The design as shown in Figure 12 has the disadvantage that soil particles will be carried up into the splined joint between the two parts of the casing. Initially, it was felt that this material could be shaken out by running the drill in the impact mode while held in the rest position above the surface. An alternate design, which is clearly superior, is shown in Figure 13. In this case the floating or sliding casing is larger in diameter than the rotating drive portion. Thus, the splined joint is not exposed directly to the soil unless the casing is driven below the surface in loose material. Since soil particles entering this joint cannot be shaken out, it is necessary to prevent entry of soil by either limiting the feed to something less than the length of the casing or enclosing the joint in a boot or bellows. It is probably best to do both of these. The only other place where soil particles might enter is the sliding interface between the drill stem and the casing. This can be minimized by incorporating a tight fitting elastomeric wiper at the tip of the casing and by having a very close fit between the drill stem and the casing. Using this alternate casing design will result in a drill mechanism which is not unduly more complex than the uncased drill and probably only slightly less reliable.

The same deployment system that was used for the uncased rotary/impact drill (E-1) can be used with this sampler mechanism. The operational sequence for this drill is given in Table IX. The weight statement is given in Table X. It is seen that only a nominal weight increase is incurred in this design over that for the uncased drill.

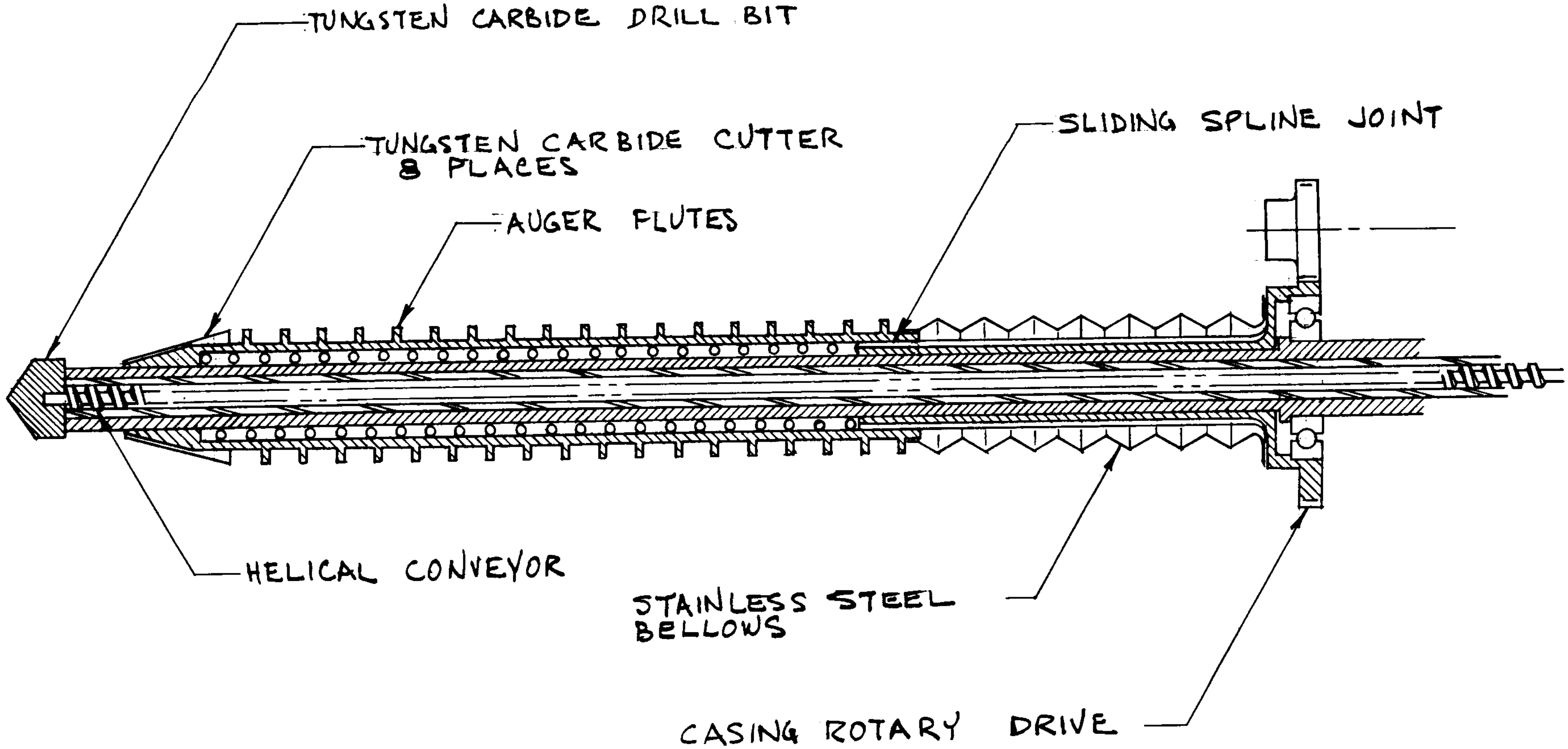


FIGURE 13. ALTERNATE CASED DRILL DESIGN

TABLE IX

CASED ROTARY/IMPACT DRILL OPERATIONAL SEQUENCE

1. Drive drill assembly to desired azimuth.
2. Deploy drill assembly until contact with the surface is sensed. This terminates power to the deployment drive motor.
3. Activate drill in simple rotary drilling mode.
4. Activate casing rotary drive in direction to cause material to feed downward.
5. Activate axial feed.
6. Monitor feed rate. If the rate drops to a very low value, indicating solid rock, reverse rotation of casing to cause material to feed up.
7. Activate rotary/impact mode of drilling.
8. Drilling is terminated when either some predetermined time has elapsed or when maximum feed has been utilized.
9. Withdraw drill from hole by reversing both the feed drive and the casing rotary drive. Reversing the casing drive will assist in lifting the drill out of the hole.
10. When the feed is completely retracted, activate deployment drive in reverse.
11. Stop retraction at rest position before sample collection lever is engaged.
12. Drive the sampler assembly to extreme right hand azimuth position.
13. Position transfer cup to receive sample.
14. Complete retraction of drill assembly to actuate sample collection chamber and position it over the sample transfer cup.
15. Activate impact hammer for a few cycles to assist in sample transfer.
16. Deploy drill assembly to rest position to close sample collection chamber.
17. Drive sample transfer cup to dump position over delivery funnel.
18. Repeat steps 11 through 15 until sample is depleted or it is desired to discard residual sample.
19. Run sampler assembly at the rest position for a predetermined time in the impact mode to transfer residual soil in the helical conveyor to the sample collection chamber.
20. Retract sampler assembly to stowed position to empty sample collection chamber. This may be done with the sample transfer cup in position if desired.
21. Run impact hammer for a few cycles to shake out collection chamber.
22. Return sampler assembly to stowed position pending the next run.

TABLE X

WEIGHT STATEMENT, CASED ROTARY/IMPACT DRILL, E-2

Item No.	Item	Mtl	Qty	Weight /Part	Weight /Assy	Total
	Drill Assy					
1	Drill Assy, E-1	-	105		5.8438	
2	Casing Drive Motor	-	1	.7500	.7500	
3	Gear .75 PD	STL	1	.0185	.0185	
4	Gear 2.00 PD	STL	1	.0612	.0612	
5	Bearing .625 Bore	STL	1	.1429	.1429	
6	Drive Casing	STL	1	.3145	.3145	
7	Drive Casing Blades	WC	4	.0020	.0080	
8	Sliding Casing	STL	1	.1401	.1401	
9	Sliding Casing Blades	WC	8	.0015	.0120	
	Subtotal		123			7.2910
	Feed Screw Assy		8			3.0279
	Deployment Assy		26			3.4911
	Total Sampler		157			13.8100

3.3 CONICAL ABRADING SIEVE CONE, E-3

This sampler design effort is based on the deep abrading cone sampler developed by JPL and tested in the field as reported in the final report covering that task. In the form tested, this sampler used a helical conveyor running in a rubber lined casing. Since this method of soil transport was used on the uncased rotary/impact drill design (E-1), it was decided to explore other approaches for transporting the sample with this sampler mechanism. No changes in the external configuration of the abrading cone were made in this design since the existing configuration performed satisfactorily in the field tests.

Two sample transport methods using pneumatic transport were considered. The first approach is shown in Figure 14. In this approach the abraded soil sample passes through the slotted openings in the cone and collects at the base. When sufficient sample has collected to a depth which covers the base of the low speed helical conveyor or auger feed, soil is carried up the short helical conveyor and spills into a closed chamber. The drive shaft to the helical conveyor is also a tube that provides a path for pressurized transport gas which emanates from the exit tubes creating an aerosolizing jet to pick up the soil particles. The flow out of the chamber up the drive shaft of the abrading cone carries the soil sample through a transfer tube to a cyclone collector. The soil in the feed auger or helical conveyor acts as a seal to prevent the pneumatic flow along the auger. The gas supply is valved intermittently to conserve the amount of transport gas required.

Another approach based on the valved sample transport system used on the breadboard model of the cylindrical abrading sieve with closed pneumatic transport developed by JPL is shown in Figure 15. In this case an internal cavity attached to a hollow shaft mounted concentrically with the abrading cone drive shaft serves as a receptacle into which the soil is deposited. This central tube is fixed to the support structure so that it does not rotate with the abrading cone drive shaft. The soil particles enter the conical head through the entry slots and fall to the bottom of the cone where it is free to enter the cavity through an opening in the bottom third of the internal cavity. An elastomeric seal bonded to the inside surface of the abrading cone closes this opening once in every rotation of the abrading cone since the internal cavity does not rotate. At the same time a valve located at the top of the abrading cone admits a quantity of pressurized gas which emerges as an aerosolizing jet from the gas outlet tube. This gas then flows up the central support tube to a cyclone collector carrying the soil sample with it.

The breadboard model tested in the field had the drive and feed mechanism gimballed at the upper end so that the abrading head could swing away from a nominal vertical position to which it was spring loaded. This was done to allow the abrading head the freedom to seek a desirable entry point

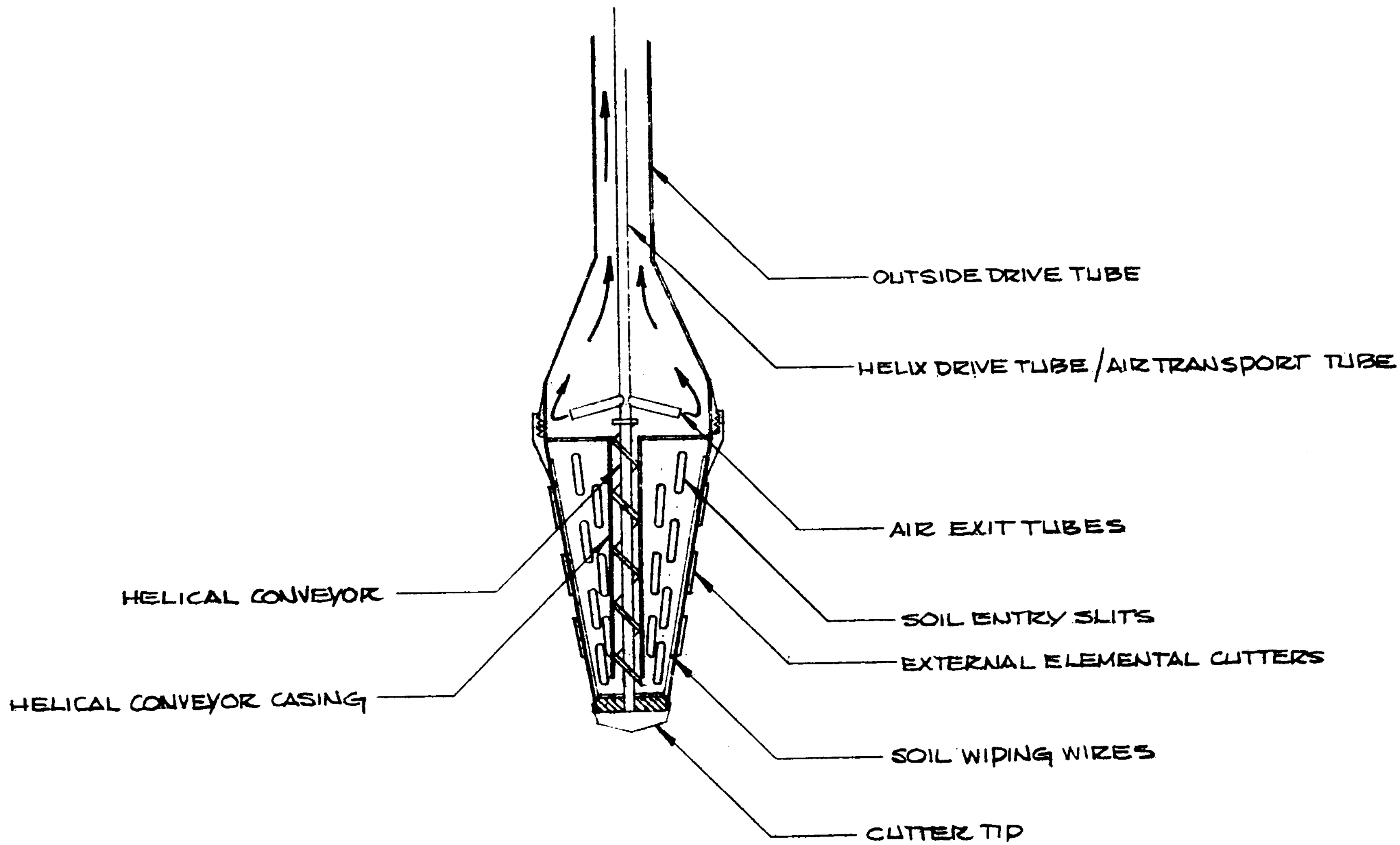


FIGURE 14. AUGER FED PNEUMATIC TRANSPORT

FIGURE 14. AUGER FED PNEUMATIC TRANSPORT

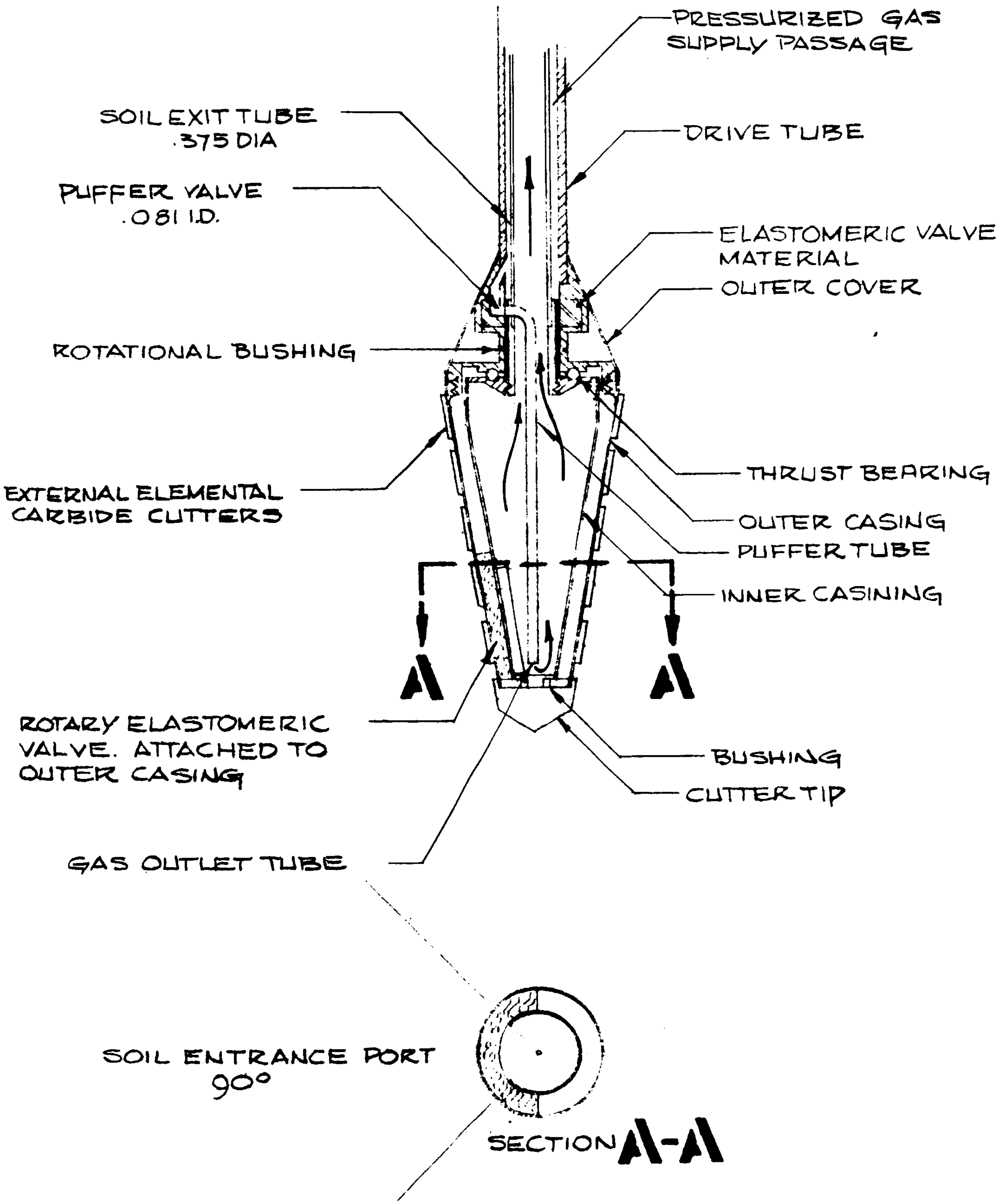


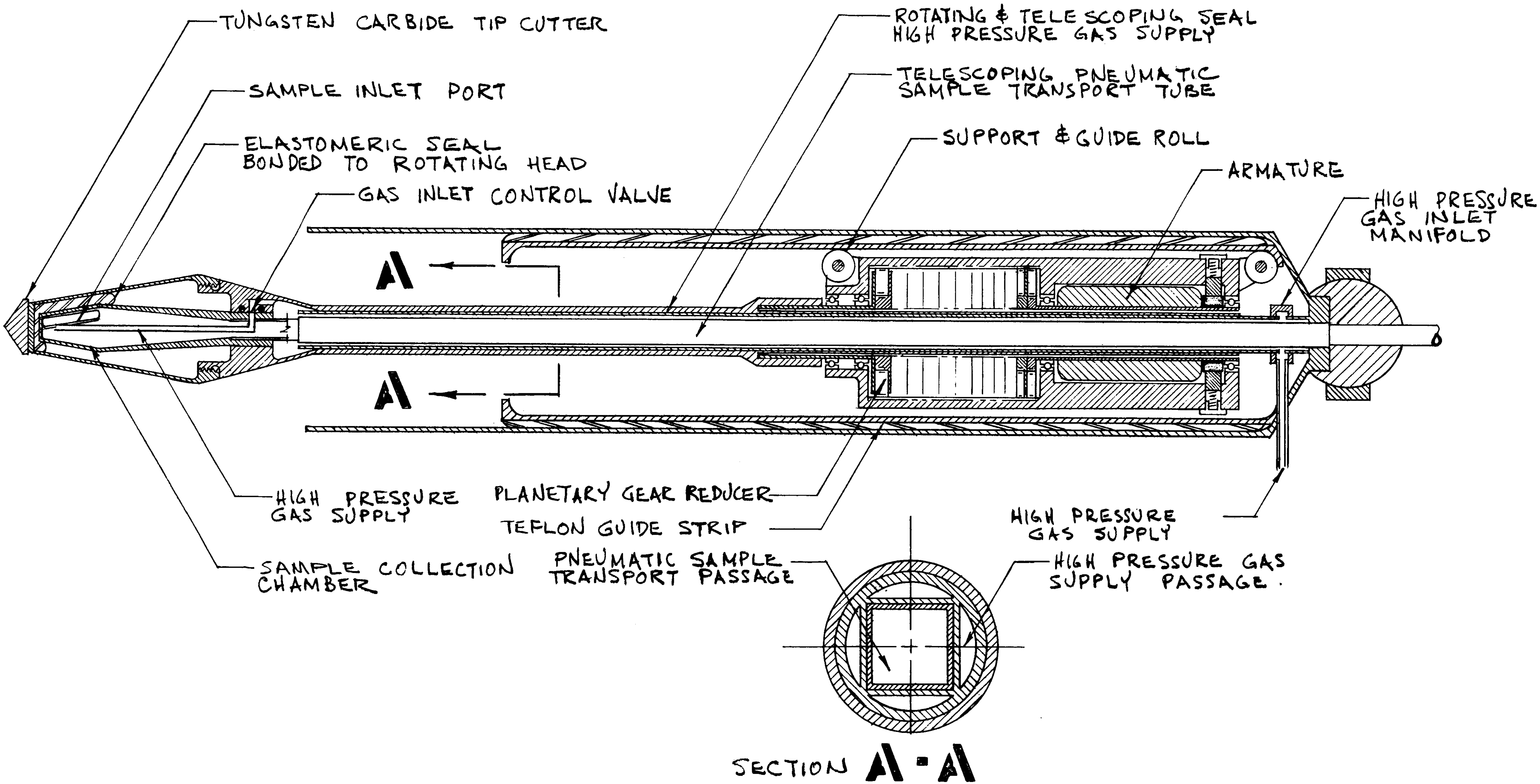
FIGURE 15. VALVED PNEUMATIC TRANSPORT

into the surface between any rocks or cobbles that might be encountered. This requires the feed to be also gimballed so that the axial feed is always applied in a direction parallel to the axis of the abrading cone and the hole it digs. This design replaces the four bar linkage and pivot which provided this freedom of movement with a spherical ball joint which is more compact. Cantilever springs arranged around the periphery of the drive mechanism serve to spring load the mechanism to the nominal vertical position. These springs also react the torque generated by the abrading head. An alternate approach would be to use a bonded rubber joint similar to shock mount suspensions to provide the gimballing action. However, consideration of ultimate dry heat sterilization requirements and the long term aging of rubber would dictate that the reliability of the all metal construction using springs should be used.

The detail of the drive assembly for this sampler is shown in Figure 16. In order to obtain a clean sample transport flow path, it was assumed that the drive motor and planetary gearbox could be fabricated with hollow shafts through which the gas transport tubes could pass.

The central pneumatic soil transport tube is a square telescoping tube which prevents rotation of the sample collection chamber and extends as required to accommodate the axial feed. Mounted concentrically with the pneumatic transport tube is another telescoping tube element. The annular space between these tubes provides the passage space for the high pressure gas supply. This supply is valved into the chamber as required through a rotary valve built into the abrading sampling head. The pneumatic sample transport tube passes out through the ball joint to a flexible tube that is carried along one of the parallel bar links to a cyclone collector.

The drive motor, reducing gearbox, drive shaft, and conical abrading head are connected together as a subassembly. This subassembly is supported by six rollers inside a tube so that axial feed can be achieved as a unit. This tube is in turn supported inside another tube. Thus, the drive motor/conical abrading head assembly moves down the intermediate tube until it picks up a stop at the end. As feed progresses, this tube is then free to move down the outer fixed support tube. The sequence is reversed during retraction. It should be noted that the intermediate tube is free floating between stops at either end so that the actual sequence of movement is arbitrary. This floating intermediate tube was incorporated into the design primarily to reduce the stowed length to a minimum while maintaining the ability to obtain a total axial feed of 25 centimeters. Some simplification in the design could be obtained by making the outer support tube longer and eliminating the floating tube element. The intermediate floating tube, as shown in Figure 16, is supported in the outer tube on teflon guide strips running between a corresponding pair mounted to the outer tube. Thus, axial movement is allowed while reaction to torque is maintained. Rollers such as used on the drive motor assembly could also be used but



FOLDOUT FRAME 1

FIGURE 16. DRIVE ASSEMBLY DETAIL CONICAL ABRADING SIEVE SAMPLER, E-3

FOLDOUT FRAME 2

would not be as compact. The lower feed rate associated with the feed does not incur significant losses due to sliding friction in this case.

Figure 17 shows the method of providing axial feed for this sampler. The drive for these tubes is provided by a perforated tape system attached to the motor/abrading head assembly. The tape passes over the drive sprocket at the upper end of the outer support tube, down the outside of the outer support tube, over an idler sprocket, and back to the attach point. Thus, a closed cable system is in effect used to provide the axial feed. A perforated tape is used since the thin tape can be made to wrap over a smaller diameter pulley than a chain or stranded cable drive.

A nominal feed rate of 2 inches per minute is used with this sampler. This is the same as was used on the JPL breadboard model tested in the field. This can be achieved with a drive sprocket speed of 1.27 rpm. The characteristics of a gearmotor that will provide the necessary output are tabulated in Table XI.

TABLE XI

FEED DRIVEMOTOR CHARACTERISTICS

Characteristics	Value
Motor number	43A114-4
Rated Torque, in oz	.45
Breakaway Torque, in oz	1.00
Rated Voltage, volts	27.0
Rated Current, amperes	.2
Speed, rpm	7750
Gear Ratio	6391:1
Gear Train Efficiency, %	32
Output Torque, in oz	300
Output Speed, rpm	1.21

The output torque is determined by the gearbox limits for continuous duty. This torque can produce a 75 pound pull on the feed drive tape which is about four times the required value of 20 pounds. In order to limit the axial thrust, a torque switch can be incorporated at the drive sprocket to turn the feed drive motor off and on as required.

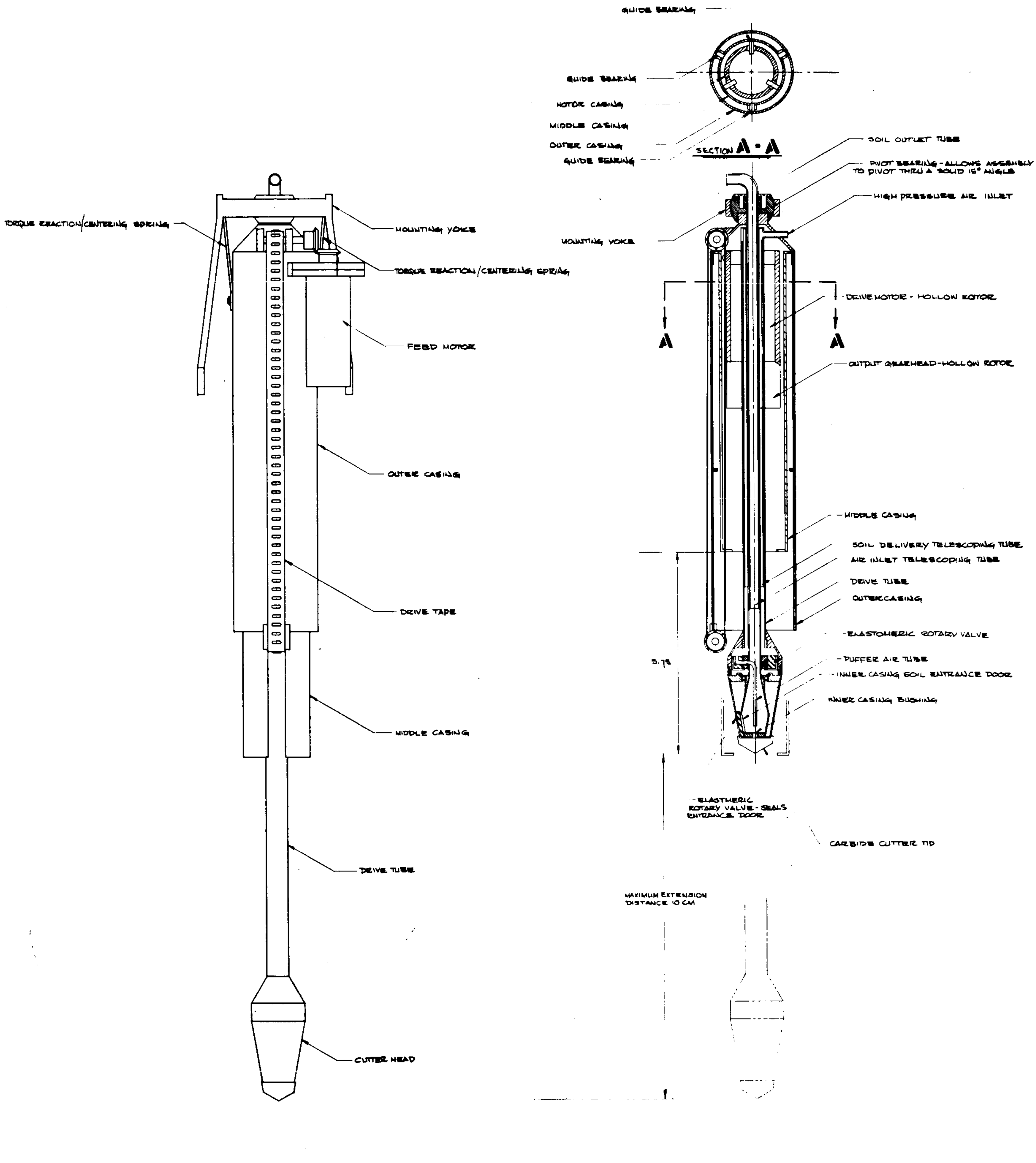


FIGURE 17. CONICAL ABRADING SIEVE CONE SAMPLER, E-3

A parallel bar linkage similar in concept to that used for the uncased rotary/impact drill is used to deploy this sampler mechanism to the surface. The complete sampler assembly and deployment mechanism is shown in the stowed position in Figure 18. Three parallel bar links are used to maintain the vertical orientation of the sampler during deployment. There are two lower links on either side of the sampler assembly and one upper link located at the sampler centerline. This link is attached to the sampler through a yoke that ties into either side of the ball support structure. The pneumatic transport tube is attached to the upper side of this link and is deployed with it. Two lead screw actuated extensible struts tie into each of the lower parallel bar links. Extension of these struts causes the sampler to be deployed until the surface is sensed or the limit of travel has been reached. After contact with the surface is sensed, this drive motor is turned off. The extensible struts together with the lower parallel bar links forms a truss support structure between the sampler and the spacecraft mounting surface. The extensible strut must extend 7 inches to achieve full deployment. If an extension rate of 2 inches per minute is used, maximum deployment is reached in 3.5 minutes which is a reasonable time. A .625 diameter acme thread lead screw is used in the extensible strut. This is a fairly large thread which should be less sensitive to dust and dirt. This thread has a lead of .1818 inches so that a total of 38.5 revolutions of the screw must be made in the 3.5 minutes. Thus, the lead screw must rotate at 11 rpm. Since the bevel gear ratio is one to one, the output of the drive motor must be 11 rpm. A type SS Globe Gearmotor 43A109-4 will provide an output speed of 10.6 rpm.

The parallel bar linkage and extensible struts are mounted to a single piece support structure which is in turn mounted on a vertical shaft. This allows the entire assembly to be rotated about this axis to any desired azimuth location between extreme positions of plus or minus 60 degrees from the nominal or center position. The azimuth drive gear and motor are mounted on the moveable support structure. This gear meshes with a sector gear fixed with respect to the spacecraft to drive the assembly to the desired azimuth. A drive rate of one degree per second will result in an operating time of 2 minutes from one extreme position to the other. This is compatible with the deployment time. The gear ratio between the fixed sector gear and the pinion is 2.28 to one resulting in a pinion speed of about .4 rpm. A type SS Globe gearmotor 43A117-4 has a gear reduction of 24400 to one. This produces an output speed of .32 rpm resulting in a total time of 2.3 minutes to sweep between the azimuth extremes.

Figure 19 shows this sampler deployed to the maximum radial position and to the maximum down position. The sampler can move radially a maximum of 8.5 inches from the stowed position and vertically a maximum of 29.6 inches including the 25 centimeters of feed.

The operational sequence for this sampler is given in Table XII. The weight statement is given in Table XIII.

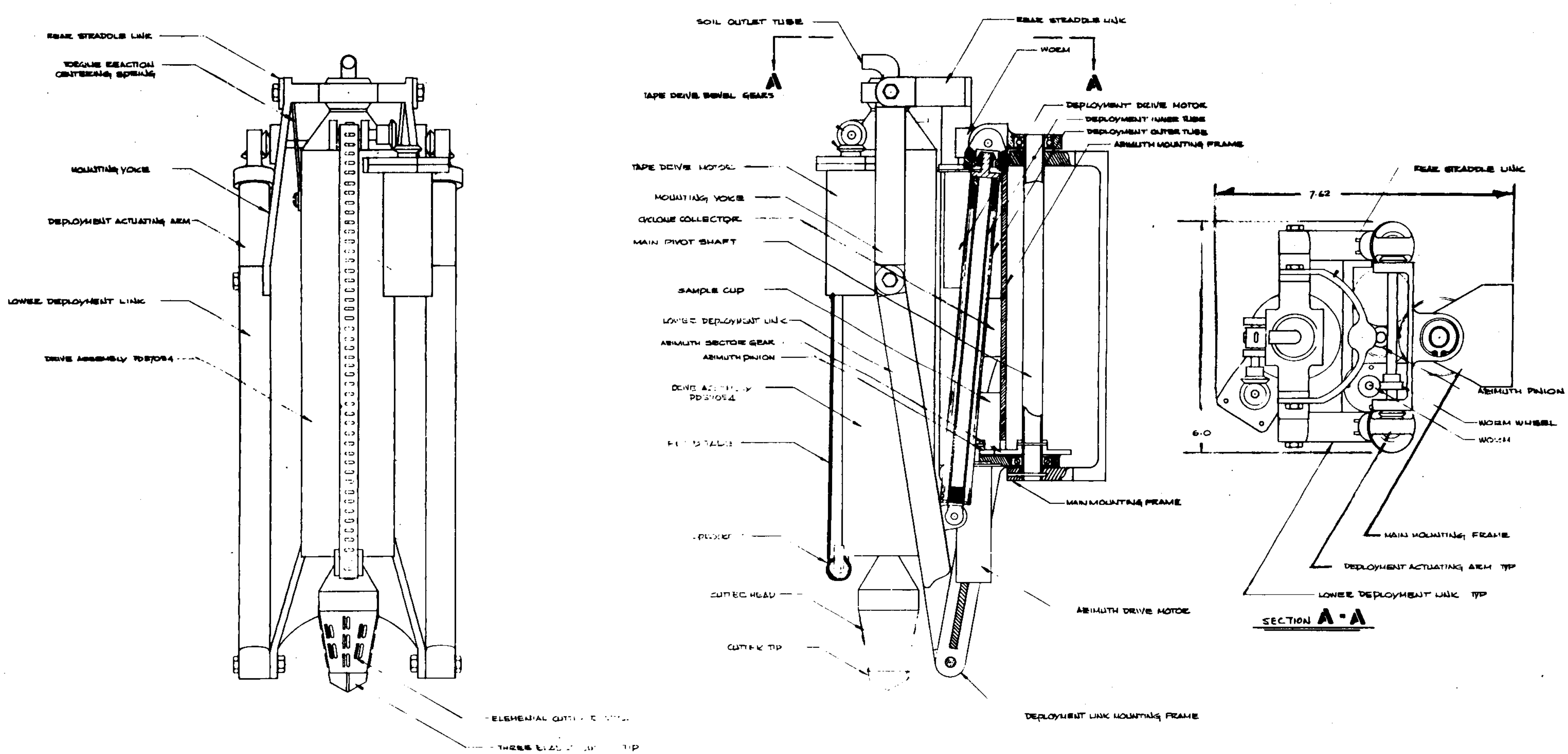


FIGURE 18. CONICAL ABRADING SIEVE CONE SAMPLER AND DEPLOYMENT ASSEMBLY

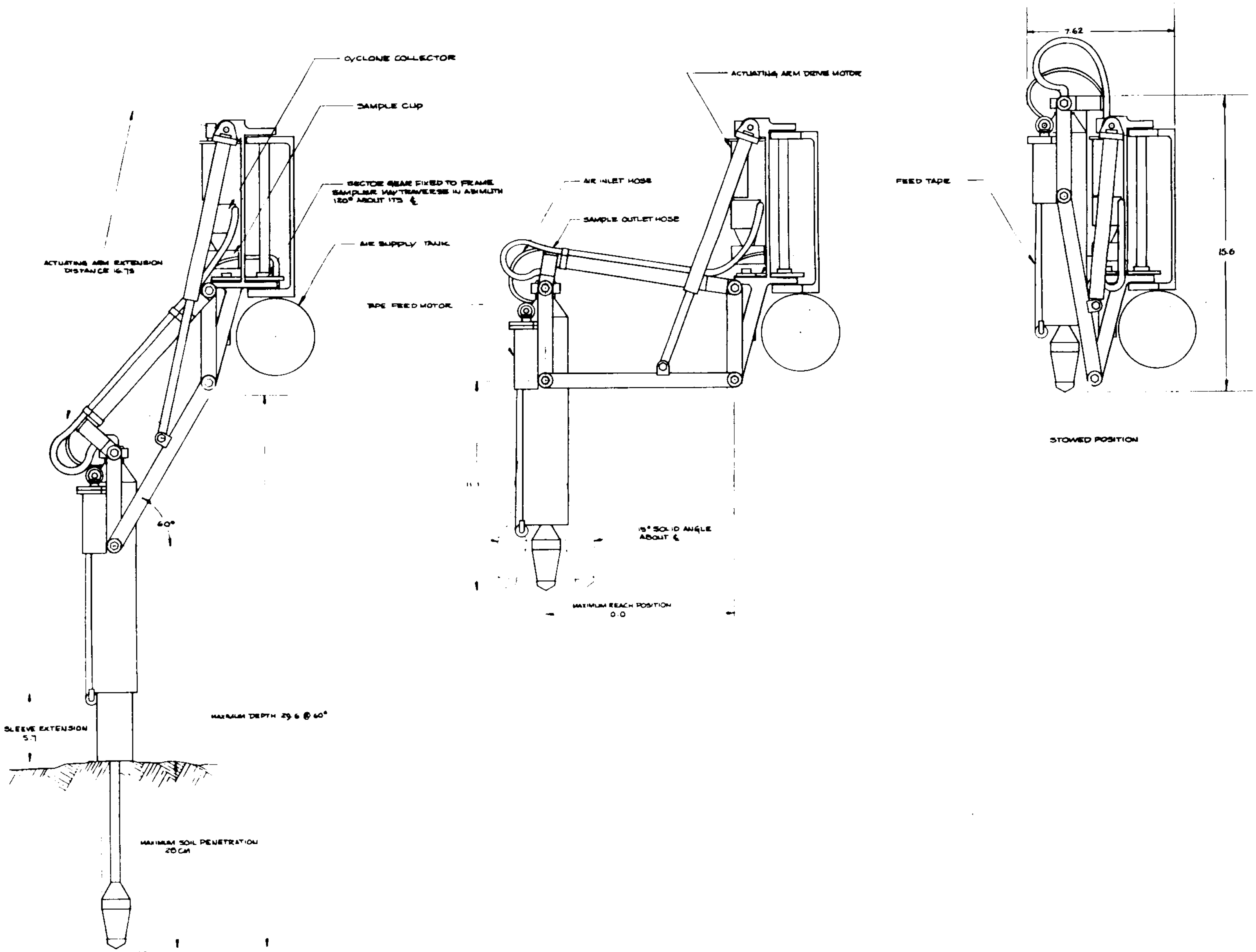


FIGURE 19. CONICAL ABRADING SIEVE CONE SAMPLER DEPLOYED

TABLE XII

CONICAL ABRADING SIEVE CONE SAMPLER OPERATIONAL SEQUENCE

1. Drive sampler assembly to desired azimuth.
2. Deploy sampler assembly until contact with the surface is sensed. This terminates power to the deployment drive motor.
3. Activate conical abrading sieve drive.
4. Activate axial feed.
5. Open valve to pressurized gas supply.
6. Sampler operation is terminated when either some predetermined time has elapsed or when maximum feed has been utilized.
7. Reverse feed drive to extract sampler from the hole.
8. Reverse deployment motor to return sampler to stowed position.
9. Operate sampler in this position with rotary drive and gas supply on to purge collection chamber.
10. Terminate purge cycle after a predetermined time and shut off gas supply.

TABLE XIII

WEIGHT STATEMENT, CONICAL ABRADING SIEVE CONE, E-3

Item No.	Item	Mtl	Qty	Weight /Part	Weight /Assy	Total
	Drive Assy					
1	Motor (Drive)	-	1	.406	.406	
2	Gearhead (Drive)	-	1	.427	.427	
3	Feed Motor	-	1	.406	.406	
4	Gearhead (Feed	-	1	.406	.406	
5	Outer Casing	Al	1	.440	.440	
6	Middle Casing	Al	1	.279	.279	
7	Drive Tube	STL	1	.101	.101	
8	Sprocket	Mg	2	.006	.012	
9	Drive Tape	Be/Cu	1	.0078	.0078	
10	Feed Bevels	Al	2	.0285	.0570	
11	Feed Motor Mount	Al	1	.018	.018	
12	Feed Drive Shaft	STL	2	.011	.022	
13	Bearings	STL	2	.014	.028	
14	Soil Tube	Al	1	.047	.047	
15	Air Tube	Al	1	.023	.023	
16	Air Inlet Tube	Al	1	.023	.023	
17	Soil Outlet Tube	Al	1	.0002	.0002	
18	Mounting Yoke	Al	1	.332	.332	
19	Pivot Bearing	STL	1	.289	.289	
20	Outer Casing Top	Al	1	.027	.027	
21	Reaction Springs	STL	2	.0061	.012	
22	Guide Bearings	STL	12	.014	.168	
	Subtotal		38			3.1350
	Cutter Head Assy					
23	Cutter Tips	STL	3	.0007	.0021	
24	Inner Casing	STL	1	.0136	.0136	
25	Puffer Tube	Al	1	.001	.001	
26	Upper Cone	Al	1	.011	.011	
27	Rotary Valve	Mg	1	.039	.039	
28	Outer Casing	STL	1	.048	.048	
29	Elemental Cutters	STL	16	.0007	.011	
	Subtotal		24			.1257
	Deployment Assy					
30	Deployment Mtg Frame	Mg	1	.754	.754	
31	Main Mounting Frame	Mg	1	.539	.539	
32	Main Pivot Shaft	STL	1	.219	.219	
33	Pin Pivot Shaft	STL	2	.0004	.0008	
34	Azimuth Drive Motor	-	1	.5	.5	
35	Azimuth Bearings	STL	2	.014	.028	
36	Azimuth Drive Pinion	Al	1	.015	.015	
37	Azimuth Sector Gear	Al	1	.007	.007	
38	Pin	STL	1	.0004	.0004	
39	Bearing Retainers	STL	2	.0007	.0014	
40	Cyclone Collector	Mg	1	.1284	.1284	
41	Deployment Motor	-	1	.5	.5	
42	Deployment Worm	STL	1	.041	.041	
43	Deployment Gear	STL	1	.039	.039	
44	Rear Straddle Link	Mg	1	.195	.195	
45	Mounting Bolts	STL	4			
46	Lower Deployment Link	Al	2	.148	.297	
47	Mtg Bolts	STL	4			
48	Deployment Bevels	Al	4	.0285	.114	
49	Deployment Bearings	STL	2	.006	.012	
50	Bearing Retainers	STL	2	.0007	.0014	
51	Deployment Drive Shaft	STL	1	.0324	.0324	
	Subtotal		37			3.397
	Deployment Link Assy					
52	Upper Head	Mg	2	.0384	.0768	
53	Bearing	STL	2	.006	.012	
54	Retainer	STL	2	.0007	.0014	
55	Bevel Bearing	STL	2	.006	.012	
56	Retainer	STL	2	.0007	.0014	
57	Outer Sleeve	STL	2	.209	.418	
58	Inner Sleeve	Al	2	.0370	.0614	
59	Lower Bearing	STL	2	.006	.012	
	Subtotal		16			.5950
	Air Supply Assy					
60	Air Supply Tank	Al	1	1.05	1.05	
61	Air Hose	PL	1	.016	.016	
62	Soil Hose	PL	1	.042	.042	
63	Mtg Clips	Al	4	.0042	.0168	
	Subtotal		7			1.1248
	Total Sampler Assy		122			8.377

3.4 HELICAL CONVEYOR SIMPLE PARTICULATE SAMPLER, E-4

This design is based on a JPL prototype using a helical conveyor mounted on a flexible shaft so that it can be run in a curved housing. This approach differs from the simple helical conveyor tested in the field in this respect. The model tested in the field had a rigid helical conveyor running in a straight casing or housing. No rubber liner is used in the housing for this conveyor, although it could be considered.

The prototype model built by JPL has two bends in the helical conveyor as shown in Figure 20. The housing is rotated about the axis as shown so that the tip of the conveyor sweeps through the arc R_1 and enters the surface of the soil. The first 5 to 6 inches of the conveyor housing is formed to a radius equal to R_1 so that it can follow the tip through the curved hole made in the soil without interference. In loose material the sampler can proceed past this point but in cemented material the penetration of the sampler is limited to this 5 or 6 inch length. The radii R_2 and R_3 are determined by the flexibility of the helical conveyor shaft. These should be kept as large as possible consistent with small size to minimize power lost in flexing the helical conveyor shaft as it rotates. The approach taken in this design is to eliminate the bend associated with the radius R_3 while retaining the ability to deliver the sample to a fixed point. Such a configuration is shown in Figure 21. In this configuration the exit is located over the receiving cup in such a manner that it sweeps across the cup symmetrically with respect to the vertical center line of the rotation point as the sampling tip traverses through a level surface. This necessitates the use of an annular bearing or support point so that the receiving cup is located at the center of rotation.

The feed of the sampler is provided by a geared drive through a slip clutch set to limit the axial thrust to a value compatible with the strength of the conveyor housing. This axial thrust at the sampler tip is designed to be limited to 2 pounds maximum.

The single bend simple particulate sampler should require less power to operate and will have a much more compact operating envelope. It should also be structurally sounder and more resistant to damage because of the reduced length of the conveyor housing and the elimination of some of the eccentric loading.

The final design configuration for this sampler is shown in Figure 22. Several features have been incorporated into this design as a result of effects noted in testing of the breadboard versions. These are primarily concerned with the tip design and the feed drive.

Breadboard testing indicated heavy wear at the entrance to the helical conveyor housing on both the housing and the conveyor helix. To reduce this wear a tungsten carbide sleeve or insert is used for the first half

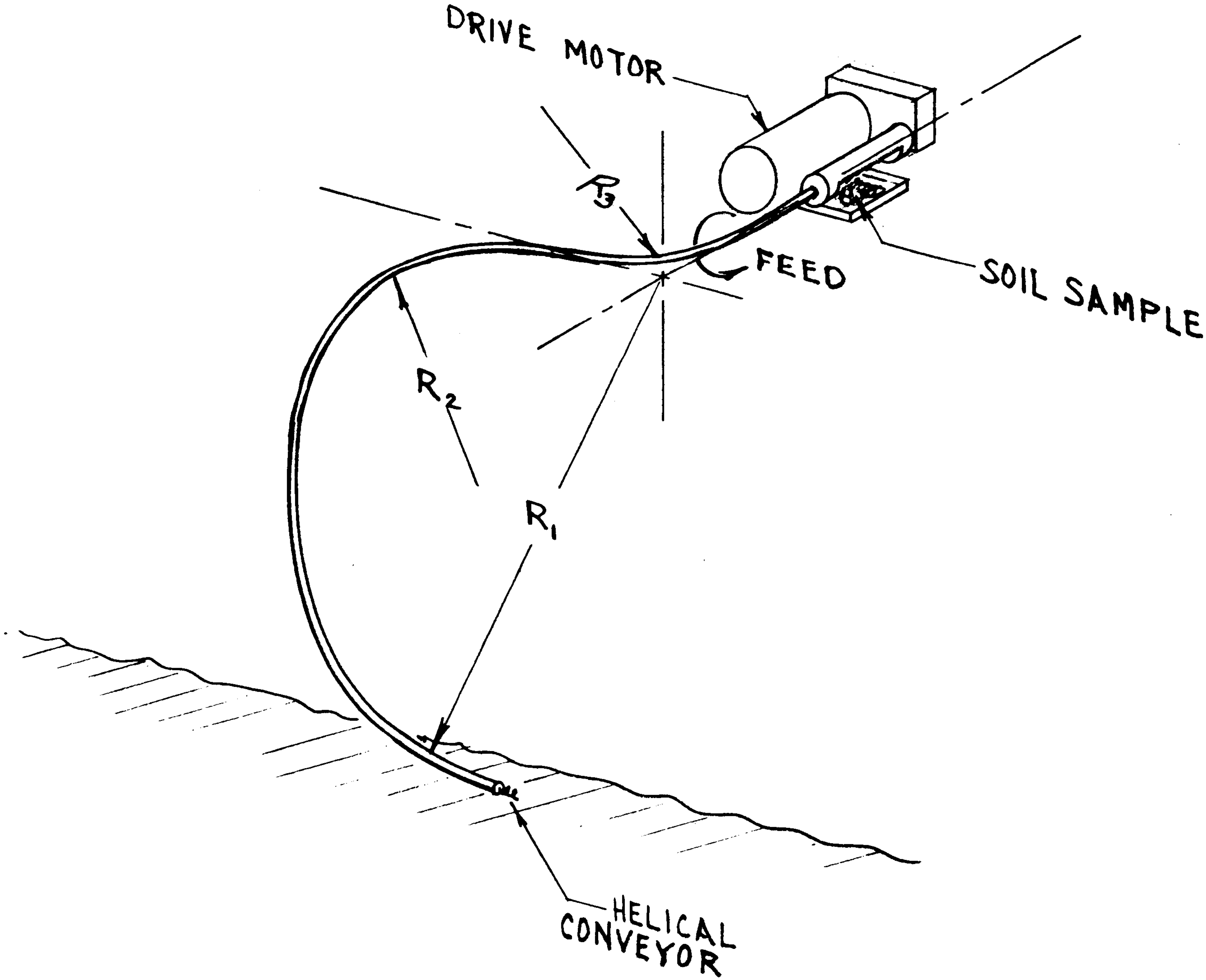


FIGURE 20. JPL PROTOTYPE GEOMETRY - HELICAL CONVEYOR SIMPLE PARTICULATE SAMPLER

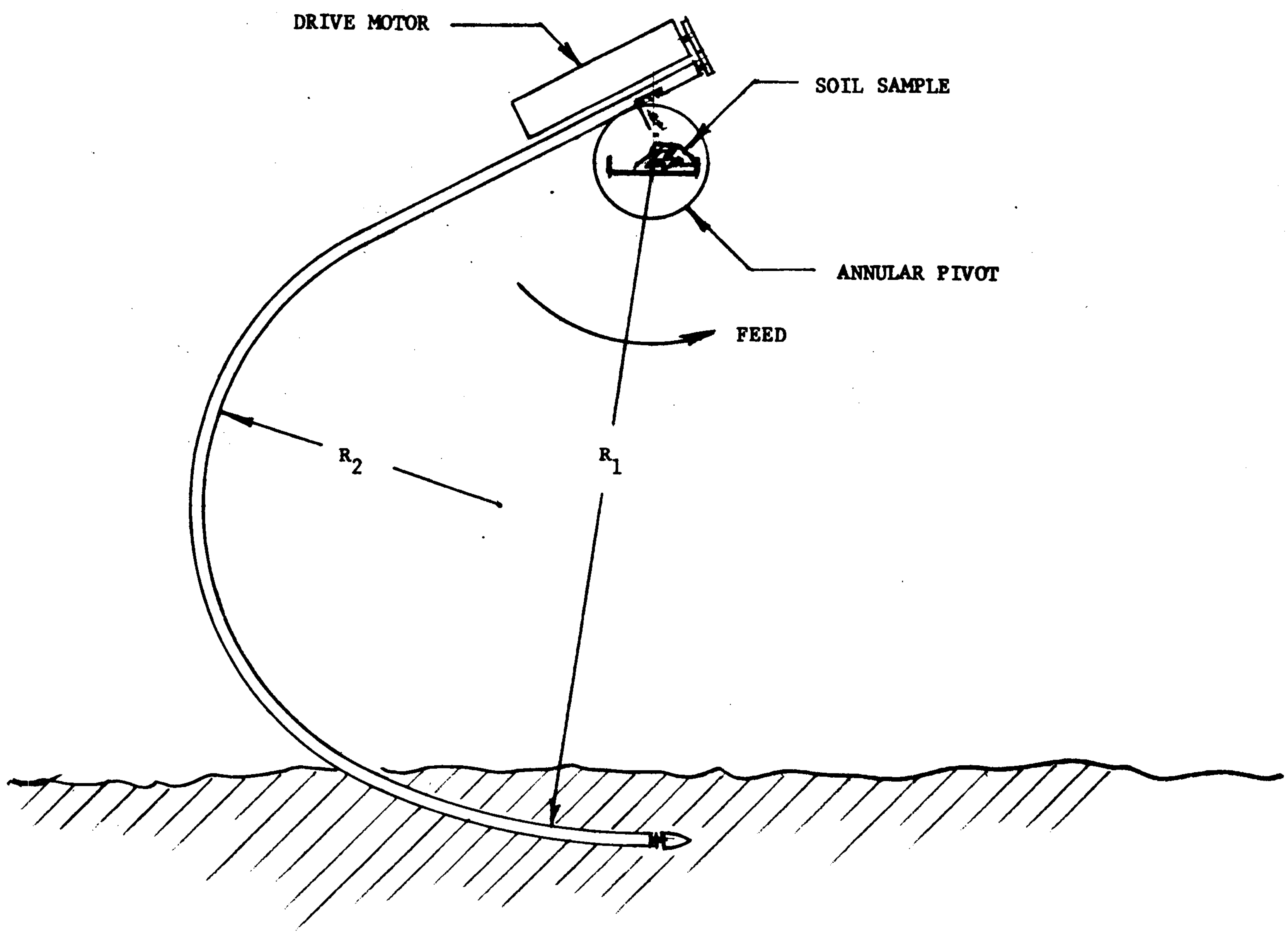


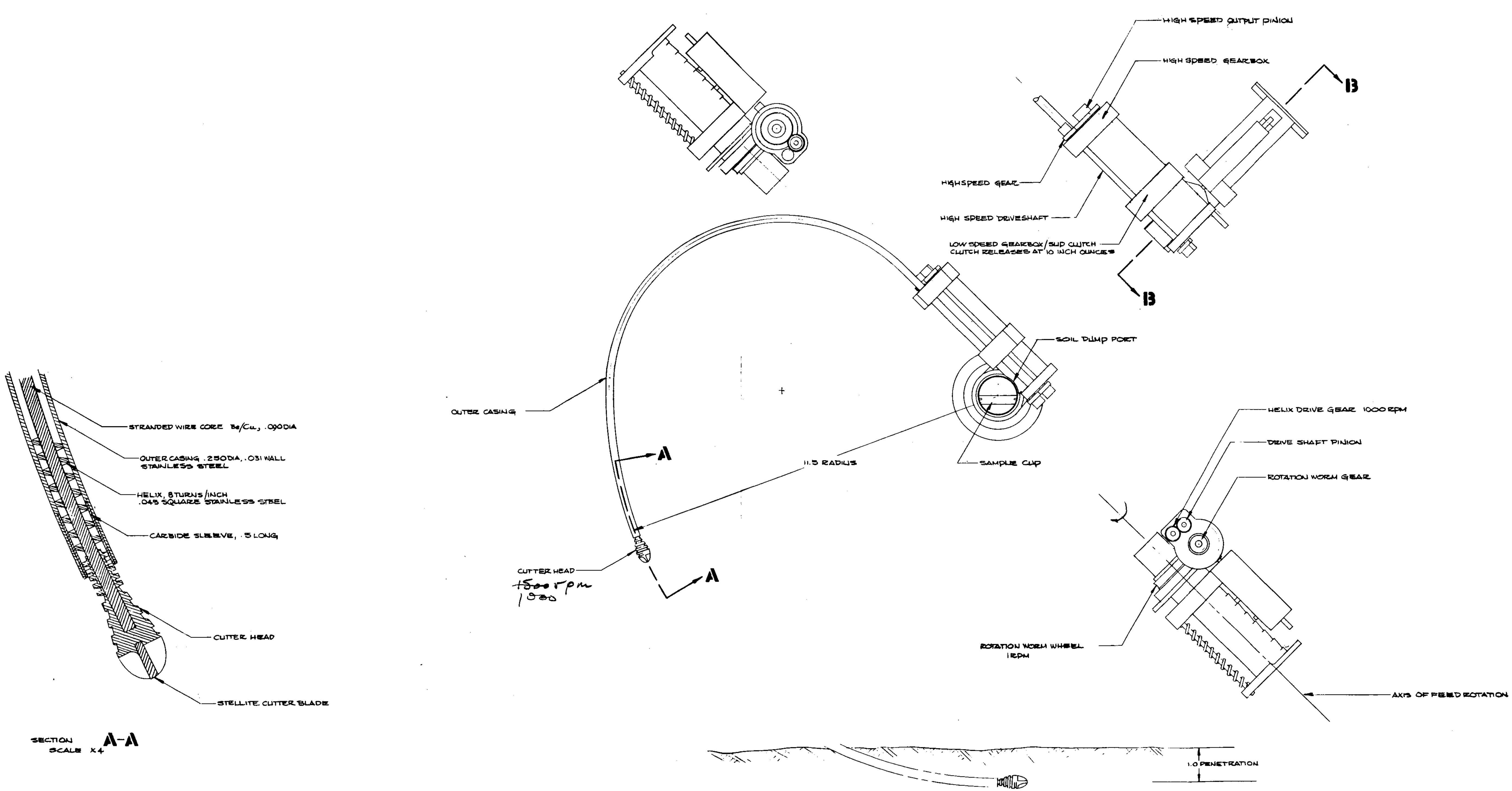
FIGURE 21. SINGLE BEND HELICAL CONVEYOR SIMPLE PARTICULATE SAMPLER

inch of the entrance into the housing. Also, the first few turns of the helix are made of this material. This detail is shown in Section A-A of Figure 22. These first few turns are machined as an integral part of the abrading tip and have a lower pitch than the main helix of the conveyor. This is done to limit the intake of sample to prevent overloading the helical conveyor which can cause it to stall or break. The JPL breadboard as presented in SPS 37-46, Volume IV did not incorporate a tip cutter. It relied on an exposed length of helical conveyor to abrade the soil. This is not very effective in cemented material. The helical conveyor tested in the field earlier in this program had a tip which was simply a wide flat blade brazed to the shaft of the conveyor. While this tip cutter improved the penetration characteristics for this sampler, it proved to be susceptible to wedging between rocks and on two occasions was sheared off the shaft. The tip cutter design used in the prototype, shown in Figure 22, was therefore made as an elliptical surface of revolution at the nose followed by a tapered section reducing to the helical conveyor diameter where it attaches to the conveyor shaft. Shallow cutting edges are ground into this elliptical surface so that the tendency to wedge between rocks is reduced since the bite of each cutting edge is small. A helix was incorporated on the tapered afterbody to assist in feeding abraded material towards the entrance to the conveyor. The maximum diameter of this cutter is slightly larger than the outside diameter of the conveyor housing. This provides a clearance between the outside of the housing and the inside of the hole when sampling in cemented material.

The main body of the helical conveyor was the same construction that was used in the breadboard models. The shaft is a wound stranded wire cable similar to a speedometer cable. A helix made of square stainless steel wire is wound over this cable and brazed to it.

The helical conveyor terminates at a support housing which is free to rotate about an annular support shaft. This support shaft is cut away at the top to allow the sample to fall out of the open end of the helical conveyor housing into a sample cup mounted inside the shaft.

Other features, which were not incorporated in the breadboard models, are a power driven feed and an arrangement whereby the sampler mechanism can be stepped over a given amount after each sampling cycle so that another sampling attempt can be made in a slightly different location on each subsequent run. A single motor with a dual output is used to drive both the helical conveyor and the rotary feed. It is assumed that a motor can be made or modified with a gearbox on each end as shown in Figure 22. A block diagram for the power train is shown in Figure 23. The gear ratios shown for the speed reducers correspond to the reduction ratio for type LL Globe gearmotors. The motor and helical conveyor combination is equivalent to Globe gearmotor 5A537-4. The motor and rotary feed drive is equivalent to Globe gearmotor 5A563-4.



FOLDOUT FRAME

FIGURE 22. HELICAL CONVEYOR SIMPLE PARTICULATE SAMPLER, E-4

FOLDOUT FRAME 2

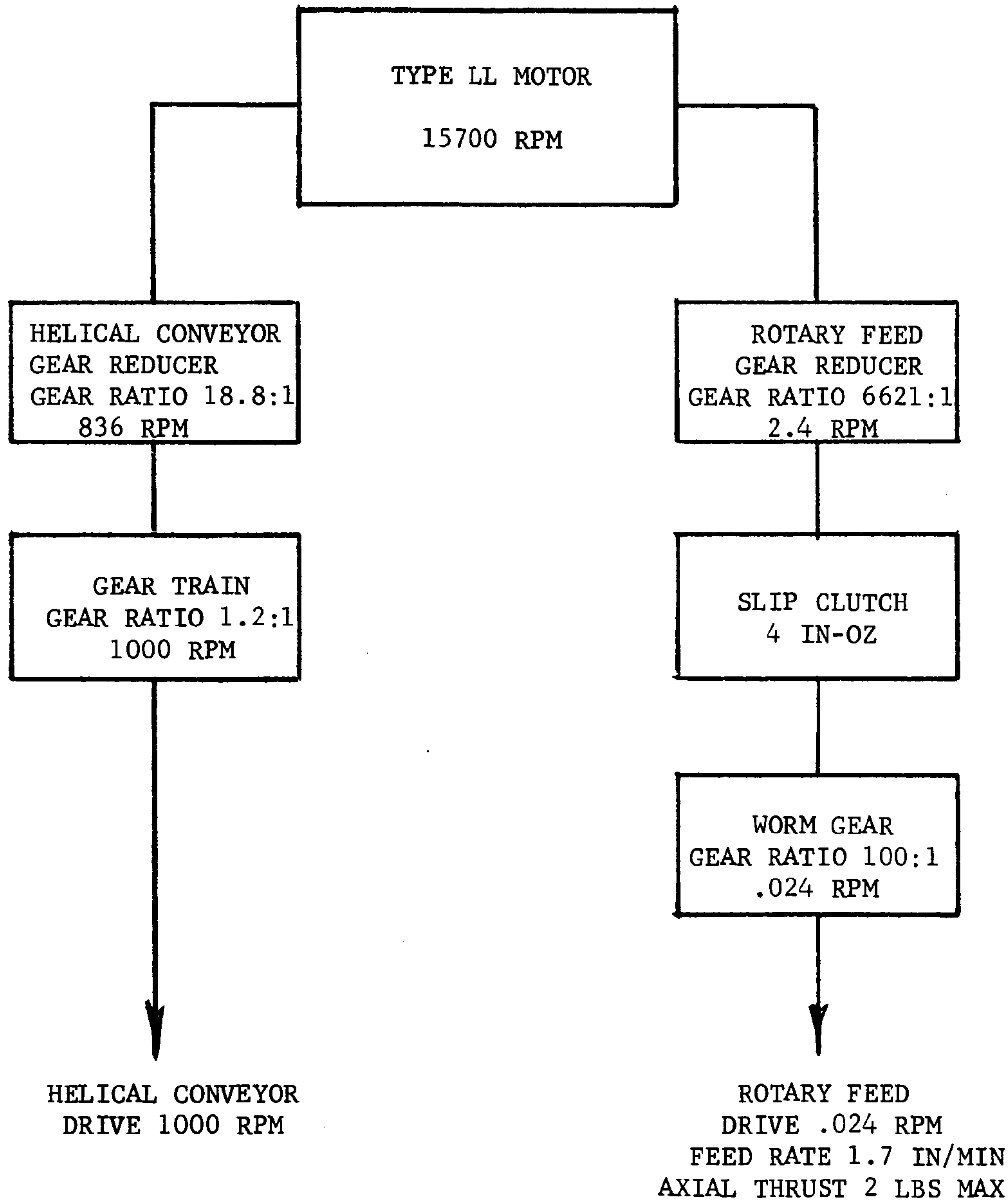


FIGURE 23. BLOCK DIAGRAM HELICAL CONVEYOR SIMPLE PARTICULATE SAMPLER POWER TRAIN

Based on results obtained with the breadboard model run by a type SS Globe motor, a larger motor was used in this design. The breadboard model would stall occasionally due to the helical conveyor loading up. The -4 armature used in this design has a rated torque of one inch-ounce compared to .45 inch-ounces for the type SS motor which is approximately twice the rated power. The breakaway torque output is 7.8 inch-ounces, almost three times as much as the best SS motor. Thus, this larger motor should provide adequate power to run both the helical conveyor and the rotary feed. The power input required for the rotary feed is approximately .2 percent of the rated output of the motor. This takes into account the gear train efficiency of 26 percent.

The worm drive is connected to the output of the rotary feed gear reducer with a slip clutch set to slip between 2 and 4 inch-ounces. This torque applied to the worm will produce an axial thrust at the abrading tip of the helical conveyor of 1 to 2 pounds. The helical conveyor housing has an outside diameter of .250 inches with a wall thickness of .031. The axial thrust applied at the tip produces a bending stress of 28,000 psi in the housing where it attaches to the support structure. If half hard stainless steel tubing is used this will provide a factor of safety of 2 based on the yield point of the material.

Figure 24 shows a sectional view taken through the escapement mechanism used to step the sampler over .375 inches after each sampling cycle. The support shaft is machined with a series of sloping steps or ramps. These are engaged by one of two dogs or pins which prevents the tension indexing spring from pulling the sampler mechanism along the support shaft. These dogs are cam activated by a cam mounted on the rotating part of the sampler. Thus, as the sampler is deployed, the cam moves off of the pin attached to the cam positioning slider. When this happens the indexing return spring pulls the cam positioning slider to the left which in turn lifts the dog engaged with the stepped shaft and lowers the other. This allows the tension indexing spring to pull the sampler assembly over until the leading dog engages the next step. This occurs at the beginning of each sampling cycle. A total of 3 to 4 lateral positions can be achieved in this way.

An alternate design which eliminates the double ended gear reducer and escapement stepping mechanism is shown in Figure 25. In this design, the rotating support structure of the sampler is mounted on a sliding support keyed to the basic support shaft. This sliding support is free to move laterally along the support shaft but is not free to rotate. At the end of this sliding support is fixed a worm gear which engages with the feed drive worm. Thus, as the feed drive worm rotates it walks along the worm gear causing the sampler structure to rotate about the axis of the shaft. The drive worm is not keyed to the shaft driving it but is connected to it through a slip clutch. A Belleville spring is used to maintain the normal force necessary to provide the friction force required to drive

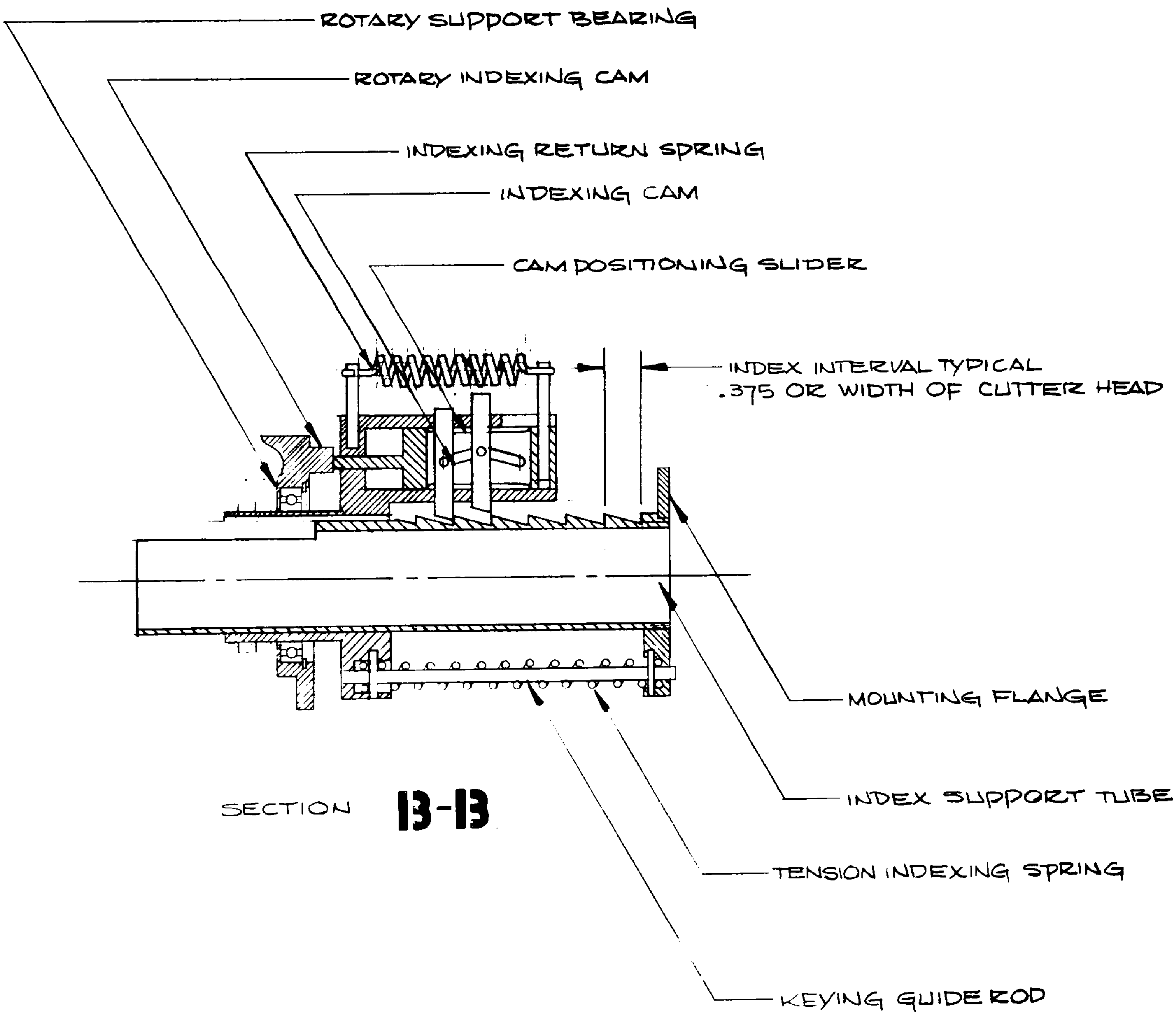


FIGURE 24. SECTIONAL VIEW THROUGH ESCAPEMENT MECHANISM

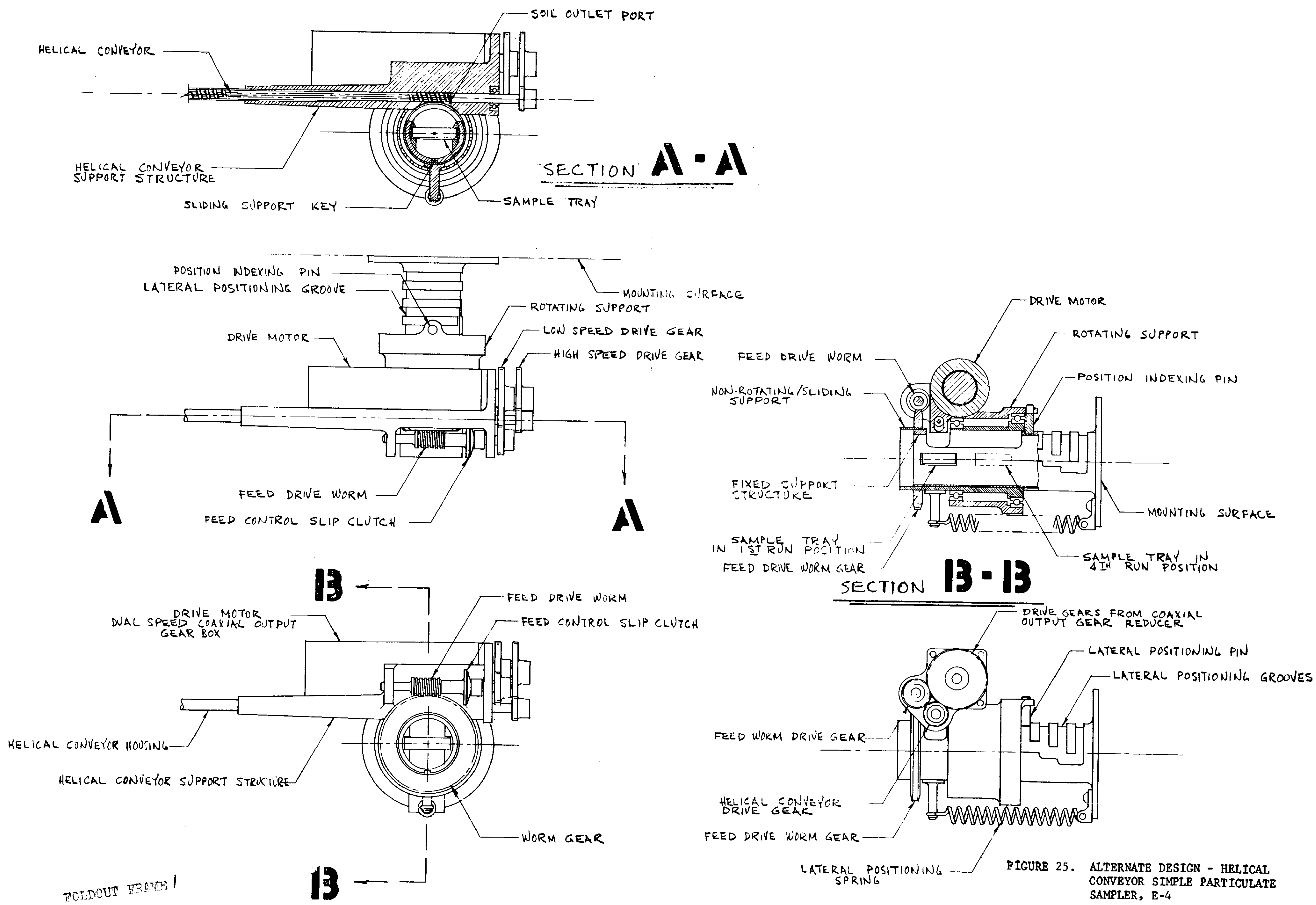


FIGURE 25. ALTERNATE DESIGN - HELICAL CONVEYOR SIMPLE PARTICULATE SAMPLER, E-4

FOLDOUT FRAME 1

FOLDOUT FRAME 2

the worm. Slip clutches of this design were used to actuate valves in the rotating wire brush sampler tested earlier in this program. No failures of these clutches in actuating the valves were noted even though the valves were exposed to a dirty environment.

Section B-B, of Figure 25, is a section through the vertical centerline of the support structure. It can be seen in this view that the primary support shaft has a series of circumferential grooves machined in it. An indexing pin mounted on the rotating sampler structure runs in one of these grooves to maintain a given lateral position. In order to move the sampler over laterally to a new sampling position, it is merely necessary to retract the sampler past its start position a few degrees. When this is done, the indexing pin runs past the end of the groove allowing the lateral tensioning spring to pull the sliding support over until the wall of the next groove is engaged by the pin. Each step moves the sampler laterally .375 inches.

It should be pointed out at this point that the mechanism required to step the sample tray into position was not considered to be part of this design; however, some means of indexing the tray position to correspond to the particular lateral position occupied by the sampler is required. This can be easily accomplished with this design by mounting a pin or projection on the keyed sliding support against which the sample tray can be positioned. Thus, as the sampler steps over to a new lateral position, the sample tray stop is also automatically moved the corresponding amount so that the tray will be in the correct position relative to the sample delivery port.

This design also utilized another approach in the power drive train. A coaxial output planetary gear reduction is used with the drive motor. The high speed output is used to drive the helical conveyor while the low speed output drives the rotary feed mechanism. Such a coaxial drive was fabricated and used successfully on the vertically deployed conical sieve sampler developed by Philco-Ford and tested earlier in this program. This appears to be a more compact design than the design shown in Figure 22 with a gear reducer at each end of the motor and should also save weight in the gear reducers by elimination of one of these.

Section A-A, of Figure 25, shows the relative geometry through the centerline of the helical conveyor. The relative positions shown are those existing when the sampler head makes initial contact with a flat surface normal to the vertical centerline of the mechanism. It is seen that the soil outlet port is positioned over the right side of the sample tray as shown in section A-A. As sampling progresses, the outlet port sweeps across the tray and will wind up in the same position on the left side of the tray when the sampling head emerges from the surface being sampled.

If cemented material is being sampled and the sampler ceases to progress when the 6 inch limit, defined by the radius R_1, is reached, the sample delivery port will be positioned over the center of the sample tray.

The feed rate was established to produce a linear velocity of the abrading head of 2 inches per minute. This is the same rate used on the breadboard version of this sampler tested in the field. If the sampler is supported over a level surface as shown in Figure 22, a maximum rotation about its axis of 120 degrees is required. At this rotation rate, 14 minutes are required to traverse this segment of arc of which approximately half is spent reaching the surface. Thus, the maximum time to complete a cycle is 28 minutes. The operational sequence for this sampler is given in Table XIV.

TABLE XIV

HELICAL CONVEYOR SIMPLE PARTICULATE SAMPLER OPERATIONAL SEQUENCE

1. Activate the drive motor. The sampler rotates about its axis requiring about 7 minutes to reach the surface.

2. The sampler penetrates the surface for another 7 minutes. If the sampler ceases to feed because of an obstruction, it runs in that position until the time to complete a maximum sweep with no impedance is completed.

3. At the end of 14 minutes the drive motor reverses to return the sampler to its stowed position.

4. When the stowed position at the start of the run is reached and over-run by 7 degrees, the sampler assembly steps into the next lateral position for the next run.

The weight statement for this sampler is given in Table XV. The weights as calculated in this table are for the configuration shown in Figure 22. The weights should be substantially the same except that the weight of the high speed gearbox can be eliminated reducing the total sampler weight to 1.5 pounds for the alternate design. This is a realistic reduction in weight since the high speed output is taken off on one of the early reduction stages in the planetary gear reducer needed to provide the low speed output.

TABLE XV

WEIGHT SAMPLE, SIMPLE PARTICULATE HELICAL CONVEYOR SAMPLER, E-4

ITEM NO.	ITEM	MTL	QTY	WEIGHT /PART	WEIGHT /ASSY	TOTAL
	Motor Drive Assy					
1	Motor	-	1	.406	.406	
2	High Speed Gearbox	-	1	.427	.427	
3	Low Speed Gearbox	-	1	.427	.427	
4	High Speed Pinion	Al	1	.011	.011	
5	High Speed Gear	STL	1	.015	.015	
6	High Speed Driveshaft	Al	1	.014	.015	
7	Bearing	STL	2	.0016	.0032	
8	Drive Shaft Pinion	Al	1	.0043	.0043	
	Subtotal		9			1.307
	Rotation Assy					
9	Worm Gear	STL	1	.041	.041	
10	Worm Wheel	STL	1	.039	.039	
11	Rotary Bearing	STL	1	.097	.097	
	Subtotal		3			.177
	Indexing Assy					
12	Indexing Cage	Mg	1	.133	.133	
13	Mounting Flange	Mg	1	.028	.028	
14	Index Support Tube	STL	1	.011	.011	
15	Indexing Spring	STL	1	.006	.006	
16	Guide Rod	Al	1	.0038	.0038	
17	Cam Slider	Al	1	.022	.022	
18	Indexing Cam	STL	2	.0066	.013	
19	Return Spring	STL	1		.003	
	Subtotal		9			.2198
	Helical Conveyor Assy					
20	Outer Casing	STL	1	.099	.099	
21	Core Wire	Be/Cu	1	.0083	.0083	
22	Helix	STL	1	.0928	.0928	
23	Carbide Sleeve	STL	1	.0007	.0007	
24	Helix Drive Gear	Al	1	.0043	.0043	
25	Cutter Head	STL	1	.014	.014	
26	Cutter Blades	STL	3	.0007	.0021	
	Subtotal		9			.2212
	Total Sampler Assy		30			1.93

3.5 BACKHOE SAMPLER, E-5

3.5.1 SCOOP CONFIGURATION

This sampler utilizes a scoop mounted on the end of an extendible boom which can be drawn across the surface to collect a sample. Repeated operation in the same location will allow trenching to some depth below the surface consistent with the strength of the soil surface being sampled. Design criterion number 2, which limits the maximum particle size to 5 millimeters, and criterion number 7, which provides that no rocks shall prevent complete closure of the scoop, are felt to be fundamental to the configuration and design of the scoop. A concept was generated in which a thin blade is mounted on the boom ahead of the scoop as shown in Figure 26.

The function of this blade is to minimize hang ups on large rocks by causing the scoop to ride up and over and to start intermediate size rocks moving to one side or the other so that they will tend to flow around the scoop. The scoop itself is configured so that the width at the inlet is compatible with the maximum size particle that is acceptable, i.e., 5 to 6 millimeters between the edge of the scoop and the divider blade. The cutting edge of the scoop is positioned so that it shears the soil at an angle of 30 degrees from the horizontal. The internal configuration of the scoop is such that the soil entering it rests on an essentially horizontal surface. Thus, if the scoop is lifted off the surface before it is closed, the soil sample will not all fall out.

In order to determine the positioning requirements for the scoop, the geometry of the boom relative to the surface being sampled must be considered. This geometry is shown schematically in Figure 27. The length of the boom, ℓ, the height of the boom support, h, and the angle between the boom and the surface, α, are related to each other by the expression

$$\sin \alpha = \frac{h}{\ell} \sin \beta.$$

It is assumed that the surface can lie anywhere between a 30 degree upslope and a 30 degree downslope from the boom support axis, the height of the support point varies from 12 to 48 inches, and that the maximum down angle of the boom is 60 degrees below the vertical.

The curves in Figures 28 and 29 show that the angle at the tip of the boom, between the axis of the boom and the surface, varies in essentially the same manner and magnitude regardless of the slope of the surface. This is true between a 30 degree upslope and a 30 degree downslope. It also indicates that the angle increases steadily as the boom is retracted to a maximum value of 60 degrees on a level surface and 90 degrees on a 30 degree downslope when the boom is in its most depressed condition. Figure 30 indicates the minimum length of boom required to reach the

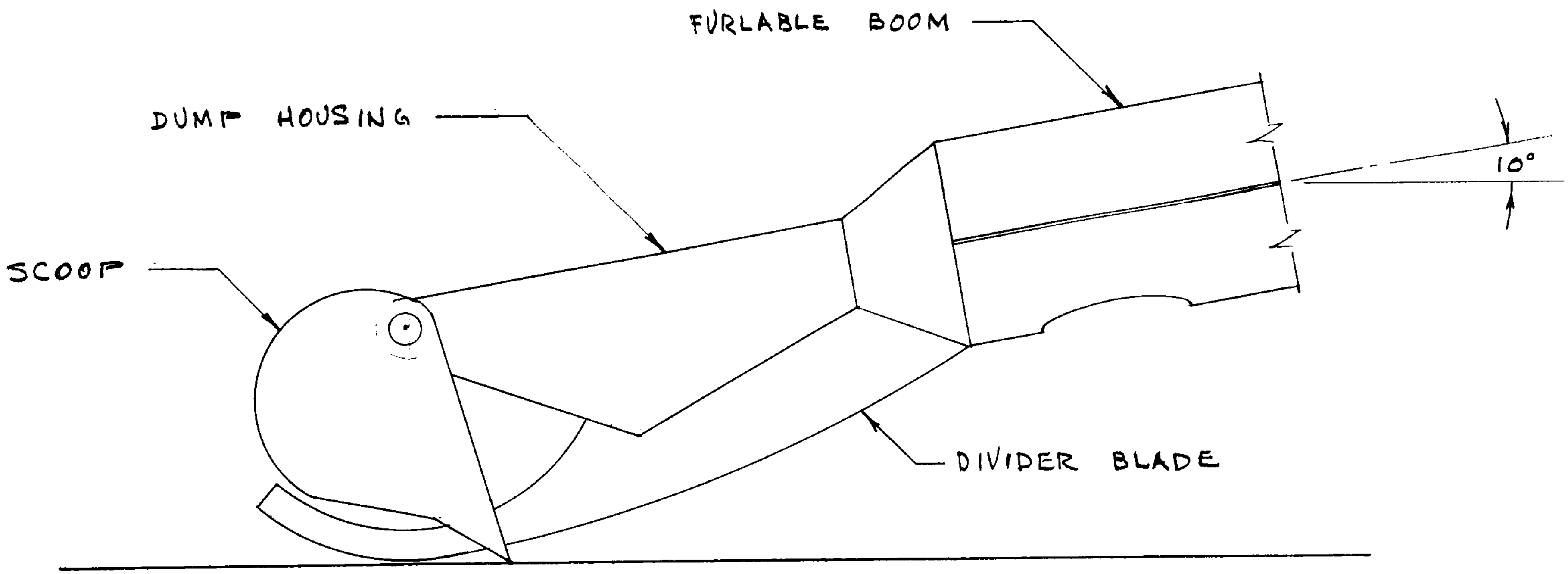


FIGURE 26. BACKHOE SCOOP CONCEPT

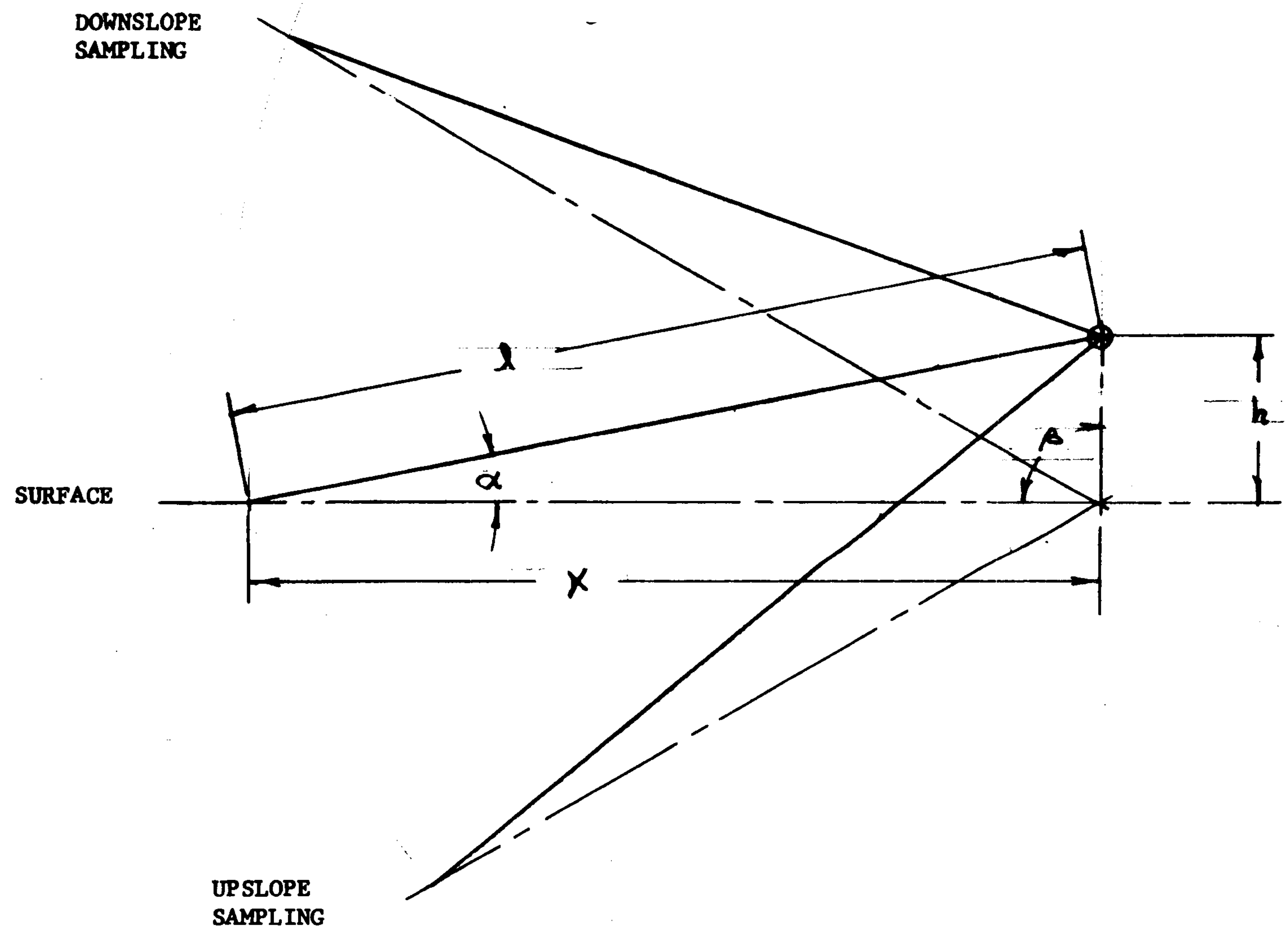


FIGURE 27. BOOM DEPLOYED BACKHOE GEOMETRY

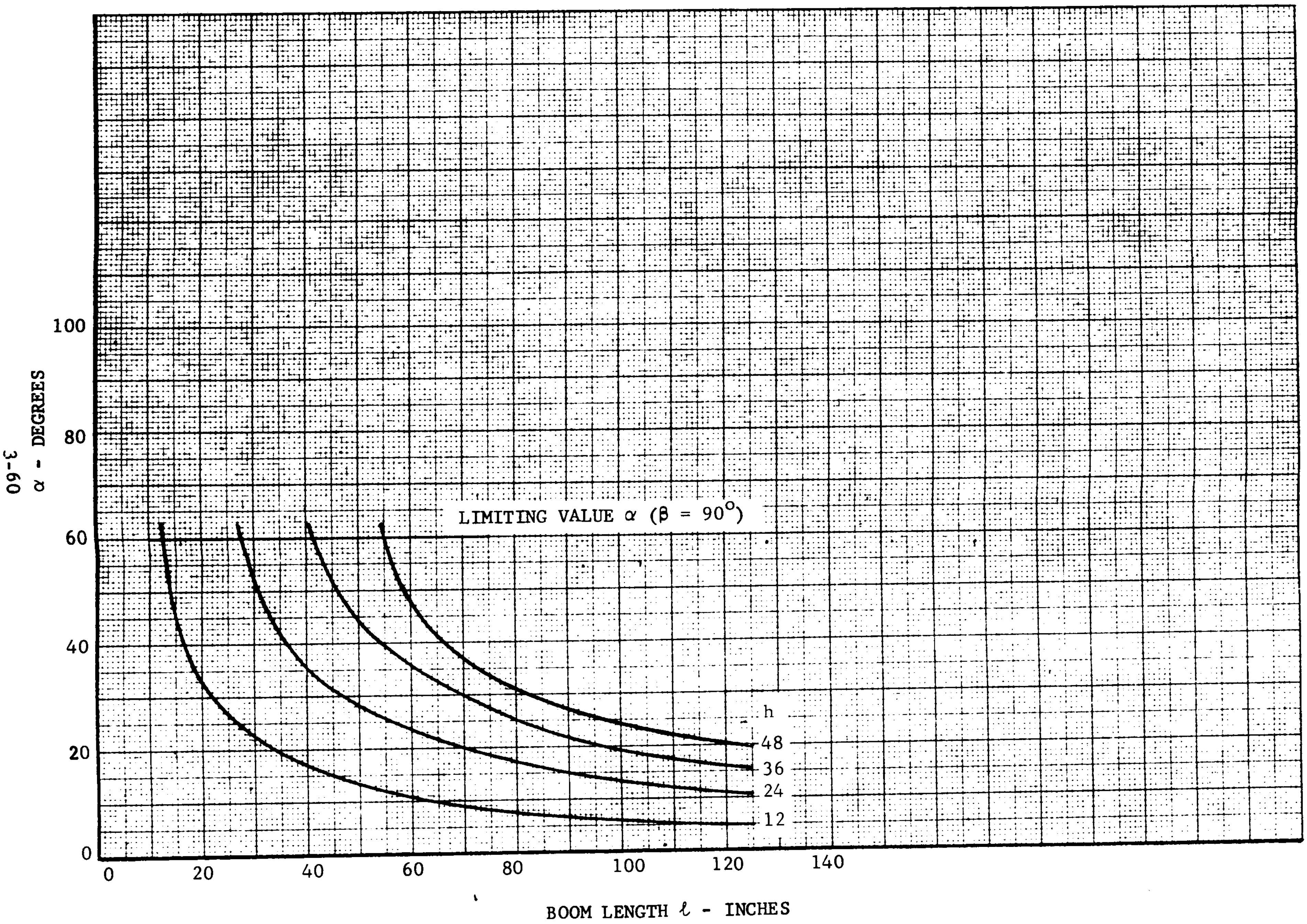


FIGURE 28. BOOM TIP ANGLE VARIATION WITH LENGTH ON A LEVEL SURFACE

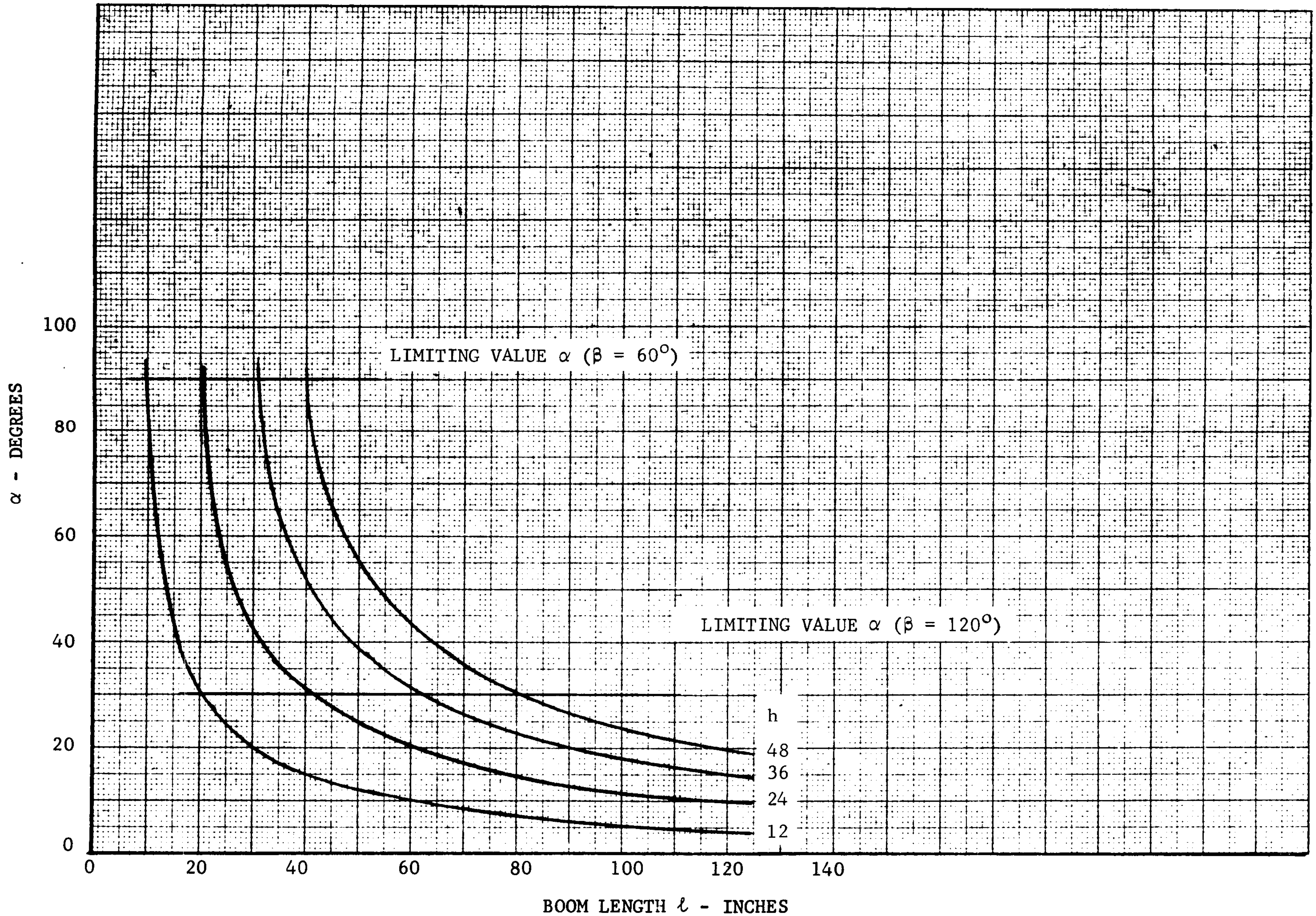


FIGURE 29. BOOM TIP ANGLE VARIATION ON A SLOPED SURFACE FOR β = 60° AND 120°

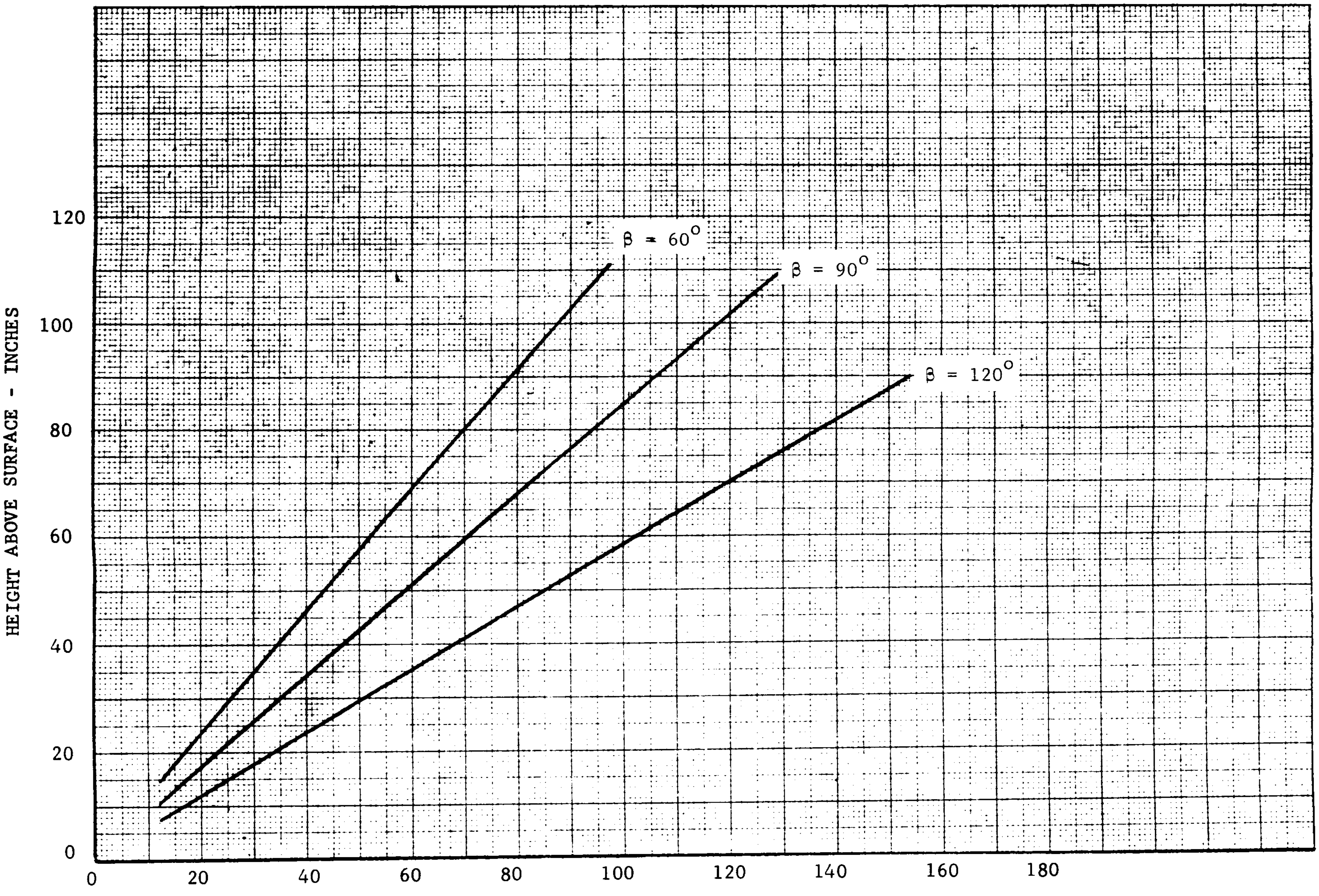


FIGURE 30. MINIMUM LENGTH OF BOOM TO REACH SURFACE AT LIMITS OF DOWN DEFLECTION OF BOOM

surface as a function of support height, h. Thus, a five foot long boom must be mounted less than 3 feet above the surface in order to collect a sample. A boom length of 10 feet was assumed initially for this sampler.

Based on the large variation in the angle α, it is desirable that the scoop is always positioned in the same way with respect to the local vertical regardless of the position of the boom. This implies some sort of positional feedback and servo to continuously adjust the scoop relative to the boom as the sampling traverse progresses. Thus, the cutting edge of the blade attacks a level surface at a 30 degree angle. As sampling progresses the soil collects in the scoop.

After collecting the sample the scoop is closed so that the sample can be transported to the payload. Two methods can be used to perform the sample transfer. One is to simply position the scoop over a dump port or funnel in the payload and then open the scoop dumping the sample. The other is to utilize the boom as a transport path by raising it to a vertical position and allow the sample to fall down the boom by the action of gravity. The latter approach was used in the design of the booms which will be described subsequently.

Two approaches for actuating the scoop were considered. The simplest to mechanize is to mount a motor at the tip of the boom as shown in Figure 31. This scheme requires positional feedback for the boom in order to servo drive the scoop to the correct attitude. With this approach powerful actuation forces are achieved through the worm gear drive. A disadvantage is that the weight of the motor is placed on the tip of the boom which increases the reaction moment required to raise the boom. For very light weight furlable booms, the increase in bending moment can be critical in terms of the strength of the boom.

The alternate approach considered was a cable system which provides mechanical feedback and actuation forces. This configuration is shown in Figure 32. This approach uses parallel cables fixed to a drum on the scoop. The other ends are wrapped around another set of drums at the base of the boom. As the boom extends or retracts, these drums rotate in opposite directions to feed out or take up the appropriate amount of cable as dictated by the length of the boom. As the boom is elevated or depressed, these cables act in a manner similar to the bands on a drafting machine to maintain the scoop in the same orientation regardless of boom attitude. To open or close the scoop, the drums at the base of the boom are rotated in unison in the appropriate direction. While this system is somewhat more cumbersome, it places less weight on the end of the boom and automatically maintains the scoop in the proper position.

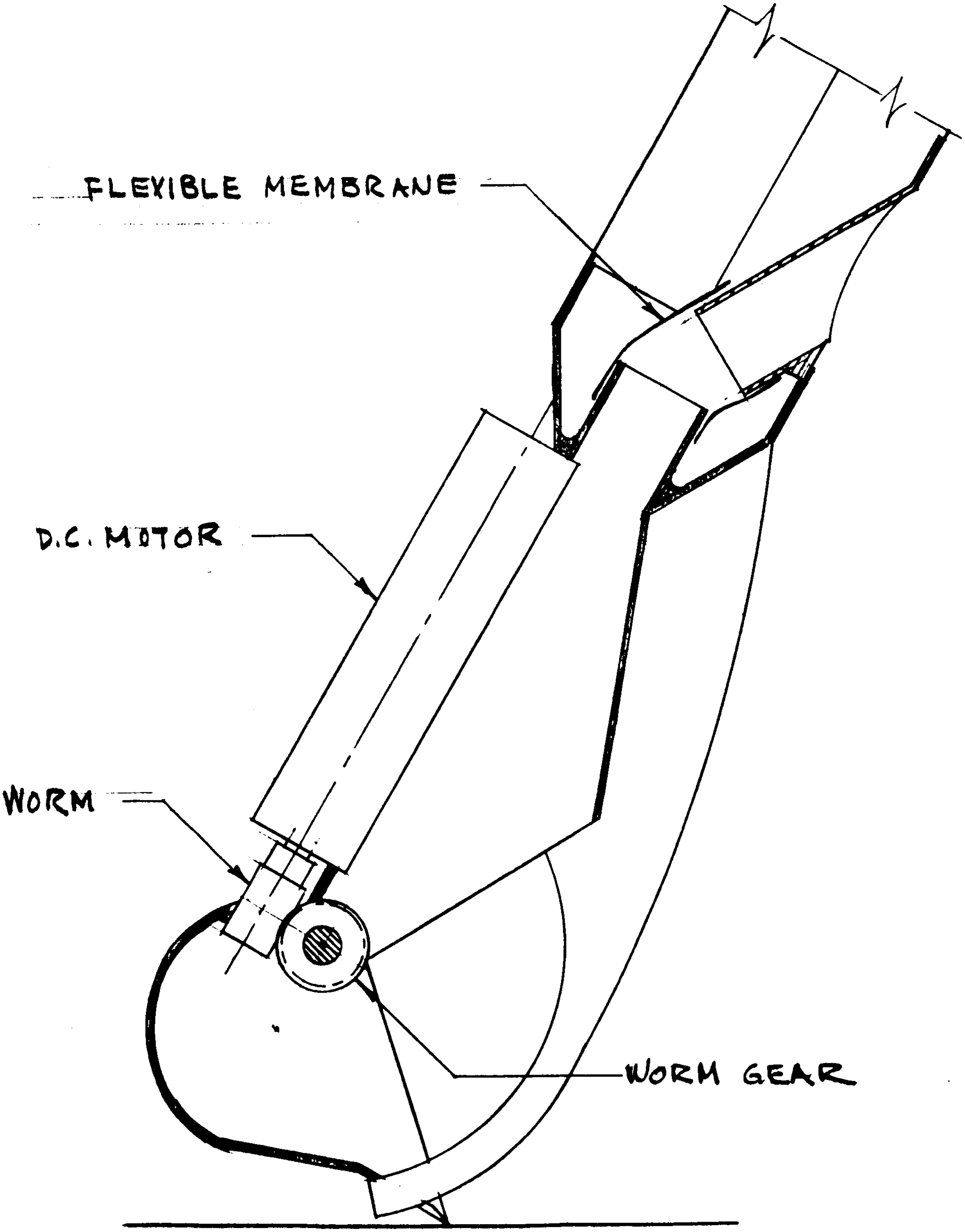


FIGURE 31. BACKHOE SCOOP - MOTOR ACTUATED

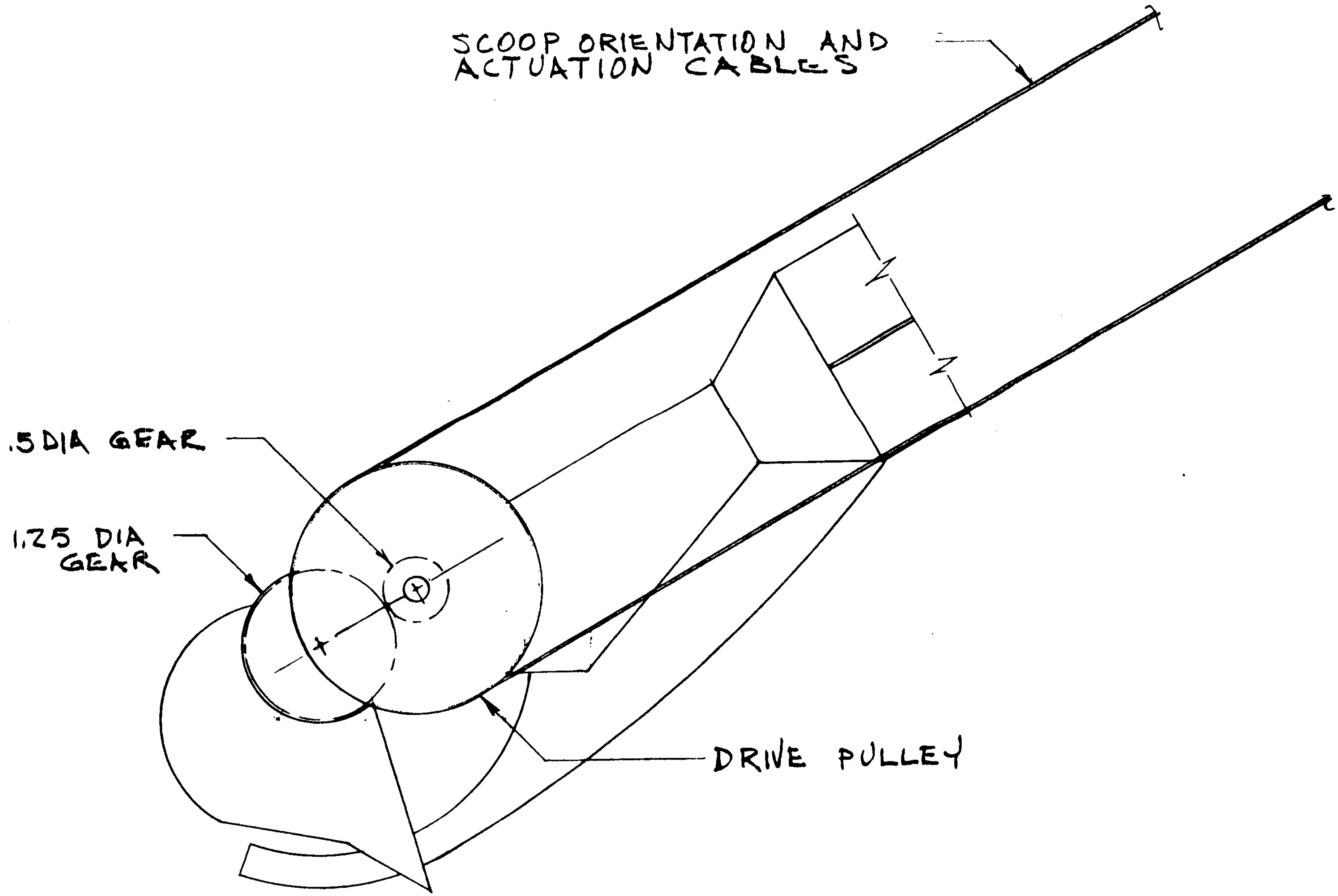


FIGURE 32. BACKHOE SCOOP CABLE ACTUATION SYSTEM

3.5.2 FURLABLE BOOM

An initial design approach was oriented towards using the gravity dump mode with the Ryan furlable boom. This boom has a closed cross-sectional shape which gives the boom more structural integrity than the DeHavilland boom (particularly in torsional rigidity) although not as much as can be achieved with a telescoping boom. Thus, an exit port for the sample must be incorporated at some point in the furlable boom that does not degrade its strength. Two possible locations are available. One near the tip of the boom in the section which is not flattened after complete retraction and the other at the base of the boom when it is fully deployed. This latter location for the exit port must be behind the boom support guides in the transition section. The disadvantages of locating the exit port at this point is that the opening in the tube must be compatible with the flexing involved in retracting the boom. Also, the interior of the boom is exposed to soil particles throughout its entire length. Any residual material would seriously degrade the life of the boom and could result in local failure of the furlable boom. Another possible disadvantage is that the boom must be fully extended during the gravity dump cycle. All these disadvantages can be circumvented if the exit port is located near the tip where a minimum of flexing occurs and the bending stresses are minimized. Also, since no flexure occurs for this portion of the boom, an auxiliary small diameter tube can be mounted inside the furlable boom to act as a soil transport chute thus eliminating the chance that soil particles will be inside the boom when it is retracted.

The furlable boom developed by Ryan Aeronautical Company consists of two preformed thin titanium alloy sheets resistance welded together at the edges which forms a cross-sectional shape as shown in Figure 33.

This boom is stowed by elastically flattening the tube and winding it onto a drum. The first consideration in using this boom to transport soil to the payload was to provide an opening in one of the tubes near the base when the boom was fully extended. When the boom is erected to the vertical the soil falls down the inside of the tube and out the opening at the base into a receptacle. The disadvantages to this approach are as follows:

(a) Sample transport to the payload can only occur when the boom is fully extended.

(b) There is no assurance that the internal cleanliness of the boom can be maintained. Soil particles inside the boom would adversely affect the flexure characteristics during stowage on the take-up drum.

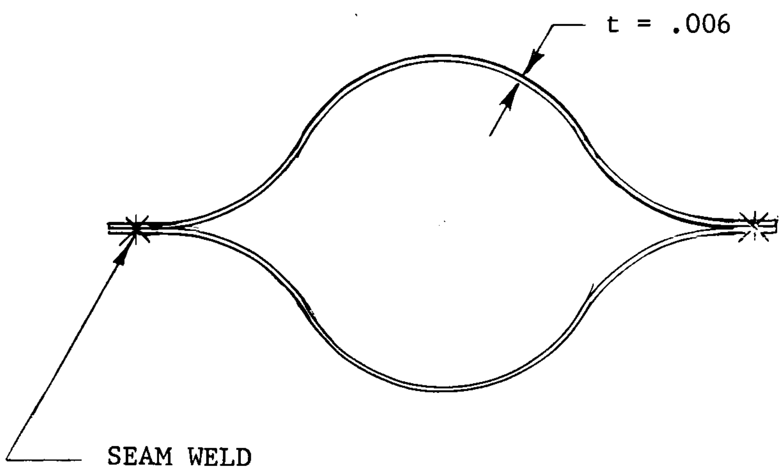


FIGURE 33. RYAN FURLABLE BOOM CROSS-SECTION

(c) The interface between the receiving receptacle and the exit hole in the boom is unduly complex.

Another approach to soil delivery with this boom was then considered and is shown schematically in Figure 34. In this approach the boom must be fully retracted to deliver a soil sample which at least eliminates extending the boom again after making a sampling attempt. The opening is now located near the tip of the boom where bending stresses are low and elastic deformation during stowage is small. Thus, a short fixed tube can be incorporated between the backhoe scoop and the opening in the side of the boom. This soil transport tube can be sealed at the outlet in the side of the boom tube thereby preventing soil particles from gaining entry into the interior of the boom. When the boom is fully retracted, this opening is aligned with a corresponding one located in the boom guide shoe. This opening is the end of a tube leading to a channel shaped passage fitted around the boom take-up drum. The fourth side of this passage is closed with two flexible tapes. The take-up spools for these tapes are mounted on either side of the sample delivery chute. These take-up spools are geared together so that as the boom is erected to a vertical position, tape is deployed from one and taken up on the other. This in effect provides an opening in the sample delivery passage that remains fixed with respect to the sample delivery chute

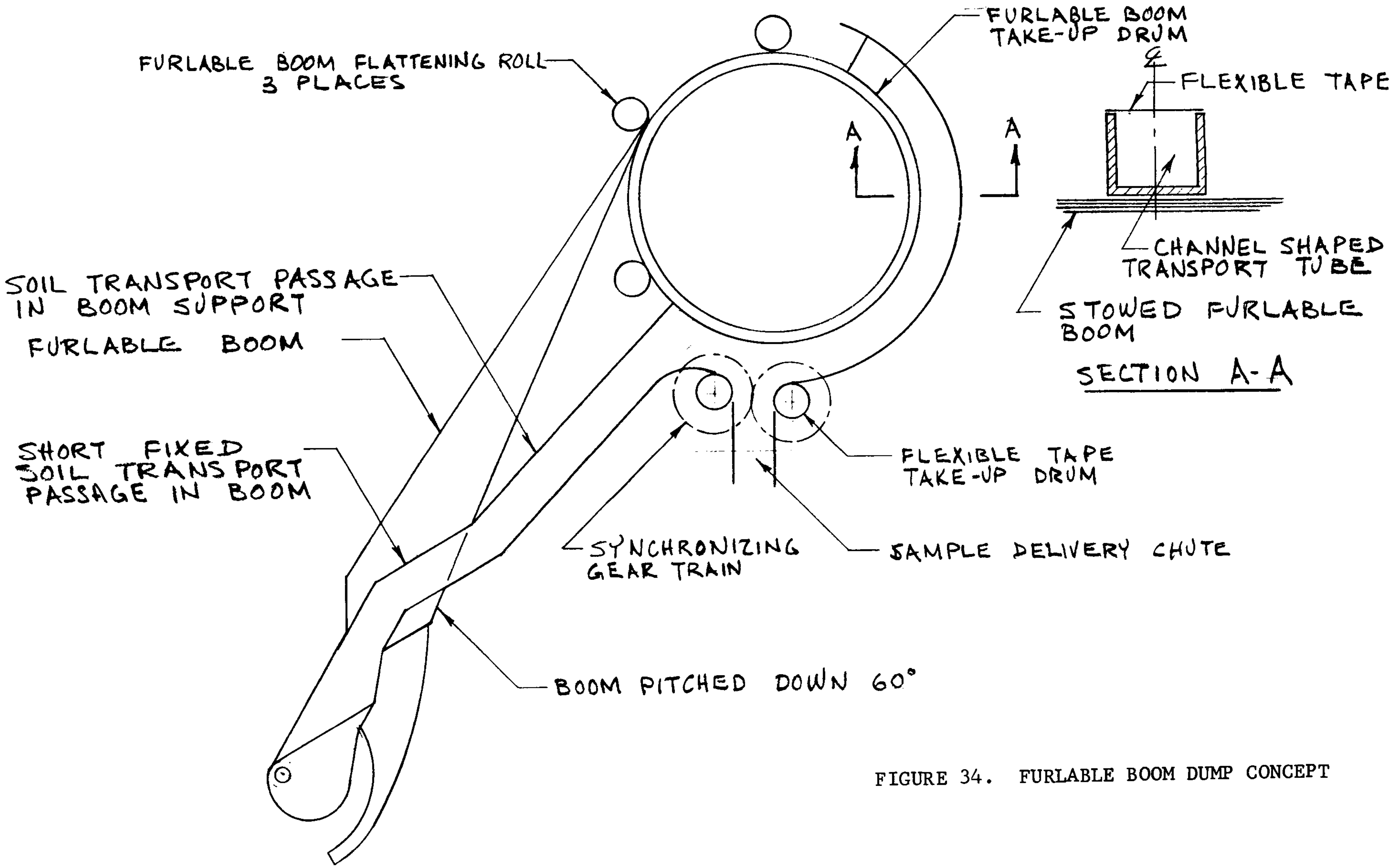


FIGURE 34. FURLABLE BOOM DUMP CONCEPT

regardless of the orientation of the boom. Any soil remaining in the channel shaped passage after the soil is dumped is progressively dumped into the sample delivery chute as the boom is lowered from the vertical. Wipers can be installed at the take-up spools to ensure that no particles remain adhering to the tape.

In attempting to mechanize this approach, primary attention was given to compactness, since the flattened boom is rather wide and requires a fairly large diameter take-up drum. The demonstration model of this boom fabricated by Ryan Aeronautical Company had the physical characteristics listed in Table XVI.

TABLE XVI

PHYSICAL CHARACTERISTICS OF RYAN FURLABLE BOOM

Characteristic	Description
Boom material	6Al 4V (Titanium Alloy)
Flattened boom width	4.5 inches
Wall thickness of boom	.006 inches
Drum diameter	7.0 inches
Boom length	120 inches

This boom was built to the specification that it be capable of supporting a one pound weight at the tip when fully extended and that it be capable of applying a downward force of two pounds under the same conditions. Thus, the cable scoop actuation system was incorporated in this approach in the interest of minimizing the boom tip weight.

It appeared that the diameter of the take-up drum might conceivably be reduced, improving the volumetric efficiency of the design. The analysis for allowable flattening stresses, presented in Section 2.3.1 of Philco-Ford Report Number UG-3962, was used to check the required take-up drum diameter. The material properties for several materials are compared in Table XVII.

TABLE XVII

MATERIAL PROPERTIES COMPARISON

MATERIAL	PROPERTY	VALUE
6Al 4V Titanium Alloy	Yield point Young's Modulus E/σ	120,000 psi 16×10^6 psi 133
PH 15-7Mo Steel	Yield point Young's Modulus E/σ	145,000 psi 29×10^6 psi 200
Fiberglass	Yield point Young's Modulus E/σ	50,000 psi 5×10^6 psi 100
Beryllium/Copper	Yield point Young's Modulus E/σ	120,000 psi 19×10^6 psi 158

The lower the value of E/σ, the more deformation that can be tolerated without yielding the material. It is seen that the titanium alloy falls between fiberglass and beryllium copper making it a good choice for the boom material when considering its relatively lower density when compared to Be/Cu or steel. An allowable ratio of tube diameter to wall thickness of 150 is reasonable for this material. Since this boom consists of two sheets welded at the edges, the effective wall thickness in terms of flexing is twice the thickness of the individual sheets. This results in a nominal tube diameter of $d = (.012)(150) = 1.8$ inches which is very near that used in the Ryan boom. Since this boom has curvatures of the tape in both directions, it must be considered in the category of a backward wound tape which is more severe in stressing the material. For this type of winding, the stress falls rapidly with increased drum diameter. A good drum to tube diameter has a ratio of 3:1. Larger drum diameters do not appreciably reduce the stowage stresses. This results in a drum diameter of $D = (1.8)(3) = 5.4$ inches. Since the analysis used is conservative, a drum diameter of 5 inches was assumed for the initial design.

In order to integrate the cable actuation system with the drum design, it was apparent that it would be desirable to make the cable take-up drum diameter nominally the same as the boom take-up drum. Since a small pulley or drive drum at the tip of the boom is desirable, a one-to-one ratio did not appear feasible. The geometry relating the drive drum response to the boom attitude for drums of different diameters is shown in Figure 35.

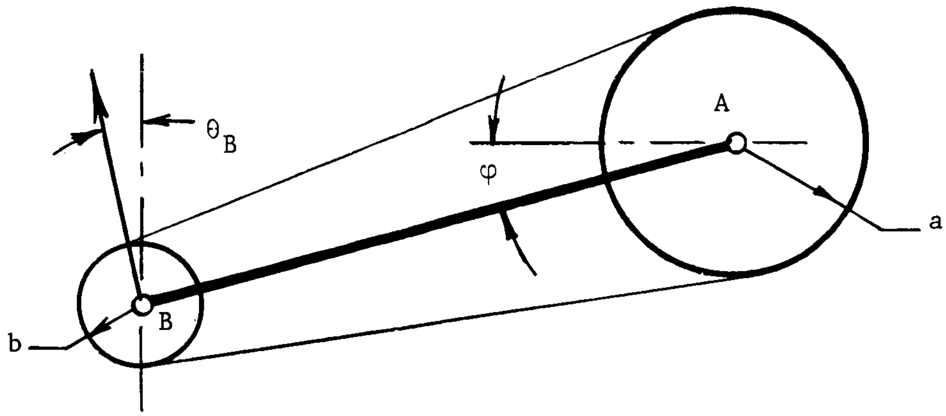


FIGURE 35. SCOOP ACTUATION CABLE GEOMETRY.

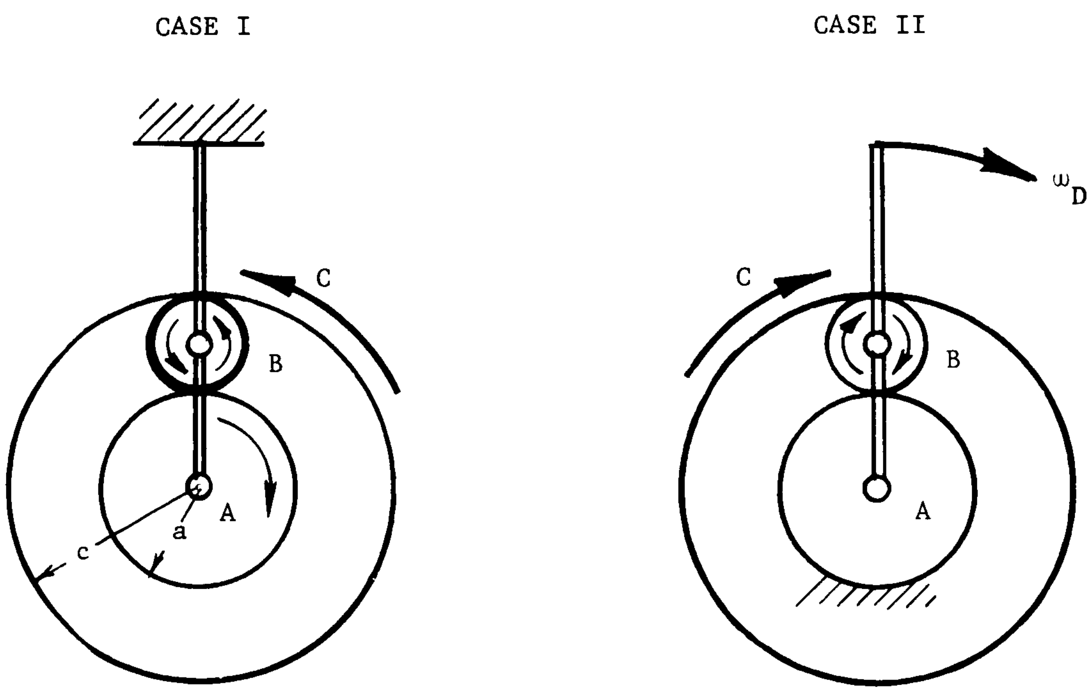
The rotation of B with respect to A is given by $\theta_B = \theta_A$ (a/b). If A is fixed and the arm AB moves, then $\theta_A = -\varphi$ and $\theta_B = \varphi - \varphi(a/b) = \varphi(1 - a/b)$. From the preceding equation, it can be seen that when a = b, the rotation of B is zero; i.e. it maintains a fixed rotational orientation with respect to A. If a larger drum diameter at A is used, then the rotation of B is given by

$$\theta_B = \varphi \left(1 - \frac{a}{b}\right) = -k\varphi.$$

That is, B rotates clockwise when φ is a counterclockwise rotation and the rotation of B differs from φ by a constant of proportionality. Thus, gearing at B can be used to restore the condition achieved when a = b by using a gear ratio to step up the response of B to equal that of A. If a diameter of two inches is assumed for B, then the required gear ratio at the scoop is

$$\frac{\theta A}{\varphi} = 1 - \frac{a}{b} = 1 - \frac{5}{2} = -1.5.$$

Since the drive drum at B is over-responding, a gear reduction must be incorporated as shown in Figure 32 between the scoop and the cable drive drum. The parallel linkage cables must extend synchronously with the boom and simultaneously be capable of actuating the scoop. The extension can be accomplished using two drums rotating in opposite directions to feed out the upper and lower cable as required by the boom extension. Rotating the two drums together will cause B to rotate thereby actuating the scoop. A dual input drive to the drums are indicated and can be achieved with a planetary gear arrangement as shown in the sketch in Figure 36.



$$\omega_C = \omega_A \frac{a}{c} \qquad \omega_C = \omega_D \left(1 + \frac{a}{c}\right)$$

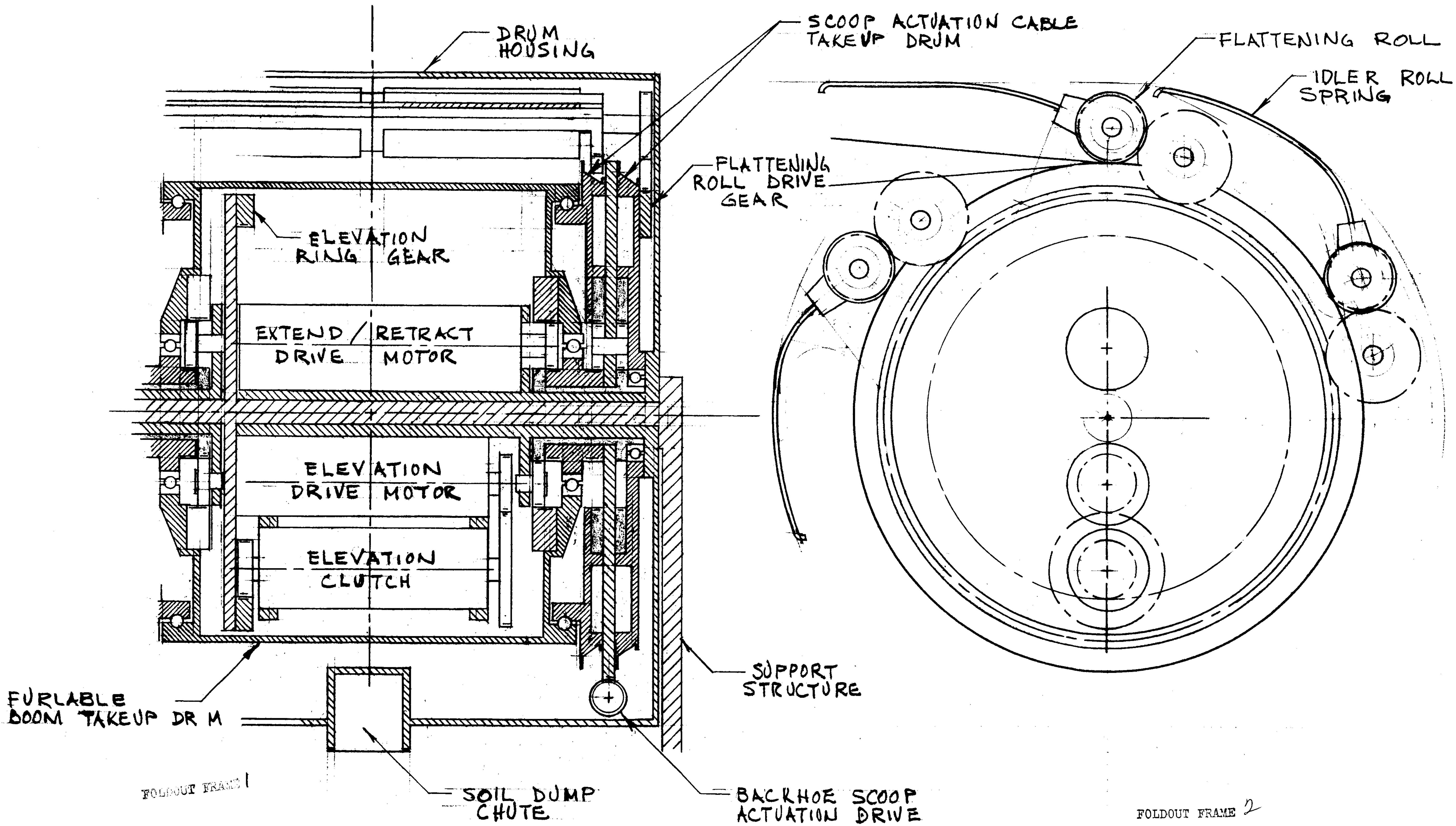
FIGURE 36. PLANETARY GEAR SCHEMATIC.

Case I represents the condition existing when the boom is being extended or retracted to drive the cable take-up drum. Since two drums rotating in opposite directions are required, two identical gear trains as shown for Case I must be employed. The rotational input at A must have an opposite sense. Case II represents the situation when A is not driving but the planetary carrier arm is driving. In this case C rotates in the same sense as the carrier arm D. If this planetary carrier arm is common to both gear trains, then both drums will rotate together. For the assumed geometry the rotation of C is given by

$$\omega_C = \omega_D \left(1 + \frac{2}{1}\right) = 3\ \omega_D.$$

Since the angle that the scoop must go through to open is 60 degrees, the planetary carrier arm is required to move through an angle of 20 degrees. The conditions pertaining for Case I and Case II can be superimposed to obtain the case where scoop actuation occurs simultaneously with boom deployment. In either case, the planetary carrier arm must move only between two fixed points with respect to the boom support structure, thus allowing the use of microswitches to sense whether the scoop is open or closed.

Figure 37 shows a partial section through the take-up drum assembly for this boom. The boom extension/retract drive motor is mounted inside the take-up drum on a hollow shaft tied to the boom housing structure. This motor simultaneously drives the boom take-up drum through a ring gear attached to the inside of the drum and a geared shaft leading to the planetary drive of the inner cable take-up drum which rotates in the same direction as the boom take-up drum and at the same rate. The ring gear inside the boom take-up drum drives the input gear to the outer cable take-up drum through a set of planetary idlers mounted on the drum housing structure. The idlers effect a reversal of input rotation causing the outer gear to drive in the opposite sense as the inner drum but at the same rate. A ring gear affixed to this outer cable take-up drum also drives the boom flattening rolls. The flattening rolls should be driven so that they keep the furlable boom tightly wrapped on the take-up drum. This can be accomplished by driving them so they try to feed slightly faster than the take-up drum. This implies that they must be elastically mounted to the shaft to allow for the differential motion if no slippage occurs. If the gear ratio is set so that the lineal surface speed of the flattening roll and take-up drum match at the start of deployment, an increasing feed rate for the flattening roll is achieved. This is due to the fact that the take-up drum diameter decreases as the boom is deployed because the number of flattened wraps on the drum is less. The flattening rolls are driven through a gear train which are hinged at the axis of the idler gear between the driving ring gear and the gear on the flattening roll. This allows the flattening roll to pivot as required to accommodate the change in drum diameter as the boom



FOLDOUT FRAME 1

FOLDOUT FRAME 2

FIGURE 37. FURLABLE BOOM DRIVE CONCEPT

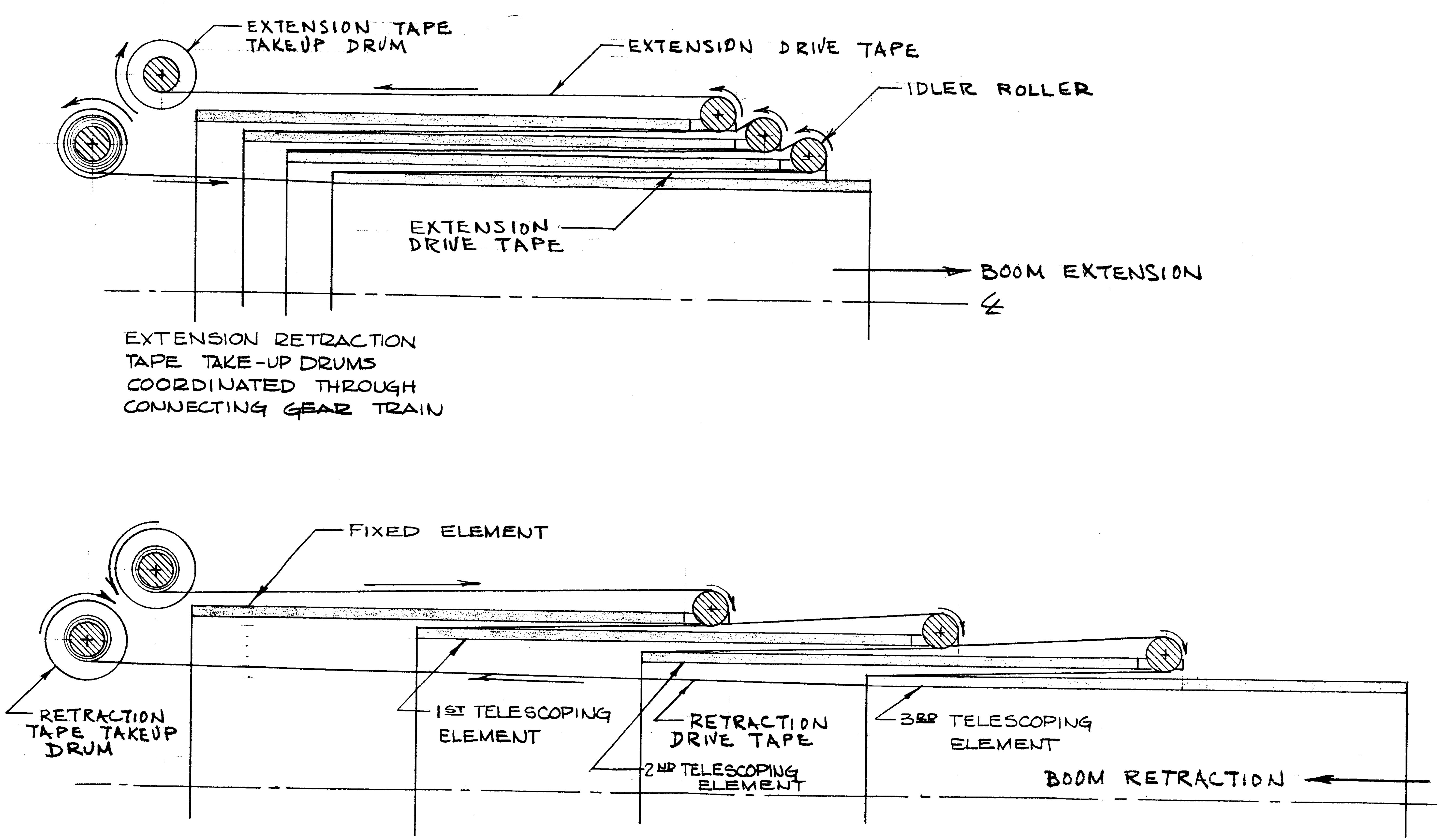
is deployed. The flattening rolls are held against the boom take-up drum by means of flat wide cantilever springs attached to the roll support structure and bearing against the housing structure.

The elevation drive motor is also mounted inside the boom take-up drum on the drum housing structure. This motor drives through a ring gear attached to the fixed support structure about which the boom housing is free to pivot. Thus, the gear output at the elevation clutch walks around the ring gear causing the boom housing structure to be carried with it to control the elevation. The elevation clutch is incorporated so that a sampling traverse can be made with only the weight of the boom acting on the scoop. Alternatively, a load sensor can be incorporated to sense the vertical force being exerted on the scoop to control the rate at which the elevation drive motor operates to maintain a limited downward preload on the scoop.

This mechanism results in a drum housing of 7 inches in diameter and 6 inches wide. The transition length of the boom is approximately one foot so that the overall length from the center of the drum to the back hoe scoop is approximately 14 inches.

3.5.3 MECHANICALLY EXTENDIBLE TELESCOPING BOOM

Another approach considered the use of a telescoping boom using a completely mechanical means of extending and retracting the boom. Such a boom is shown schematically in Figure 38. In this concept, an idler roller is mounted on the tip of all but the last tube segment. A thin tape or small diameter flexible cable is attached to the preceding tube segment, passed over the idler roller back between the tube segments, and is attached to the base of the next tube segment. This is repeated with each tube segment until all the telescoping elements are interconnected. The tape from the first telescoping element is carried back and attached to a tape take-up drum. When power is applied to the take-up drum to rotate it, the tape pulls on the base of the first telescoping segment causing it to extend relative to the fixed element. This in turn causes a tensile force to be applied to the tape connected to the base of the second telescoping segment causing it to also extend. The same occurs for the third telescoping segment. Thus, all segments are being extended simultaneously at the same rate relative to the adjacent segment. A closed cable system can be made by attaching another tape or cable to the base of the third or final telescoping segment and attaching the other end to another take-up drum for retraction. By appropriately sizing the drums and interconnecting the extension take-up with the retraction take-up through a gear train, the deployment and take-up of the tapes can be coordinated so that the tapes or cables will always be taut during the extension or retraction of the boom. A boom of this type offers the advantages of higher strength and rigidity and will also allow a gravity dump to be made at any point in the extension cycle. It also allows the use of various cross-sectional shapes for the telescoping elements, such as square or triangular, if desired. A preliminary configuration of this boom as applied to the backhoe sampler is shown in Figure 39.



FOLDOUT FRAME 1

FOLDOUT FRAME 2

FIGURE 38. MECHANICAL EXTERNAL TELESCOPING BOOM

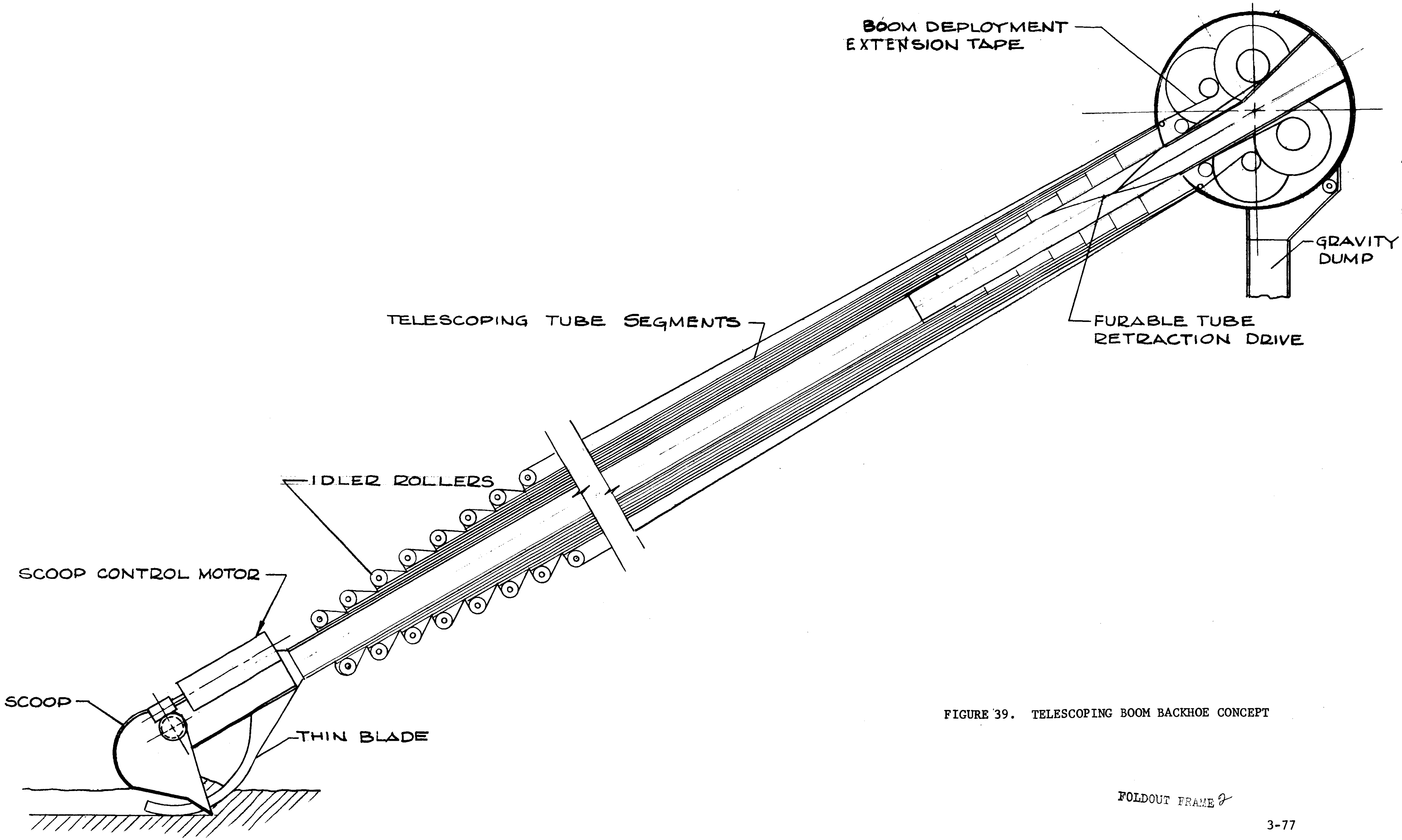


FIGURE 39. TELESCOPING BOOM BACKHOE CONCEPT

FOLDOUT FRAME 2

FOLDOUT FRAME 1

The retraction tapes are furlable tubes utilizing the principle of the DeHavilland furlable tubes. As the boom extends, these two tapes form a secondary tube inside the telescoping boom structure. This boom offers the following advantages:

(a) The internal tube can be used to transport soil at any extended position of the boom simply by raising the boom to a vertical position.

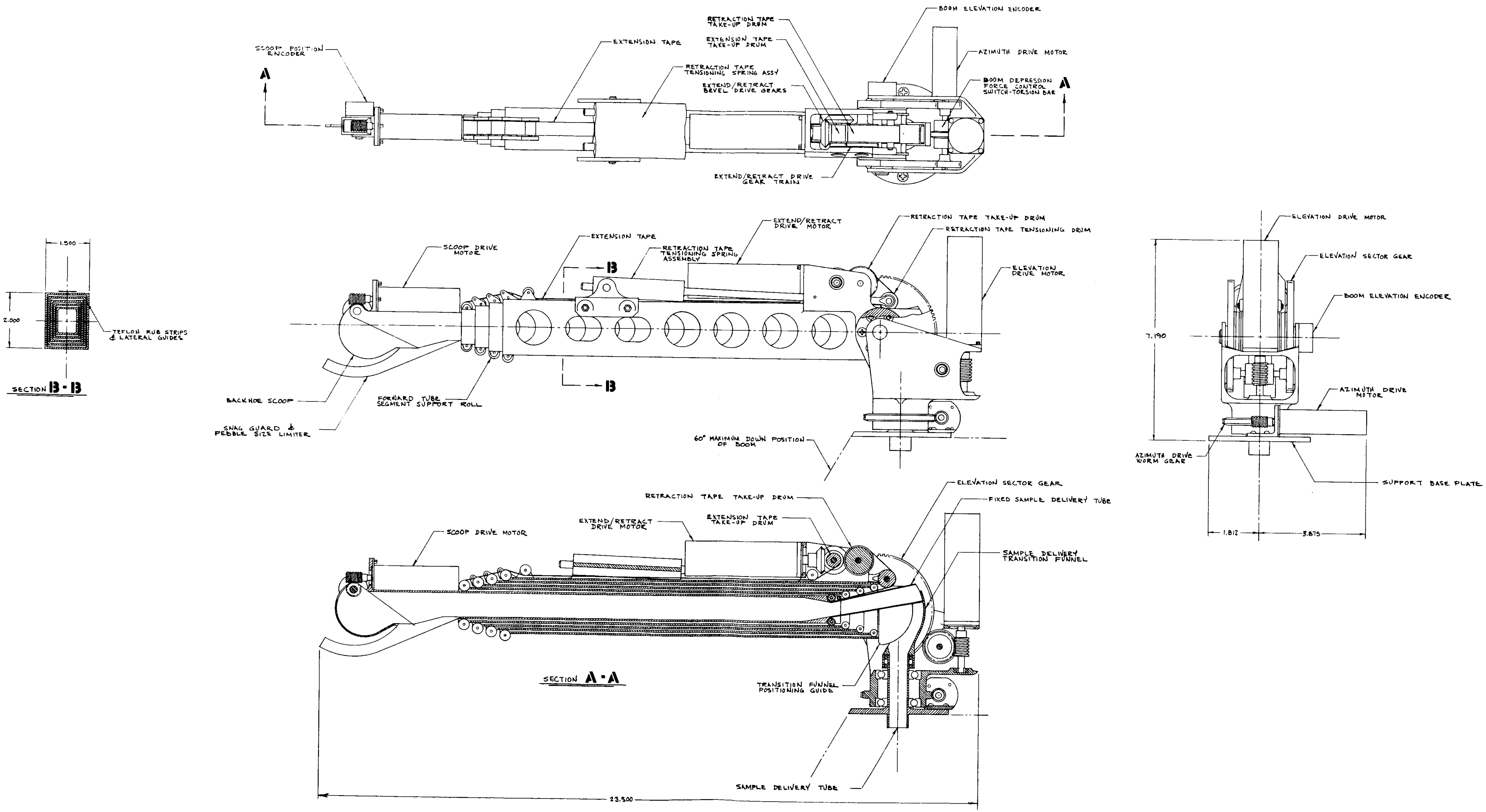
(b) The telescoping boom is inherently stronger than a furlable boom since there are no restrictions governing wall thickness, as there is for the furlable tube.

(c) The secondary internal tube protects the sliding segments of the boom internally from soil particles during the dump mode.

(d) Depending on the total extension required, this boom can be more easily configured to occupy a small volume when stowed.

It should be pointed out that the cable scoop actuation system can be used with this boom also; however, the greater simplicity associated with the motor drive would probably dictate its use even with the penalty of an increased reaction torque in the elevation mechanism. The increase in bending moment in the boom would not be critical from a structural viewpoint. The use of flat tapes rather than cables will allow a more compact design to be made in that smaller clearances can be used between telescoping segments and smaller diameter idler rolls can be used. The extension tape take-up drums are coordinated through a geared drive with the retraction tape take-up drums so that these tapes are deployed simultaneously at essentially the same rate. It may be necessary to mount the take-up drums on their shafts with springs to provide a preload to take up small differences in extension rates since the extension tape is much shorter than the retraction tape resulting in differences in the variation of the effective take-up drum diameters. This configuration is felt to be much simpler to mechanize and should occupy a smaller stowed volume than the furlable boom concept described earlier.

The final prototype design configuration evolved for the backhoe sampler is shown in Figure 40. At the request of JPL, the maximum extension for this boom was reduced to 5 feet resulting in fewer telescoping segments. The cross-sectional shape used for the telescoping segments is rectangular. This eliminates the requirement to key the segments together to obtain torsional strength and rigidity.



FOLDOUT FRAME 1

FIGURE 40. BACKHOE SAMPLER, E-5

 FOLDOUT FRAME 2

In order to achieve further simplicity in the design, the use of the DeHavilland type furlable tapes to act as both a soil transport tube and boom retraction tapes were eliminated. In so doing, it was decided to also eliminate the capability of dumping the soil sample at any intermediate extension of the boom. This was done to minimize exposure of the telescoping elements to soil particles which might wedge between the segments. The sample dump at intermediate boom extensions could be maintained with this design by eliminating the lightning holes in the telescoping segments and enlarging the entrance to the fixed sample delivery tube at the base of the boom. The configuration as shown in Figure 40 performs the soil dump only after the boom is completely retracted. This provides good control of the sample during the dump without exposing the retraction tape and telescoping segment support rolls. The use of rollers to support each segment also eliminates sliding friction between the telescoping segments.

To effect a sample transfer, the boom is first fully retracted and then elevated to a vertical position. As the boom elevates the end of the fixed delivery tube swings down into sample delivery transition funnel. As the boom nears the vertical position, the soil sample falls through the funnel and down the vertical shaft which supports the sampler. The sample delivery funnel is mounted on the end of the vertical support shaft so that it is free to rotate with the boom as it is moved to different azimuth locations. The funnel is always engaged with the fixed sample delivery tube structure by means of the funnel positioning guides. This is true regardless of the amount of elevation or depression that the boom may have. Thus, as the boom rotates in azimuth it carries the sample delivery funnel with it allowing the sample to be delivered in any azimuth position.

The sampling scoop is actuated by a servo motor which responds to an input from the boom position encoder to maintain the scoop in the desired orientation during sampling. This scoop is closed before the boom is elevated to trap the sample in the scoop and support housing for the scoop. Raising the boom to the vertical position causes the sample to fall down the inside of the smallest telescoping segment and be delivered.

Again, in the interest of obtaining maximum design simplicity, a single closed system of extension/retract tapes were used. These were located on the upper side of the boom to shield them from contact exposure with surface features such as boulders as well as to locate the drive motor and tape take-up drums in a location which does not interfere with the elevation of the boom. This is particularly important in achieving the maximum depression of the boom to an angle 60 degrees below the horizontal.

Each telescoping segment is 13.5 inches long and when full extended overlaps the adjacent segment by 2 inches. This is approximately the same amount of overlap as was used on the 10 foot telescoping boom used to deploy the rotating wire brush sampler developed by Philco-Ford which proved to be satisfactory from a strength point of view. Thus, each segment extends 11.5 inches resulting in a total added extension of boom length of 46 inches. The tape drum for the extension tape is required to take up 11.5 inches of tape while the retraction tape must feed out 46 inches in the same time. Since the tapes are wound on the take-up drums in an Archimedes spiral, the rate that they deploy or take up is not a constant. Thus, if a one-to-one gear ratio is used between the extension and retraction tape take-up drums, different size drums must be used so the take-up and deployment rates are compatible. The drums can be sized so that no slack in the tape system occurs at either the extreme retracted or extended positions of the boom; however, slack will occur in the retraction tape in varying amounts between these positions. For the one-to-one gear ratio assumed the maximum length differential is .46 inches at mid extension. For this case the retraction tape take-up drum is nominally four times the dimater of the extension take-up drum. In order to obtain a more compact design, a two-to-one gear ratio between these drums was investigated. By turning the retraction tape take-up drum at twice the rate as the extension drum, the nominal drum size for the retraction tape can be reduced by a factor of two. This causes an increase in the maximum length differential between the boom and the retraction tape to .625 inches. While this amount of slack could possibly be tolerated, smoother and more trouble-free operation can be achieved by incorporating some method of automatically tensioning the tape to eliminate the slack. In this design, the idler roll necessary to reverse the direction of the retraction tape to feed it onto the take-up drum is mounted on a pair of extendible rods which are axially spring loaded. Thus, this idler roll is free to move fore and aft as required to take the slack out of the retraction tape. This roll must have a total axial movement of .312 inches to take up the .625 inches of slack. With this tensioning system, maximum tension in the tape occurs at either the fully retracted or fully extended positions. A spring is provided for each of the two idler roll support rods. These springs are sized to produce a minimum tension in the tape system of 50 pounds and a maximum tension of 75 pounds. Each of these springs are steel with an outside diameter of .625 inches. The wire diameter is .091 inches and the spring has 18 coils. Each of these springs have a preload deflection of .625 inches at 25 pounds and a total deflection of .94 inches at 37.5 pounds.

If a .003 inch thick Be/Cu tape is used that is .500 inches wide, the tape has a total load capability of 216 pounds. Based on strength tests run with the Be/Cu tapes for the rotating wire brush sampler, only 50 percent of this strength can be practically realized at the end connections resulting in a maximum pull of 108 pounds. Thus, if no friction force or binding occurs between the telescoping segments, a maximum axial pull of 33 pounds is available for sampling at full extension and 58 pounds at mid extension of the boom. This should be more than adequate for the type operations anticipated with this sampler.

A nominal extension rate of one foot per minute was assumed for this boom. The block diagram for the power train to extend and retract the boom is given in Figure 41. The drive motor is rated at one inch-ounce of torque and the gear reducer has an efficiency of 26 percent. This produces an output torque of 108 inch-pounds which yields an extension pull capability of 432 pounds and a retraction pull capability of 108 pounds. It is seen that the drive capability in retraction exactly matches the allowable strength of the retraction tape. The fact that the pull capability on the extension tape exceeds the allowable strength of the extension tape by a factor of four does not mean that it is likely to break the tape during extension unless the telescoping segments bind up. If all telescoping segments bind up equally, the maximum resistance that can be overcome between each segment is 11 pounds at either full extension or retraction and is 19 pounds at mid extension. Similarly, if only one segment binds or the boom is used to push an object, a maximum force of 33 pounds can be applied at full extension or retraction and 58 pounds at the midpoint. It should be noted that the available forces for extension or retraction is based on the 50 to 75 pound tensioning preload in the tape. If this were reduced the maximum available forces could be increased. Conversely, an increase in tape thickness or width could increase the allowable strength as could improved efficiency in the joints at the tape ends.

Separate drives are used for the boom elevation and azimuth orientation. The block diagrams for these power trains are given in Figure 42. A type LL motor was used for the elevation drive as shown in Figure 40; however, a more accurate assessment of the power requirements indicates that the type SS motor is adequate. The maximum torque that must be reacted in elevation is estimated to be 100 inch-pounds for a five foot long boom in an earth gravity environment. The particular gear motor called for in Figure 42 has a rated output of 45 inch-ounces. The 100 inch-pound elevation moment is reflected at the planetary gearbox as 13.3 inch-ounces so that the available power exceeds the demand by a factor of three. Two elevation sector gears four inches in diameter are used resulting in a tooth load per gear of 25 pounds. This load is low enough so that either a narrower face on the gears can be used or one sector gear can be eliminated. Another alternative is to make these gears out of aluminum.

This sampler mechanism is shown in various stages of deployment in Figure 43. The primary factor governing the time required to complete a sampling cycle is the time required to extend and retract the boom since the elevation and azimuth drives are faster and may be activated

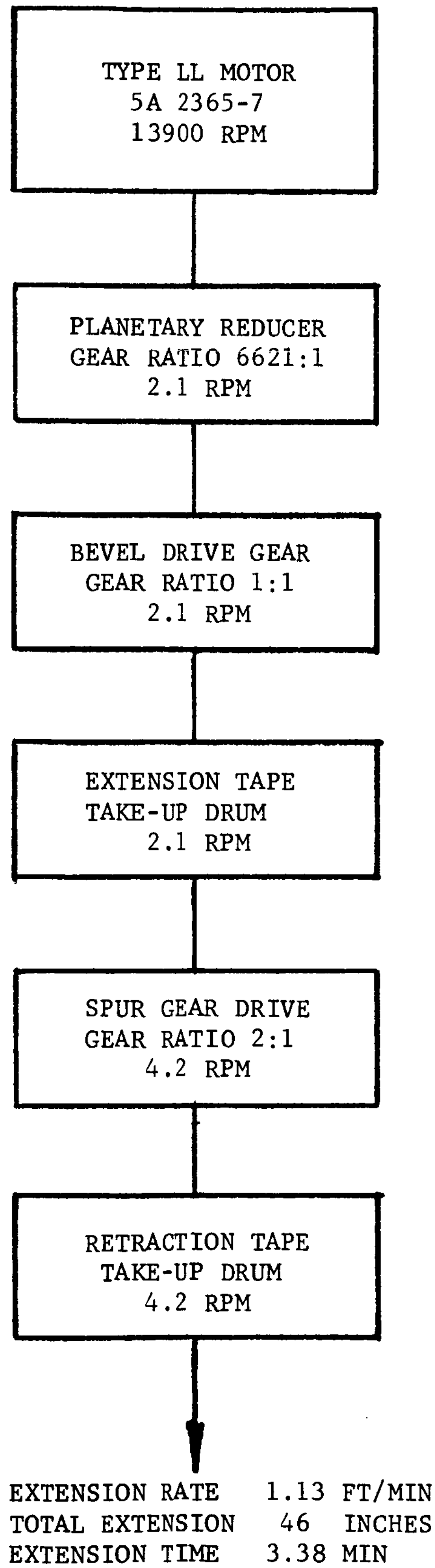


FIGURE 41. BLOCK DIAGRAM - BACKHOE BOOM EXTENSION POWER TRAIN

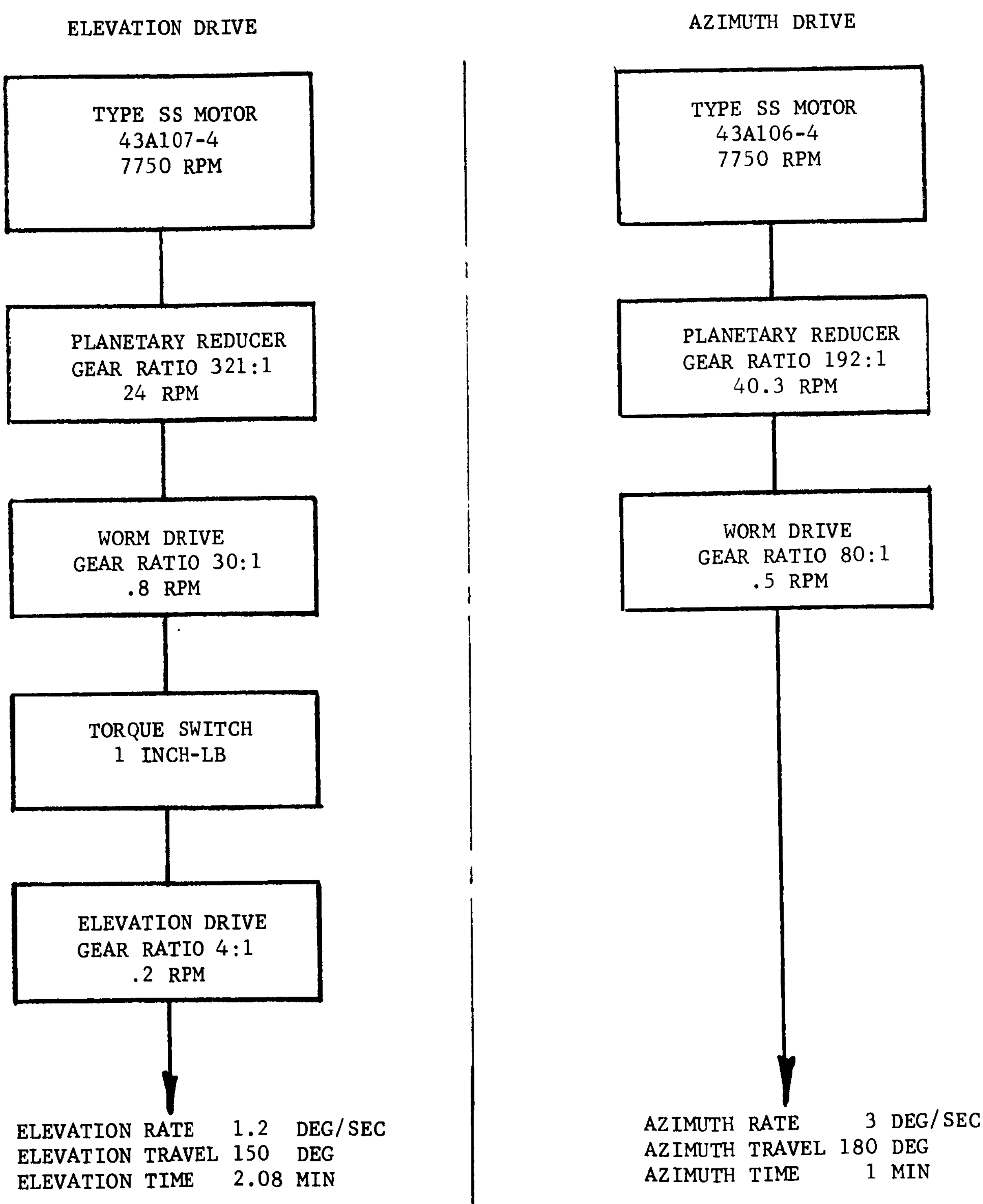
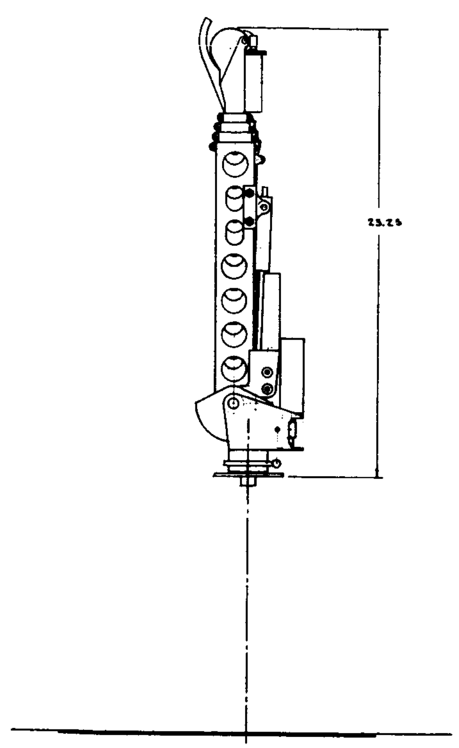
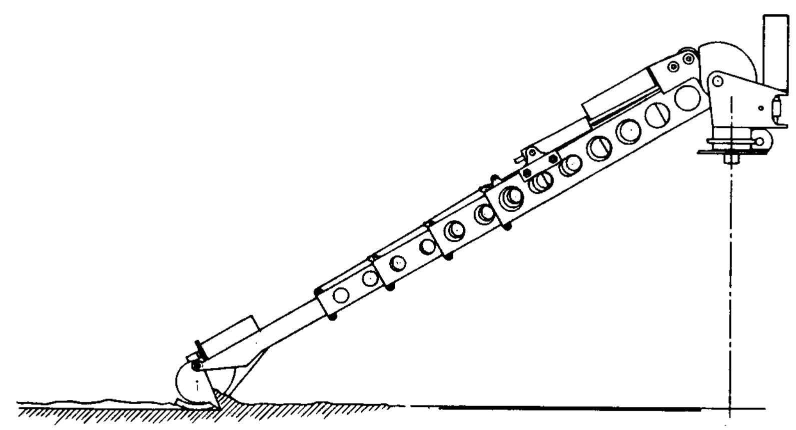


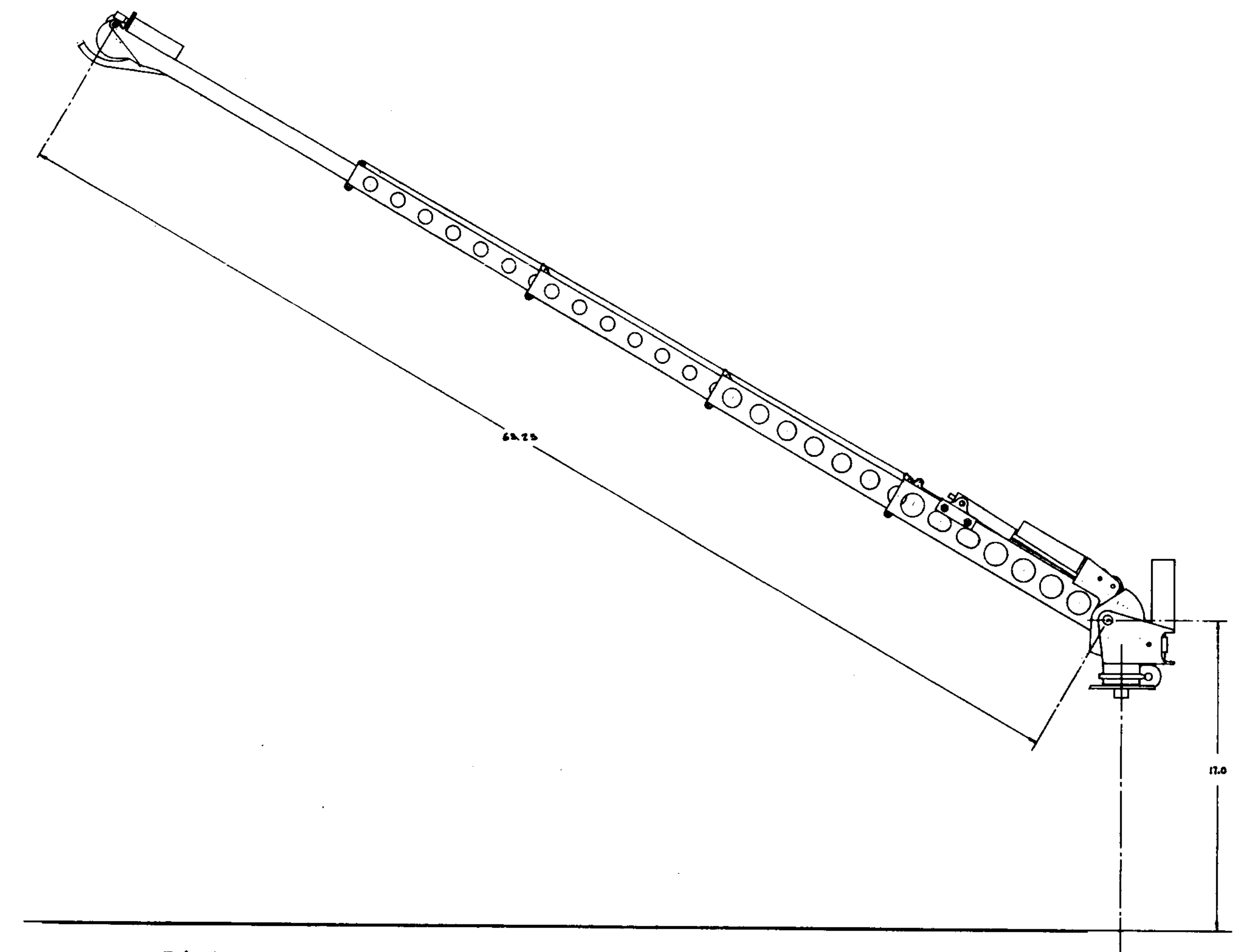
FIGURE 42. BLOCK DIAGRAM - BACKHOE BOOM ELEVATION AND AZIMUTH POWER TRAINS



BACKHOE IN GRAVITY DUMP POSITION



BACKHOE SAMPLING - BOOM DEPRESSED 30°



BACKHOE EXTENDED FULL LENGTH AND ELEVATED 30°

FIGURE 43. BACKHOE SAMPLER DEPLOYED

FOLDOUT FRAME 1

FOLDOUT FRAME 2

simultaneously with the boom extension. A sampling cycle using full extension of the boom requires about 6.5 minutes. This is a reasonable length of time. The operational sequence is outlined in Table XVIII.

The weight statement for the backhoe sampler mechanism is given in Table XIX.

TABLE XVIII

BACKHOE SAMPLER OPERATIONAL SEQUENCE

1. Activate the azimuth drive to position boom at desired azimuth location.

2. Activate elevation drive to erect the boom to a vertical position.

3. Activate the boom extension drive to extend the boom. These first three steps may be performed sequentially or simultaneously as desired.

4. Activate backhoe scoop drive to open scoop.

5. When the desired boom extension is achieved, reverse the elevation drive to place the backhoe scoop in contact with the surface. The load sensing torque switch in the elevation gear train turns off the elevation motor when the desired preload on the scoop is reached.

6. Reverse the extension drive motor to retract the boom and collect a sample. The scoop motor servo drives the scoop to maintain the desired orientation of the scoop.

7. After completing the desired length of sampling run, activate the elevation drive to lift the boom off of the surface. Stop elevation 20 degrees above the horizontal.

8. Activate the scoop drive motor to close the scoop. This step may occur before, after, or simultaneously with step 7.

8\. Complete retraction of the boom.

10\. Elevate the boom to the vertical position to transfer the soil sample. The sampler may be left in this position until next sampling cycle is initiated.

TABLE XIX

WEIGHT STATEMENT, BACKHOE SAMPLER, E-5

Item No.	Item	Mtl	Qty	Weight /Part	Weight /Assy	Total
	Backhoe Scoop Assy					
1	Scoop	STL	1	.0488	.0488	
2	Drive Motor	-	1	.4375	.4375	
3	Worm	STL	1	.0156	.0156	
4	Worm Gear	STL	1	.0175	.0175	
5	Position Encoder	-	1	.0251	.0251	
6	Snag Blade	STL	1	.0278	.0278	
7	Scoop Housing	Mg	1	.0293	.0293	
8	Scoop Shaft	STL	1	.0121	.0121	
	Subtotal		8			.6137
9	Segment #1	Mg	1	.1197	.1197	
10	Segment #2	Mg	1	.1352	.1352	
11	Segment #3	Mg	1	.1666	.1666	
12	Segment #4	Mg	1	.1856	.1856	
13	Segment #5	Mg	1	.1924	.1924	
14	Guide Roll .375 OD	Mg	10	.0034	.340	
15	Guide Roll .25 OD	Mg	8	.0014	.0112	
16	Guide Roll Shaft	STL	18	.0005	.0090	
17	Extend/Retract Motor	-	1	.7500	.7500	
18	Tensioning Rod	Ti	2	.1346	.2692	
19	Tensioning Spring	STL	2	.0869	.1738	
20	Spring Housing	Mg	1	.1307	.1307	
21	Housing Support	Mg	2	.0225	.0450	
22	Bolts 8-32 x .31	STL	4	.0033	.0132	
23	Nuts 8-32	STL	4	.0026	.0104	
24	Gear Support	Mg	1	.1512	.1512	
25	Retract Drum	Mg	1	.0251	.0251	
26	Extend Drum	Mg	1	.0062	.0062	
27	Bevel Gear	STL	2	.0632	.1264	
28	Spur Gear .562 PD	STL	1	.0077	.0077	
29	Spur Gear 1.125 PD	STL	1	.0341	.0341	
30	Shaft	STL	2	.0126	.0252	
31	Tensioning Drum	Mg	1	.0089	.0089	
32	Drum Shaft	STL	1	.0106	.0106	
33	Elevation Gear	STL	2	.0961	.1922	
34	8-32 x .375 Screws	STL	6	.0033	.0198	
35	Elevation Encoder	-	1	.0251	.0251	
36	Sample Tube	Mg	1	.0335	.0335	
37	8-32 x .5 Screws	STL	4	.0042	.0168	
38	Extension Tape	Be/Cu	4	.0062	.0248	
39	Retraction Tape	Be/Cu	1	.0276	.0276	
40	Tape Pins	STL	6	.0043	.0258	
41	Bearing .1875 Bore	STL	4	.0109	.0436	
	Subtotal		97			3.0506
	Base Assy					
42	Base	STL	1	.2223	.2223	
43	Bearing .750 Bore	STL	2	.1409	.2819	
44	Bearing .750 Bore	STL	2	.2599	.5200	
45	Transition Funnel	Mg	1	.0486	.0486	
46	Trunnion Support	Mg	1	.3130	.3130	
47	Boom Shaft	STL	2	.0075	.0150	
48	Elevation Shaft	STL	1	.0381	.0381	
49	Elevation Gears	STL	2	.0935	.1875	
50	Elevation Motor	-	1	.4375	.4375	
51	Torque Switch	-	1	.0625	.0625	
52	Worm Gear	STL	1	.0793	.0793	
53	Worm	STL	1	.0382	.0382	
54	Bearing .1875 Bore	STL	2	.0109	.0218	
55	Azimuth Gear	Al	1	.0441	.0441	
56	Azimuth Worm	STL	1	.0149	.0149	
57	Azimuth Motor	-	1	.4375	.4375	
58	Motor Bracket	Mg	1	.0150	.0150	
59	8-32 x .31 Screw	STL	3	.0033	.0099	
	Subtotal		25			2.7871
	Total Sampler Assy		130			6.4514

3.5.6 SOIL AUGER SAMPLER, E-6

This sampler mechanism consists of an auger 1 inch in diameter and about 4.5 inches long. The auger is rotated slowly while thrust is applied to cause it to penetrate the soil. After penetration is achieved, the soil auger rotation is stopped and the auger is withdrawn without rotation to its initial stowed position. When it reaches this point the auger is spun at a high rotational speed to spin the soil off of the auger flights. Two basic approaches to mechanizing this sampler were taken as discussed in the following.

A variation of the canted feed roller system, developed by Philco-Ford under a previous contract, was applied to this sampler as shown in Figure 44. In this approach, one drive motor with a coaxial output from the gearbox is used to simultaneously drive a high speed gear train for the spin dump and a low speed gear train to drive the axial feed and rotate the auger as required. The low speed gear is connected to a hollow shaft with flats on the outer surface as shown in section A-A of Figure 44. The feed assembly housing has a hole shaped to fit this shaft so that the low speed gear train is always driving the feed housing regardless of direction of rotation or axial position along the shaft. When the direction of rotation is such that downward axial feed is achieved, the over-running clutch engages the shaft of the auger causing it to rotate with the housing. The feed is achieved by six rollers mounted to the feed assembly housing in sets of three rollers at the upper and lower end of the housing. These rollers are canted at a small angle and are mounted on torsion bars in such a manner that they are pressed tightly against the outer support tube. Thus, rotation of this assembly causes the canted rollers to feed the auger up or down, depending on the direction of rotation. These rollers are mounted on the ends of torsion bar supports so that, as the axial thrust builds up, the reaction forces acting on the feed rollers causes them to rotate to a smaller cant angle thereby reducing the feed rate. Thus, if a strong cohesive surface is encountered the axial thrust is built up to some maximum value determined by the torsion bar deflection characteristics. As the auger shears the soil the load is reduced and the canted feed rollers immediately move to an angle which will again cause the auger to feed into the soil. In this manner the feed rate always accommodates itself to a rate at which the auger can penetrate while maintaining the requisite axial thrust to cause penetration. To withdraw the auger the polarity of the power supplied to the motor is reversed thereby reversing the rotation of all gear trains. When this happens, the over-running clutch releases the auger shaft so that it is not driven while vertical feed is applied. When the auger mechanism reaches the initial stowed position, the conical tip of the high speed drive shaft is driven into a mating socket thereby engaging the high speed drive to spin the soil off of the auger and into the annular chamber at the base of the outer support tube. Some features of this design approach are as follows:

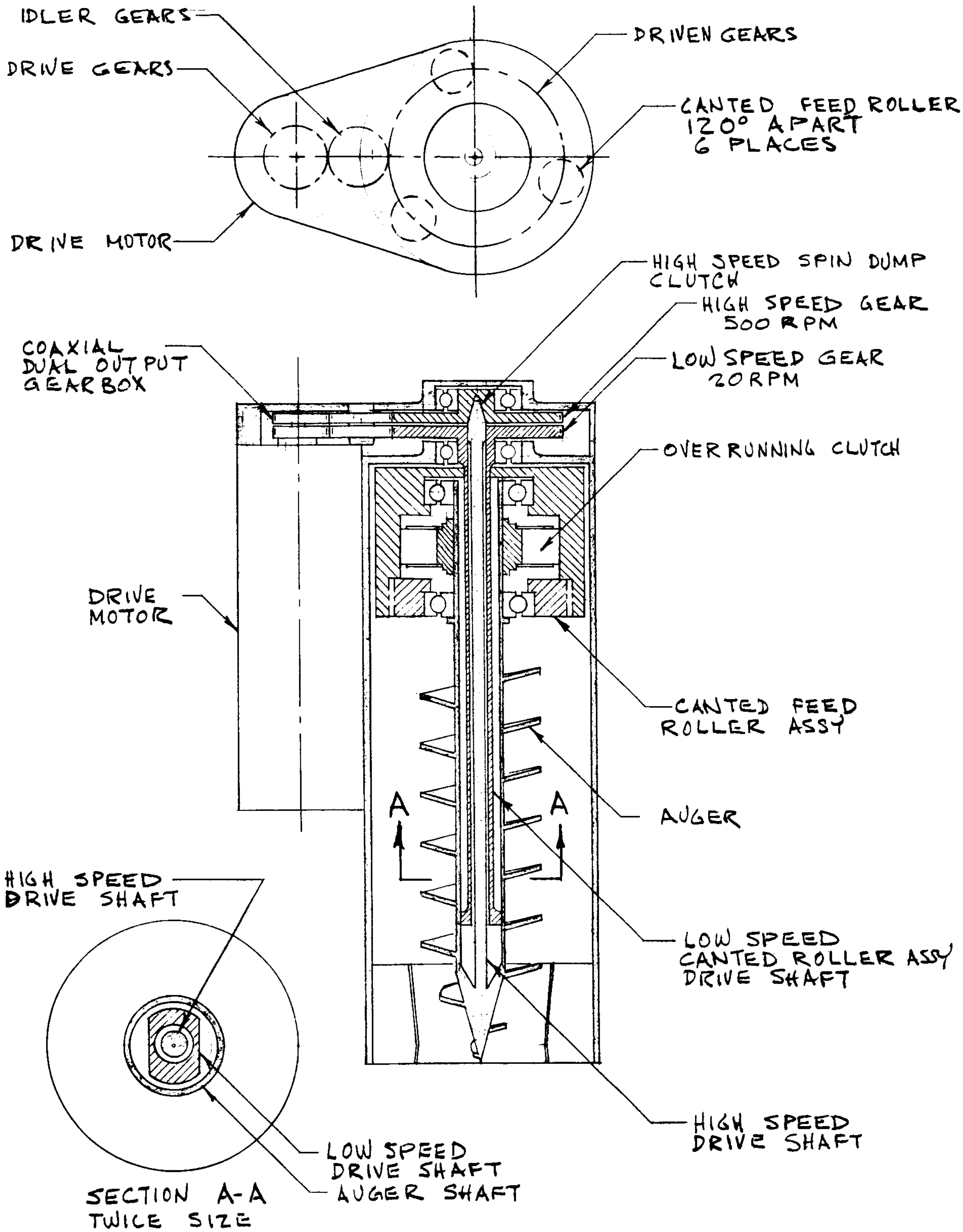


FIGURE 44. SOIL AUGER - CANTED FEED ROLL DRIVE

(a) The mechanization is simple and requires only one drive motor.

(b) Except for reversing the polarity of the power to the drive motor, the operational steps are automatically executed.

(c) Because the direction of rotation is dictated by the over-running clutch, initiation of the high speed spin will tend to cause the sample to feed down along the auger flights; however, the rapid increase of centrifugal force should spin the soil off before much downward feed takes place. Embossing the surface with shallow radial saw tooth slots would inhibit the downward feed.

The second design approach uses a system more closely related to the breadboard version built by JPL as shown in Figure 45. Two motors are required, one to drive the axial feed screws and another with a coaxial output gearbox to drive the auger at low speed for digging and at high speed for the spin dump. The thrust exerted by the axial feed screws are reacted through springs at the top of the screws. As the thrust builds up the screws are lifted in proportion to the thrust. At some predetermined thrust limit, the ends of the screws actuate one or both of the snap action switches connected in series to turn off the power to the feed drive motor. As the thrust falls off the springs push the screws down until the switches are again closed, thereby providing power to the feed drive motor. Thus, axial thrust is maintained while the feed rate is adjusted to accommodate itself to the feed rate of the auger. The auger is engaged with the appropriate speed drive gear train through two over-running clutches mounted on the auger drive shaft. The relative rotation and clutch engagements are shown schematically in Figure 46. The drive shaft is square in cross-section as shown in section A-A of Figure 45. This shaft is fitted into a square hole in the auger so that rotational power is applied as the auger advances. Some features of this design approach are as follows:

(a) The shaft of the auger can be smaller in diameter than for the canted feed roller approach.

(b) The operational reliability of the over-running clutch to engage the high speed spin is probably better than the cone friction clutch.

(c) More programmed inputs are required to complete an operational cycle; however, more flexibility for altering the operational mode exists.

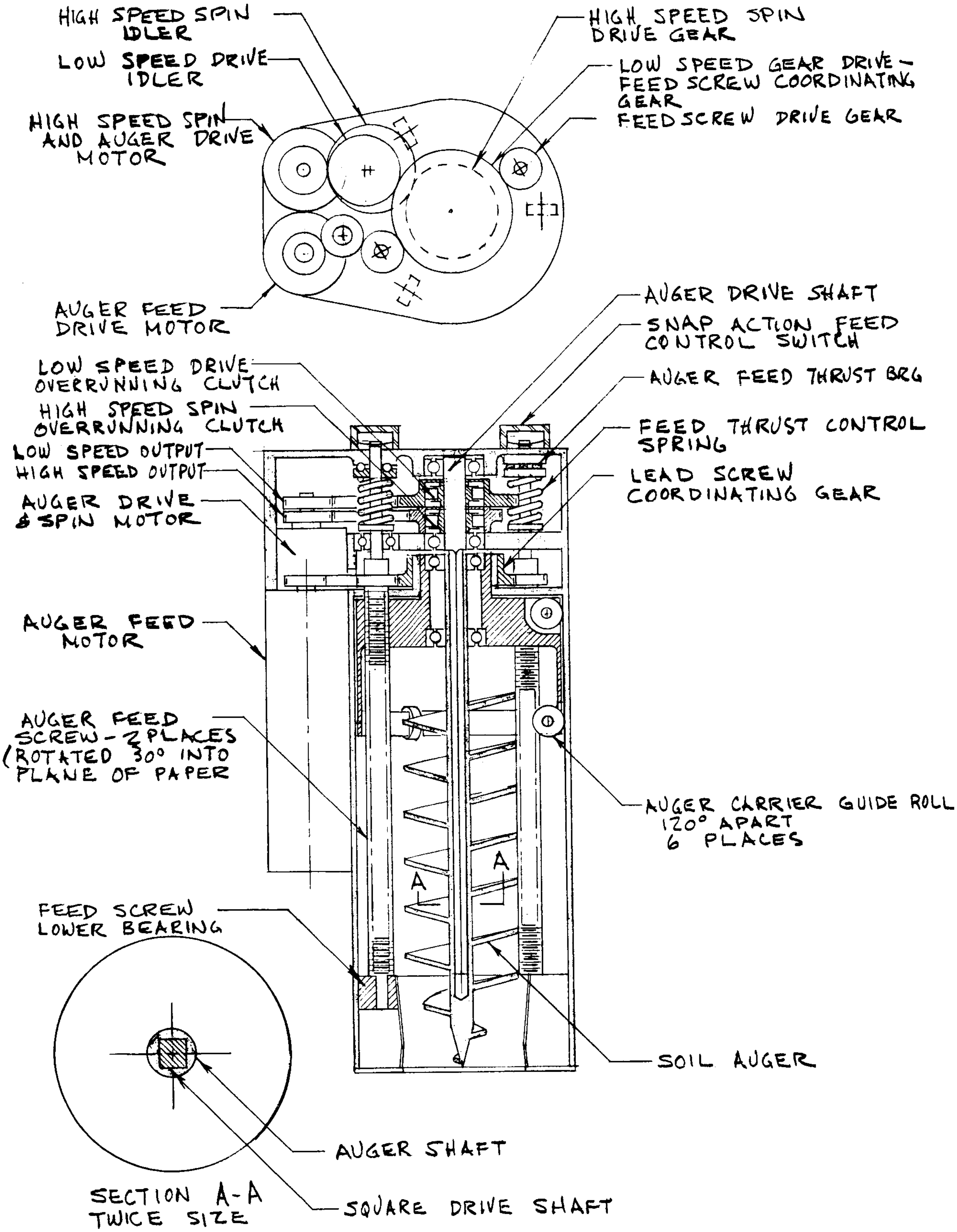
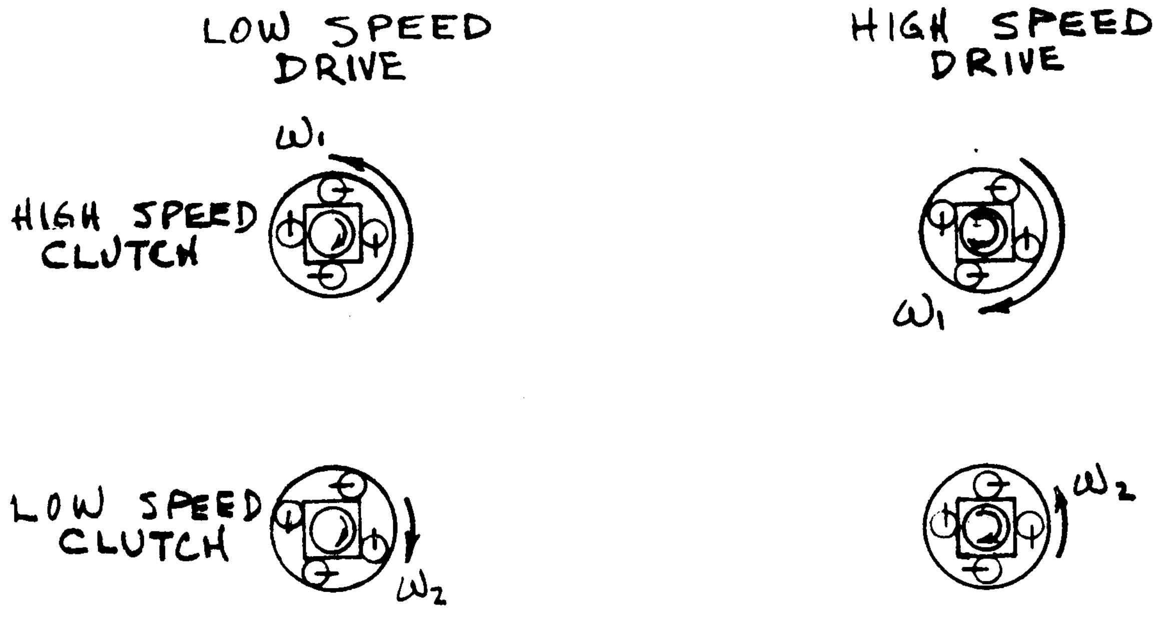


FIGURE 45. SOIL AUGER - LEAD SCREW FEED



$\omega_1 = 500$ RPM $\quad \omega_2 = 10$ RPM

SCHEMATIC - CLUTCHING ARRANGEMENT
SOIL AUGER DRIVE - LEAD SCREW FEED

1. COAXIAL OUTPUT GEARBOX DRIVES BOTH HIGH SPEED AND LOW SPEED GEAR TRAIN
2. DURING LOW SPEED DRIVE OVERRUNNING CLUTCH ALLOWS HIGH SPEED GEAR TRAIN TO RUN IDLE
3. DRIVE MOTOR TURNED OFF DURING RETRACT CYCLE — CLUTCHES PREVENT AUGER ROTATION IN REVERSE DIRECTION DURING RETRACT
4. DURING SPIN DUMP OVER-RUNNING CLUTCH ENGAGES HIGH SPEED GEAR TRAIN AND RELEASES LOW SPEED GEAR TRAIN

FIGURE 46. SCHEMATIC OF CLUTCH ACTION

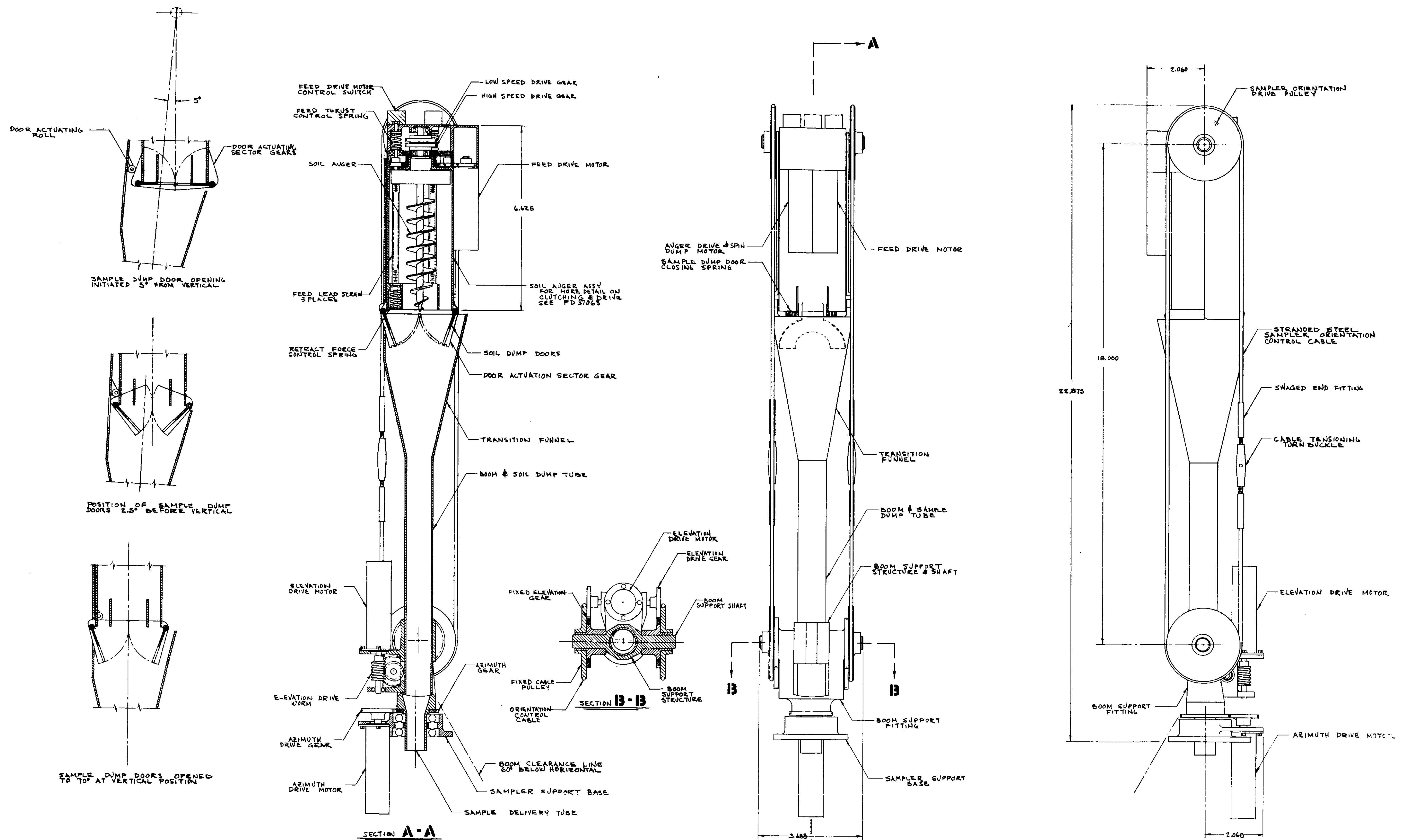
(d) The axial feed lead screws are exposed to the soil during the spin dump. Some sort of shielding may be required.

(e) The auger carrier guide rolls could be eliminated by using three feed screws driven synchronously.

As originally presented to JPL, this auger was to be mounted on a telescoping boom. At their suggestion a short rigid boom will be used. The orientation of the soil auger with respect to the local vertical can be achieved by either a parallel bar linkage or a closed cable system. The closed cable system will probably provide the lightest configuration. Final transfer of the soil from the annular chamber in the outer support housing will be achieved by erecting the boom to a vertical position. Since the soil auger orientation is always maintained in a vertical position, this will position the bottom of the support housing or annular soil chamber over the end of the tubular boom. Two spring loaded doors closing the annular soil chamber are opened as the soil auger reaches the vertical position to dump the soil down the boom to the payload sample entry port.

These doors are shown in the open position of section A-A of Figure 47 and the adjacent views indicating the operational sequence of these doors. The two doors are interconnected by means of sector gears. These are actuated by a roller mounted on the boom structure during the last 5 degrees of rotation of the boom to the vertical position. At the same time, erecting the boom to the vertical position rotates the open end of the boom structure at the base into position over the support shaft. Thus, when the sample dump doors open, the sample is free to fall down the boom through the vertical support shaft to some delivery point in the spacecraft payload.

Although either of the sampler drive approaches discussed earlier can be used with this sampler, the lead screw feed was shown with some modifications in section A-A of Figure 47. These modifications were suggested by JPL and consist of eliminating the guide rollers, adding a third lead screw, and incorporating the ability to sense the axial thrust during withdrawal of the auger as well as during the feed of the auger. The latter feature can be achieved by mounting each axial feed screw between springs at either end. This allows the lead screws to float vertically in either direction as the axial thrust builds up. The end of the lead screws can then actuate the drive motor control switch. This switch must be double acting to turn the motor off when the plunger is moved in either direction from the nominal on position. If desired, the lead screw support springs can be sized to produce a higher withdrawal force than feed force. A nominal axial feed force of 10 pounds appears to be adequate based on results obtained in the field test phase with the breadboard model of this sampler.



FOLDOUT FRAME /

FIGURE 47. SOIL AUGER SAMPLER, E-6

FOLDOUT FRAME 2

The block diagram for the power train driving the lead screws and auger are shown in Figure 48. The auger in the breadboard model was driven through a clutch rated at 160 inch-ounces. Assuming this to be adequate, a load of 89 inch-ounces is applied at the drive gear. The rated torque output of the SS gearmotor is 70 inch-ounces. This is determined by long life operation of the gearbox rather than the torque capability of the motor. The gearbox efficiency for this motor is 41 percent which would yield a potential torque output capability of 89 inch-ounces. The break-away torque for this motor is twice the rated torque yielding a maximum drive torque of 180 inch-ounces. Thus, the SS motor should be adequate to drive the auger.

The soil auger assembly is mounted on the end of a short rigid boom as shown in Figure 47. Two sets of steel cables are used to maintain the vertical position of the soil auger for any deployment angle of the boom. Two pulleys are fixed to the soil auger assembly through the shaft supporting the auger assembly on the end of the boom. Another set of pulleys are mounted to the fixed base of the support structure so that they cannot rotate with respect to the mounting surface. The steel cables passing over these pulleys then cause the pulley mounted on the sampler to remain aligned with the fixed pulley thereby maintaining the vertical orientation of the soil auger.

The block diagrams for the elevation and azimuth drive power trains are shown in Figure 49. The elevation drive motor is mounted on the boom structure and drives a spur gear through a worm drive. This spur gear meshes with a spur gear fixed to the support structure. It elevates or depresses the boom by walking around the fixed gear. The azimuth drive has a spur gear fixed to the rotating support structure which is driven by the drive gear mounted on the output shaft of the drive motor.

The boom on this sampler can also be depressed to a maximum of 60 degrees below the horizontal. The sampler is shown deployed in Figure 50 for this position and an intermediate position. The operational sequence for this sampler is given in Table XX.

The weight statement is given in Table XXI.

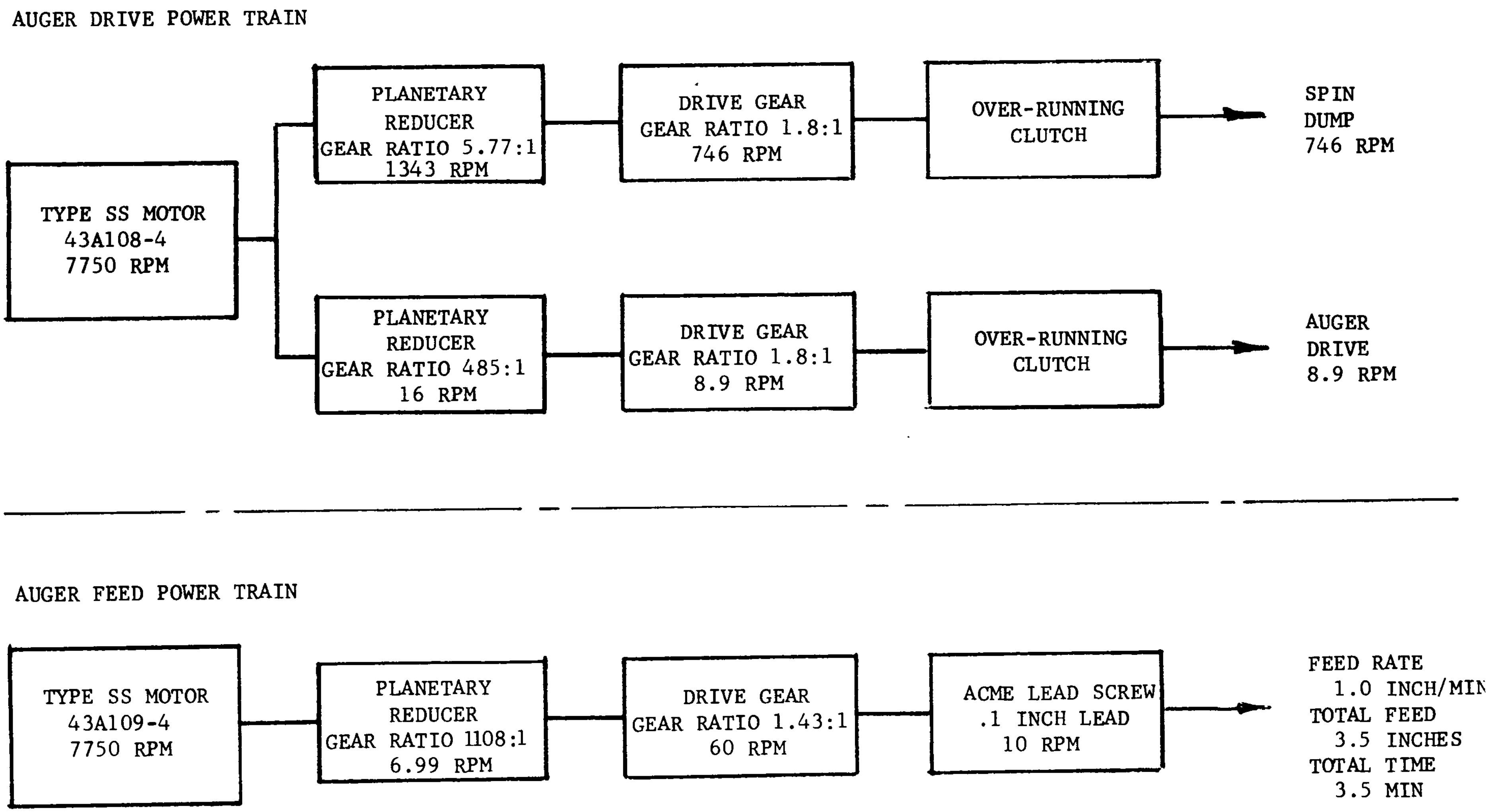


FIGURE 48. BLOCK DIAGRAM - SOIL AUGER POWER TRAINS

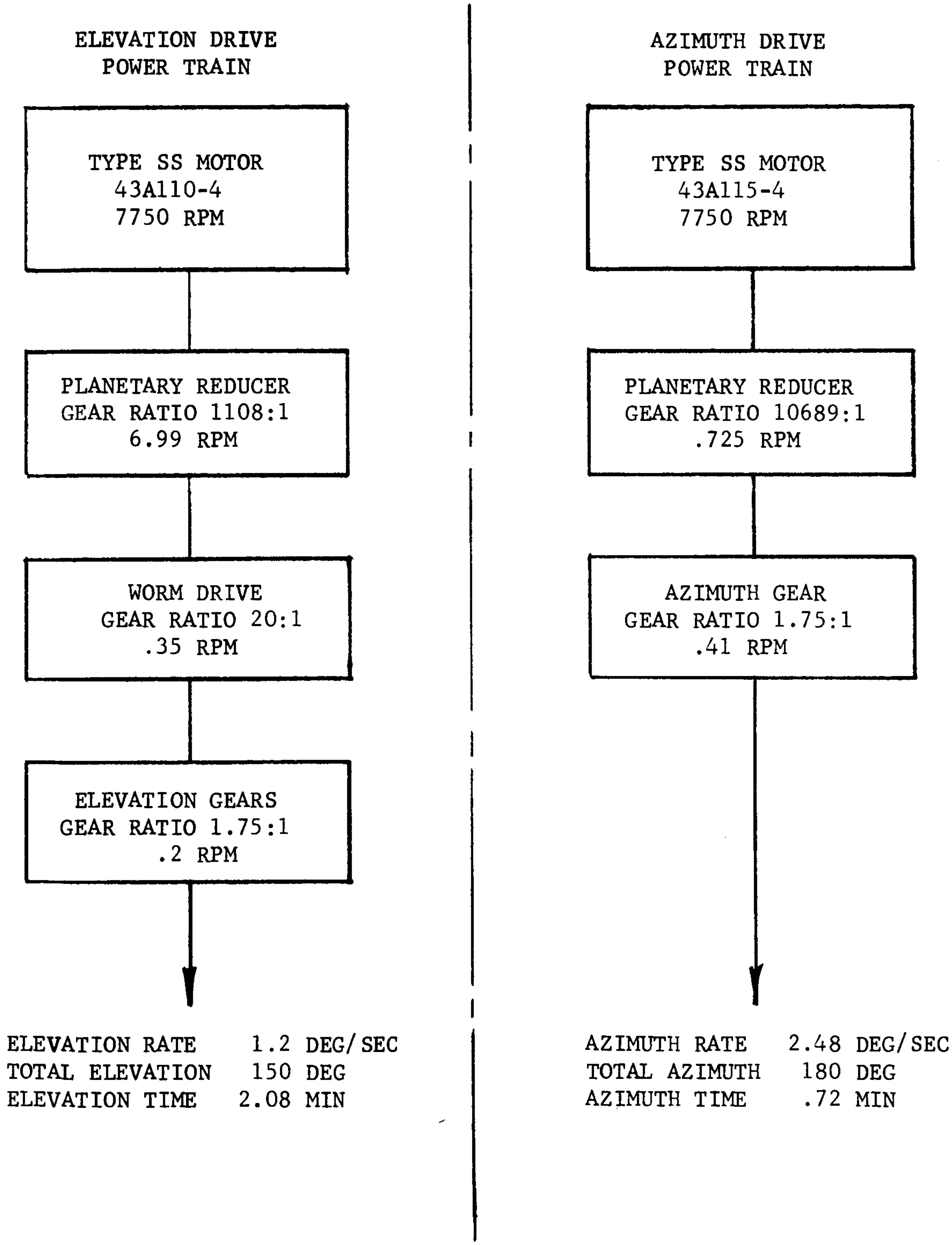
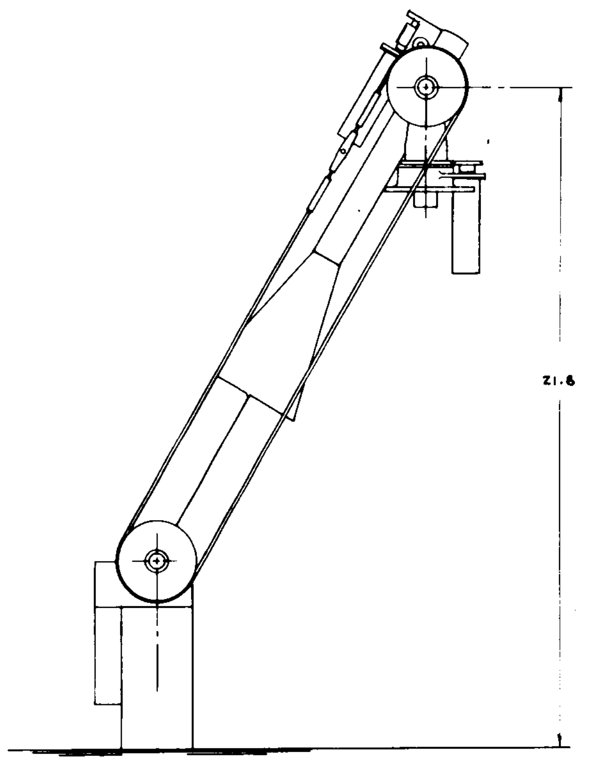
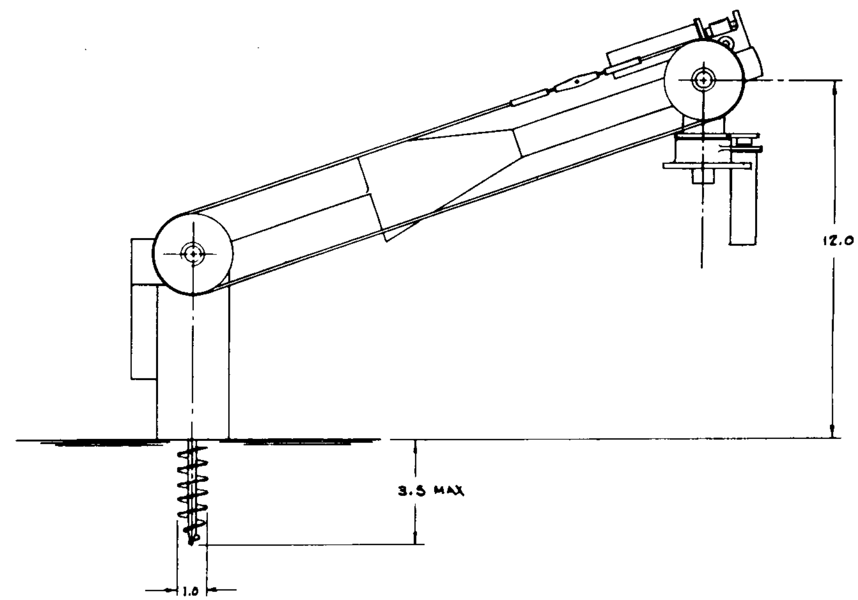


FIGURE 49. BLOCK DIAGRAM - SOIL AUGER ELEVATION AND AZIMUTH POWER TRAINS



SOIL AUGER IN MAXIMUM DOWN POSITION
60° BELOW HORIZONTAL



SOIL AUGER IN INTERMEDIATE POSITION

FIGURE 50. SOIL AUGER SAMPLER DEPLOYED

TABLE XX

SOIL AUGER SAMPLER OPERATIONAL SEQUENCE

1. Activate the azimuth drive motor to drive the sampler to the desired azimuth.

2. Activate the elevation drive to lower the soil auger to the surface. A load sensor terminates power to the elevation drive motor.

3. Activate the soil auger drive motor.

4. Activate the soil auger feed.

5. The soil auger digs into the surface for a predetermined time or until the limit of feed travel is reached. Power is then terminated to the auger drive motor.

6. Power to the feed drive motor is reversed causing the auger to be withdrawn from the surface without rotation.

7. When auger retraction is complete power is terminated to the feed drive motor.

8. The auger drive motor is activated in reverse. This causes the low speed gear train over-running clutch to release the auger and the high speed gear train clutch to engage the auger and to spin dump the soil into the annular collection chamber at the base of the soil auger housing. Power is terminated to the auger drive motor after a predetermined time.

9. Activate the elevation drive motor to return the boom to the vertical position. The last 5 degrees of boom travel causes the soil dump doors at the bottom of the soil auger assembly to open dropping the soil down the hollow boom to a terminal delivery point.

TABLE XXI

WEIGHT STATEMENT, SOIL AUGER SAMPLER, E-6

Item No.	Item	Mtl	Qty	Weight /Part	Weight /Assy	Total
	Soil Auger Assy					
1	Bearing .125 Bore	STL	9	.0109	.0981	
2	Bearing .187 Bore	STL	8	.0109	.0872	
3	Over-running Clutch	STL	2	.0203	.0406	
4	Feed Drive Motor	-	1	.7500	.7500	
5	Auger Drive Motor	-	1	.7500	.7500	
6	Feed Control Switch	-	3	.0312	.0937	
7	Gear .437 PD	STL	3	.0150	.0450	
8	Gear .625 PD	STL	4	.0175	.0700	
9	Gear .750 PD	STL	2	.0185	.0370	
10	Gear 1.125 PD	STL	2	.0329	.0658	
11	Gear 1.312 PD	STL	1	.0682	.0682	
12	Feed Lead Screw	STL	3	.0086	.0258	
13	Feed Control Spring	STL	6	.0035	.0210	
14	Auger	STL	1	.1703	.1703	
15	Auger Drive Shaft	STL	1	.0447	.0447	
16	Auger Carrier	Mg	1	.1072	.1072	
17	Upper Housing	Mg	1	.1669	.1669	
18	Lower Housing	Mg	1	.1732	.1732	
19	Soil Dump Door	Mg	2	.0076	.0152	
20	Door Sector Gear	Al	2	.0130	.0260	
21	Dump Door Spring	STL	4	.0022	.0088	
22	Support Shaft	STL	2	.0151	.0302	
23	8-32 x 1.12 Screw	STL	6	.0074	.0444	
24	Door Shaft	STL	2	.0053	.0106	
	Subtotal		68			2.9499
	Deployment Assy					
25	Boom	Mg	1	.0879	.0879	
26	Boom Support	Mg	1	.1268	.1268	
27	Bearing .187 Bore	STL	3	.0109	.0327	
28	Elevation Gear	STL	2	.0225	.0450	
29	Elevation Shaft	STL	1	.0204	.0204	
30	Worm Gear	STL	1	.0175	.0175	
31	Worm	STL	1	.0156	.0156	
32	Worm Shaft	STL	1	.0116	.0116	
33	Elevation Drive Motor	-	1	.7500	.7500	
34	Orientation Pulley	Mg	4	.0604	.2416	
35	Orientation Cable	STL	2	.0501	.1002	
36	Cable End FTG	STL	4	.0239	.0956	
37	Turnbuckle	STL	2	.0552	.1104	
38	Fixed Elevation Gear	STL	2	.0416	.0832	
39	Boom Support FTG	Mg	1	.1499	.1499	
40	Bearing 1.062 Bore	STL	2	.0526	.1052	
41	Gear 1.562 PD	Al	1	.0153	.0153	
42	Gear 1.125 PD	Al	1	.0084	.0084	
43	Azimuth Drive Motor	-	1	.7500	.7500	
44	Support Base	Mg	1	.0938	.0938	
	Subtotal		33			2.8611
	Total Sampler Assy		101			5.8110

3.5.7 MINIATURE ROTARY ROCK CRUSHER, E-7

This design was initially defined as a miniature jaw crusher which was later redefined as a miniature rotary rock crusher at the request of JPL. A small amount of work was expended on the jaw crusher in preliminary analysis of the crushing mechanism.

The effort expended on this mechanism consisted of making an estimate of the forces required to crush a pebble with a maximum diameter of 5 millimeters. Two basic mechanisms can be used to develop a high mechanical advantage. These are a differential screw drive and a toggle mechanism. At first glance, the toggle mechanism appears more desirable since the friction forces in the differential screw threads could be high. Some typical properties of three rock types are given in Table XXII.

TABLE XXII.

TYPICAL ROCK PROPERTIES

	Rock Type		
Property	Quartzite	Basalt	Granite
Poisson's ratio	.10	.25	.09-.20
Young's Modulus, psi	-	-	$5.8\text{-}8.7 \times 10^6$
Compressive strength, psi	$1.42\text{-}2.84 \times 10^4$	$2.84\text{-}4.98 \times 10^4$	$1.42\text{-}3.98 \times 10^4$
Tensile strength, psi	-	-	400-700
Shear strength, psi	-	-	2100-4300

Based on these values, a value for Young's modulus of 9×10^6 psi and a compressive strength of 50,000 psi, was assumed in the calculations. A maximum force of 1600 pounds was calculated to be required on the jaw to crush a disc 5 millimeters in diameter. A design jaw force of 2000 pounds was used in all subsequent calculations. The jaw deflection required was calculated based on a sphere being compressed between two flat plates sufficiently to develop the maximum compressive stress across the cross section at the major diameter of the sphere. This should give a conservative value for the required deflection to cause fracture since the maximum stress at the point of contact is 460,000 psi. Timoshenko in Volume II "Strength of Materials" on page 357 quotes a typical value of 530,000 for a crucible steel ball. An order of magnitude less could be expected from rock based on the relative strength values for rock and steel. A required deflection or jaw movement of .04 inches was obtained from these calculations.

The jaw crusher mechanism envisioned in these calculations is shown schematically in Figure 51.

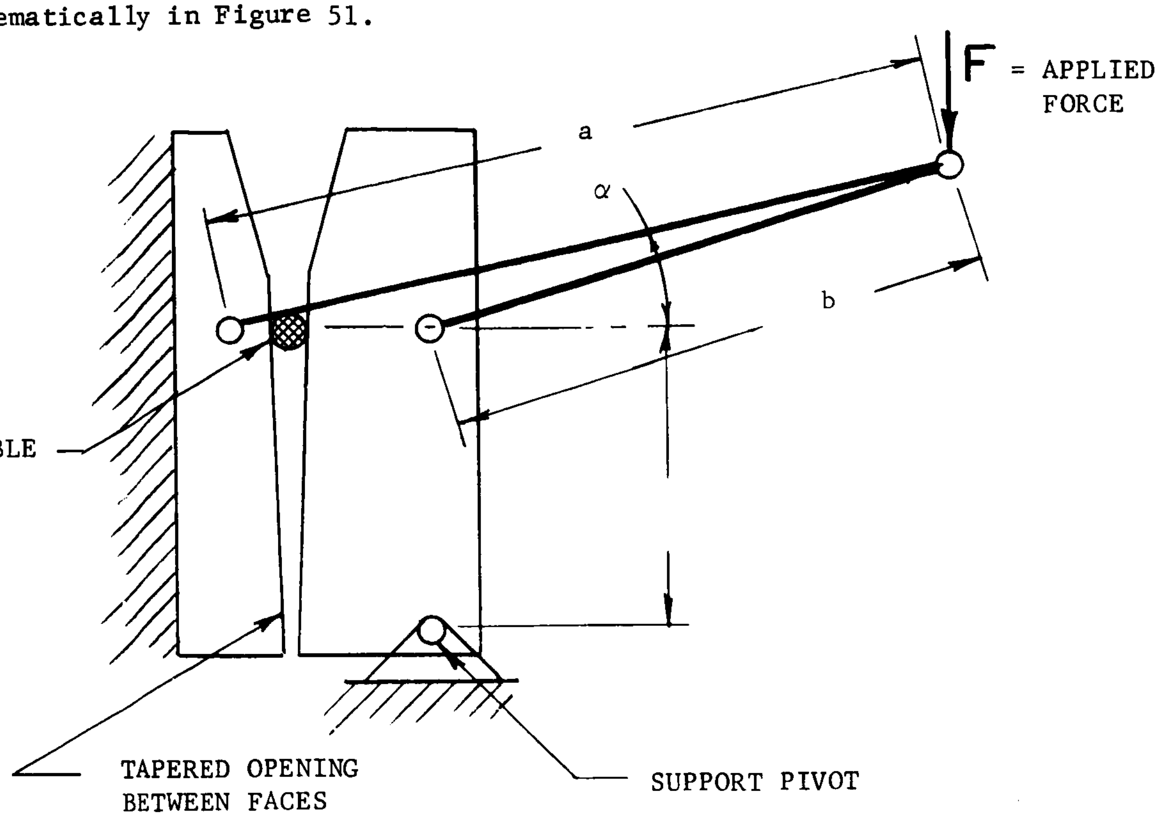


FIGURE 51. SCHEMATIC OF JAW CRUSHER MECHANISM

Equations were derived relating the required actuation force, the jaw force, and the geometry. Using these equations, the variation in required actuation force, F, and the required jaw movement, δ, were calculated for a set of values defining the geometry that might be typical of a miniature crusher.

These assumed values were a = 5 inches, b = 4 inches, c = 2 inches, and a jaw force of 2000 pounds. The variation of the required actuation force and jaw deflection with the toggle link angle α is given in Figure 52. It is seen that for the estimated properties of the pebble being crushed, an actuation force of 535 pounds and an initial toggle link angle of 14.4 degrees are required to produce a jaw movement of .04 inches with sufficient force to crush the pebble. It is also seen that reducing the required jaw

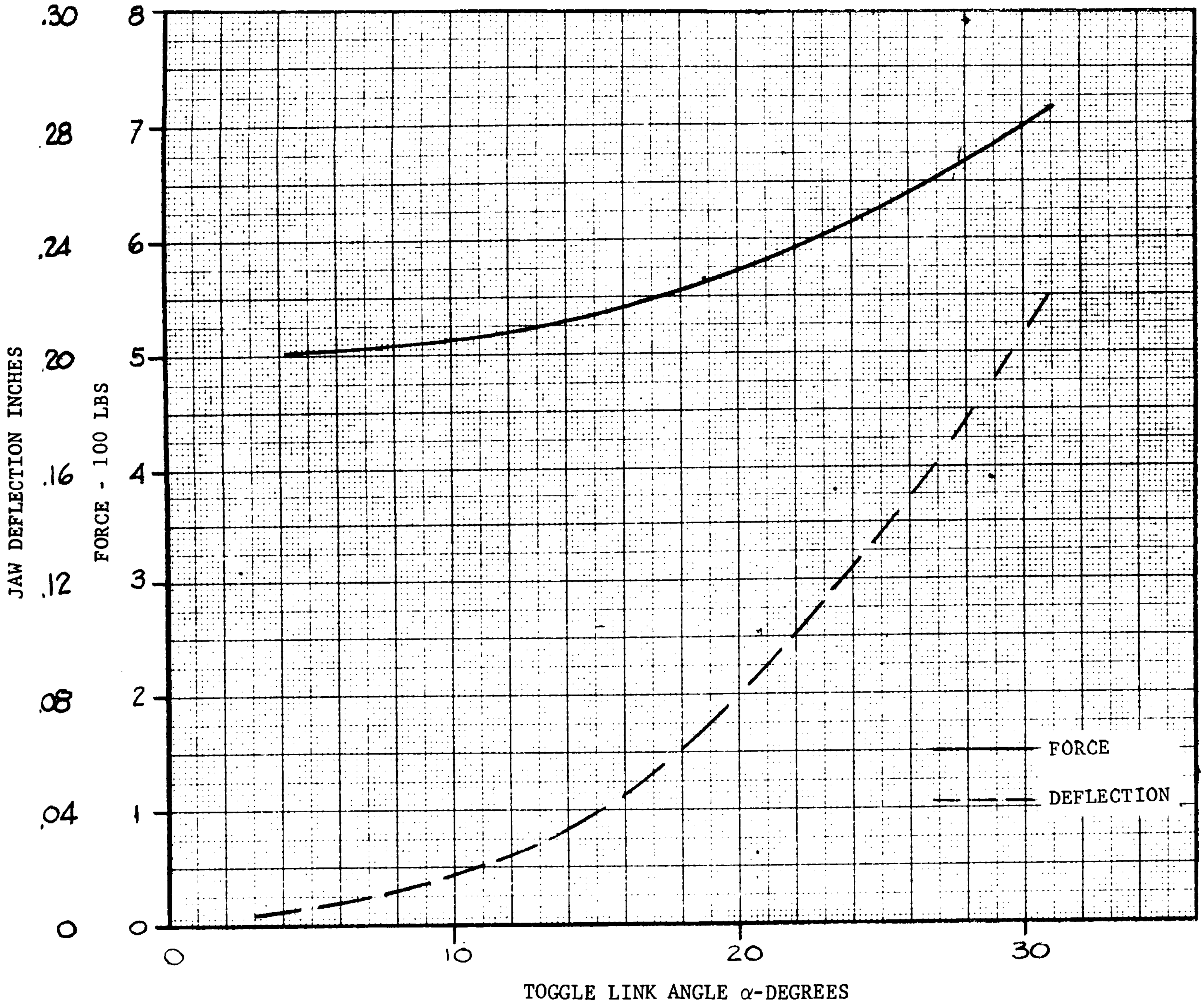


FIGURE 52. FORCE/DEFLECTION CHARACTERISTICS FOR JAW CRUSHER

deflection does not significantly reduce the required actuation force. The values obtained indicate such a mechanism is feasible; however, it should be pointed out that the assumed configuration does not represent an optimum. Other combinations of link-jaw geometry could produce more favorable relationships.

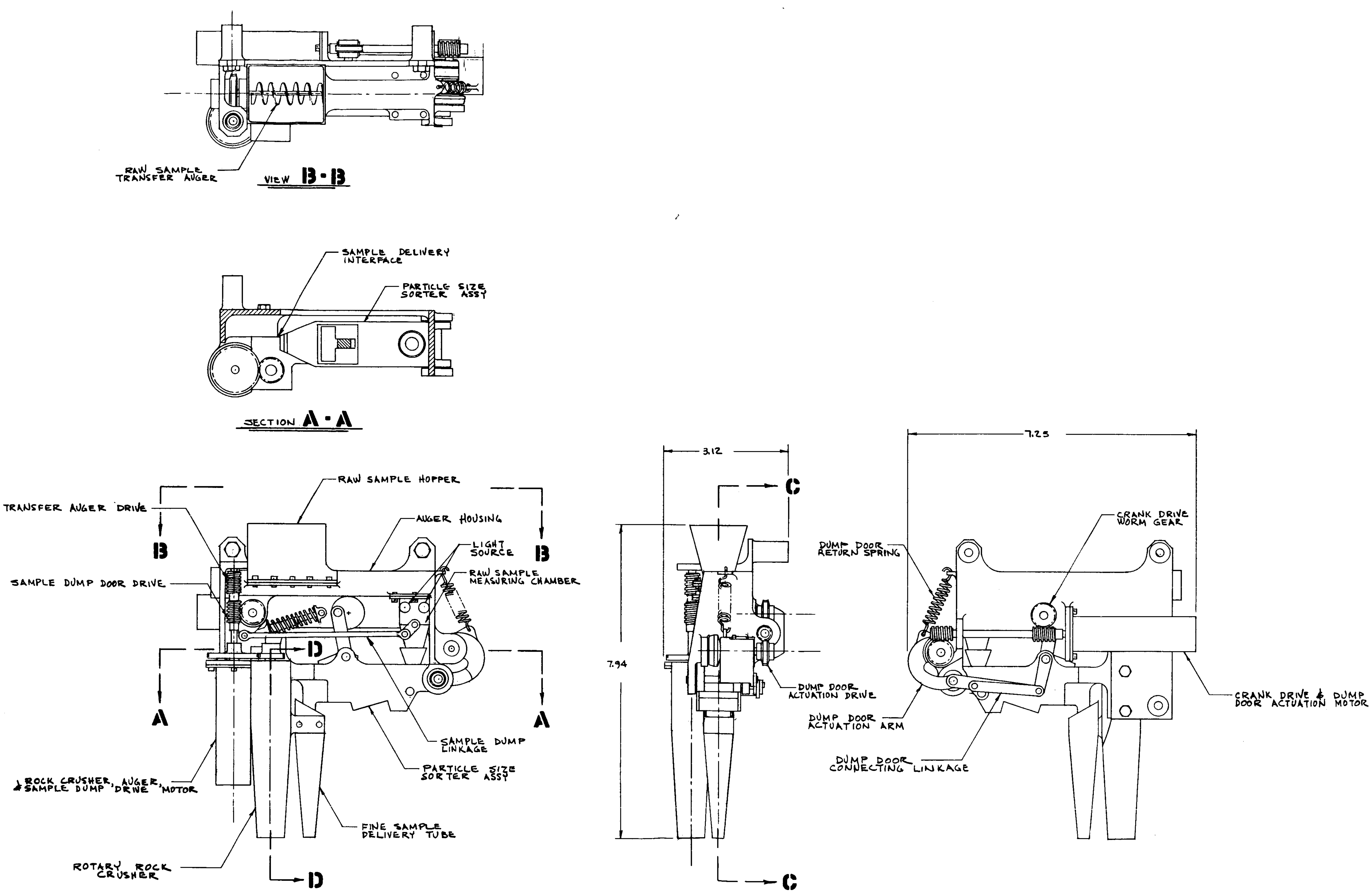
Since this initial work on the jaw crusher, the effort was redirected to pursue a design for a miniature rotary crusher based on a breadboard model built and tested by JPL. The design criteria given in Section 2.0 for this mechanism reflect this redirection. Initially it was thought that the jaw crusher would be less susceptible to jamming by any malleable metallic material. This can be prevented in the rotary crusher, which is much more compact, by separating material with high magnetic permeability before it enters the crusher. Thus, a preliminary separation of material fed to the crusher will catch and hold the magnetic particles, which are presumed to be metallic meteorites. It will also separate the fine material below 500 microns in size and bypass these around the crusher to keep from loading the crusher and reduce the production of the very fine particle size population to a minimum.

The design intent of this crusher is ideally to produce particles of a given mean grain size of 300 microns with as narrow a distribution of particles on either side of the mean as possible. In actual practice, it turns out that almost any rock crushing mechanism can be made to cut off rather sharply above some given size but that it is almost impossible to obtain a sharp cutoff in the fine particle size population. There is always a high production of fine material which tests conducted by JPL and others indicate to have a distribution which is nearly always the same regardless of the method used to crush the material. Thus, it is assumed in this design that the output of the rotary crusher will be subsequently sieved by another mechanism if it is desired to eliminate or reduce the fine population. In order to minimize the production of fine material, it was assumed that no naturally occurring fine material contained in the raw sample would be allowed to go through the crusher but would be bypassed around it. The rotary crusher, as developed in breadboard form by JPL, was designed to accept a maximum particle size of 4 to 5 millimeters in diameter. Since sampler mechanisms such as the backhoe or the soil auger are only partially selective in collecting a sample, the raw sample fed to the rock crusher could contain material somewhat larger; i.e., up to 8 to 10 millimeters in diameter. Thus, a design requirement was established that not only the fine material would be separated and bypassed around the rock crusher, but that all particles above 5 millimeters in diameter would be separated and discarded.

These requirements then dictated that some sort of metering and separating process be incorporated ahead of the rock crusher. It was decided to adapt the basic principle of an oscillating sieving mechanism, developed at JPL to perform part of the function of preparing sample slides for the petrographic microscope breadboard, to perform the particle size separation ahead of the rotary rock crusher. The general configuration of the prototype design generated for this mechanism is shown in Figure 53. The basic part of this design is a raw sample hopper which receives the sample delivered by the sampler mechanism. Running laterally out of the base of the hopper is a soil transfer auger sized to accept the largest particles contained in the raw sample. This auger is used to transfer raw sample at a controlled rate to a sample measuring chamber. A volumetric determination of the amount of sample transferred to the chamber is made by means of light sources and photo sensitive cells which scan across the chamber. This measured sample is then transferred into the receiving chamber of the oscillating particle size sorter where it separates the raw sample into fines with particle diameters less than 500 microns, the material to be crushed with particle sizes between 500 microns and 5 millimeters in diameter, and the residual material larger than 5 millimeters in diameter. A permanent magnet is incorporated in the particle size sorter to trap large metallic particles which may be malleable and prevent them from entering the crusher. The fine material separated from the sample is continuously delivered during the separation cycle to a fine sample delivery tube for collection and use if desired. At the termination of the sorting process, the material to be crushed is delivered to the rotary crusher and the residual material is discarded.

In order to more fully explain the details, reference is made to Figure 54 which is section C-C taken from Figure 53. Two drive motors are used to perform all the basic functions previously summarized. This is accomplished by the use of over-running clutches and motor rotation reversal to engage the appropriate mechanical drives.

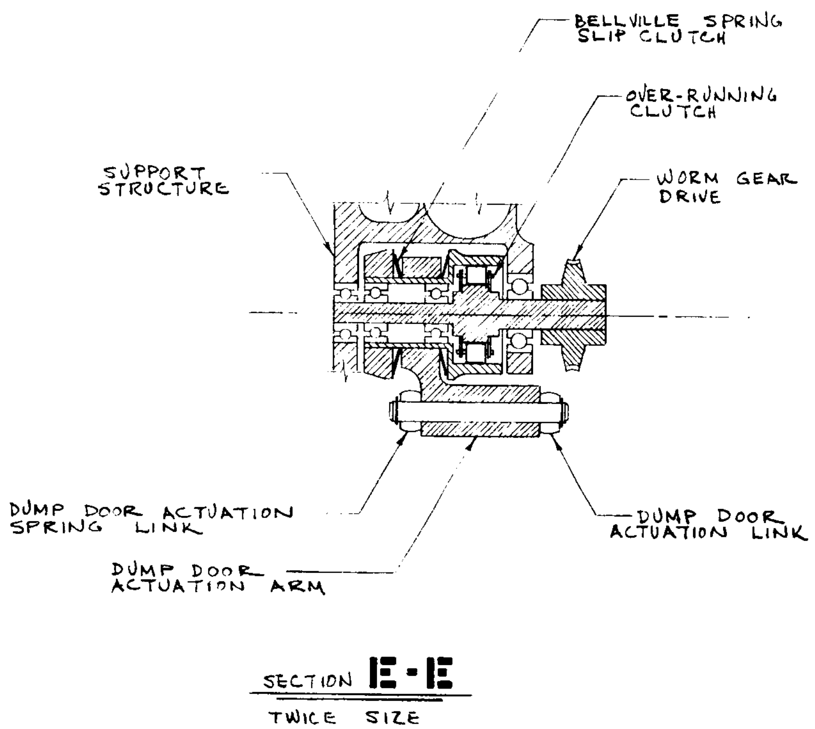
One motor is used to drive the crusher rotor, the raw sample transfer auger, and to hold the dump door to the sample measuring chamber closed. In this mode, the motor drives the crusher rotor in reverse while simultaneously driving the soil transfer auger through an over-running clutch. The dump door housed inside the sample measuring chamber is also driven to the closed position through another over-running clutch and slip clutch in combination. Section E-E of Figure 54 shows the details of this clutching arrangement. Thus, the over-running clutch mounted on the shaft from the worm gear drive picks up the clutch housing causing it to rotate with the shaft. This in turn carries the dump door actuation arm with it, closing the dump door in the measuring chamber and simultaneously compressing a return spring. Since the dump door actuation arm has a limited travel, it is mounted on the clutch housing shaft so it is free to rotate. The necessary torque required to hold the dump door closed is maintained by the slip clutch consisting of two Belleville springs acting between the dump door actuation arm and the over-running clutch housing.



FOLDOUT FRAME 1

FIGURE 53. MINIATURE ROTARY ROCK CRUSHER, E-7

FOLDOUT FRAME 2



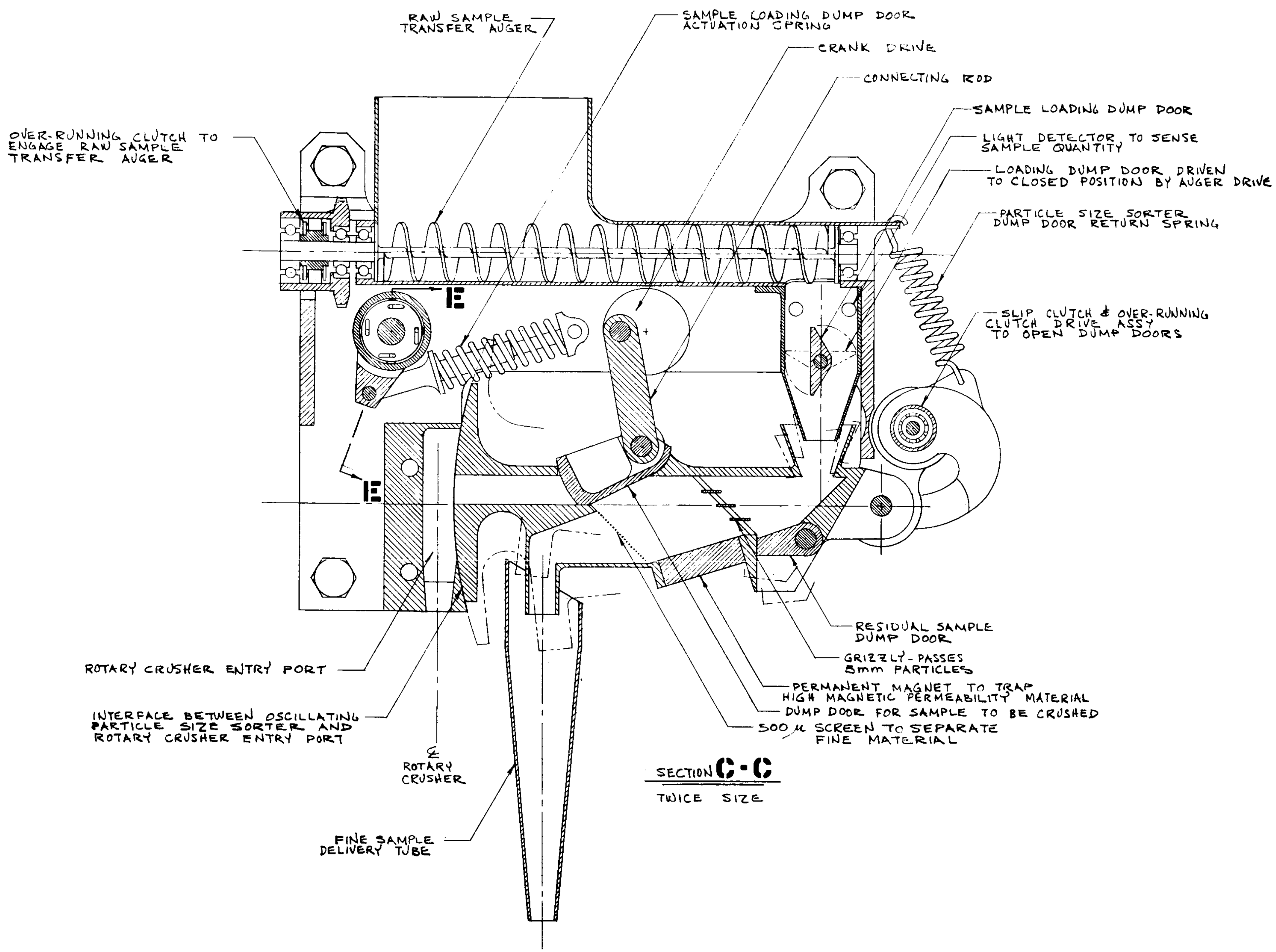


FIGURE 54. SECTIONAL VIEW OF PARTICLE SEPARATOR

FOLDOUT FRAME 1

FOLDOUT FRAME 2

Soil is transferred into the measuring chamber by the transfer auger and collects on top of the dump door. When the level of soil is high enough to intercept both light beams and cut off the output from the photo-sensitive cells, the crusher drive motor is reversed. This action stops rotation of the soil auger terminating raw sample transfer and also allows the return spring to bring the dump door actuation arm to its original position opening the sample dump door dropping the measured sample into the receiving chamber of the particle size separator.

At this point power is terminated on the crusher drive motor and turned on to the oscillating sieve drive motor. When this motor is driven in one direction it drives only the crank connected to the particle separator. When it is driven in the other direction it oscillates the separator and also actuates the dump doors through an over-running and slip clutch combination as described earlier for the actuation of the sample measuring chamber dump door. The particle size separator is basically a hollow rectangular body hinged at the right end as shown in section C-C of Figure 54. The oscillating crank connecting rod ties into the separator body at the center of percussion to minimize vibratory reactions at the hinge point. Within the separator body are two screens which have a mesh of the required size to separate the particles into the required cuts. These screens are inclined 45 degrees to the longitudinal axis of the separator so that as oscillation proceeds the vertical or normal velocity component of the screen can be utilized to aid in causing particles to pass through the screen. The centrifugal force component, a result of the rotational velocity about the hinge axis, imparts an axial velocity to the particles causing them to move away from the hinge axis. The fine material less than 500 microns in diameter is delivered continuously to the sample delivery tube during the sieving part of the separation cycle. The first coarse screen or grizzly excludes all particles greater than 5 millimeters in diameter. Thus, the chamber formed between these screens holds all particles between 500 microns and 5 millimeters in diameter. A permanent magnet mounted in the base of this chamber traps all material with a high magnetic permeability to prevent large malleable metallic particles from being delivered to the rock crusher. A permanent magnet is used for simplicity since the probability of acquiring this kind of material is very low resulting in a low rate of accumulation on the magnet.

After a finite period of oscillation the drive motor to the rock crusher is turned on with the direction of rotation such that only the crusher rotor is driven. The oscillating drive motor is then reversed engaging the linkage to the dump doors on the receiving chamber and the intermediate chamber containing the material to be crushed. The residual material is discarded and the material to be crushed is transported out the end of the particle separator into the entry passage to the rock crusher. Figure 55 shows sectional view D-D taken through the rotary crusher. The rotor configuration shown is essentially that used in the

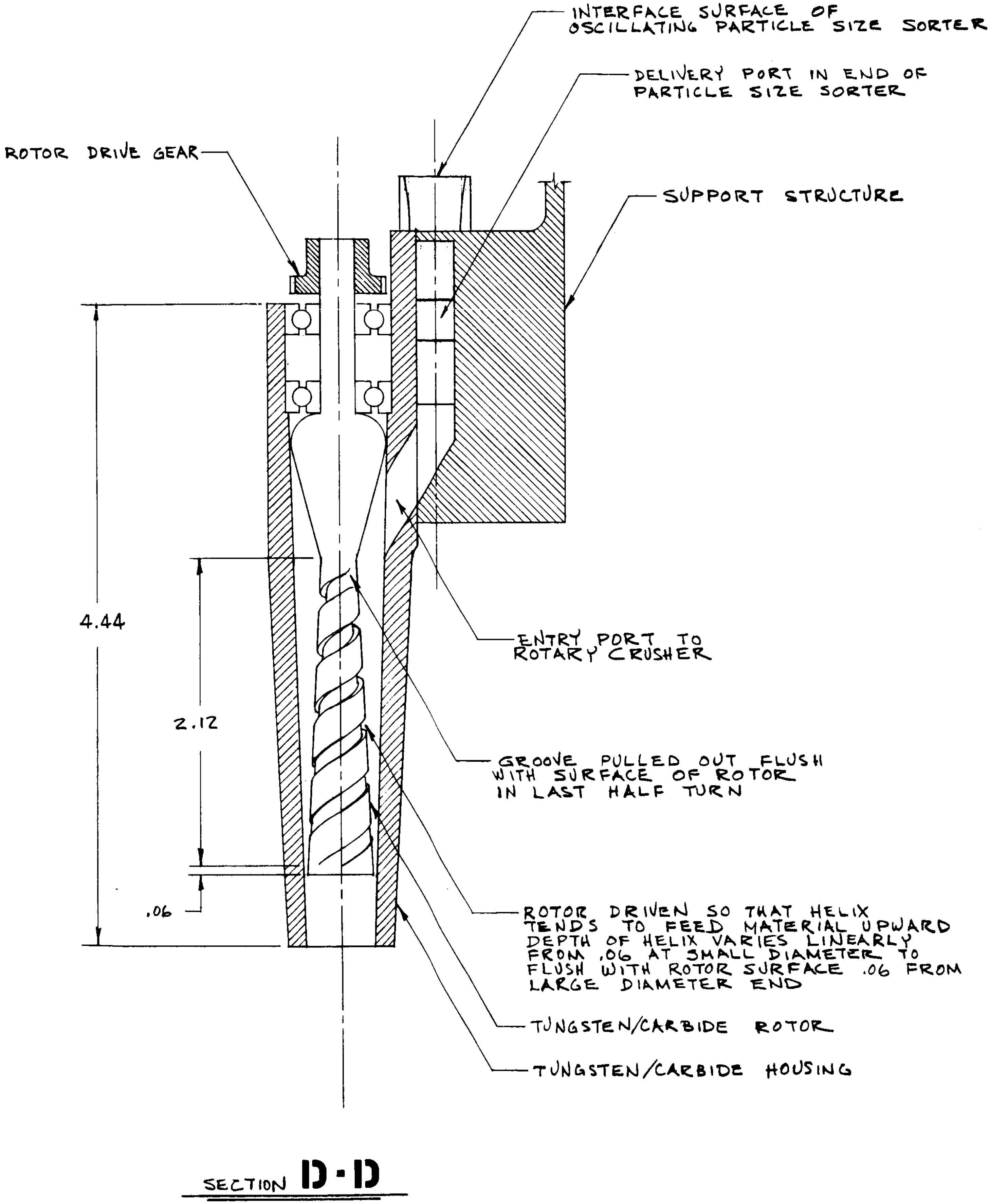


FIGURE 55. SECTION VIEW OF ROTARY ROCK CRUSHER

breadboard model built by JPL with some minor modifications. The helical spiral cut into the rotor is rotated in such a direction that it will tend to feed the material upward. It was found in preliminary testing at JPL that if this upward feed is not employed, the material tries to pass through the crusher too rapidly causing it to choke up. The helix machined into the breadboard rotor had a constant root diameter to simplify fabrication. It was found that it was necessary to fill the helix near the base of the rotor in order to obtain proper crushing action and feed. Based on these facts, this design is shown with a helical groove that varies linearly in depth from .03 inches at the top to flush with the rotor surface near the base.

Another modification made in the rotor design is the addition of the tapered section at the top of the rotor. This tapered section in conjunction with the entry port sizing is intended to limit the rate at which large particles enter the crusher and also to ensure a smooth entry by virtue of the rolling action of the rock between the rotor and the wall of the crusher housing.

The clearance between the rotor and the housing wall at the base or outlet determines the maximum size of the crushed material. Since the crusher is supported only at the upper end, the lower end is free to deflect laterally. Thus, the actual clearance between the rotor and the housing wall must be something less than the maximum size of particle desired. In this design, the upper cut off in particle size is 300 microns. This clearance can be sized to produce coarser material, but there is a limit to how small it can be made and still perform properly. This limit has been established as near the 300 micron size in tests. Such a crusher will produce an output with a mean grain size near 60 microns.

The block diagram for the power trains used in this mechanism are shown in Figure 56. Because of the low speed output of the crusher rotor drive motor and the high reduction associated with worm gears, the drive gear to the measuring chamber dump door actuation linkage is shown as helical gears with a one-to-one gear ratio in the block diagram. This gives a closing time for the dump door of 1.6 seconds. Because the rotation of the transfer auger is very slow, this door actuation is probably fast enough to prevent an appreciable amount of sample being delivered before the door is closed. Because of the high rotational rate of the oscillation drive motor, a helical gear drive is also used to drive the oscillation crank so that the input speed to the door actuation mechanism is minimized. These doors actuate in .084 seconds which is a reasonable time. In both cases the worm drives shown in Figure 53 can be easily replaced with helical gears.

Since the operational sequence for this mechanism has been completely defined in describing the operational features of the mechanism, no detailed sequence is given for this mechanism as was done for the preceding mechanisms. The weight statement for this mechanism is given in Table XXIII.

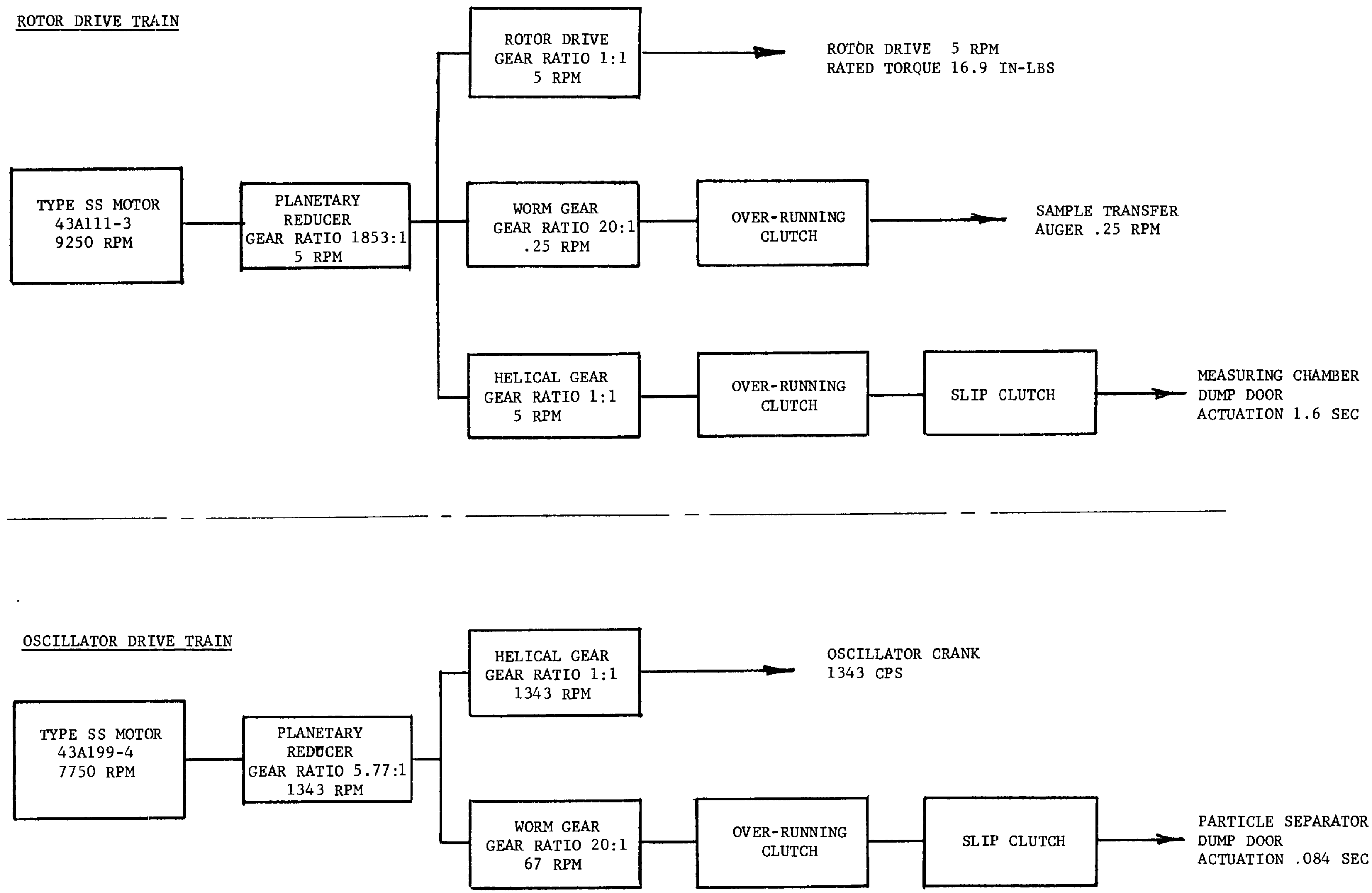


FIGURE 56. BLOCK DIAGRAM - MINIATURE ROTARY ROCK CRUSHER POWER TRAIN

TABLE XXIII

WEIGHT STATEMENT, MINIATURE ROTARY ROCK CRUSHER, E-7

Item No.	Item	Mtl	Qty	Weight /Part	Weight /Assy	Total
	Rotary Crusher Assy					
1	Rotor	STL	1	.1377	.1377	
2	Housing	STL	1	.4876	.4876	
3	Bearing .250 Bore	STL	2	.0168	.0336	
4	Gear .625 PD	STL	1	.0175	.0175	
5	Drive Motor	-	1	.7500	.7500	
6	Gear 1.250 PD	STL	1	.0275	.0275	
	Subtotal		7			1.4539
	Particle Sorter Assy					
7	Shaker Body	Mg	1	.0698	.0698	
8	Grizzly	STL	1	.0038	.0038	
9	Magnet	Al	1	.0552	.0552	
10	Door Shafts	STL	2	.0136	.0272	
11	Pivot Shaft	STL	1	.0121	.0121	
12	Bearing .187 Bore	STL	18	.0109	.1962	
13	Residual Dump Door	Mg	1	.0062	.0062	
14	Sample Dump Door	Mg	1	.0146	.0146	
15	Sample Delivery Tube	Mg	1	.0116	.0116	
16	Connecting Rod	Mg	1	.0037	.0037	
17	Crank	STL	1	.0206	.0206	
18	Over-running Clutch	STL	3	.0203	.0609	
19	Clutch Housing	STL	2	.0424	.0828	
20	Auger Clutch Housing	STL	1	.0277	.0277	
21	Auger Worm Gear	STL	1	.0253	.0253	
22	Worm	STL	4	.0156	.0624	
23	Worm Gear	STL	3	.0175	.0525	
24	Clutch Shaft	STL	2	.0163	.0326	
25	Belleville Springs	STL	4	.0025	.0100	
26	Sample Load Crank	Mg	1	.0259	.0259	
27	Link Shaft	STL	1	.0038	.0038	
28	Sample Dump Crank	Mg	1	.0381	.0381	
29	Link Pin	STL	3	.0019	.0057	
30	Dump Door Linkage	Mg	3	.0021	.0063	
31	Load Door Link	Mg	1	.0034	.0034	
32	Load Door Spring	STL	1	.0023	.0023	
33	Spring Link	STL	1	.0106	.0106	
34	Spring Link	STL	1	.0059	.0059	
35	Dump Door Spring	STL	1	.0023	.0023	
36	Load Door	Mg	1	.0020	.0020	
37	Load Door Shaft	STL	1	.0036	.0036	
38	Load Door Crank	Mg	1	.0012	.0012	
39	Load Chamber	Mg	1	.0112	.0112	
40	Support Structure	Mg	1	.3470	.3470	
41	Sample Hopper	Mg	1	.0178	.0178	
42	Auger	STL	1	.0582	.0582	
43	Oscillator Motor	-	1	.7500	.7500	
44	Worm Shaft	STL	1	.0272	.0272	
45	Worm Shaft	STL	1	.0155	.0155	
	Subtotal		73			2.1152
	Total Crusher Assy		80			3.5691

3.5.8 PARTICLE SIZE SORTER, E-8

Several concepts for performing this processing operation were generated under the fundamental assumption that the size sorting is intended to prepare the sample for use in analytical instruments such as the petrographic microscope or the X-ray diffractometer. On this basis, the particle size sorter has to separate the raw sample into three cuts. These were defined in the design criteria for this mechanism to be $d > 500\mu$, $125\mu < d < 500\mu$, and $d < 125\mu$. Two basic approaches are available to perform a size sorting operation. One is to pass the material through a set of screens, as is conventionally done. The other is to utilize the effect of terminal velocity variation with particle size to separate particles using a pneumatic flow system. A disadvantage of the latter system is that size sorting is not accomplished independently of the material density. In addition to these two approaches either batch processing or continuous flow processing can be used. On this basis, four design concepts were generated and are shown schematically in Figure 57.

Concept (1) is a recirculating closed flow pneumatic system. It was assumed that for the low atmospheric pressures existing on Mars that this would have to be a sealed and pressurized system to operate. In operation the raw soil sample is metered out of the hopper in a dispersed condition and falls into the central tube. The upward flow velocity in this tube is sufficiently large to entrain all particles 500 microns in diameter or less. These are carried through the blower into the annular outer return chamber. This chamber is sized to reduce the flow velocity to a very low value allowing the particles to settle to the bottom of the chamber. Only the very small particles obeying Stoke's Law remain entrained in the flow and continue to recirculate as long as the processing continues. When all the raw sample has been processed, the coarse cut and intermediate cuts are removed or dumped. A brush traversing the bottom of the annular chamber sweeps the intermediate cut to the dump port. After removing these two cuts, the exit ports are closed and the fine material is allowed to settle out, the majority of which will be on the floor of the annular chamber because of the larger area involved. This fine cut can then be swept to another dump port and collected. It should be noted that the pressurized atmosphere of the system should be maintained during the dump or collection of the various cuts of sample.

Concept (2) is a conventional sieving system vibrated in a vertical direction. The sieves are mounted in a cylindrical body which has a close sliding fit inside a housing. The raw sample is introduced into the top chamber and the lid on the housing is closed. Vertical oscillation is then initiated and continued until sieving is complete. The housing is then pivoted to a horizontal position to transfer or dump the various cuts in each of the sieving chambers. The oscillating cylindrical body has ports cut into the wall which are blocked or closed by the close fitting housing during the sieving operation. To effect the sample dump, the

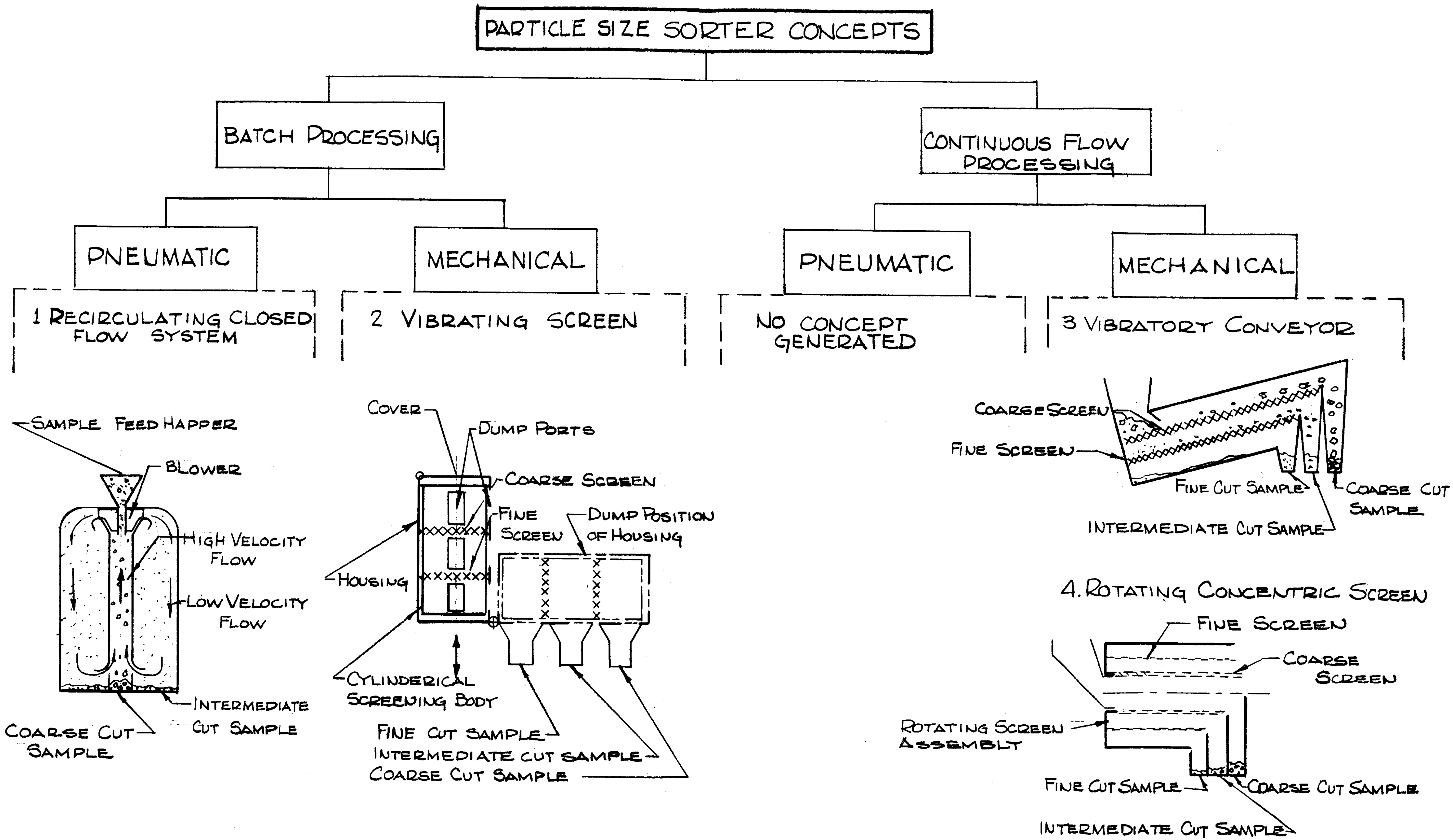


FIGURE 57. PRELIMINARY PARTICLE SIZE SEPARATION CONCEPTS

FOLDOUT FRAME 1

FOLDOUT FRAME 2

cylindrical body is rotated inside the horizontal housing until the ports align themselves with corresponding ports in the housing. Oscillation of the cylindrical body continues during this operation causing the various cuts of sample to be shaken into the receiving hoppers or sample containers. The assembly is then returned to the vertical position for the next sieving cycle.

Concept (3) utilizes the concept of a vibrating conveyor to simultaneously provide the agitation required for sieving and the energy necessary to move the sample along the screen. The basic mechanism consists of a coarse screen mounted over a fine screen inside a rectangular container which is vibrated both vertically and horizontally in harmonic motion. The raw sample is metered onto the coarse screen which is then conveyed toward the higher end. As the sample is transported along the screen the finer material falls through. The completeness of sieving is determined by the dwell time on the screen. The various cuts are ultimately transported off the higher end of the screens or lower surface into their respective sample containers.

Concept (4) utilizes a series of concentric screens and an outer cylinder which is slowly rotated. The raw sample is metered into the inner cylindrical screen which is the coarse screen. The entire assembly is sloped at some slight downward angle so that as the sample is tumbled inside the screens, it not only falls through the screen but progresses along it toward the dump end where it falls into the appropriate containers. A wiper blade running inside the outer cylinder would prevent the fine sample from tumbling and aid in causing it to transport to the delivery end. An advantage of this system is that the surface area of the fine screen is much larger than the coarse screen thereby making more mesh openings available for the fine material. This should improve the sieving rate of the finer material which is known to become more difficult as the mesh size decreases. A disadvantage of this concept is that it is probably sensitive to orientation which might require a gimbal system to provide final orientation with respect to the local vertical.

These initial concepts were generated early in the task. After a review by JPL, it was indicated that this mechanism should be based on the existing breadboard particle size sorter used with the petrographic microscope being developed at JPL. The basic design approach used for this mechanism was to retain the fundamental features of the JPL breadboard design using screens canted at a 45 degree angle. Oscillation of the sieving chamber about a fixed axis produces not only vertical motion but also provides a centrifugal acceleration which will cause the particles to drift towards the sieving screen. The prime intent of this mechanization was to mechanize it in a more compact form and to expand its capability to serve other instruments as well as the petrographic microscope. To accomplish the latter goal, additional doors were incorporated into the bottom of each of the sieving chambers as well as those existing in the top which serve the petrographic microscope. This device is controlled through a mechanical programmer using cam actuation. This program can be subdivided into two subprograms, one which controls a sieving and dispensing cycle for the petrographic microscope and the other cycle controls a general purpose sieving cycle to serve some other instrument. The sequence of events for these cycles are given in Table XXIV. The proper cam program is engaged by means of over-running clutches depending on the direction of rotation of the drive motor.

After a review by JPL, it was indicated that the need to service the petrographic microscope should be deleted making this only a general purpose particle size sorter. Thus, the A part of the sequencing program can be deleted and the mechanism considerably simplified by the elimination of processing steps. Two approaches were used for the soil feed mechanism to load the sample weighing cup. One method used a vibratory conveyor to feed the sample to the cup. The other used an auger to feed the sample to the cup. JPL indicated that the vibratory feed mechanism would probably be too sensitive to orientation and that the auger feed appeared to be preferable. The auger feed was used in the final design of this mechanism. The general configuration of this mechanism is shown in Figure 58.

The fundamental features of this design are similar in many respects to the particle size separator used with the miniature rotary rock crusher previously discussed. Two drive motors are used, one to drive the oscillation mechanism and the other to drive the soil transfer auger and cam actuation mechanism that operates the dump doors. These motors are arranged parallel to the soil transfer auger along each side of it. The motors and auger structure are located above the oscillating particle size separator body. To the right of the separator body is located a series of cams that are used to actuate the raw sample measuring dump doors and the individual doors to each of the chambers for the various particle size cuts as defined in the design criteria, Section 2.8. Sequential opening of the sample dump doors was incorporated in this design to provide greater versatility in operation although it represents a more complex mechanization.

In operation, the cam and auger drive motor is started with a direction of rotation such that an over-running clutch engages the auger. The cams are not driven when the auger is driven since they are connected to the drive motor through another over-running clutch that engages during the opposite sense of rotation of the drive motor. Soil is fed into the sample measuring chamber which uses the same light system, as was used for the rotary rock crusher, to sense the amount of soil in the chamber. When the amount of soil in the chamber is sufficient to cut off the output from the photo-sensitive cells, the polarity is reversed to the auger/cam drive motor reversing its rotation. When this happens the transfer auger

TABLE XXIV

PARTICLE SIZE SORTER SEQUENCE OF EVENTS

A. PETROGRAPHIC MICROSCOPE SIEVING CYCLE

1. Raw sample hopper is loaded from sampling mechanism, by external comand.
2. External command actuates soil feed mechanism to transfer sample to the weighing cup.
3. Weight sensor provides input to terminate soil feed when 1 gram is in cup and actuates main drive motor to start cam program and sieve oscillation.
4. Cam opens weighing cup dump doors dropping sample into sieve mechanism.
5. Cam closes dump doors (20 seconds).
6. Sieving cycle continues for 2 minutes.
7. Cam opens upper fine cut dump port and holds it open for 10 seconds to effect sample transfer to slide.
8. Cam closes upper fine cut dump port (20 seconds).
9. Cam opens upper intermediate cut dump port and holds it open for 10 seconds to effect sample transfer to slide.
10. Cam closes upper intermediate cut dump port (20 seconds).
11. Cam actuates switch to reverse polarity to main drive motor. (This releases petrographic microscope sieving program cams and engages general sieving program cams.)
12. Cam actuates dump port to transfer residue out of sieving mechanism.
13. Cam closes dump port and terminates operation.

B. GENERAL PURPOSE SIEVING CYCLE

1. External command actuates soil feed mechanism to transfer sample to weighing cup.
2. Weight sensor provides input to terminate soil feed when 5 grams are in cup and actuates main drive motor to start cam program and sieve oscillation.
3. Cam opens weighing cup dump doors dropping sample into sieve mechanism.
4. Cam closes dump doors (20 seconds).
5. Sieving cycle continues for 2 minutes.
6. Cam opens lower fine cut dump port and holds it open 10 seconds to effect sample transfer to receiving container.
7. Cam closes lower fine cut dump port (20 seconds).
8. Cam opens lower intermediate cut dump port and holds it open for 10 seconds to effect sample transfer to receiving container.
9. Cam closes lower intermediate cut dump port (20 seconds).
10. Cam actuates dump port to transfer residue out of sieving mechanism.
11. Cam closes dump port and terminates operation.

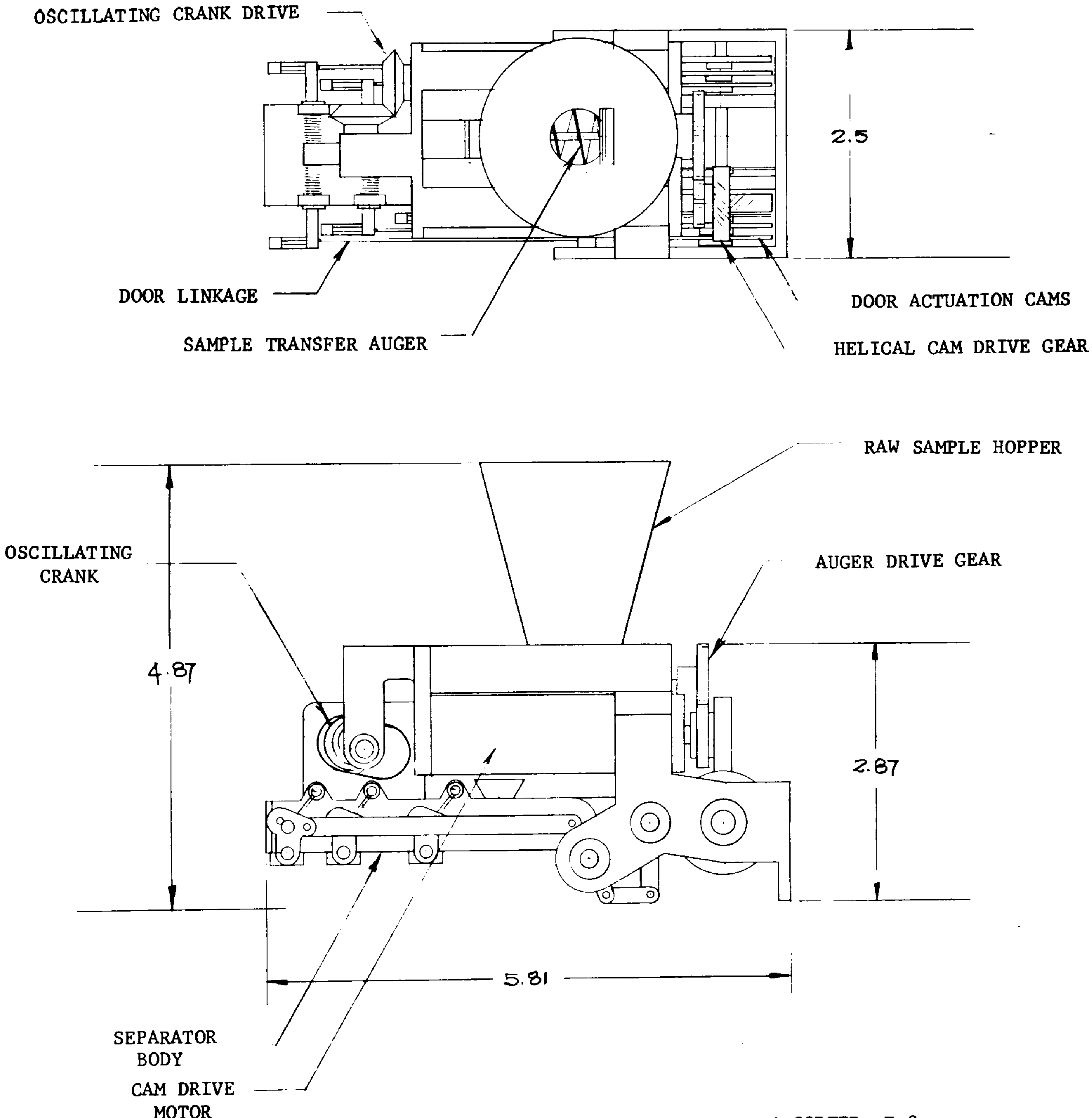


FIGURE 58. PARTICLE SIZE SORTER, E-8

is released and the cams begin to be driven. One cam located at the center line of the sampler then actuates the sample measuring chamber dump doors and starts the oscillating drive motor. This cam linkage mechanism is shown in Figure 59. The cam is designed to drop the cam follower suddenly to obtain fast actuation of the doors. The power to drive the linkage and cam follower is provided by a spring. This arrangement also makes it possible to use a gentle rise on the cam to lift the cam follower since speed is not essential in closing the doors. Sample separation is then continued for two minutes at which time another cam actuates the dump door to the fine particle size cut, 50 microns and less in diameter. The dump door is held open for 10 to 15 seconds and is then closed and the next door, for the particle size cut with diameters between 50 and 300 microns, is opened. The same sequence is followed to open the dump door to the residual sample with particle diameters greater than 300 microns. It is noted at this point that the dump doors have been located in the bottom of the oscillating separator rather than in the top, as was done on the breadboard built by JPL. This was done since the delivery requirements for the petrographic microscope were uniquely determined by the interfacing mechanisms. This mechanism is intended as a general purpose device and as such will probably deliver the sample to cups or trays similar to those used by the X-ray diffractometer or the α-scattering spectrometer.

The linkage mechanisms to the sample delivery dump doors are shown in Figure 60. The basic cam and spring driven actuating link is similar to that used for the raw sample dump doors. An additional feature is incorporated in the manner of driving the dump doors to the open position as shown in the schematic and freebody diagram in Figure 60. The door is essentially supported by a toggle linkage which is spring loaded to a position which pulls the door down against the body of the separator. This guarantees that a tight fit exists between the door and the separator body to prevent loss of sample during the oscillation cycle. The actuation link is tied to the end of the link which attaches to the door. When the actuation force P is applied the other link,which is spring loaded, it resists movement causing the door to try to swing against the stop on the separator housing producing the reaction R_1. Thus, the initial motion of the toggle linkage holds the door against the stop but because the linkage is approaching dead center the center distance between the door attach point and the spring loaded link attach point increases. This causes the door to be lifted off of the separator housing. When the toggle linkage reaches dead center, the two links come together at a stop causing them to lock together. From this point the locked linkage acts as a rigid link rotating about the support point of the spring loaded link. This causes the door to be pulled to one side removing all obstructions to the opening allowing the sample to be delivered. This action is reversed during the closing of the door. The primary advantage of this mechanism is that it is quick acting, involves no sliding motion, and acts to pull the door down snugly on the housing making a good seal.

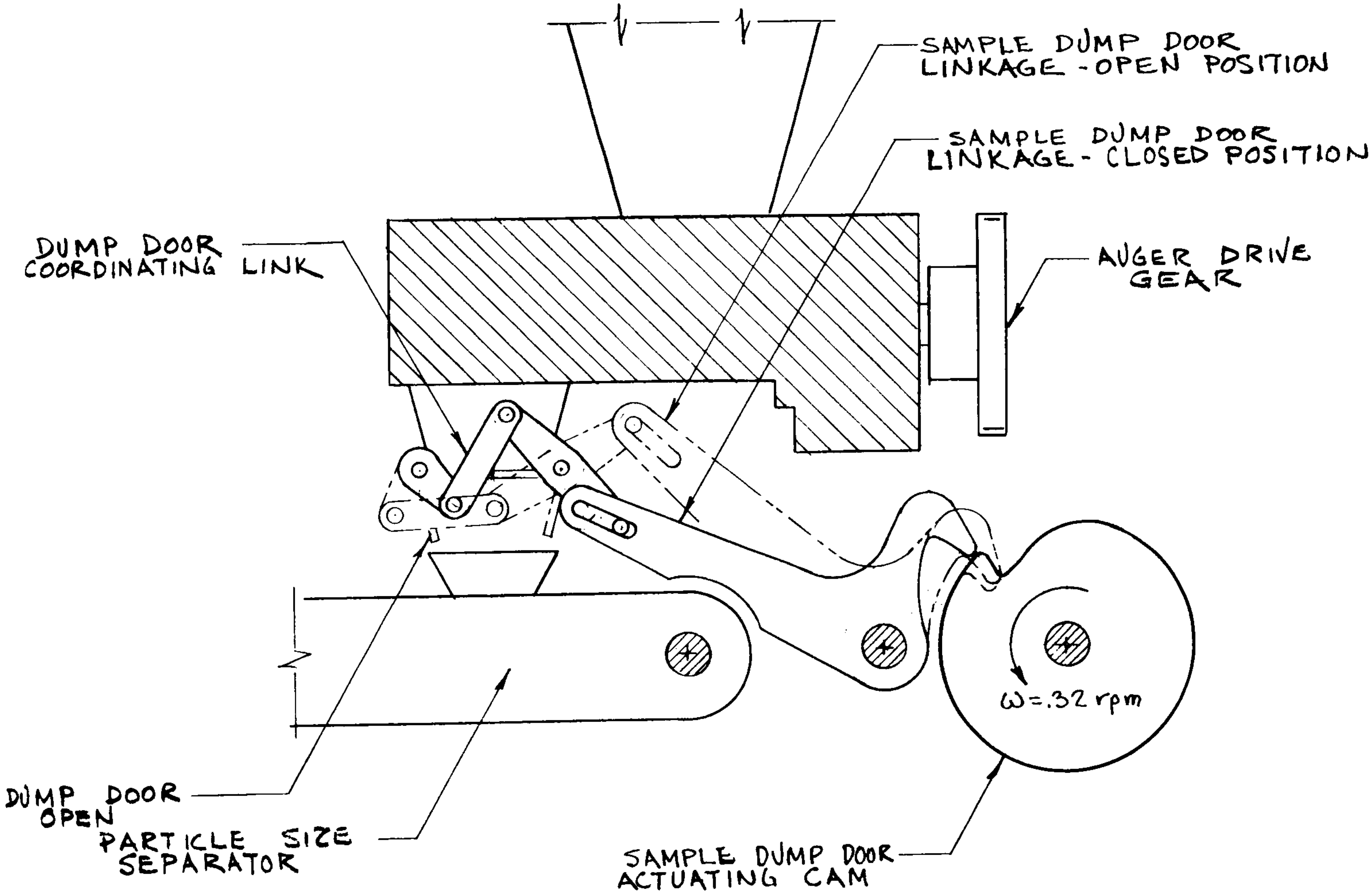


FIGURE 59. RAW SAMPLE DUMP DOOR ACTUATION LINKAGE

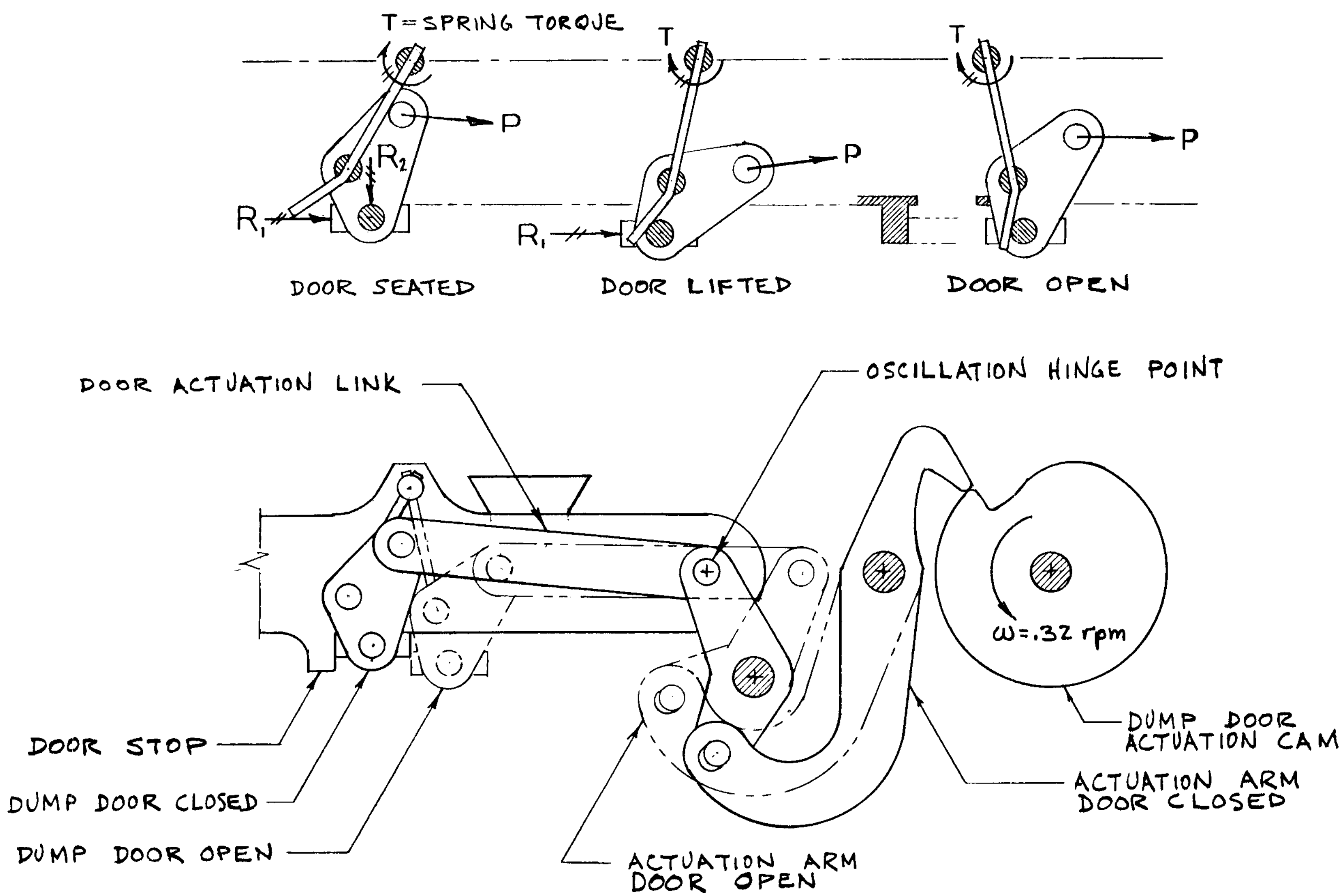


FIGURE 60. SAMPLE DELIVERY DUMP DOOR ACTUATION LINKAGE

Since the downward sample delivery is used, the separating screens were mounted in an opposite sense to those in the breadboard model. This was done to take advantage of the downward velocity component imparted to particles impacting the screen during oscillation which should aid delivery of the sample out of the chamber. The internal detail of this chamber is shown in Figure 61.

Another feature used in this design in order to make it more compact, was the elimination of the connecting rod between the oscillating drive crank and the particle size separator body. The crank pin has a ball bearing mounted on it which runs in a slot machined into the structure attached to the separator body. In effect, this is a Scotch crosshead drive mechanism which also produces harmonic motion response of the oscillating body. As in the case of the connecting rod for the separator on the rotary rock crusher, this drive is located at the center of percussion to minimize vibratory reactions at the hinge axis.

The block diagram describing the power train for this mechanism is shown in Figure 62. The total time to complete one cycle or cam rotation is 3.15 minutes. If a two minute shake cycle and 10 second door opening cycle are used, this leaves 29 seconds which can be utilized as dwell times between delivery cycles if desired.

The operational sequence for this mechanism is given in Table XXV. The weight statement for this mechanism is given in Table XXVI.

TABLE XXV

PARTICLE SIZE SORTER OPERATIONAL SEQUENCE

1. Activate auger drive motor to start raw sample transfer.
2. Sense volume of sample transferred to sample measuring chamber. When output of photo-sensitive cell cuts off, reverse polarity to auger drive motor.
3. Cam drive starts actuating sample dump door.
4. Activate oscillating drive motor and continue oscillation for 2 minutes.
5. Actuate sample delivery dump door to fine cut sample. Hold open 10 seconds and then close.
6. Actuate sample delivery dump door to intermediate sample. Hold open 10 seconds and close.
7. Actuate residual sample dump doors. Hold open for 10 seconds and then close.
8. Terminate cycle by turning off both drive motors. This is accomplished with the same cam that initiates the oscillation cycle.

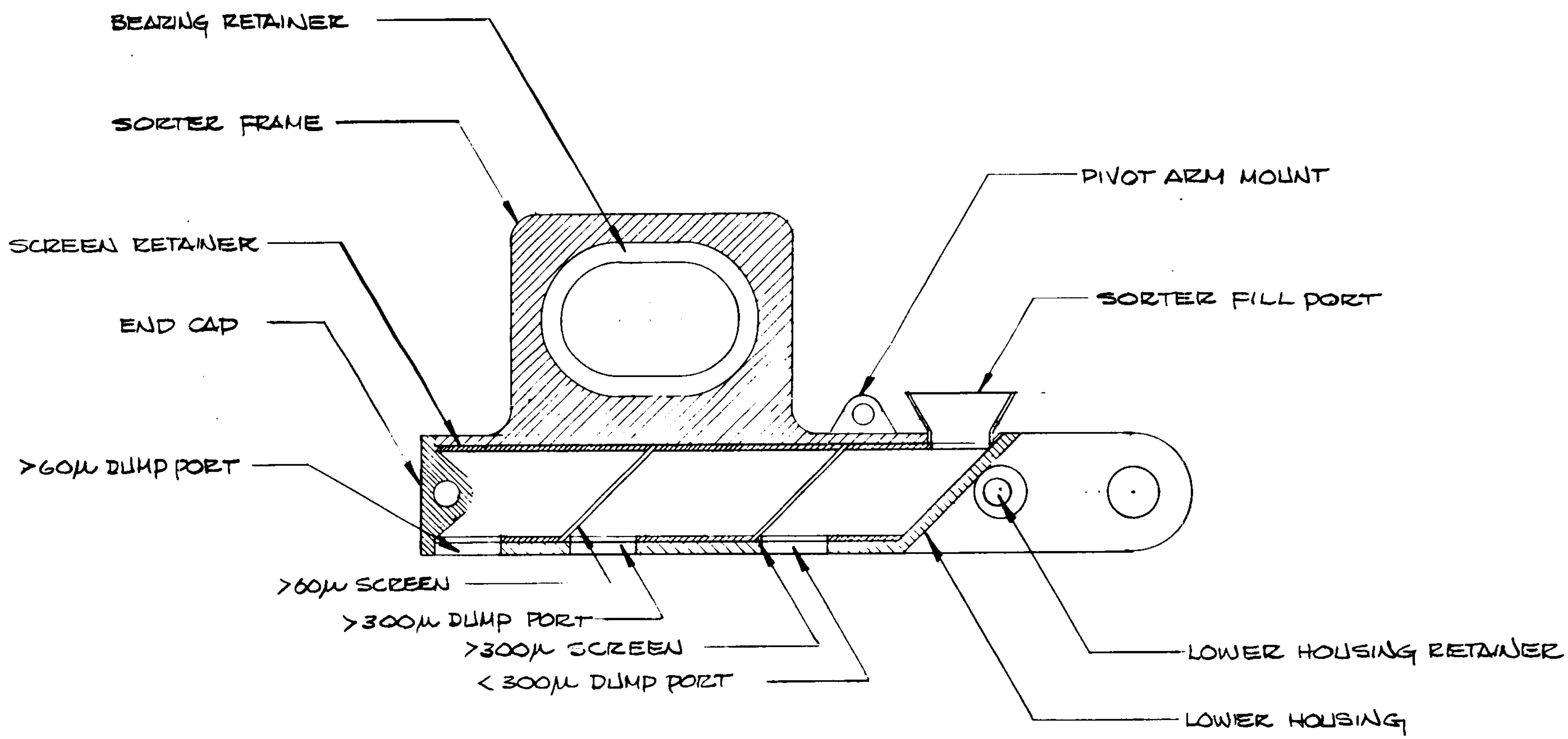


FIGURE 61. INTERNAL DETAIL OF PARTICLE SIZE SORTER

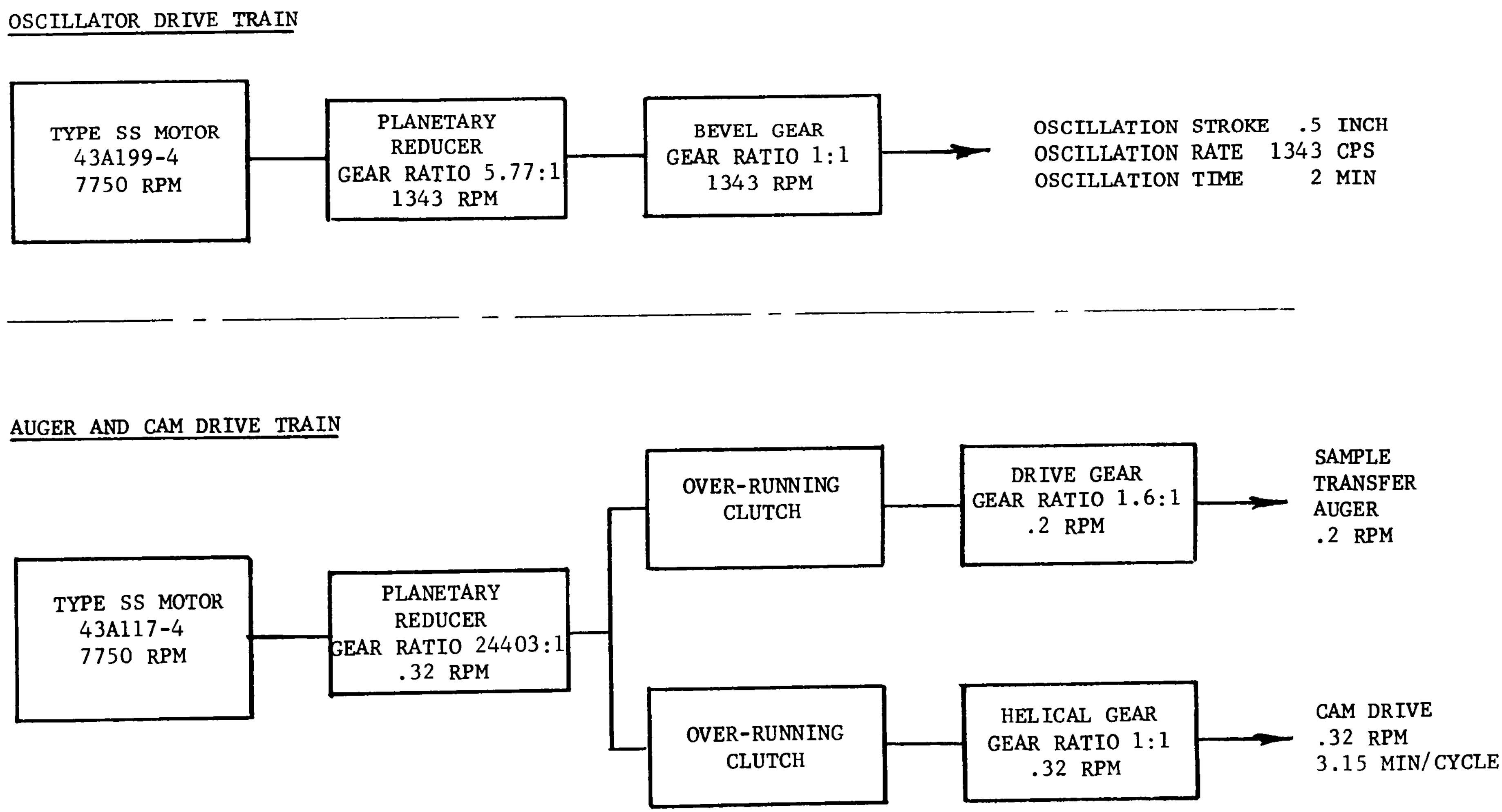


FIGURE 62. BLOCK DIAGRAM - PARTICLE SIZE SORTER POWER TRAIN

TABLE XXVI

WEIGHT STATEMENT, SAMPLE SIZE SORTER, E-8

Item No.	Item	Mtl	Qty	Weight /Part	Weight /Assy	Total
	Sorter Assy					
11	Sorter Frame	Mg	1	.0425	.0425	
22	End Cap	Mg	1	.0114	.0114	
3	Fill Port	Mg	1	.0023	.0023	
4	Screen Retainer	STL	1	.0175	.0175	
5	> 60μ Screen	STL	1	.0100	.0100	
6	> 300μ Screen	STL	1	.0100	.0100	
7	Lower Housing	Mg	1	.0120	.0120	
8	Screw	STL	4	.001	.004	
9	Retainer	Al	2	.0001	.0002	
10	Pivot Arm	STL	6	.0011	.0066	
	Subtotal		19			.1165
	Port Assy					
11	Arm	Al	6	.0029	.0174	
12	Cover	Mg	6	.0052	.0312	
13	Shaft	STL	3		.0175	
14	Spring	STL	6	.002	.0120	
15	Link < 60μ	Al	2	.003	.006	
16	Link < 300μ	Al	2	.0025	.005	
17	Link > 300μ	Al	2	.0015	.003	
	Subtotal		27			.0921
	Auger/Hopper Assy					
18	Hopper	Mg	1	.0126	.0126	
19	Auger	STL	1	.023	.023	
20	PE Cells	-	2			
21	Dump Doors	STL	2	.0016	.0032	
22	End Bushing	Al	1	.0014	.0014	
23	Bearing	STL	1	.0009	.0009	
24	Light Source		2			
25	Clutch	STL	2	.029	.058	
26	Bushing	Al	1	.010	.010	
27	Dust Seat	Mg	1	.0005	.0005	
28	Pin	STL	2	.0004	.0008	
29	Bearing	STL	1	.006	.006	
30	Drive Shaft	STL	1	.001	.001	
31	Retainer	STL	1	.0007	.0007	
	Subtotal		19			.1181
	Door Link Assy					
32	Link A	Al	1	.0001	.0001	
33	Link B	Al	1	.0003	.0003	
34	Link C	Al	1	.0003	.0003	
35	Cam Follower	Al	1	.0018	.0018	
36	Door Cam	Al	1	.0018	.0018	
	Subtotal		5			.0025
	Frame Assy					
37	Motor		2	.552	1.1040	
38	Auger Pinion	Al	1	.010	.010	
39	Auger Gear	STL	1	.0259	.0259	
40	Cam Drive Gear	STL	1	.0414	.0414	
41	Cam Drive Pinion	STL	1	.0609	.0609	
42	Port Cams	STL	6	.0169	.1014	
43	Cam Support Shaft	Al	1	.0116	.0116	
44	Cam Follower Shaft	Al	1	.0116	.0116	
45	Actuating Link	Al	6	.0008	.0048	
46	Pivot Arm	STL	6	.0042	.0252	
47	Cam Follower	Al	6	.0043	.0258	
48	OSC Bearing	STL	1	.0264	.0264	
49	Frame	Mg	1	.1768	.1768	
50	Bearing	STL	6	.014	.0840	
61	Sorter Pivot Mount	Mg	2	.0124	.0248	
52	Bevel Gear	STL	2	.0285	.0570	
53	OSC Mtg Shaft	STL	1	.0025	.0025	
	Subtotal		45			1.794
	Total Sampler Assy		115			2.123

SECTION 4

CONCLUSIONS

This was primarily a design task to develop engineering prototype designs on a variety of soil sampling and soil sample processing mechanisms, each with its own specific criteria. No mission constraints or requirements were identified with these mechanisms. On this basis, it is difficult to present any meaningful conclusions and no attempt is made to order the mechanisms in terms of relative merit or importance. These designs do provide some insight into the probable complexity, size, or weight that might be associated with mechanisms that could be candidates for inclusion into an unmanned planetary payload. These parameters are summarized for a quick overall view in Table XXVII. The number of items gives a measure of the different kinds of parts required which combined with the total number of parts in the assembly allows some sort of qualitative judgment to be made of the complexity of the mechanism and indirectly the potential reliability of the mechanism. The importance of weight and stowed volume can only be assessed in terms of the mission requirements and constraints in the final analysis. In a general sense, mechanisms E-4 through E-8 appear to be light enough and compact enough to warrant inclusion in an early payload.

TABLE XXVII

SUMMARY OF DESIGN CHARACTERISTICS

	Prototype Mechanism	No. of Items	No. of Parts	Weight lbs	Stowed Vol. in.^{3}
E-1	Uncased Rotary/Impact Drill Sampler	65	139	12.4	598
E-2	Cased Rotary/Impact Drill Sampler	74	157	13.8	598
E-3	Conical Abrading Sieve Cone Sampler	63	122	8.4	635
E-4	Helical Conveyor Simple Particulate Sampler	26	30	1.9	59
E-5	Backhoe Sampler	59	130	6.4	241
E-6	Soil Auger Sampler	44	101	5.8	240
E-7	Miniature Rotary Rock Crusher	45	80	3.6	179
E-8	Particle Size Sorter	53	115	2.1	71