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DESIGN AND DEVELOPMENT OF A CONICAL BORING DEVICE TO ENLARGE A PILOT HOLE THROUGH ROCK

Dikrun DerMardercsian

Foster-Miller Associates, Incorporated

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13. ABSTRACT

This report summarizes Foster-Miller Associates effort completed during the first half of the second year of a proposed three year program to design and develop a conical, self-advancing and self-rotating boring machine. The conical borer will use a proven and energy effective mechanical fragmentation system - roller cutters - and will operate as a reaming device to enlarge an existing 8 3/4 inch pilot hole to a final 38 inch bore.

The general research approach has been to continue the development efforts begun under Contract H021044 during which a prototype single stage conical borer was fabricated and "proof" tested.

The scope of work for the current program has been divided into four phases:

- Phase I - Test and Cutter Refinement;
- Phase II - Final Design of the Conical Borer;
- Phase III - Mucking Studies; and
- Phase IV - Partial Fabrication of the Conical Borer.

Detailed testing and refinement of the cutting structure of the prototype borer under Phase I has been completed. The results of this research showed

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	ROLE	WT	ROLE	WT	ROLE	WT
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Conical Borer						
Rapid Excavation						
Mine Rescue						

Ib

a plexiglass shell. The model will be rotatable to ensure close simulation of the flow of air and cuttings around the conical borer.

No work has been done on the partial fabrication of hardware under Phase IV.

The basic conclusions reached thus far in the program are as follows:

1. The experimental program has confirmed the main advantage claimed for the conical borer i.e., that it will require significantly reduced thrust loads (the required load is less than 10 percent for bits) than conventional roller cutter bits of the same diameter and at the same penetration rate.

2. The principles of operation of the conical borer that had been proposed have been experimentally verified. Based on these tests and the parameters that were established the design may proceed with confidence.

3. Mechanical implementation of the design concepts is feasible and the final design of the conical borer is well underway.

4. Supply and discharge requirements for the mucking system are practical but careful research in the vicinity of the conical cutters will be required to establish the overall system requirements.

13. Cont.

a marked reduction in the values of the thrust loads which were obtained during the proof tests. Thrust loads of approximately 10 percent that required for a conventional tri-cone bit were observed. The specific energy required (inch-pounds per cubic inch of rock removed) was only 13 percent higher than that of a comparable tri-cone bit. This is considered very good in view of the extensive development history of tri-cone cutters.

The detailed design of the borer under Phase II is proceeding along lines as indicated by the results of the Phase I and Phase III efforts. Various preliminary design layouts have been prepared to establish the basic mechanical requirements for each of the conical borer subsystems. Present effort is being devoted to the integration of these layouts into an overall borer design. Workable designs have been established for the drive system, mainframe, cutters, and rotary union. A combination pressurized air and vacuum suction system has been chosen for the overall mucking system.

Initial effort under Phase III was spent on the preparation of a preliminary mainframe layout to establish air flow passages through and around the borer. Analyses of the pressure-vacuum system have yielded design parameters for the pressurized air supply and the vacuum discharge system. The transport of the cuttings is critical in several areas, particularly in the vicinity of the conical cutters and at the joints between the stages of the overall borer. Various concept solutions to overcome these potential transport problems have been generated and will be tested experimentally to determine the most efficient system.

A test model, currently being fabricated, will be used to study the mucking system behavior and characteristics. The test system will include a half-scale, two stage model of the borer encased in

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Contractor: Foster-Miller Associates, Inc.
135 Second Avenue
Waltham, Massachusetts 02154

Project Engineer: Dikrun DerMarderosian (617) 890-3200
Program Manager: Hans A. Hug (617) 890-3200

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1. Summary

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4. Supply and discharge requirements for the mucking system are practical but careful research in the vicinity of the conical cutters will be required to establish the overall system requirements.

2. Introduction and Background

The U.S. Government, acting through the Advanced Research Project Agency, ARPA, and its agent, the Bureau of Mines, Department of the Interior, is seeking improvements in all elements of underground rock excavation through its Military Geophysics Program for Rock Mechanics and Rapid Excavation. Rock disintegration has been singled out as a key element of this program.

A great many novel rock disintegration processes have or are being proposed and studied and it can be expected that this research will produce one or more processes that will ultimately be simpler, more reliable, more flexible, and more economical than the presently available tried and proven mechanical fragmentation methods. It can be expected that some of these more "exotic" methods will move from the laboratory to the field by 1980. In the meantime, every effort should be made to improve on the existing state-of-the-art boring and drilling methods so that the expected benefits can be practically realized in the field in the next two-three years. To accomplish this, Foster-Miller Associates submitted to ARPA a comprehensive program for the development of a new boring machine utilizing a <u>proven fragmentation principle involving roller cutters.</u> The overall machine concept was described in the final report under Contract No. H021044(1)*. The principle of operation of the skewed cutter, conical borer is presented in Appendix A. The concept promises substantial and prompt improvements over present performance especially in hard rock where today's boring devices are particularly limited by excessive thrust requirements.

The present program will develop a roller cutter boring device having a novel, conical cutter head geometry which completely eliminates the need for external thrust. As discussed in Appendix A, the proposed boring device generates all necessary excavation

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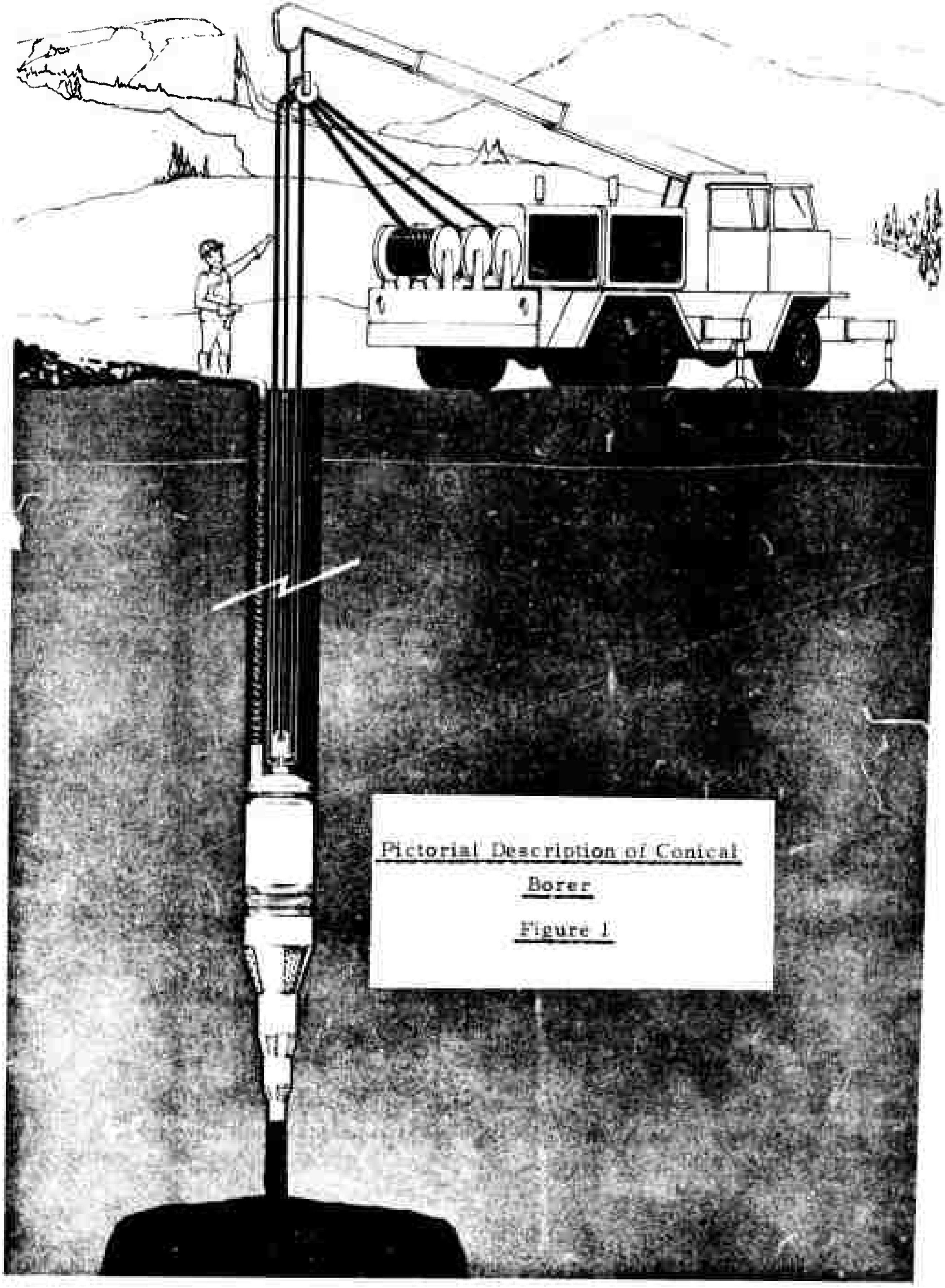
* Numbers in parentheses refer to references listed in the Bibliography.

forces directly from the rock surface being excavated and does not require a rigid link to the surface. A pictorial description of the proposed conical borer is shown in Figure 1.



Pictorial Description of Conical Borer

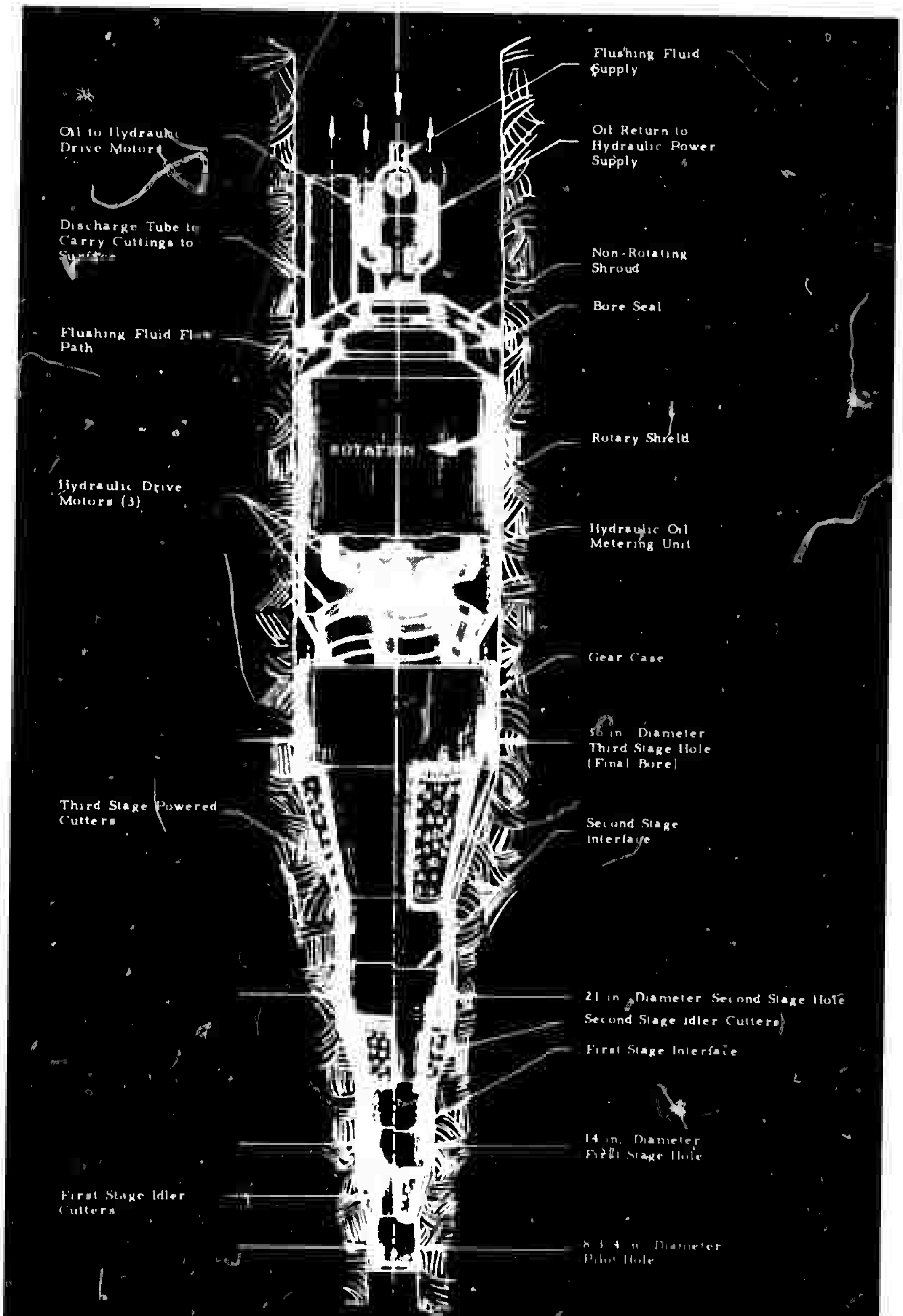
Figure 1

3. Objectives and Plan of Research

Effort under this contract is being devoted to the detailed design and development of the conical boring device. This is a continuation of research conducted during Contract H021044, where a preliminary design of the device was completed as shown in Figure 2. The conical borer will operate as a reaming device to enlarge an 8 3/4 inch pilot hole to a 38 inch bore at penetration rates equal to or greater than those obtainable with existing boring devices, but with substantially lower thrust load requirements. A pneumatic mucking system will be used to clean the cutting area and transport cuttings to the surface.

The basic plan of research, as specified in Section 1.1 of the subject contract, has been divided into four phases of effort as follows:

Phase I	Testing of the nose section of the conical borer developed under Contract H021044 to optimize its cutting performance and establish basic torque and thrust load requirements.
Phase II	Final design of the conical borer in accordance with the principles given in Appendix A of our Technical Proposal No. 71161.[2]
Phase III	Performance of mucking studies required in the pneumatic flushing and transport system, including design, analysis and model testing.
Phase IV	Partial fabrication of long lead time parts for the anticipated assembly of the conical borer.



Preliminary Overall Design

Figure 2

Progress under each of these phases of effort is discussed in the following sections.

4. Phase I - Cutter Development and Testing

The Phase I effort, as described in Section 1.2.1 of the subject contract, has been completed. During this phase the nose section prototype conical borer developed under Contract H021044 was revised and tested to optimize its cutting performance and to establish its performance characteristics. The tasks of this effort were as follows:

(a) Further study of the previous test data (Contract H021044) and closer examination of the prototype nose section.

(b) Rebuilding of the cutters including redistribution of teeth and changing tooth density.

(c) Testing, modification, and retesting of the prototype nose section.

(d) Data reduction, analysis and establishment of performance characteristics.

A detailed discussion of the research conducted during Phase I is given in our Phase I Test Report dated, 23 August 1972.[3] A summary of that research with important data and results is presented in the following sections.

4.1 Review of Previous Data

Further study of the previous data and closer examination of the prototype nose section revealed two areas of concern:

(a) at the higher drilling rates the rate did not remain constant over any significant depth interval, while at the design load the rate was quite low; and

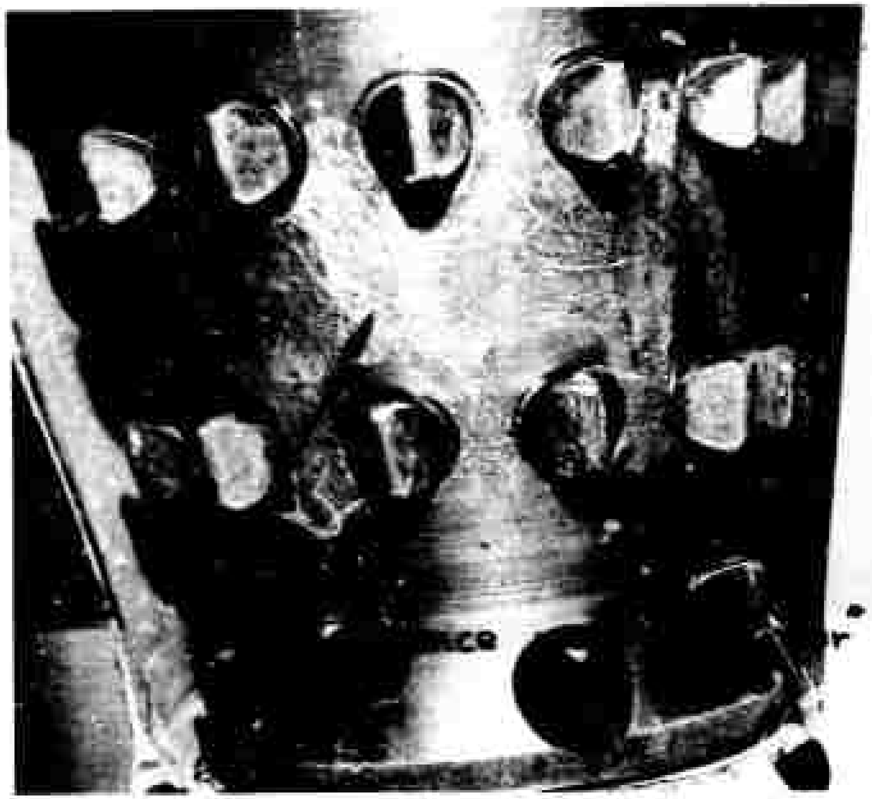
(b) the cutter cone-bodies were rolling and scraping on the rock due to a "rock-gear" formation on the hole-bottom, shown in Figure 3.

A detailed review of the drilling data indicated that at a constant load the rate usually continued to decrease with increasing penetration. This implied that most of the excess load (above the original design value) may have been attributable to cutter-body contact on a developing rock-gear hole-bottom. Discussions with the observers of the tests at Hughes Tool Company indicated that the rock-gearing was observed at several intervals during the drilling of the one rock sample.

The appearance of the cutter cone-bodies and pictures of the hole-bottom showed that rock-gearing was only occurring within the span of 3 or 4 tooth rows that had nearly equal tooth spacing and which comprised the lower third of the conical cutters. This probably caused the cutter teeth to fall "into step" at an integral number of revolutions per borer rotation. This phenomena is unique with the conical borer in that one row of teeth on one cutter can advance into the pattern of a lower row on another cutter.

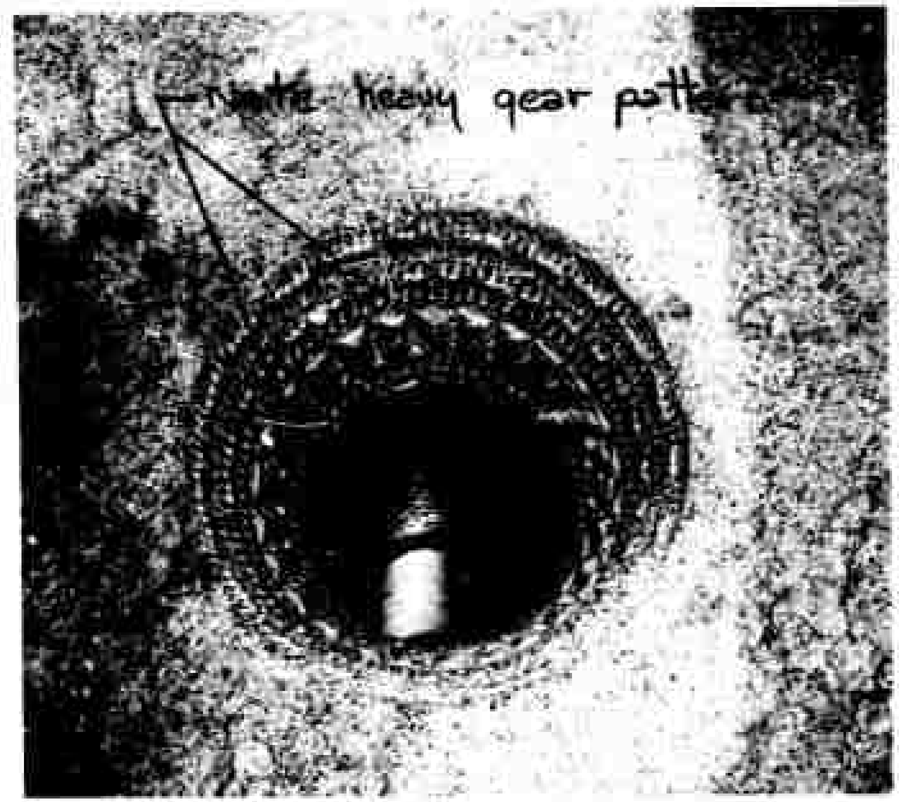
4.2 Modifications to the Borer

From the review of previous data it was concluded that further optimization of the borer performance would first require elimination of the rock-gearing. Then the number of teeth could be reduced as originally planned in pursuit of the desired relationship between load and drilling rate. It seemed inadvisable to remove any teeth before a good rock bottom-hole pattern was observed in test.



Prototype Cutter Cone After Tests

(a)



Hole Bottom Configuration

(b)

Photographs of Cutter Cone and Hole Bottom to Show the Effect of "Gearing" During the Tests of 14 March 1972

Figure 3

The design approach used for modifying the cutters to avoid rock-gearing was as follows:

(a) The imprint of each tooth row was drawn to show the nearest integral number pattern in one bit revolution, i.e., the likeliest rock-gear pattern for that tooth row;

(b) The imprints of a few teeth added above a particular tooth row were arranged to fall between all the original imprints during the passage of a few bit revolutions to "knock-out" the rock-gear;

(c) Where possible the spacing of teeth added on one cutter was made to be out of step with teeth on the other cutters near the same axial position;

(d) The axial, circumferential, and cutter-to-cutter distribution of teeth was balanced closely to minimize vibrational loads on the bit.

The total number of teeth on the bit was increased by only 14 percent to accomplish the plan just described.

The cutters were modified by inserting tungsten carbide teeth in the same manner as for the original construction. In order to optimize the use of our time at the Hughes test facility in Houston, dummy blocks with carbide inserts were constructed to test the planned deletion of teeth. An electric carbon arc, air-jet, reverse polarity, 250 amp. welder was quite effective for melting the teeth without excessive heating. This procedure was used to delete the teeth during the tests at Houston.

The deletion of specific teeth was planned prior to testing. Teeth were selected from clusters or dense regions, but not from the gage rows or from the group of teeth that were correctively added.

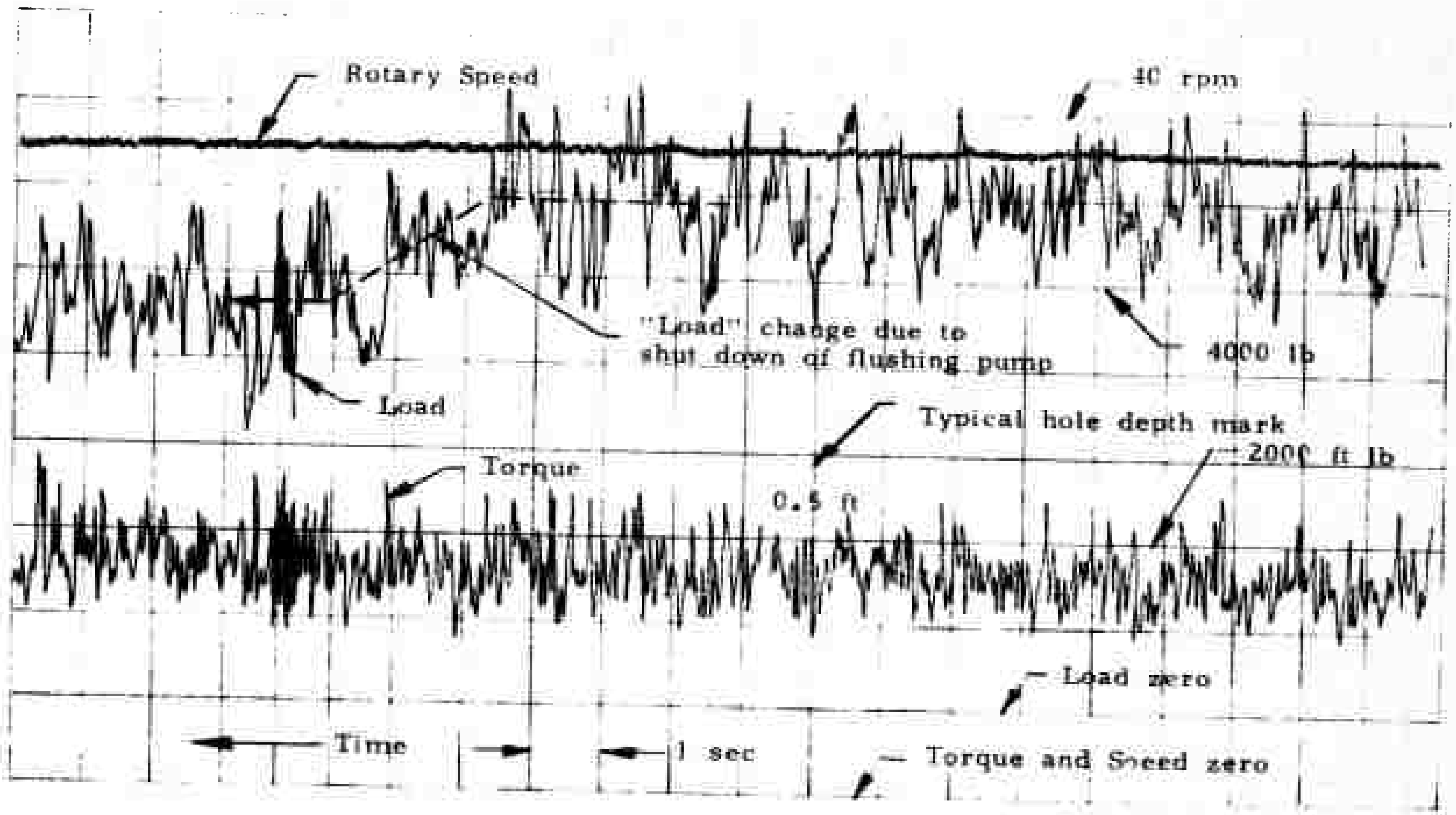
4.3 Test Results

Three rock samples were drilled. Of these, two were drilled with the cutters which had the added teeth to eliminate rock-gearing while the last sample was drilled with cutters from which approximately 8 percent of the teeth were discriminantly deleted.

Data was monitored automatically on a Visicorder for load, torque, speed, and time. Load and torque were measured in a strain-gage "sub" immediately above the bit. Net zeros were established before and after drilling, and penetration marks were manually noted on the chart record. The signals were filtered out above 10 Hz. A sample of the record is given in Figure 4.

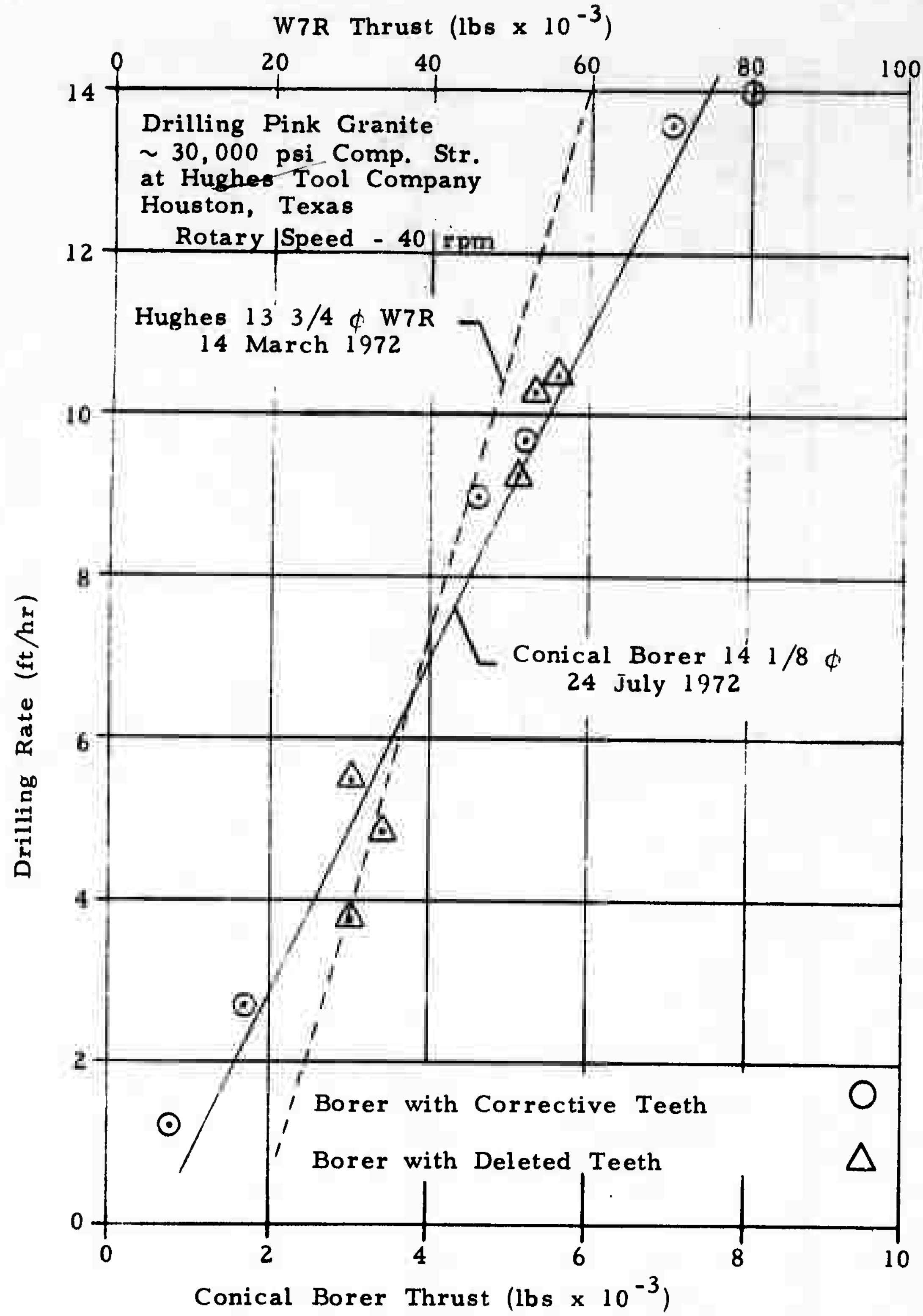
From the test records, average levels of drilling parameters were estimated for those drilling intervals in which performance was constant for more than about 0.3 ft. Only these data values were used in establishing the average bit performance shown in Figures 5, 6 and 7. Figure 5 shows the penetration rate at 40 rpm achieved as a function of total thrust on the rock surface by the bit cutting structure (i.e., thrust equal to drill stem and bit weight plus all external forces). Figure 6 shows the bit torques required to achieve the drilling rates.

Both Figures 5 and 6 show that the performance of the bit with deleted teeth was about the same as that of the borer with the teeth added to eliminate rock-gearing. In both cases the thrust load required at a given penetration rate was about 1/9th that of a comparable tri-cone bit, while the required torque was about 20 percent greater.



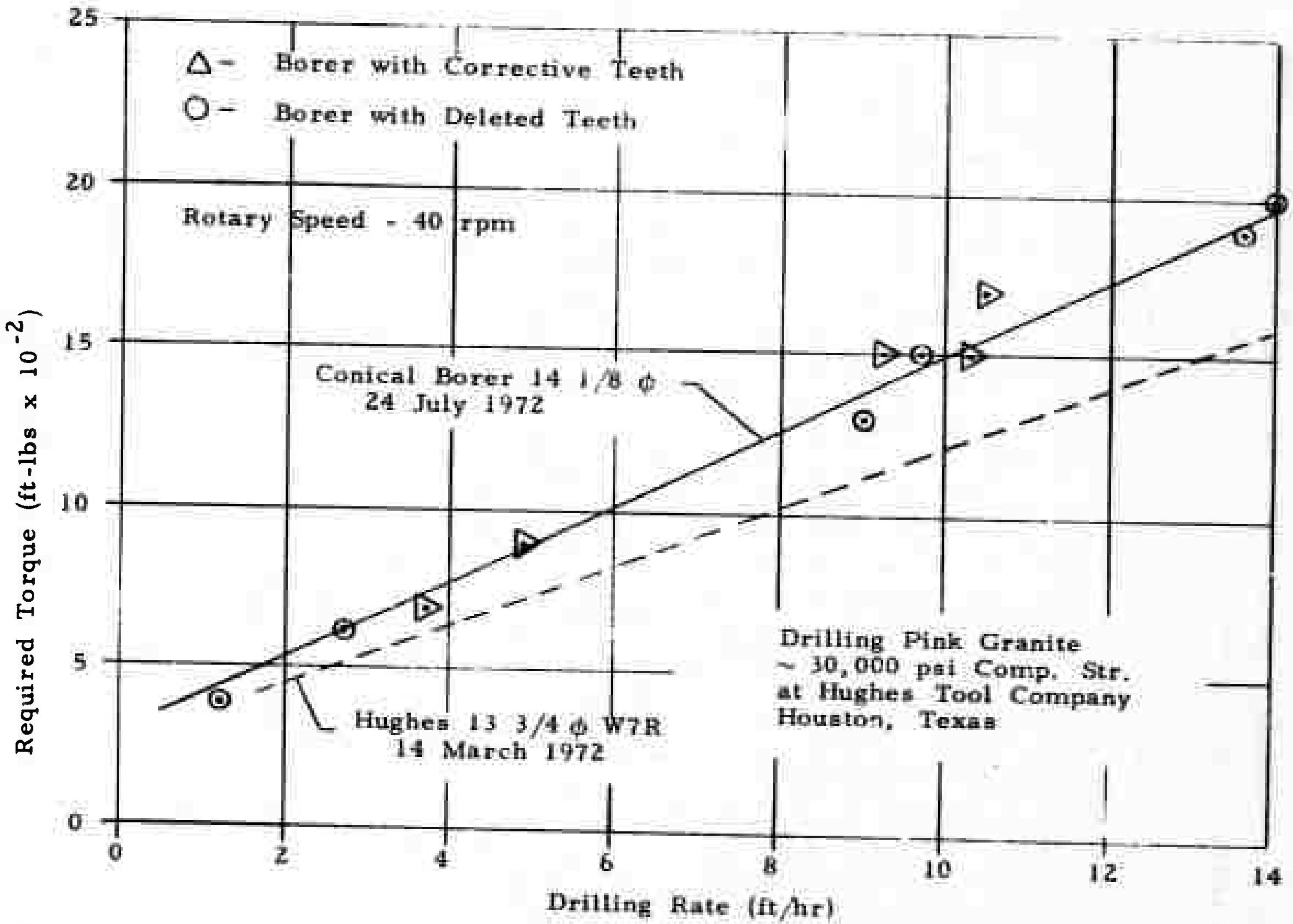
Sample of Visicorder Record of Drilling Data

Figure 4



<u>Thrust vs. Drilling Rate for the</u>
<u>Nose Section Prototype Conical Borer</u>

<u>Figure 5</u>



Torque vs. Drilling Rate for the Nose Section Prototype Conical Borer

Figure 6

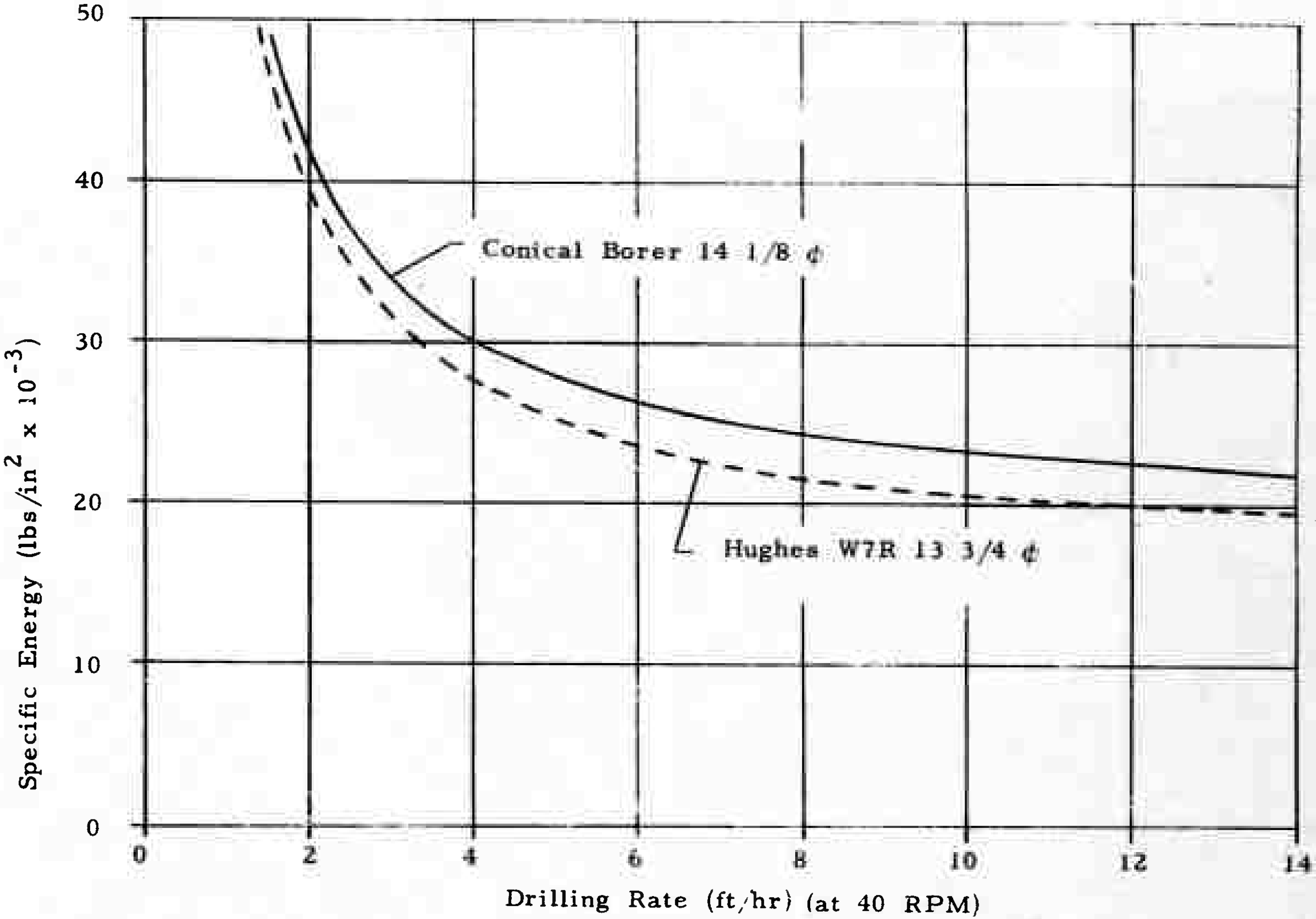
More precisely determined from rock volume rate and input power, the specific energies of rock removal are compared in Figure 7. The conical borer consumed only 13 percent more power than the flat bottomed tri-cone bit at the usual drilling rate, and both bits appeared to behave the same parametrically.

Several subjective observations made during testing are worth mentioning since there seemed to be agreement between personnel from Hughes Tool Co., the Bureau of Mines, and Foster-Miller.

The conical bit seemed to be "quiet" running, better than comparable flat-bottom bits. It behaved well during a difficult run-in when the pilot hole entry was broken away asymmetrically. Reference to Figure 4 shows that although the relative variation of load and torque are large, their absolute variation is not. Records show the absolute variation to be comparable to or less than that of the pilot bit.

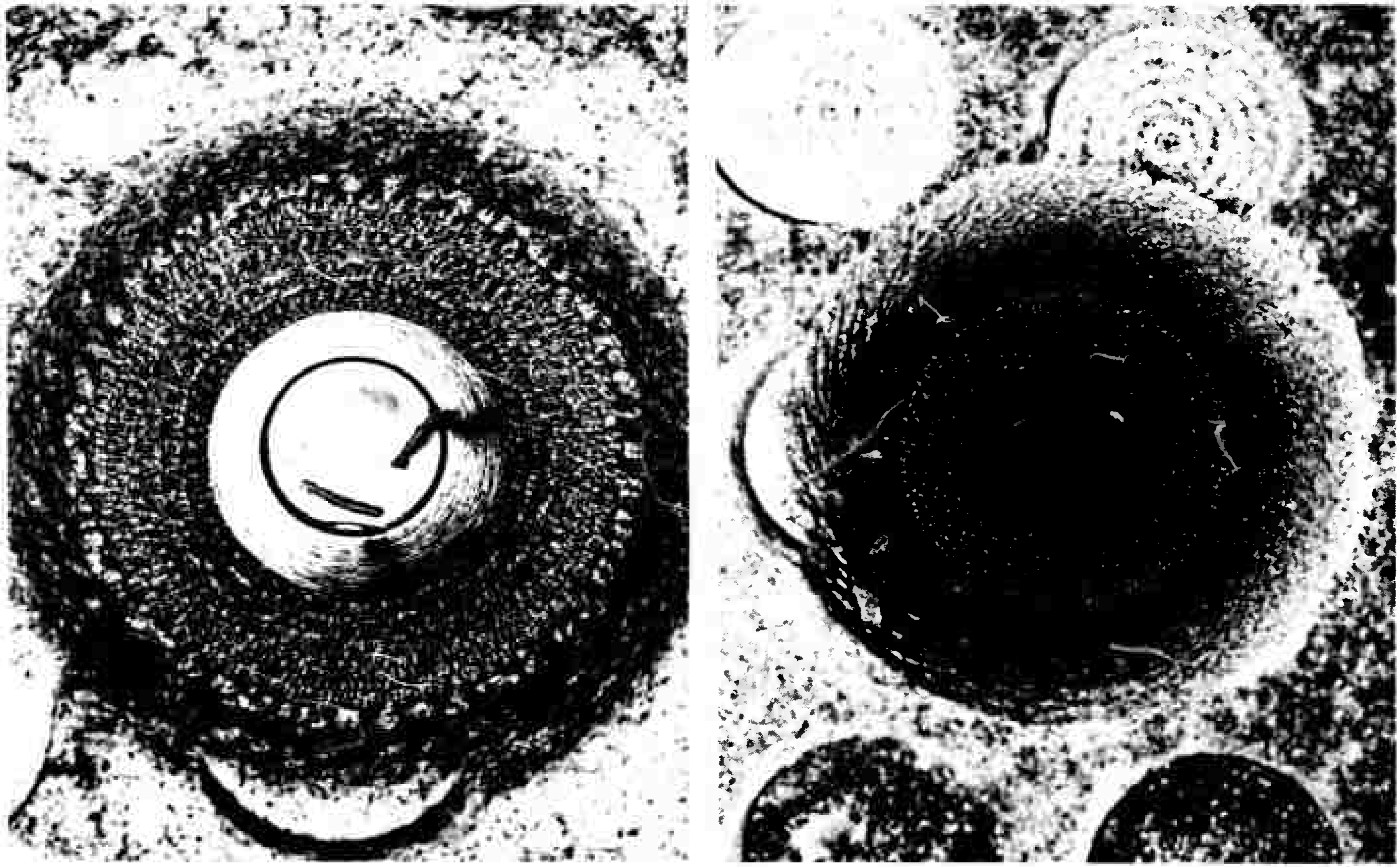
The bottom-hole pattern (actually side-hole pattern) showed the surface to be randomly lumpy (often considered to indicate efficient rock cutting). Figure 8 shows typical bottom hole patterns made by the conical borer. The bored hole shows some gage rifling and a slight rock-gear pattern at the top of the hole where the entry of the pilot hole was broken away asymetrically. This was an extreme case; all other observations showed no gearing or rifling.

The chips made by the conical borer were similar to those made by usual bits. There was some milled dust, a pre-dominance of coarse rock grains, and a noticeable fraction of penny-size and larger chips. Figure 9 compares sample cuttings from conical borer and pilot holes. During hand sifting of the dried cuttings, those from the borer feel coarser, with less mill dust present. The cuttings made with the conical borer after teeth were deleted were not apparently different from those made before.



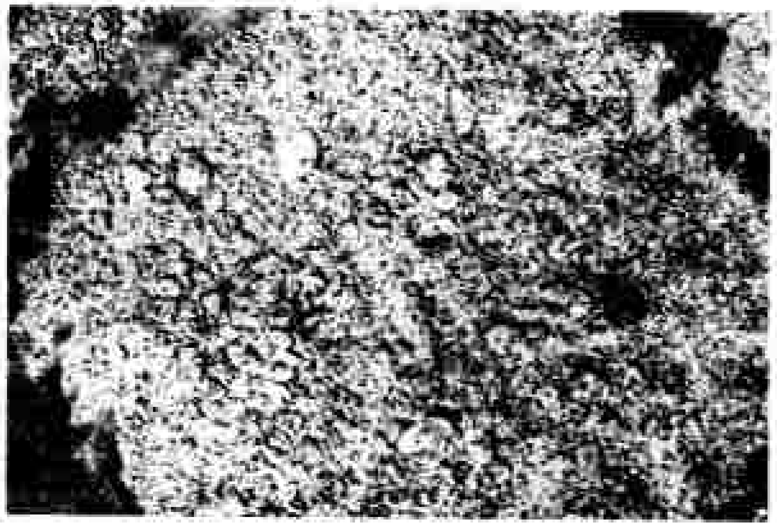
Specific Energy vs. Drilling Rate for the Nose Section Conical Borer Prototype

Figure 7

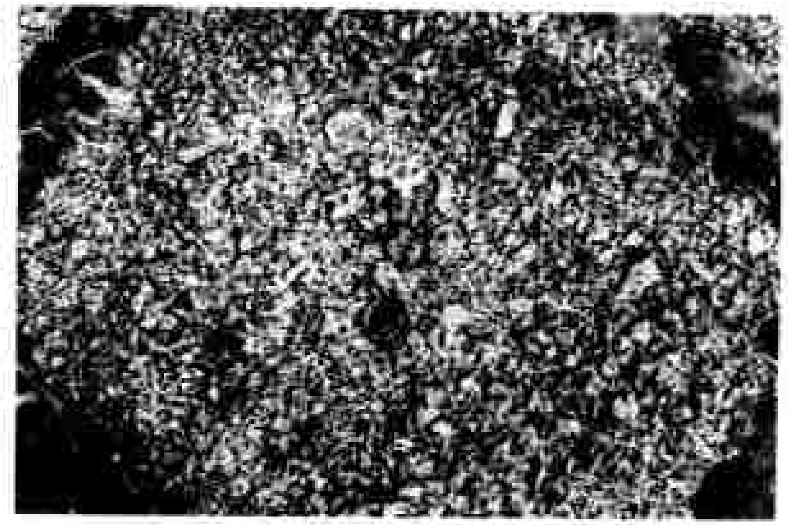


Typical Bottom Hole Patterns Made by the Conical Borer During Tests of 24 July 1972

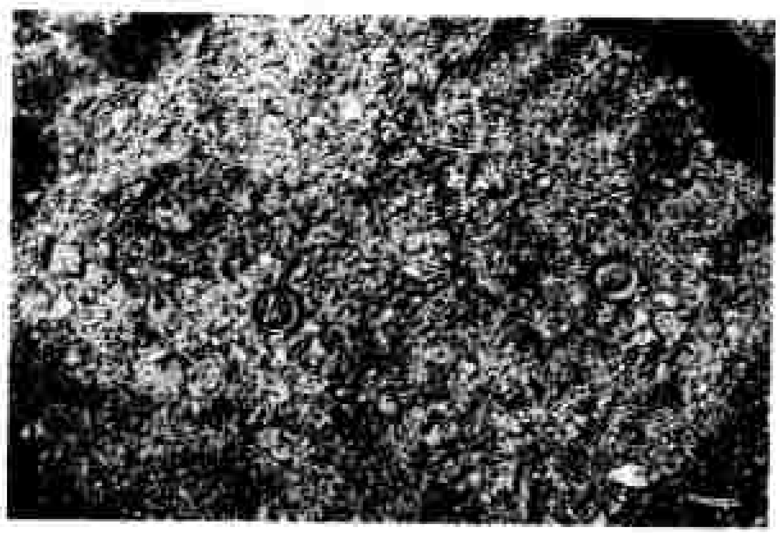
Figure 8



Rock No. 1 Conical Borer



Rock No. 3 Conical Borer



Rock No. 1 Pilot Hole Bit

Rock No. 3 Pilot Hole Bit

A Pictorial Comparison of Rock Chips Generated During The Drilling Tests of 24 July 1972

Figure 9

4.4 Conclusions on Phase I Efforts

The conical borer operated with no damage to the external structure. No tooth breakage or wear occurred, and no further scraping of the cones was apparent. The most significant achievment was the drilling of several feet of hard rock with a load of about 5000 lbs for a penetration rate of 10 ft/hr at 40 rpm. (The earliest design estimate for comparable drilling was about 4000 lbs.)

As predicted, the addition of a relatively few teeth, judiciously placed, corrected the problem of rock-gearing observed in the original proof test. Except that it required significantly less thrust load, the borer behaved similarly to a normal, flat-bottomed roller bit; it produced a good pattern of cutting and drilled smoothly up to 14 ft/hr at 40 rpm, indicating that higher speeds and penetration rates are possible. The load-penetration characteristic of the borer was approximately linear, similar to other bits (except that the load level was an order of magnitude less). Furthermore, comparatively less load was required to drive the borer past the "threshold" of low penetration that is characteristic of other hard rock bits. The deletion of a modest number of teeth had very little effect on the drilling characteristics. It was concluded that the tooth density was sufficiently close to optimum for the purposes of this research.

5. Phase II Final Design Effort

Considerable design effort has been devoted to each of the five critical design areas of the conical borer. These areas include the drive system, mainframe, mucking system, rotary union and roller cutters.

A preliminary mainframe design layout, generated under Phase III effort (see Section 6), was carefully scrutinized with respect to each of the critical areas above. "First cut" detailed design layouts of each area were prepared and integrated into the overall preliminary layout.

Although the final design of the Conical Borer will depend on the Phase I and Phase III results, preliminary design concepts had to be explored in order to establish basic configurations, space limitations, and subsystem interractions. The basic approach, therefore, has been one of continued design iteration where new sub-system concepts were integrated into the framework of the current system design.

5.1 Overall Design Considerations

The basic design of the conical borer is dependent upon a number of functional requirements including the size of the finished bore hole, the penetration rate and the bearing life requirements. The bore diameter has been set at approximately 38 inches; the design penetration rate is approximately 10 feet per hour; and the design life of the bearings has been set at approximately 500 hours.

The penetration rate of the borer will be fixed by the line loading on the conical cutters which in turn will be a function of the net weight of the borer, the cone half-angle, and the frictional thrust generated by the skewed cutters, as given by:

$$F_n' = \frac{2\ W}{3\Delta D \left[1 - \frac{F_s}{F_n \tan \alpha}\right]} \tag{1}$$

where

F_n' = cutter line loading, [lbs/inch]

$\frac{F_s}{F_n}$ = effective frictional thrust coefficient for skewed cutters

W = weight of the borer, [lbs]

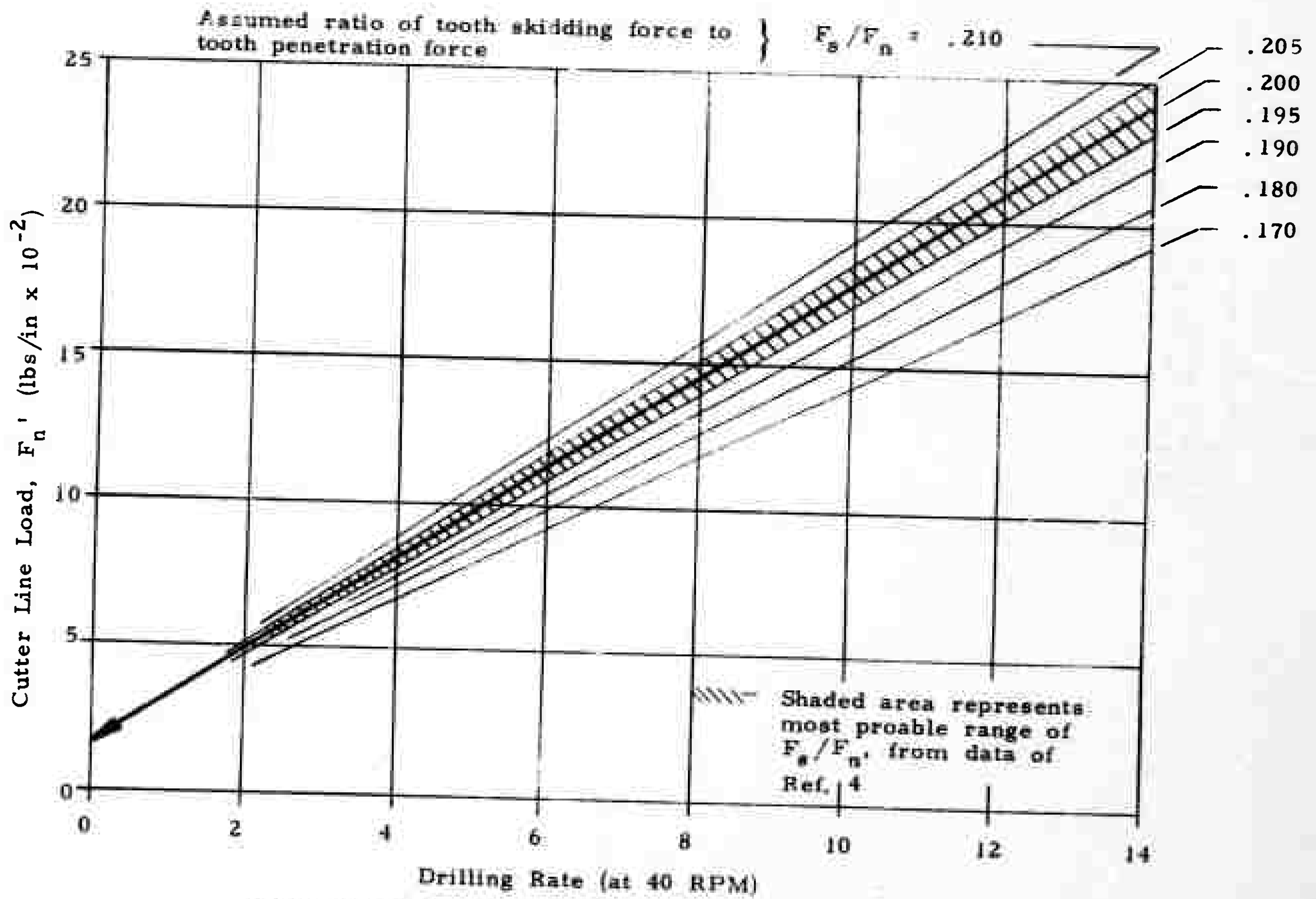
ΔD = differential diameter of rock removed, [inches]

α = borer half cone angle, [degrees]

(A rigorous discussion of the principles of operation of the conical borer is given in Appendix A.)

Based on the preliminary design parameters established under the previous contract, Equation (1) yielded a line loading of approximately 1400 lbs per inch, which was used in the preparation of a preliminary layout of a two stage borer with fairly reasonable bearing life. A two stage design was initially considered desirable since it would simplify the design and minimize the possibility of losing traction in regions of soft rock.

The test results of the Phase I effort, however, have indicated a line loading of approximately 2000 pounds per inch will be required at the desired drilling rates. The variation in line loading as a function of drilling rate, based on the data of Figure 5, is given in Figure 10. The results seriously reduced the expected bearing life for the two stage preliminary design and, thus, final design effort is being devoted to a three stage device.



Estimated Cutter Line Load, F'_n, vs. Drilling Rate For the Conical Borer in Pink Granite

Figure 10

Equation (1) indicates that the penetration rate (directly proportional to F_n') is a function of the weight of the borer and the half cone angle. Preliminary weight estimates have indicated a total of approximately 15,000 pounds for the three stage borer versus the original estimate of 18,000 to 20,000 pounds. Therefore, in order to retain the desired penetration rate, the half cone angle has been changed from 18 degrees to 15 degrees for the current design. Further analysis of the Phase I test data has indicated that this can be done without approaching a self locking condition during operation.

5.2 Drive Train

The powered stage roller cutters will be driven by means of (3) separate high torque, low speed Vickers hydraulic motors. Expensive gearing was eliminated through the use of Schmidt parallel offset couplings. Every effort was made to pack the maximum torque capability into the available space, resulting in a torque safety margin of 100 percent and a speed reserve of 150 percent.

Since it is conceivable that one or two of the powered roller cutters might become partially unloaded when entering a soft-rock formation, a triple flow divider will be incorporated to assure traction under adverse conditions and to prevent a run-away motor condition. Motor case drain flow will be collected in a common sump and pumped to the surface by a scavenge pump.

5.3 Cutter Design

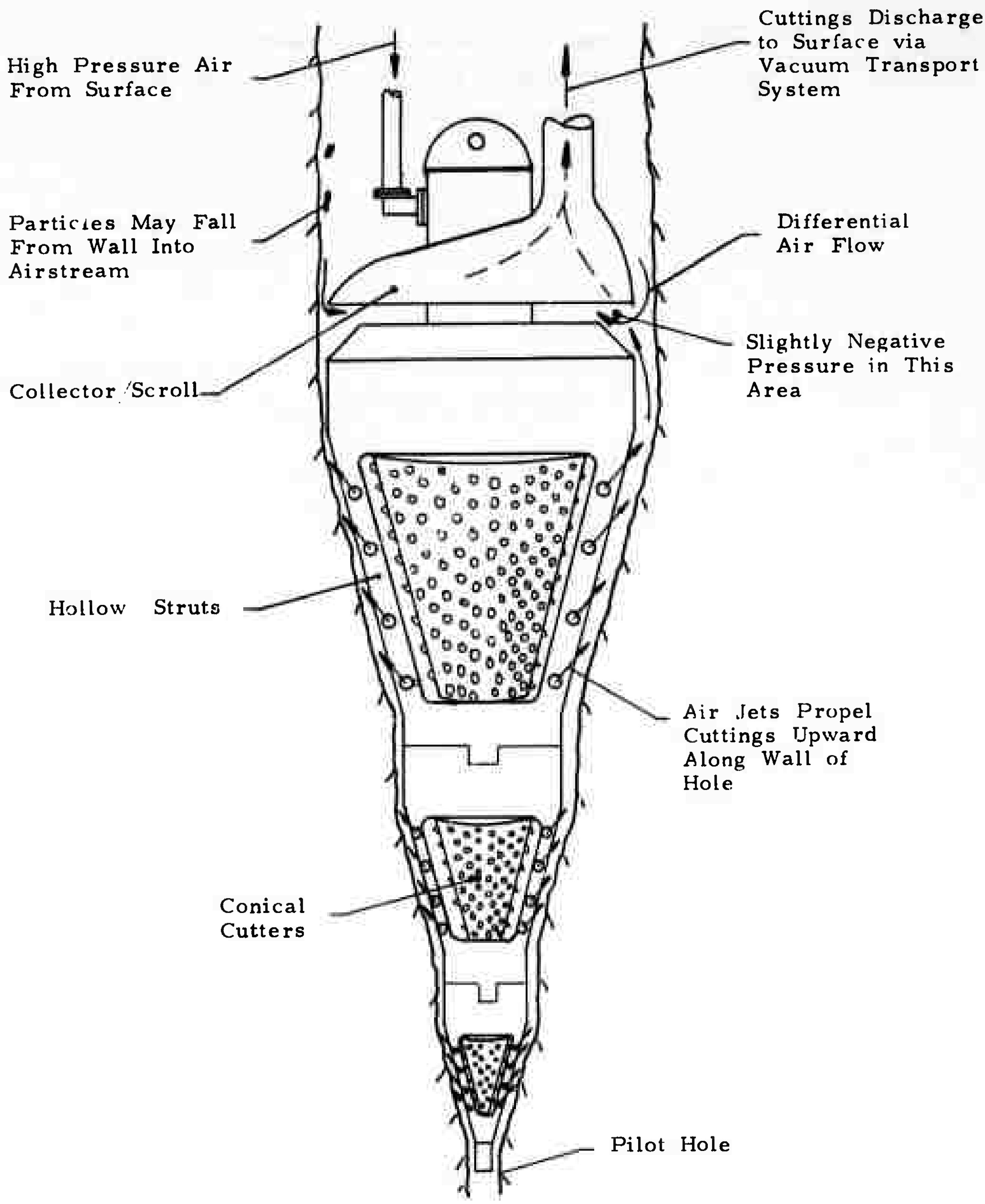
The preliminary design of the conical cutters for the three stages has been completed, including bearing selection, seal design and lubrication requirements. Extensive discussions with the bearing manufacturer (SKF, Philadelphia, Pa.) revealed that a grease packed bearing stands a better chance of reaching the expected life figures than one lubricated with hydraulic oil. To protect the bearing cavity against dirt and sand penetration, metal to metal wiper seals and "O" ring seals will be used throughout the machine. No work has been done on the actual tooth distribution; this will be done toward the end of the program.

5.4 Mucking System Design

The Conical Borer is to incorporate a pneumatic mucking system to flush chips away from its cutting faces and transport them to the bore-hole surface. Various concepts for this system have been generated and analyzed to determine their operating parameters and requirements. Concepts utilizing vacuum and/or pressurized air in combination with various jet locations, baffles, collectors, scrolls, seals and rotary-union designs were studied. Design requirements for each of the most promising concepts have been reviewed and tentative solutions have been worked into the overall borer layouts. (see Section 6 for details).

The final overall system concept chosen will incorporate a combination of pressurized air and vacuum. This concept system, as shown in Figure 11, will utilize compressed air fed through hose from the surface, through the rotary union and directed to nozzles in the struts of the borer main frame. The high velocity air jets will "scrub" the surface just behind the cutter cones and agitate and propel the chips upward. Air and chips will be directed to the discharge pipe and carried to the surface by the vacuum transport system. The flow rate induced by the vacuum system will be maintained slightly higher than that of the pressurized air. The additional air volume will come from above and around the upper section of the borer housing.

This system has a number of distinct advantages over pure vacuum or pressure systems as listed below.



Overall Mucking System Concept

Figure 11

(a) The slight down hole vacuum created by the differential flow rates will pull the borer against the cutting face rather than lifting it off.

(b) The high pressure air insures maximum jet velocity to clean cuttings from the face and direct them as desired.

(c) The pressure provides a positive means for eliminating a blockage if one should occur.

(d) No seals will be required between the borer and the rough rock surface. Chips falling from above the borer can pass by the housing and into the flushing stream.

It is anticipated that the major problem areas in the mucking system design are in the vicinity of the conical cutter sections of the borer. Abrupt changes in cross-sectional area from stage to stage and the very large flow area near the top of the powered stage are not consistent with the requirements for smooth and uniform particle transport. Reasonable flow and pressure levels have been resolved for the discharge pipe but these are not consistent with the net flow required in the cutter region. Various conceptual solutions for particle transport from the cutters to the discharge pipe are being studied and tentative design solutions are being worked into the overall borer layout. A detailed discussion of this problem area along with a summary of the technical requirements of the proposed mucking system is given in Section 6.

5.5 Rotary Union Design

Various preliminary layouts of the rotary union have been completed based on the following service requirements:

- oil in at 4000 psi
- oil out at 0-4000 psi
- flushing air in at 100 psi, minimum
- flushing air and granite chips out (optional design)
- case drain flow rate of 5.0 gpm at 40 psi

The stator of the rotary union, which contains the various fluid passages, will be connected to a heavy duty wire rope hoisting cable. All service lines will be attached to and supported by the cable via quick connect/disconnect hangers.

Four different layouts of the rotary union were prepared differing mainly in the routing of the various fluid passages and the choice of axial and radial bearings. The final design chosen will use a large diameter radial needle bearing and an oversized turntable bearing for combination radial and axial loads. A straight central passage of 5" dia. is provided in one preliminary design for the outgoing flushing air and entrained cuttings. This passage will have a removable liner to prevent erosion of the stator. Rotor and stator will be cast either of meehanite or steel.

Three hydraulically loaded pressure rollers will be incorporated at the top of the borer to prevent rotation of the stator and twisting of the service lines leading to the surface. A tubular strut framework will connect the rotor of the rotary union to the main frame of the borer and will also convey the compressed air from the union to the hollow mainframe struts.

5.6 Borer Mainframe

The borer mainframe will consist of three subframe assemblies (cutter stages) which are rigidly joined together by bolted connections. Each of the subframes will contain a set of three removable roller cutter assemblies.

The power section, drive motors, scavenge pump, rotary union, and all hydraulic and pneumatic controls will be enclosed and sealed from stone blast and chips by a cylindrical steel vessel.

All cutters including those of the power stage will be removable for servicing and/or replacement without removing the protective steel vessel and without disturbing the hydraulic prime movers.

Preliminary conservative stress analysis has indicated reasonable stresses in the borer mainframe. Further detailed analysis will be conducted as the design progresses.

It is quite possible that all subframes could be combined in one casting, using high strength Meehanite cast iron or if necessary higher strength steel alloy. No final choice has been made yet.

5.7 Summary of Phase II, Final Design Effort

At this time, some of the basic conical borer parameters can be summarized as follows:

- 3 stages, one of which is powered
- cone half angle 15°
- skew angle 4°
- approximately weight 14,500 lbs
- maximum available torque 12,000 ft lbs. at 4,000 psi
- rotary speed range 20 to 60 rpm

6. Phase III, Mucking System Studies

The initial effort under Phase III was devoted to the generation of a preliminary mainframe design layout. This layout was used to establish the basic configuration and sizes of air passages through and around the borer.

The proposed mucking system has been analyzed on the basis of the system model shown in Figure 12. Air is used as the working fluid in a combination pressure-vacuum system.

The pneumatic transport system has been divided into three discrete areas to facilitate the analysis:

1. air-cuttings flow up the discharge pipe

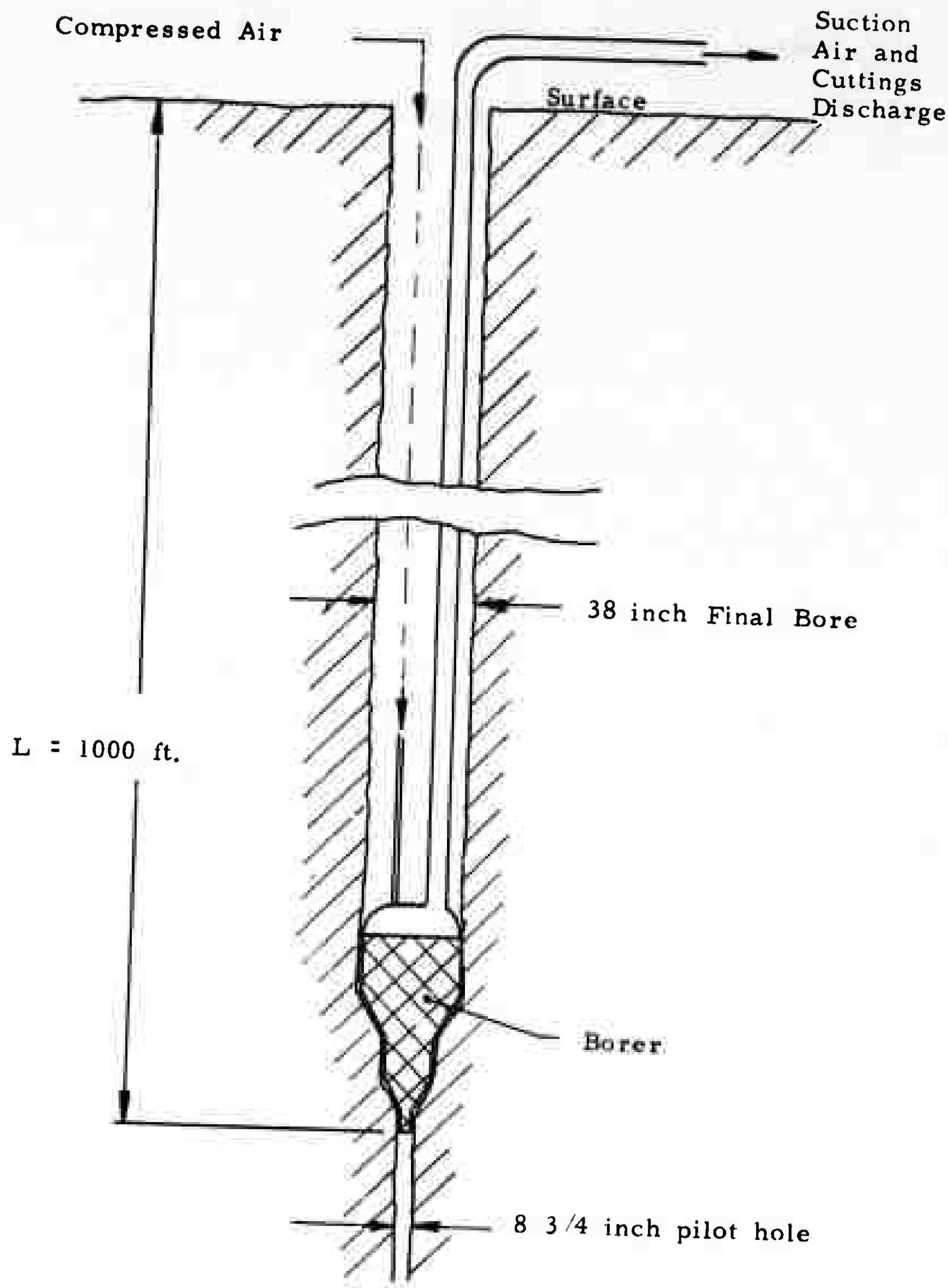
2. distribution of air in the borer and around the cutting faces

3. down flow of clean air under pressure

6.1 Discharge Pipe Analyses

The calculations for pressure loss in the return duct were based on incompressible flow relationships and atmospheric pressure at the inlet to the return duct. From Zenz[5], the relationship for pressure drop ΔP, in a vertical pipe is given by,

$$\Delta P = \frac{v_a^2 \rho_a}{2g} + \frac{W v_p}{g} + \frac{2f \rho_a v_a^2 L}{g D} \; 1 + \frac{f_p v_p}{f v_a} \; \frac{W}{v_a \rho_a} + \frac{W L}{v_p} \qquad (2)$$



Schematic of Pneumatic Transport System

Figure 12

where

v_a = velocity of the transporting air

ρ_a = density of the transporting air

g = gravity constant

W = solids mass flow rate

v_p = particle transport velocity

f = Fanning pipe friction factor

L = tube length

D = tube diameter

f_p = fluid to solids friction loss in conveying

Particle transport velocity, v_p, is the difference between the air velocity, v_a, and the particle terminal velocity in air, v_t, where,

$$v_p = v_a - v_t \tag{3}$$

Rewriting the pressure drop relation using the above expression yields,

$$\Delta P = \frac{v_a^2 \rho_a}{2g} + \frac{W (v_a - v_t)}{g} + \frac{2f \rho_a v_a^2 L}{g D} \quad 1 + \frac{f_p (v_a - v_t) W}{f v_a^2 \rho_a} + \frac{W L}{(v_a - v_t)} \tag{4}$$

According to Zenz the fluid to solids friction loss factor may be determined from,

$$f_p = \frac{3 \rho_a C_d D}{2 \rho_p d_p} \left(\frac{v_t}{v_a - v_t}\right)^2 \qquad (5)$$

where:

C_d = drag coefficient

ρ_p = density of the particle

d_p = diameter of the particle

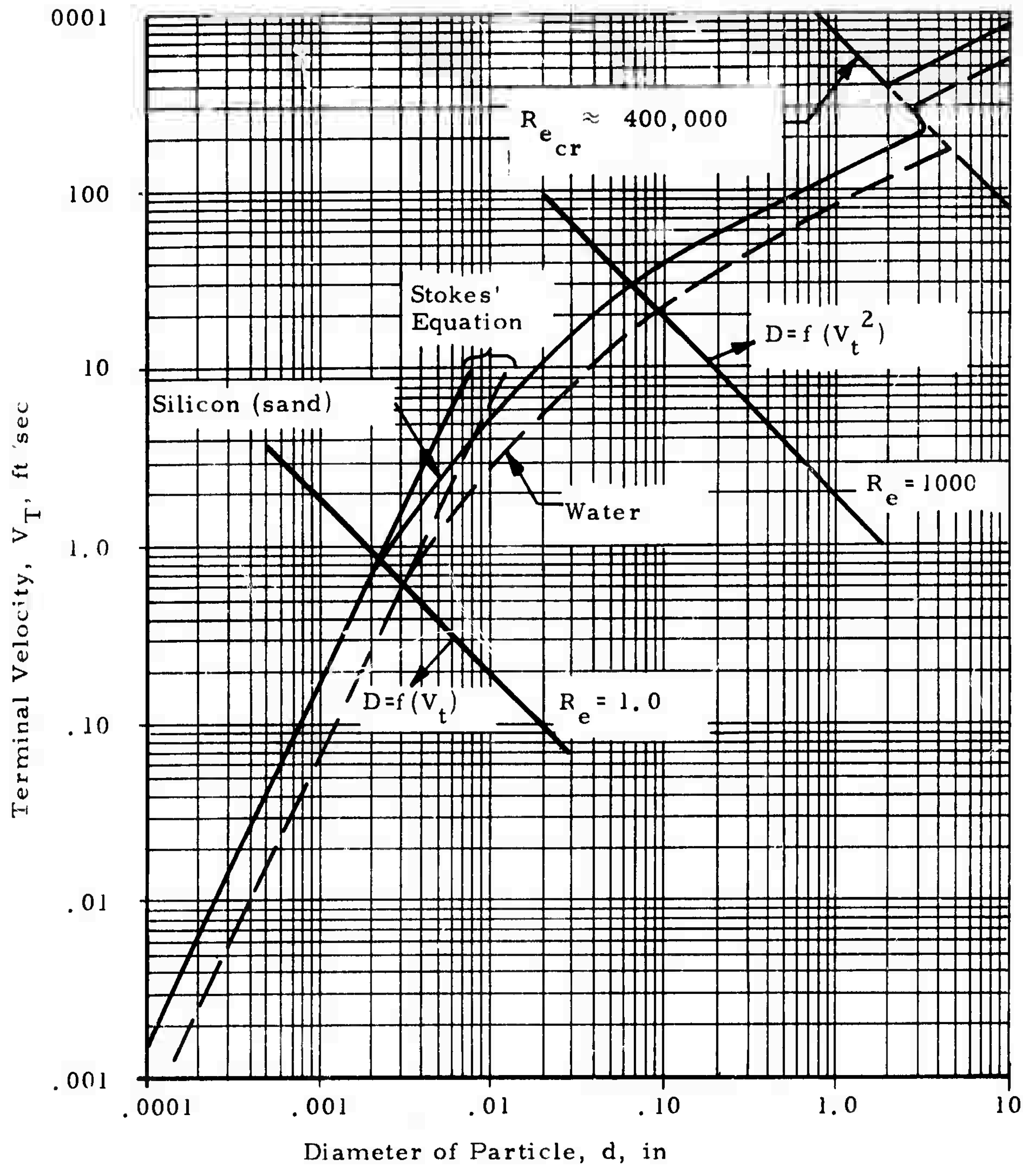
As recommended by Hinkle[6], the factor

$$\frac{f_p (v_a - v_t)}{f\, v_a} = 1 \qquad (6)$$

was used whenever calculations indicated a value greater than unity.

The terminal velocity, v_t, for the cuttings was determined from Figure 13, which is based on particles of relatively uniform cross-section. Examination of the cuttings produced during the Phase I tests (see Figure 9 Section 4.3) showed (as expected) that the particles were more like flakes rather than uniform cubes or spheres. However, since the particles tend to tumble during transport, we may assume that the area projected to the flow stream would average out to that of a uniform particle of the same weight. Average particle diameters from .065 to .250 inches were used in the analysis.

Based on the previous relationships, computations were made to predict the total pressure drop in the discharge pipe for the following parametric values,



Terminal Velocity of Particles vs. Particle Diameter (7)

Figure 13

L = 1000 feet

D = 4", 6", 8", 10"

W = lbs/sec = 2.25 lb/sec

f = 0.004, 0.005, 0.006, 0.008, 0.010

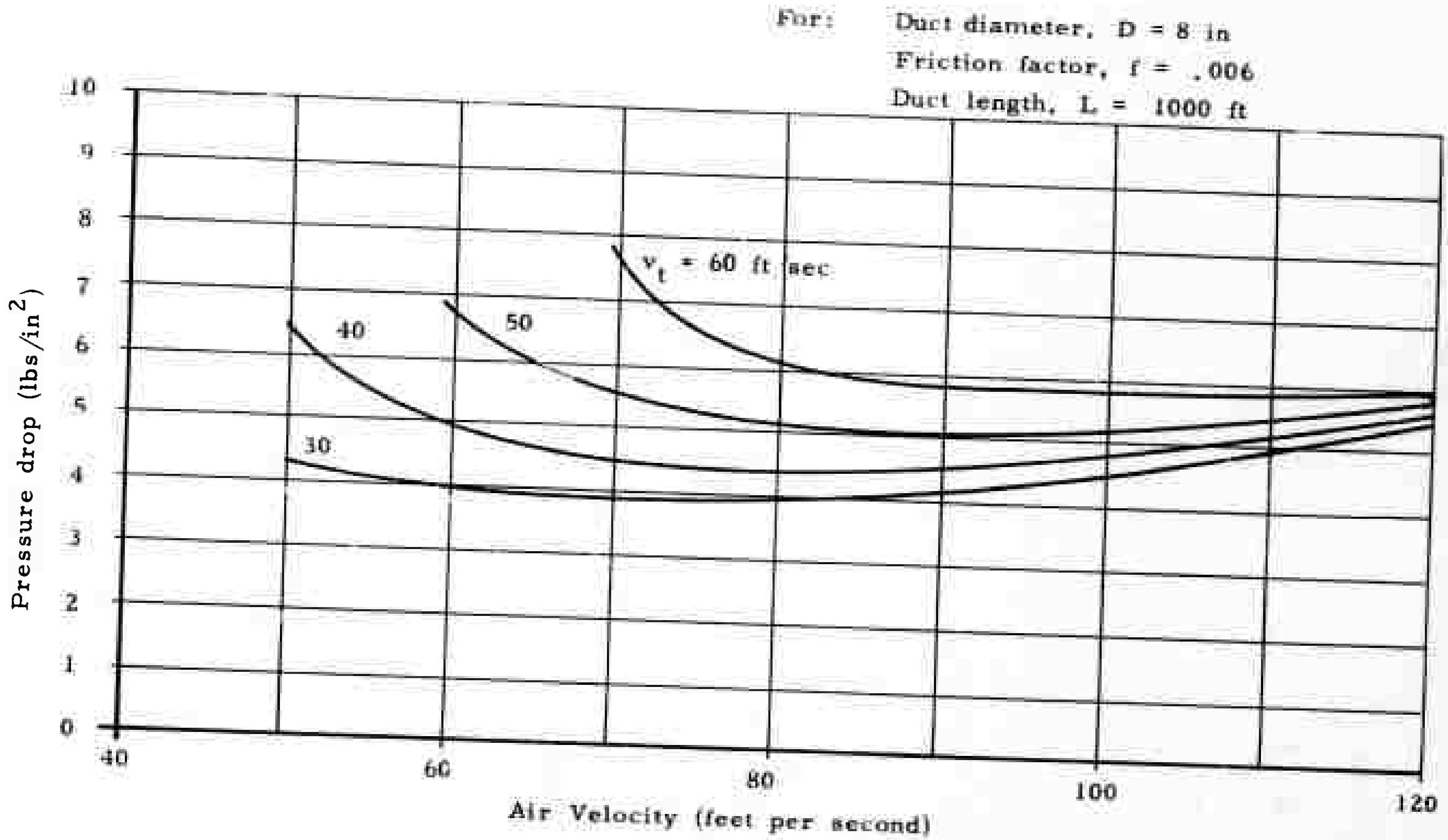
v_t = 30, 40, 50, 60 ft/sec

v_a = 50 to 120 ft/sec

ρ_a = 0.075 lb/ft^3

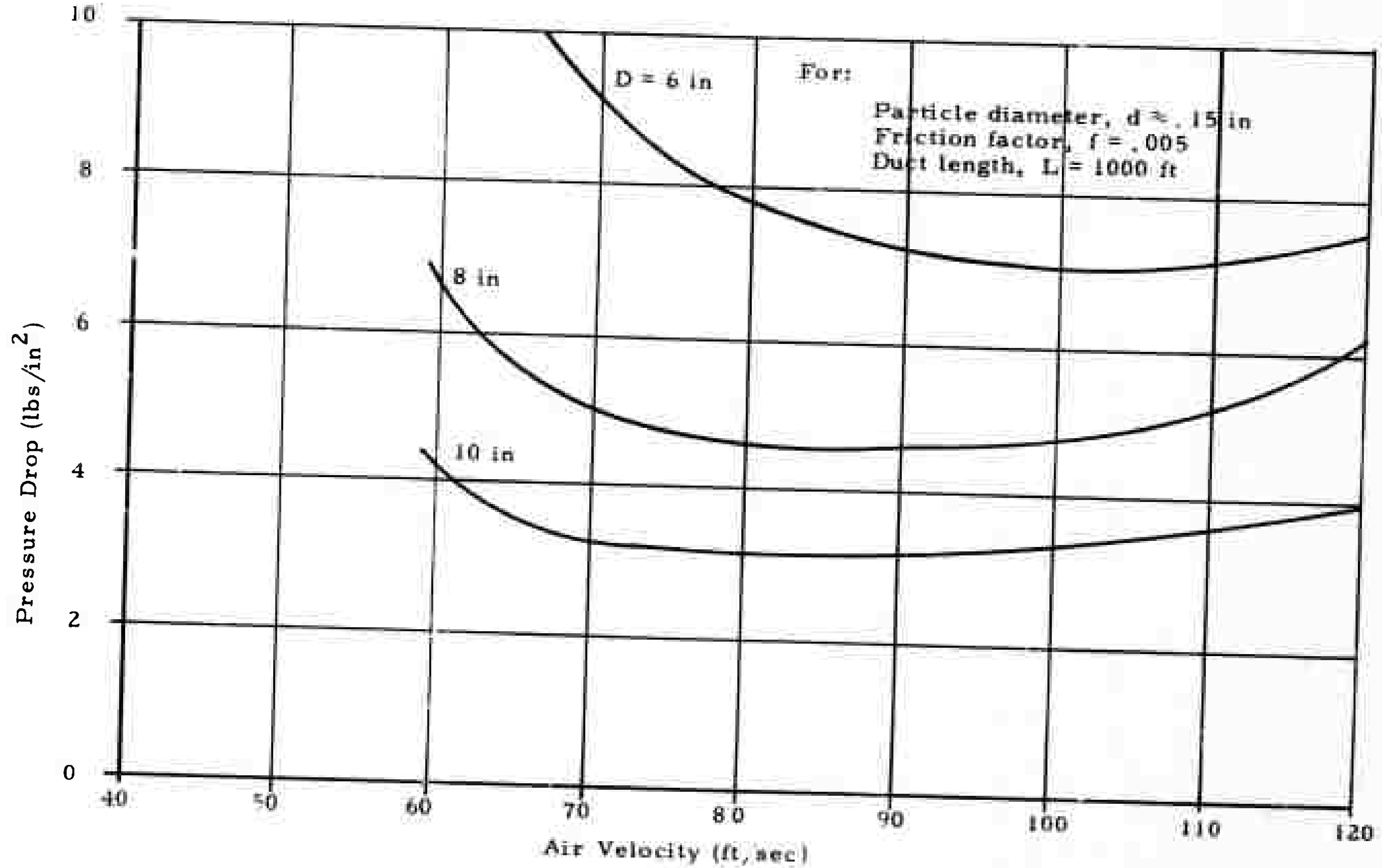
Figure 14 shows the pressure drop as a function of velocity (based on an 8 inch diameter discharge pipe, 1000 feet long with a friction factor, f = .006) for various particle sizes, i.e., terminal velocities. The curves indicate that the pressure drop would be higher for large particles than for small ones. This is reasonable since the higher terminal velocity means lower transport velocity and hence a larger mass of cuttings in the discharge pipe at any time. For the same reason, the pressure drop for the majority of particle sizes decreases to a minimum at approximately 90 feet per second. Above 90 feet per second the effects of friction predominate and the pressure drop increases.

The proposed transport system will convey a range of particle sizes and the resulting pressure drop will be a function of the net mass loading of the discharge pipe as described above. Assuming an even distribution of particle sizes, by weight, the average terminal velocity would be approximately 45 feet per second. Using V_t = 50 feet per second, the pressure drop vs. air velocity has been plotted for various pipe diameters, as shown in Figure 15.



Pressure Drop versus Air Velocity for Various Particle Sizes

Figure 14



Pressure Drop versus Air Velocity for Various Duct Diameters

Figure 15

These curves indicate the effect of increasing the diameter of the discharge duct. Since a vacuum system is proposed to supply the required pressure drop it would be desirable to minimize the pressure drop to whatever extent possible.

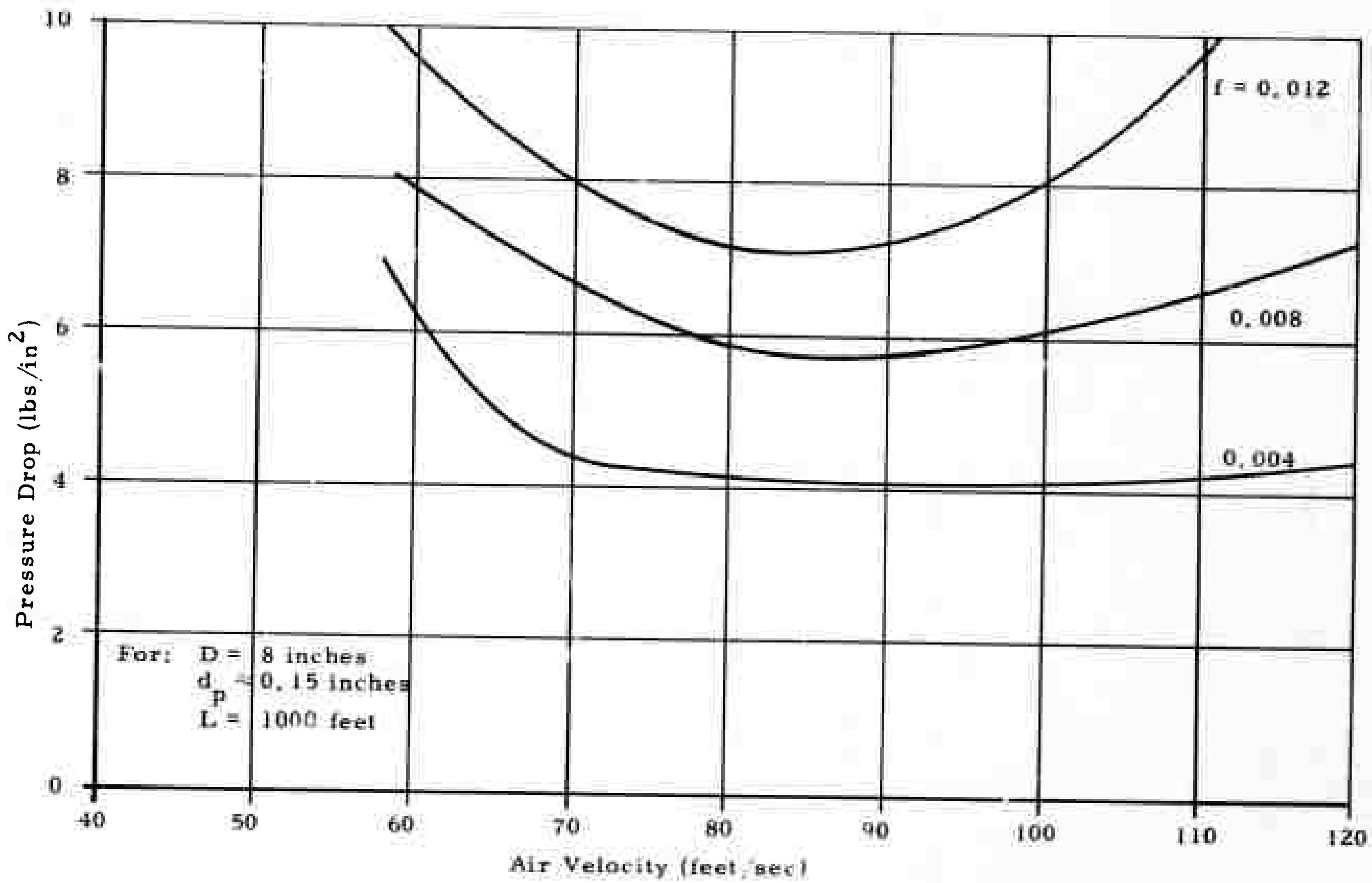
The curves of Figure 15 assume a friction factor typical of standard steel pipe or flexible hose of the type likely to be employed in the application. The inclusion of water in the bore-hole, however, may result in a marked increase in the effective wall roughness due to "cementing" of the cuttings to the duct wall. Figure 16 shows the variation in pressure drop through the return duct as a function of friction factor. The range of friction factors presented spans the entire range of wall roughnesses which might be encountered. It is not likely for this application, however, that the friction factor will exceed $f = 0.008$, the value used to compute the intermediate curve.

Among other considerations in sizing the return duct is the power required of the vacuum system. The power requirements have been determined analytically by treating the operating fluid as an ideal gas. The ideal power required to obtain the pressure drops of Figure 15 was determined using the relationship,

$$P = w \, c_p \, T_1 \left(\frac{T_2}{T_1} - 1\right) \tag{7}$$

where,

P	=	ideal power
w	=	mass rate of air flow
c_p	=	specific heat of air at constant pressure
T_1	=	temperature of inlet to blower
T_2	=	outlet temperature



Pressure Drop versus Air Velocity for Various Friction Factors

Figure 16

To simplify the determination of the temperature at the outlet of the blower we use

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}} \qquad (8)$$

where

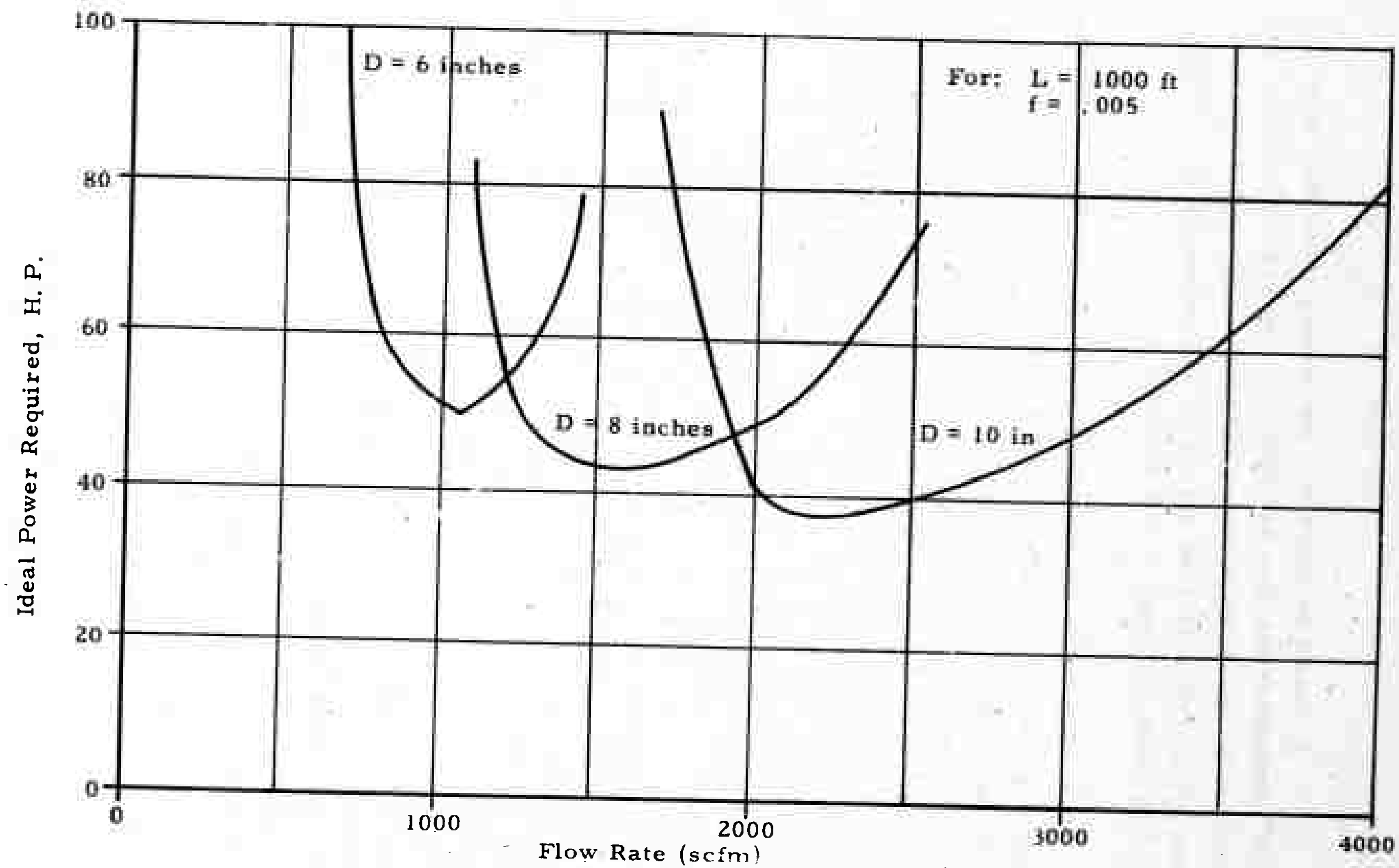
k = ratio of specific heats

The results of the computation are presented graphically on Figure 17 where the horsepower required of the vacuum system is presented as a function of air flow rate for several return duct diameters. These curves permit the selection of an optimum return duct size for a particular air flow rate.

The asymptote for low flow rate at each duct size corresponds to an upward gas velocity equal to the terminal velocity of the particles being carried. At this velocity no transport will occur, the duct will load with particles and the pressure head due to these particles will approach infinity. From this lower velocity limit the power decreases with increasing flow rate as the mass loading in the duct drops due to higher transport velocity. The required power passes through a minimum and then increases as frictional losses become the dominant factor.

For the prototype borer, where required flow rates are somewhat indefinite, the relatively flat characteristic of the 8" and 10" ducts present obvious advantages. For these duct sizes variations in flow rate produce less significant changes in required power than for the 6" duct.

These power requirements do not include the effects of inefficiencies of the equipment. To size a suction system it will be necessary to include the efficiency of the particular unit under consideration.



<u>Ideal Suction System Power Required versus Flow Rate for Various Duct Sizes</u>

<u>Figure 17</u>

It should be noted that the above calculations were based on incompressible flow assumptions. Several data points have been checked by more rigorous and complex compressible flow calculations which indicated pressure drops approximately 10 percent higher than those indicated on the curves.

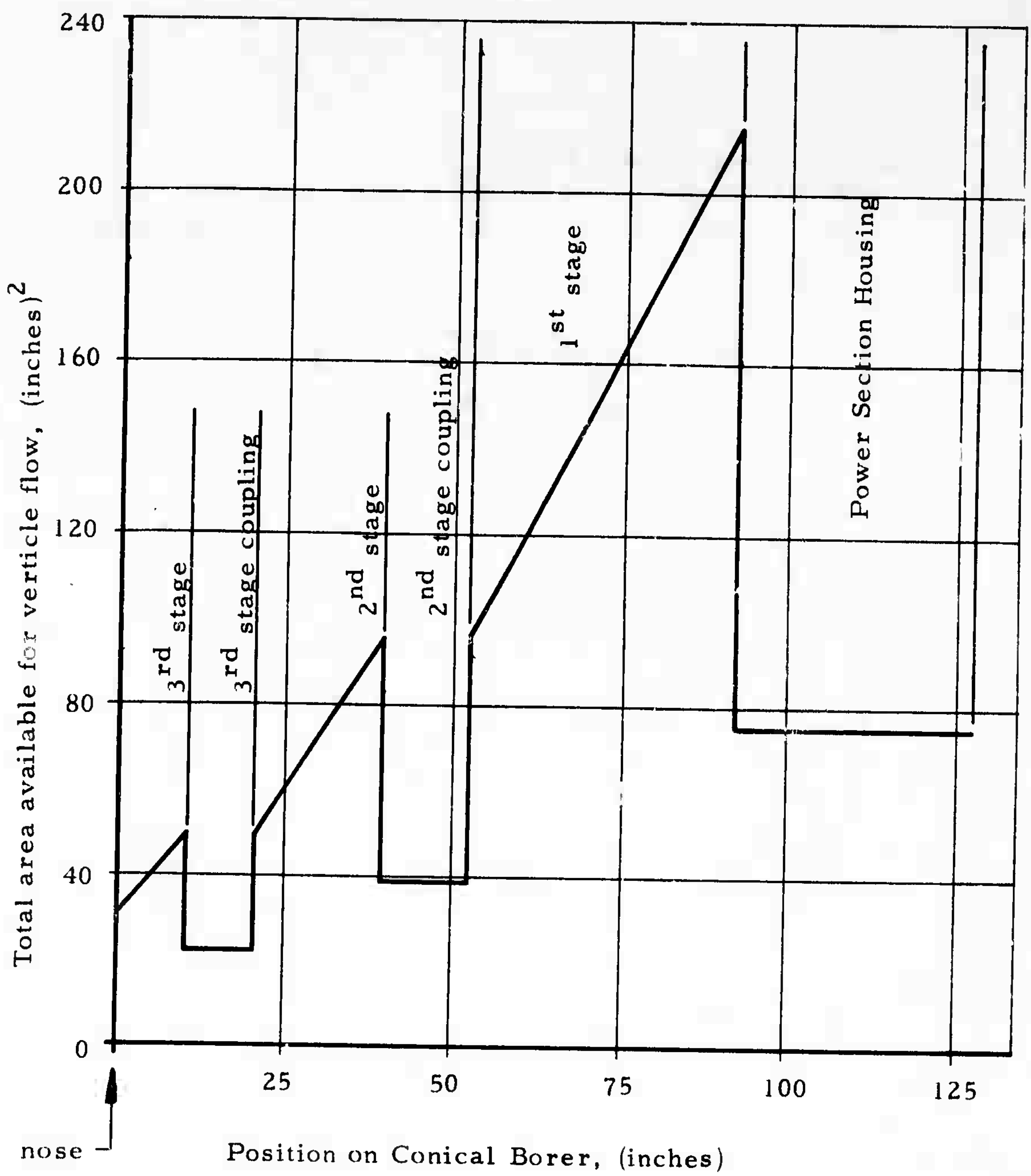
Basically, the discharge pipe analyses have established reasonable guides for the design of the pneumatic mucking system. Air velocity in the discharge pipe of approximately 90 feet per second will result in minimum pressure drop and power requirements for the suction system. A pipe diameter of 8 inches or greater would be desirable, if the logistics of transport and handling can be tolerated.

6.2 Transport in the Area of the Cutters

As stated in Section 5.4 it is anticipated that the major technical problems in the mucking system design will occur due to velocity discontinuities encountered in the vicinity of the conical cutter sections of the borer. The problems are created by abrupt changes in the air-cuttings flow paths between cutter stages and by the very large area at the top of the powered cutter stage. The areas available for vertical air flow were obtained from the current design layouts and are shown in Figure 18. The typical configuration of these flow paths in the area of the conical cutters is shown in Figure 19. The configuration in the area of the inter-stage couplings and power section housing are simply annular rings between the borer and the hole wall.

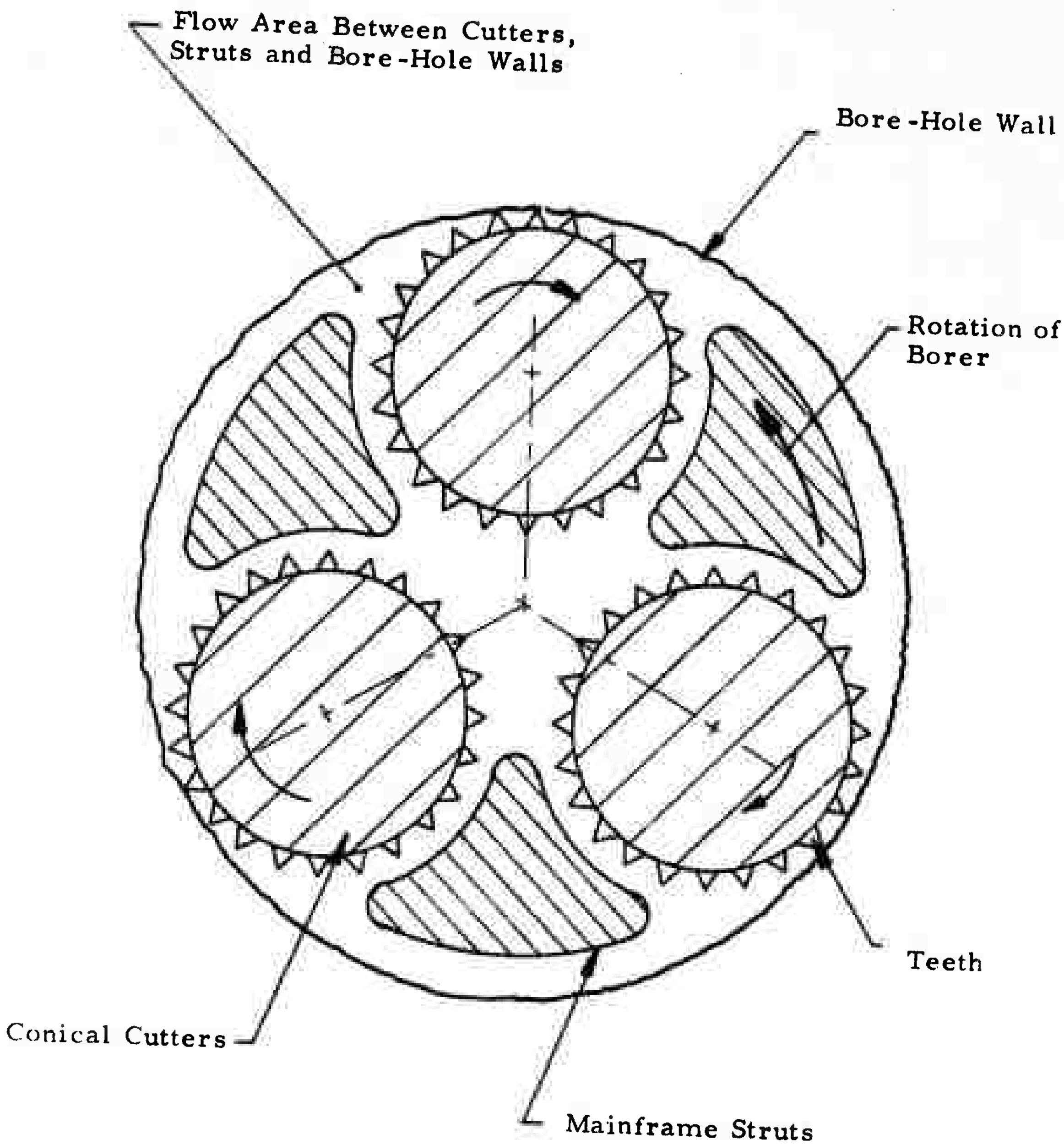
Several potential problems became apparent after studying these flow paths:

(a) The divergent areas along the cutter sections would result in decreasing air velocities and possible subsequent particle separation.



Total Projected Area Available for Vertical Air Flow at Various Positions on the Conical Borer

Figure 18



<u>Typical Configuration of Flow Paths in the Area of the Conical Cutters</u>

<u>Figure 19</u>

(b) The abrupt change in area at the couplings implies rapid changes in air-particle direction and velocity which could result in excessive pressure drops and abrasive wear on local members.

(c) Flow around and between the cutters would be largely affected by the cutting teeth. Although there might be little restriction to air flow, considerable particle interference with the teeth would drastically reduce the transport efficiency.

(d) Particle transport in the area of the inter-stage couplings would be inhibited by the rough bore hole wall.

(e) To insure sufficient air velocity (based on the total available flow area) near the top of the first stage cutters would require flow rates of four to five times the optimum for the discharge pipe.

Various conceptual solutions to the above problems have been generated and reviewed and are discussed below.

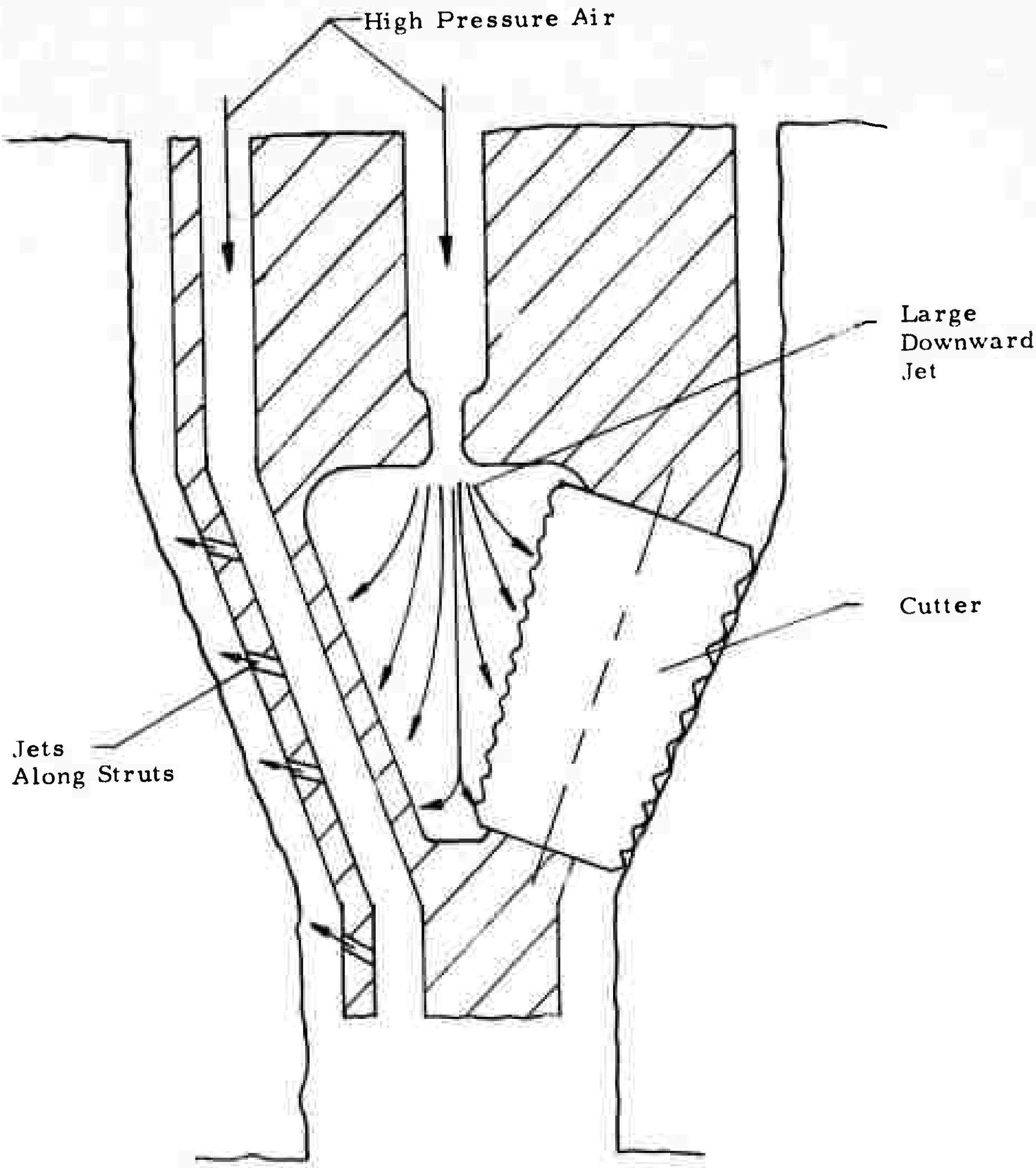
The problems associated with the divergent areas along the cutters could be overcome by distributing the incoming air through nozzles along the length of the borer struts as shown in Figure 11. The volume of air induced at each level would be proportioned to maintain uniform air velocity along the cutters. The high velocity jets, in this concept, would be directed upward and spaced properly to create a high velocity upward stream between the borer and the rock wall. It was assumed that the relatively close clearances between the cutter teeth and the struts would provide a sufficient flow barrier to prevent diffusion of the stream and recirculation of the cuttings. The total volume of flow would, therefore, be confined to the outer regions of the cutter stages and be more consistent with the discharge pipe requirements.

However, the effectiveness of the tooth-strut flow barrier, the transition of flow from stage to stage and the tendency of the cutters to drag air and particles into the center of the cutter stages are all areas of concern which could severely affect the efficient performance of the above concept.

Another concept solution is shown in Figure 20. A relatively large, downward pointed air jet would be incorporated above the center of each cutter stage in addition to the upward pointed jets along the mainframe struts. The large jet would be designed to diffuse in such a way as to prevent upward flow between the cutters and to result in a net radial outward flow between the cutters and the struts. The total air volume would be sufficient to carry the cuttings upward along the hole wall since inward flow would be blocked by the outflow of air from the large jet.

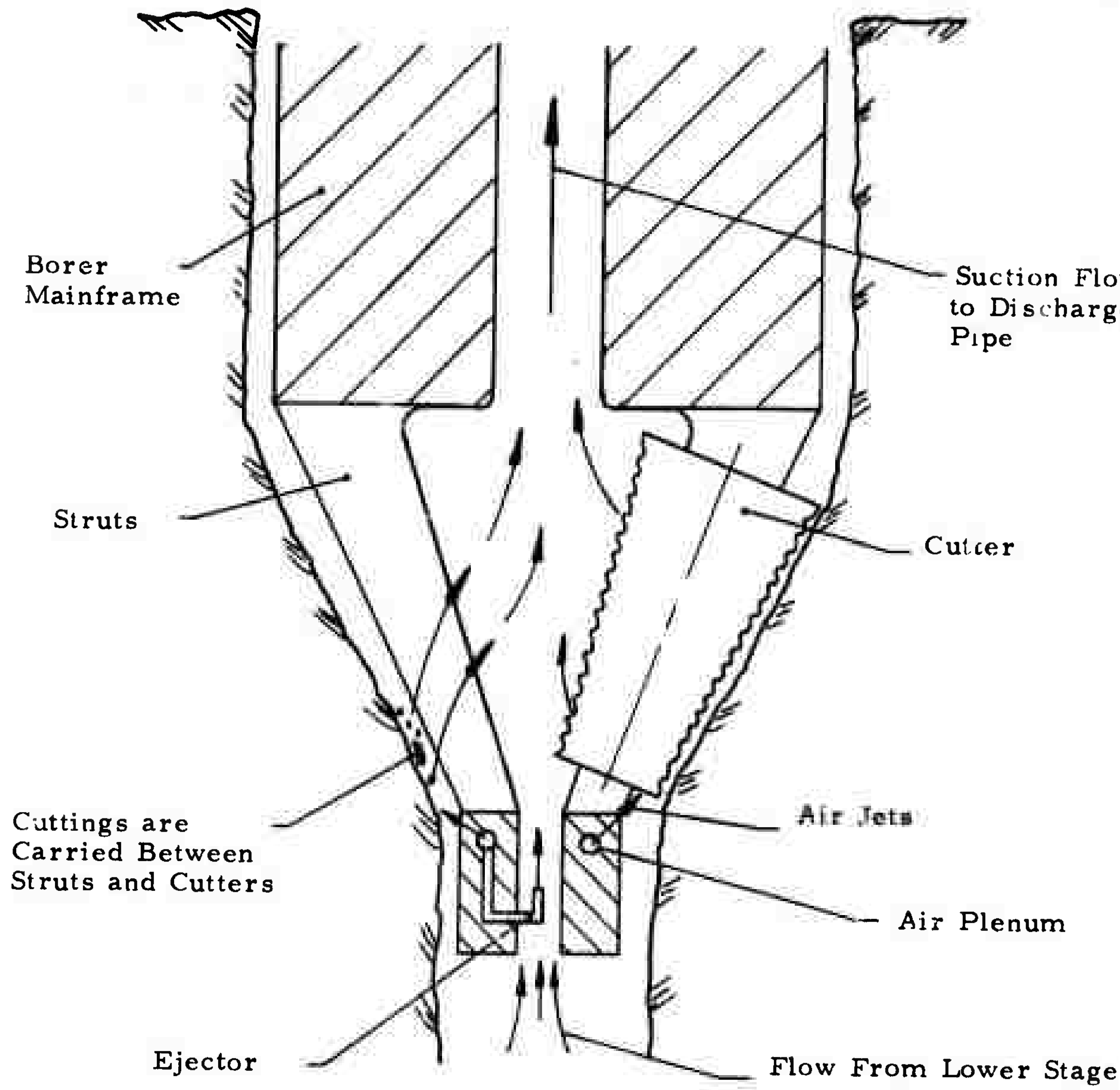
The design of the large jet would be critical, of course, and the flow required to make it effective would detract from the effectiveness of the small jets along the struts. Also, the efficiency of transporting cuttings along the rough walls of the hole would be relatively low.

A third concept would incorporate a suction pipe above the center of each cutter stage as shown in Figure 21. Suction at the top of the idler cutter stages would be generated by ejectors which would be powered by the high pressure air supply for upward pointed jets at the bottom of each cutter stage. The jets would propel the cuttings upward along the hole wall where they would be



<u>Downward Jet Concept for Transport in the Area of the Conical Cutters</u>

<u>Figure 20</u>



Suction Pipe Concept for Transport in the Area of the Conical Cutters

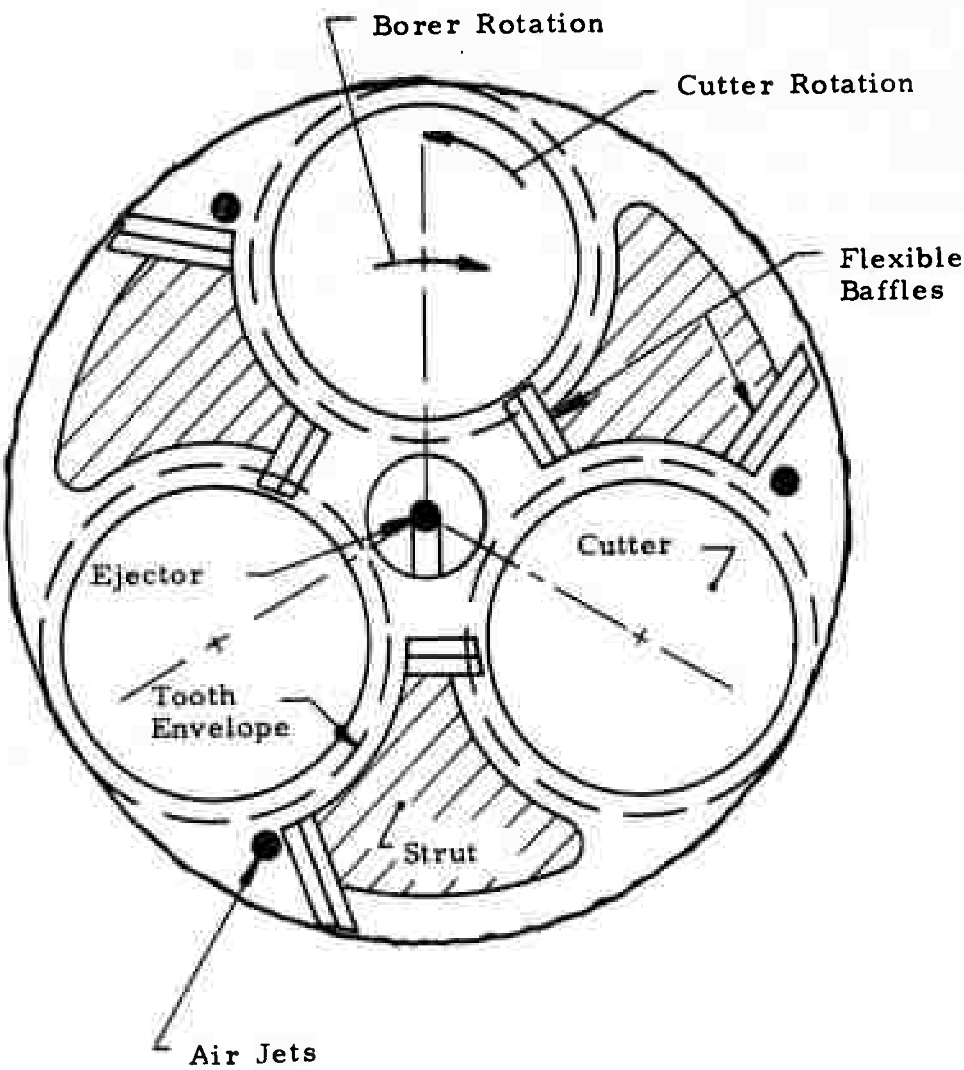
Figure 21

drawn into the center of the stage by the cutter rotation and the vacuum pipe air stream. The central air stream would be continuous from stage to stage and pass through the rotary union and discharge pipe to the surface vacuum system. Flexible baffles as shown in Figure 22 would control the air stream and prevent major recirculation and regrinding of the cuttings.

The advantage of this concept is that the air and cuttings would be immediately collected in a confined cylindrical, smooth pipe between stages, which would afford efficient and smooth transfer from stage to stage and up through the rotary union to the discharge pipe. There are, however, several disadvantages as listed below:

a. Since all of the air flow would be directed upward and a major portion of the total volume would be required to establish the central upward stream, the volume afforded to the jets would be significantly reduced and the resultant air velocity outside of the cutters might not be sufficient to transport the cuttings.

b. Assuming that the cuttings in the annulus between the cutters and the rock wall could be propelled upward by the high velocity jets (even though the net air velocity might be low), the particles must then pass between the cutter teeth and the struts to get into the suction pipes. Although the cutter rotation might tend to draw the cuttings into the center, the surface speed of the teeth would be considerably lower than the terminal velocity of the particles and hence would tend to block the transfer rather than assist it.



Cut-away of a Typical Stage to Show Baffles and Air Jet Locations for the Suction Pipe Concept

Figure 22

(c) The incorporation of pipes between stages would be difficult since most of this space is required for the cutter bearings and structure.

(d) The incorporation of baffles which would effectively control the flow stream, withstand the environment and not contribute to or cause jams would require considerable design effort.

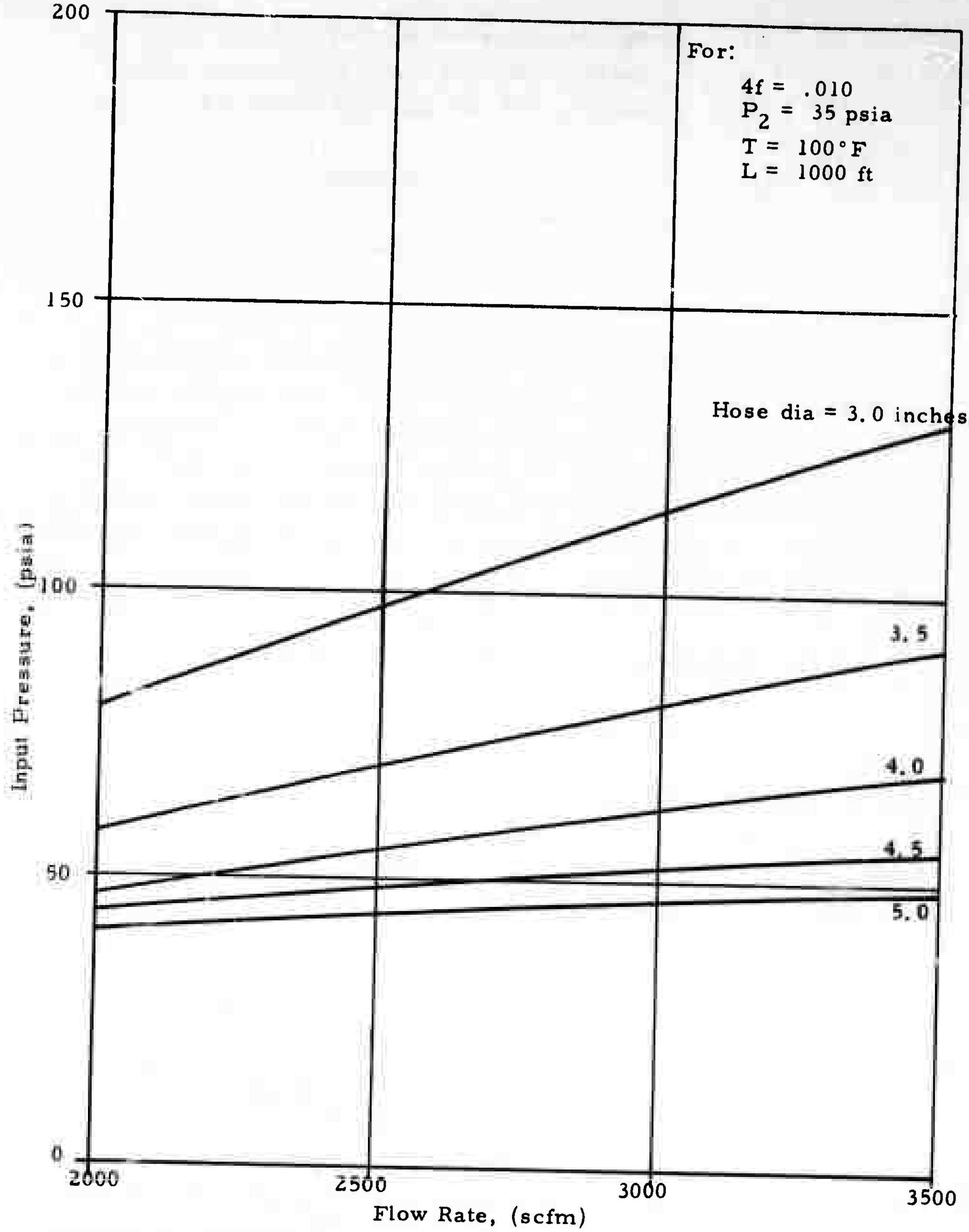
Each of the concepts described above has significant advantages and disadvantages which could not be quantitatively compared by simple analytical means. A mockup system is therefore being built to test the various concepts and establish the most promising one or combination which best satisfies the overall system requirements. A detailed discussion of the proposed mock-up model tests is given in Section 6.4.

6.3 Analysis of Compressed Air Requirements

The compressed air requirements for the proposed mucking system have been analyzed to determine the effect of various hose sizes and flow rates. Calculations were based upon the relationships for steady flow (adiabatic) assuming a discharge pressure of 35 psia at the borer, a temperature of 100°F and a hose length of 1000 feet.

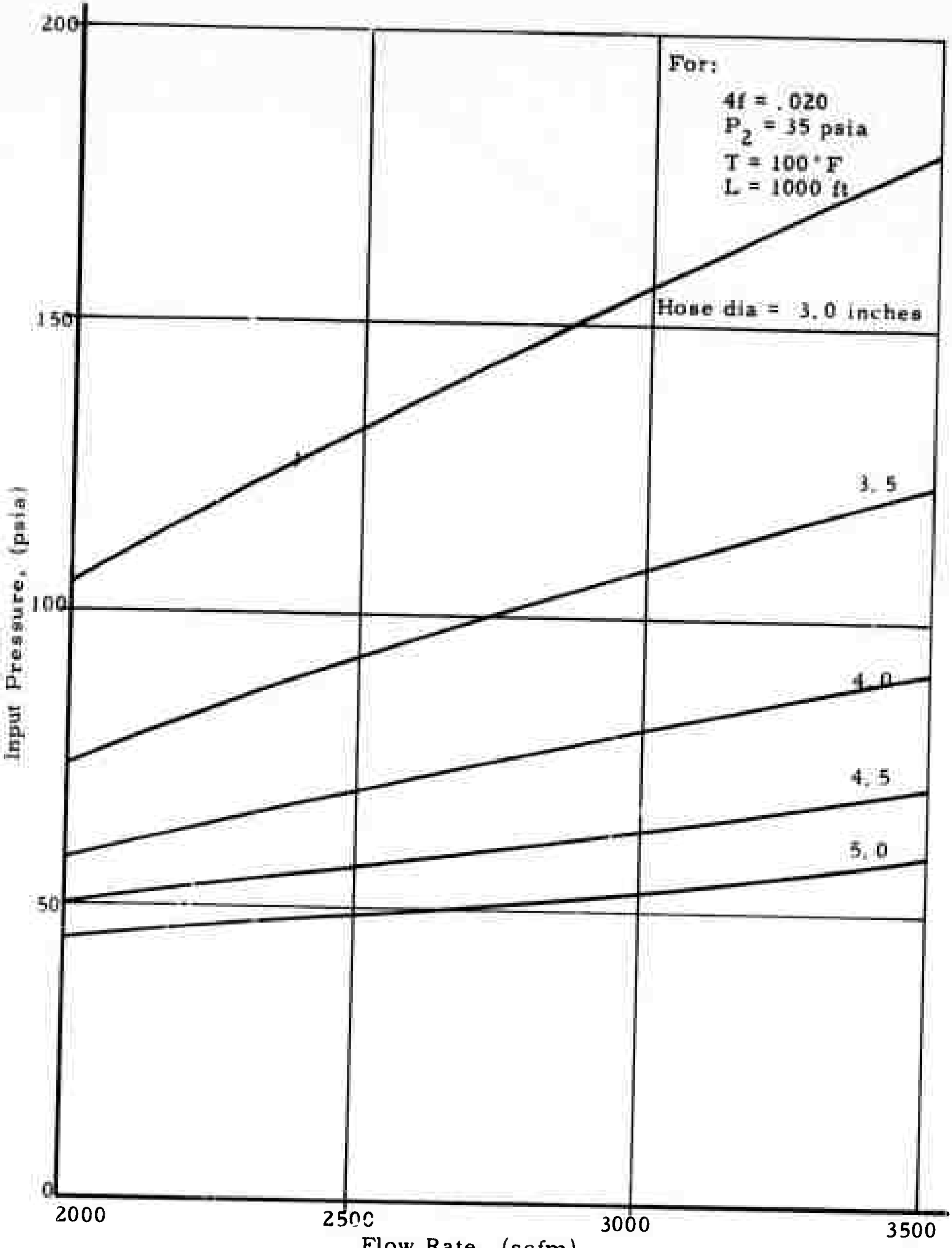
Figures 23, 24 and 25 show the inlet pressure required at the compressor as a function of flow rate for pertinent values of friction factor and hose diameter.

This data will define the compressor requirements for the mucking system once the actual flow requirements in the area of the conical cutters has been determined.



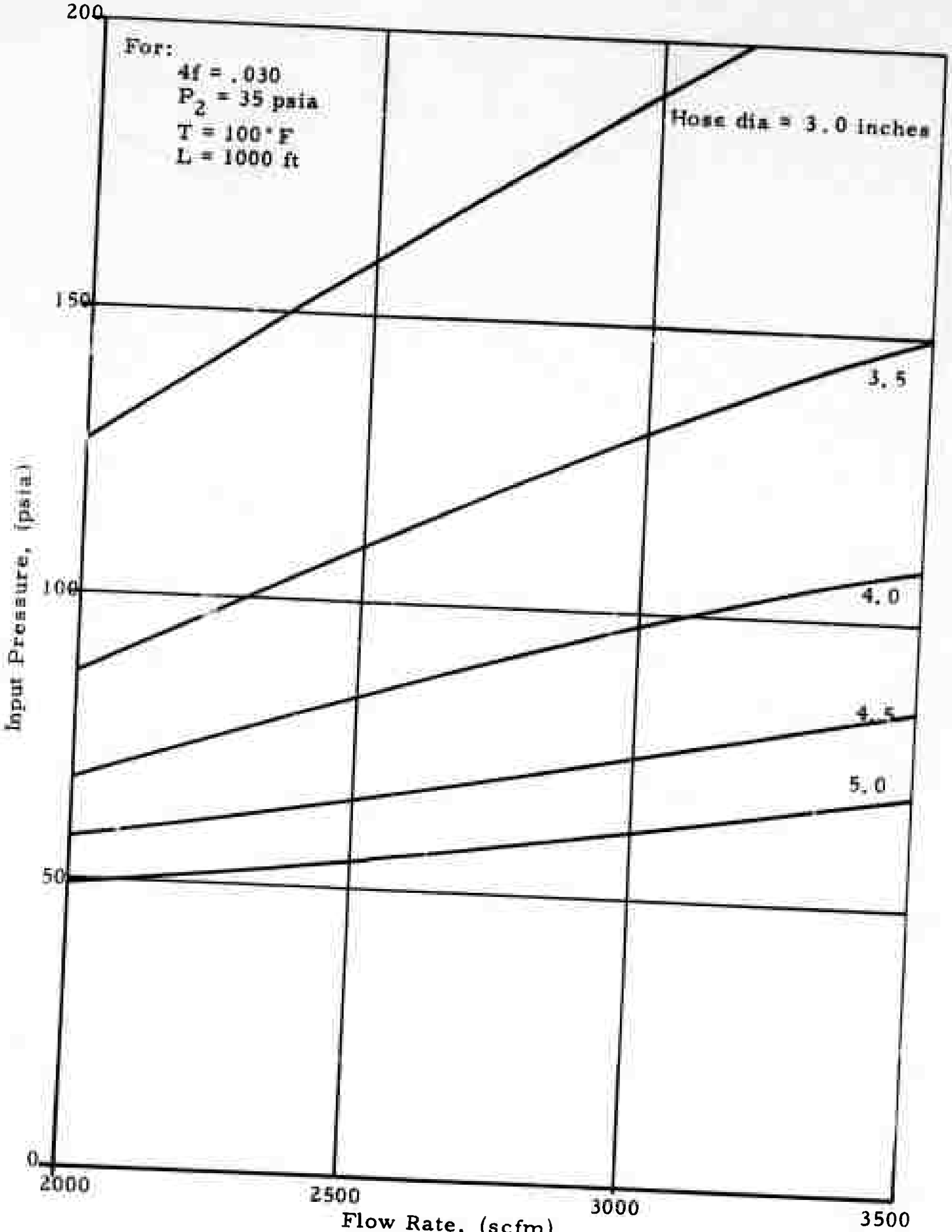
Compressor Input Pressure versus Flow Rate for Various Hose Sizes

Figure 23



Compressor Input Pressure versus Flow Rate for Various Hose Sizes

Figure 24



Compressor Input Pressure versus Flow Rate for Various Hose Sizes
Figure 25

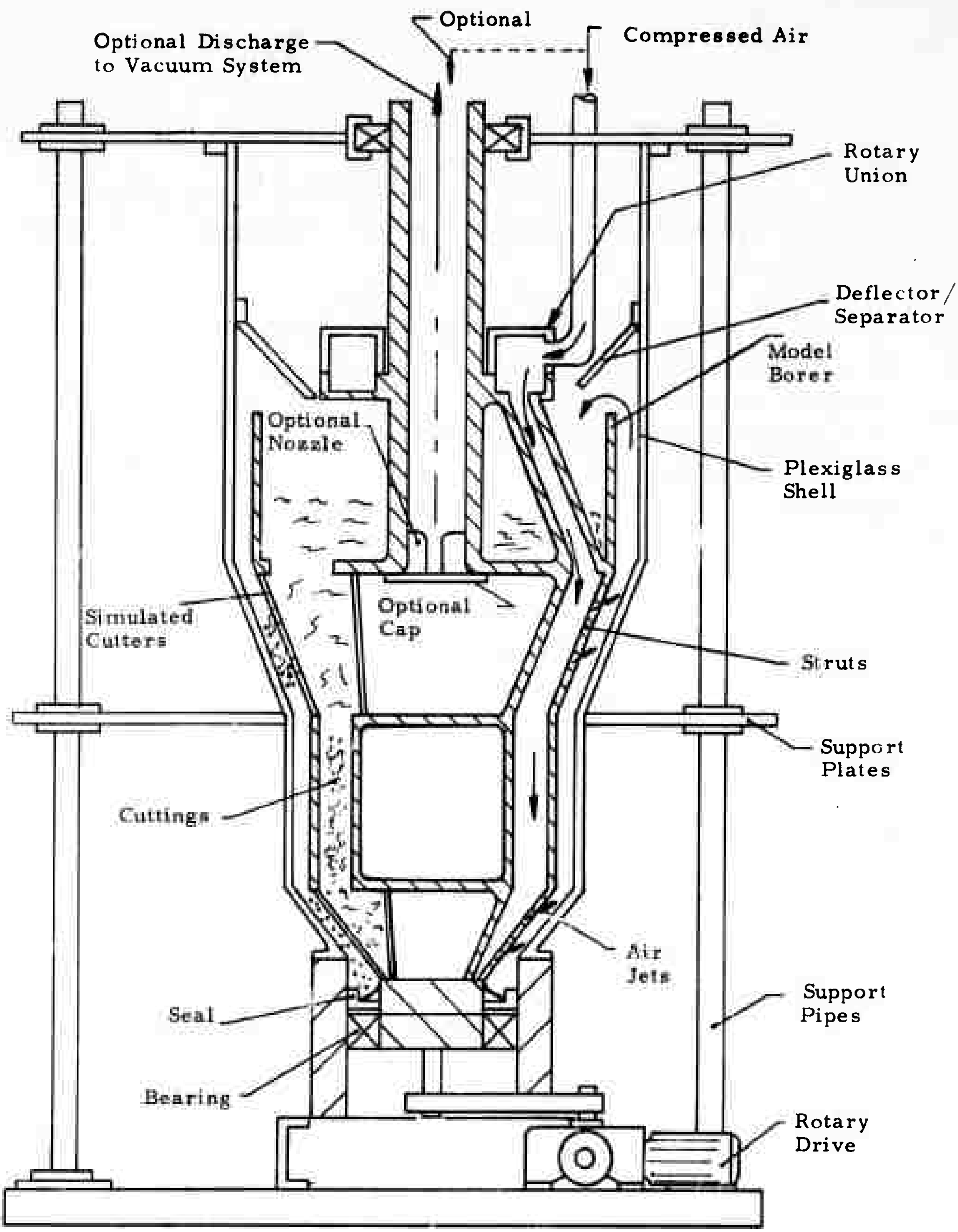
6.4 Mucking System Model Studies

Test studies of the proposed mucking system concepts are to be conducted to determine the optimum system and its operating requirements.

A test model, currently being constructed, will consist of a two stage, half scale, mock-up of the borer encased in a plexiglass shell which will simulate the bore-hole. The model will be rotatable within the shell with provisions for pressurized air supply, cuttings fed from within simulated cutters, and air-cuttings discharge and separation.

This model represents a departure from the contract statement but was considered necessary to resolve the potential problems in the area of the conical cutters as discussed in Section 6.2. The transfer of cuttings around the borer will be highly dependent upon the design and position of local air jets. It was believed necessary, therefore, to allow the jets to sweep over the bore hole surface (as they would in the real case) in order to establish their true working behavior.

A schematic description of the proposed mock-up model is shown in Figure 26. Compressed air will be fed through a rotary union and pipes to the struts of both stages. The struts have been cast of aluminum to provide a thick wall for the drilling of positively directed air jets and to withstand the high pressure of the supply air. Cuttings will be stored and collected above the upper stage and will flow down through the simulated cutter cones in both stages. Holes in the outer surface of the cones will allow cuttings to flow out at a prescribed rate. The conical sections of the plexiglass shell will be of two piece construction which will allow convenient access to the model borer for modifications or examination. The model borer will be suspended on bearings and be rotated at



Schematic of Mucking System Model

Figure 26

controlled speed by a variable speed rotary drive. Optional features incorporated in the design will provide sufficient flexibility for testing the various systems discussed in Section 6.2.

The model was designed half scale with two stages to minimize costs and complexity. The two stages will allow careful study of the interstage transition. It is believed that the data obtained from the half scale model testing can be confidently used in the design of the full scale system and that the need for closer simulation of the actual cutter area transport far outweighs the discrepencies in half size scaling.

7. Conclusions and Recommendations

The basic conclusions reached thus far in the program are as follows:

1. The experimental program has confirmed the main advantage claimed for the conical borer i.e., that it will require significantly reduced thrust loads (the required load is less than 10 percent for bits) than conventional roller cutter bits of the same diameter and at the same penetration rate.

2. The principles of operation of the conical borer that had been proposed have been experimentally verified. Based on these tests and the parameters that were established the design may proceed with confidence.

3. Mechanical implementation of the design concepts is feasible and the final design of the conical borer is well underway.

4. Supply and discharge requirements for the mucking system are practical but careful research in the vicinity of the conical cutters will be required to establish the overall system requirements.

It is recommended that the proposed effort be continued according to the scope of work outlined in the previous sections. More specifically, the following tasks should be completed:

(a) Continued detailed design of the mainframe, drive system and cutters based on the design parameters established by the Phase I results.

(b) Assembly of the mucking study model and testing research to determine the best solution to problems in the vicinity of the conical cutters.

(c) Integration of the mucking study results into the overall borer design.

(d) Final detailed design of the conical borer and ordering of long-lead-time parts.

It is also recommended that the third year effort to build and test the conical borer be planned and budgeted. This should be accomplished soon in order that the entire program can be completed within the calendar year 1973. A detailed breakdown of the required effort and costs is being prepared and will be submitted under separate cover.

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APPENDIX A

THE CONICAL BORER PRINCIPLE

A.1 Simple Conical Geometry

The operating principle of the conical borer is best illustrated in terms of a simple conical roller-cone bit. It must be noted that such a bit would not be practical, at least in a self-advancing form, but this has no bearing on the principle of operation. Let us compare a conventional bit that cuts a flat bottom hole with a bit of the same diameter that cuts a conical hole.

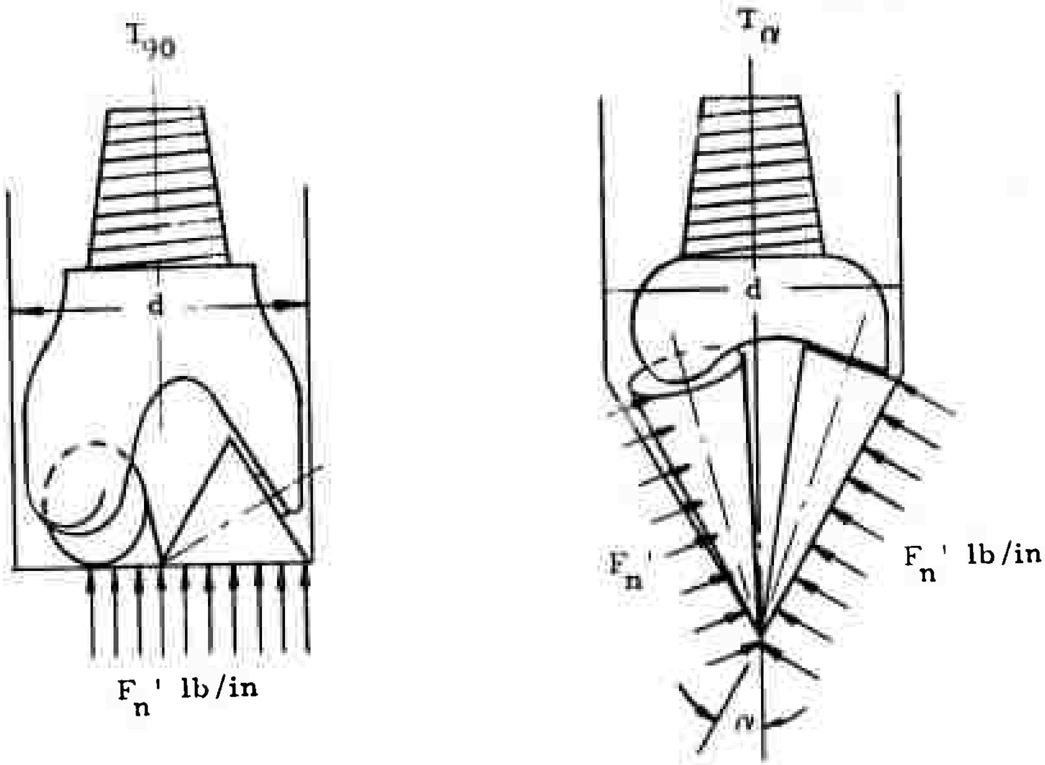
The conventional bit, shown in Figure A-1a, experiences a distributed line load, F_n' pounds per inch, along the rolling contact of each cone. Load distribution is assumed constant along the line length for convenience in illustration only. Since there are three rollers, the total roller length is $\frac{3}{2}d$, and the thrust required to generate the loading is simply

$$T_{90} = \frac{3}{2} d\ F_n' \qquad \text{(A-1)}$$

where the subscript 90 signifies a 90° hole-bottom half angle (flat bottom).

Consider next the three-roller conical bit of Figure A-1b, loaded to the same line load F_n', and having the same tooth geometry but at a half angle α. Simple statics indicates that, whatever the angle α, the thrust to generate this load remains

$$T_{\alpha} = \frac{3}{2} d\ F_n' \qquad \text{(A-2)}$$



(a) Flat-bottomed (90 deg) Bit (b) Conical (α) Bit

Flat and Conical Bit Comparison

Figure A-1

Thus there has been no thrust reduction. However, if we examine a representative unit length of cutter at the same bore radius on each of the bits, they are essentially identical, with the same tooth geometry and load. The slight rock surface curvature of the conical bit should have no influence on fragmentation behavior except, perhaps, very near the tip. Hence, in one revolution, a unit length of conical bit roller should fragment the same quantity of rock as a unit length at the same mean radius on the conventional bit. But the conical bit contains more cutter length, in the ratio $\frac{1}{\sin \alpha}$, and hence will advance faster in this same ratio. Conversely, if we wish to advance at the same rate as the conventional bit, the required load can be reduced. With the usual assumption that, with good cleaning, penetration per revolution is proportional to cutter load per inch, the thrust can be reduced in inverse proportion to the ratio of cutter lengths. Thus, in comparing the performance of conventional and conical bits run at the same speed, we can write the following relationships for advance rate, R, and thrust, T:

$$R_\alpha = \frac{1}{\sin \alpha} R_{90} \quad \text{at constant } T \tag{A-3}$$

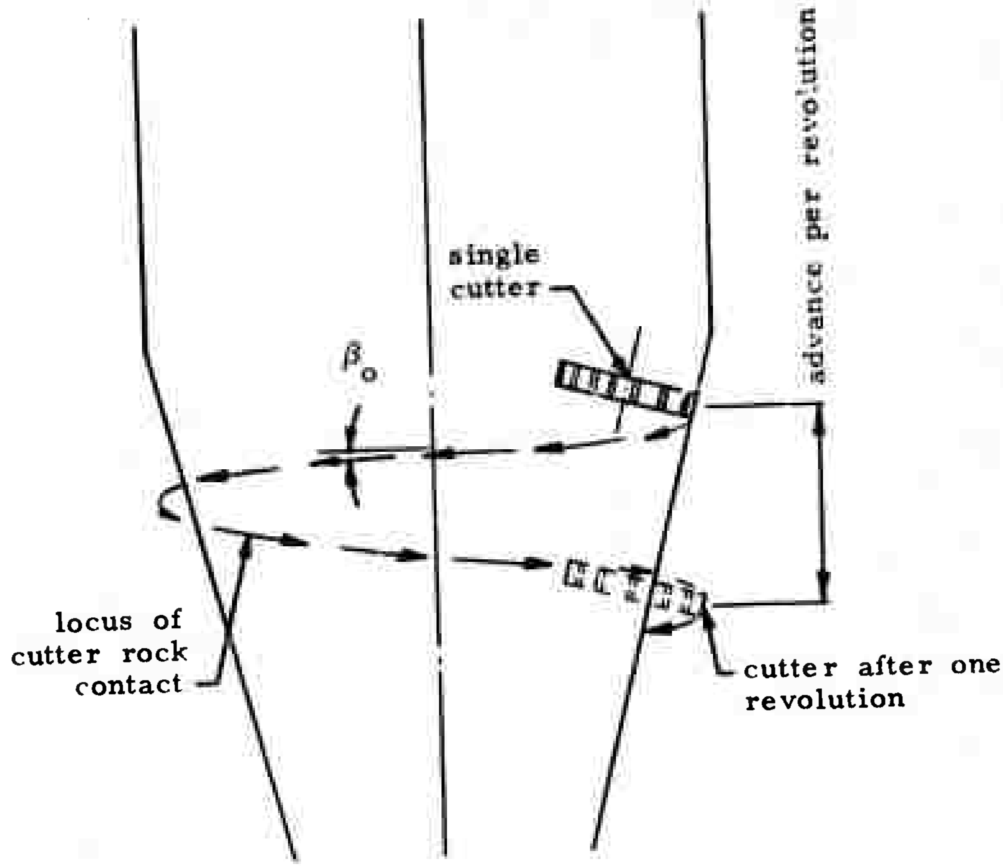
$$T_\alpha = \sin \alpha \; T_{90} \quad \text{at constant } R \tag{A-4}$$

In passing we note that Figure A-1b illustrates the impracticality of simple conical bits, i.e., the impossible bearing and stress situation presented by long, thin, cantilever-mounted rollers. This is certainly true for small α as would be required for self advancement (see Section A. 3), but it is possible that, with sufficient development, simple bits could be constructed to approximately halve the thrust required by conventional bits. Thus the self-advancing borer, the subject of this effort, is presently considered as a reaming device. For rescue service, where a pilot hole is necessary to locate and communicate with the victims, this is not considered a handicap.

A.2 The Helical Cutter Path

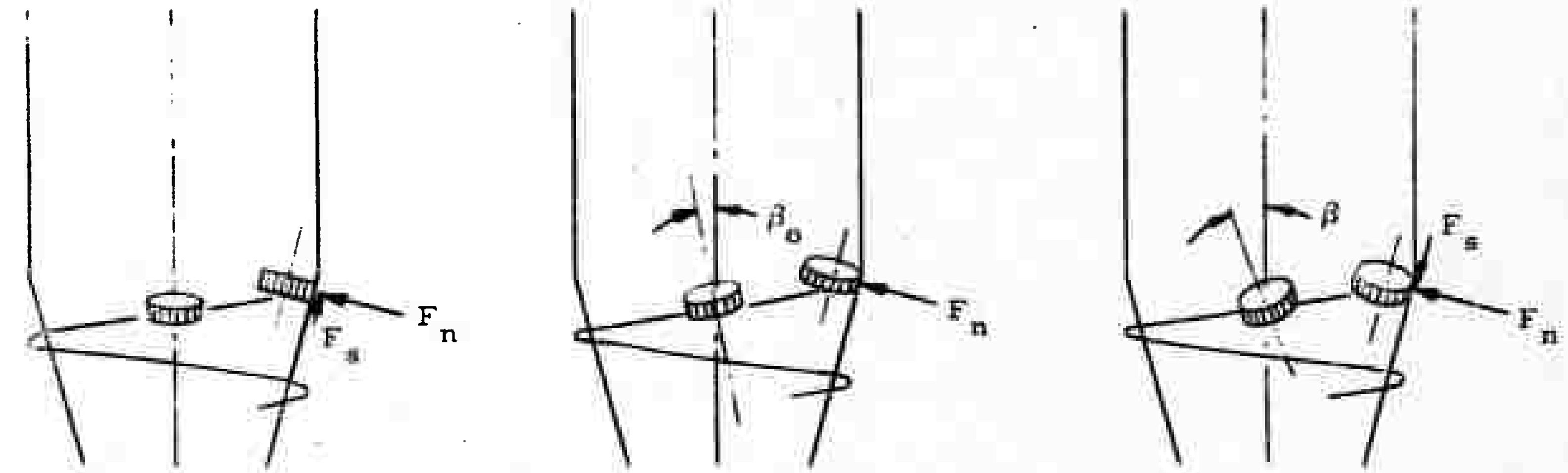
The simple description of Section A.1 is based upon an examination of a normal force at the cutter-rock interface. There is, in addition, a rolling force which requires the application of torque to rotate the bit and through which energy is transmitted to fragment the rock. In addition to these well-known forces, there may be a side force on the cutter, depending upon whether the cutter is or is not in pure rolling contact with the rock.

For simplicity, let us consider the motion of a single row of cutter teeth on one roller, as shown in Figure A-2. As the bit rotates and advances, the locus of the cutter-rock contact point is not a simple circle but is a helix as shown for greatly exaggerated advance per revolution. If the roller is to follow this helix in pure rolling contact, clearly it must be tilted or skewed forward to the helix angle, β_o. If the cutter is not skewed, the axial component of its motion will be accomplished by skidding, and accompanied by a side force on the cutter, as shown in Figure A-3a. This side force has an axial component which would be undesirable in that it would require the application of additional thrust to cause bit advance. Skewed to β_o, the cutter would experience pure rolling and the simple situation of Figure A-2, shown again in Figure A-3b would prevail. If, now, the cutter can be skewed beyond the advance helix angle β_o, as shown by Figure A-3c, the cutter teeth will experience a rearward skidding motion as they contact the rock.



Helical Cutter Element Advance

Figure A-2



(a) No Skew $\beta = 0$

(b) Neutral Skew $\beta = \beta_o$

(c) Forward Skew $\beta > \beta_o$

Skewed Cutter Geometries and Forces

Figure A-3

Qualitatively, the cutters attempt to roll ahead of the advance helix and, in so doing, they develop a side force which assists, rather than hinders, the advance of the bit. The action is somewhat like that of a screw, but not precisely in that the side force in question is frictional in origin and in no way dependent upon the cutter following a prescribed path.

For any reasonable advance per revolution, the angle β_o is very small except very near the tip of the hole. At the tip, β_o is equal to 90°. However, as stated previously, there are stress considerations which make the tip of the conical hole inaccessible anyway. In addition, a skewed roller of finite length geometrically cannot reach the center of the hole. For a reaming device of moderate advance rate as proposed here, β_o is very small (less than typical machining tolerances). Its variation over the length of the borer is of no consequence because all cutters will be skewed to angles substantially greater than β_o in order to achieve self-advancement.

A. 3 Skewed Cutter Forces and Self-Advancement

Skewed cutters have been used on conventional bits for some time, particularly on those for soft and medium rock. In such applications, the teeth are said to display a "gouging and scraping" action which considerably enhances penetration rate. For a conventional bit the existence of a side force on skewed cutters is of no overall consequence, since such forces are radial and self-canceling (although, of course, they do affect individual cutter bearing loads).

The beneficial effect of cutter side force is of major concern in the operation of the proposed conical borer. For, if the side force is large enough, or if the hole-bottom angle is small enough, the borer can be made self-advancing, that is, no external thrust will be required.

The desired condition is very simply shown in Figure A-4. Considering any one cutter-rock contact point, no external axial force will be required if the axial component of the side force, F_s, can be made equal to or greater than the axial component of the much larger normal force F_n. That is,

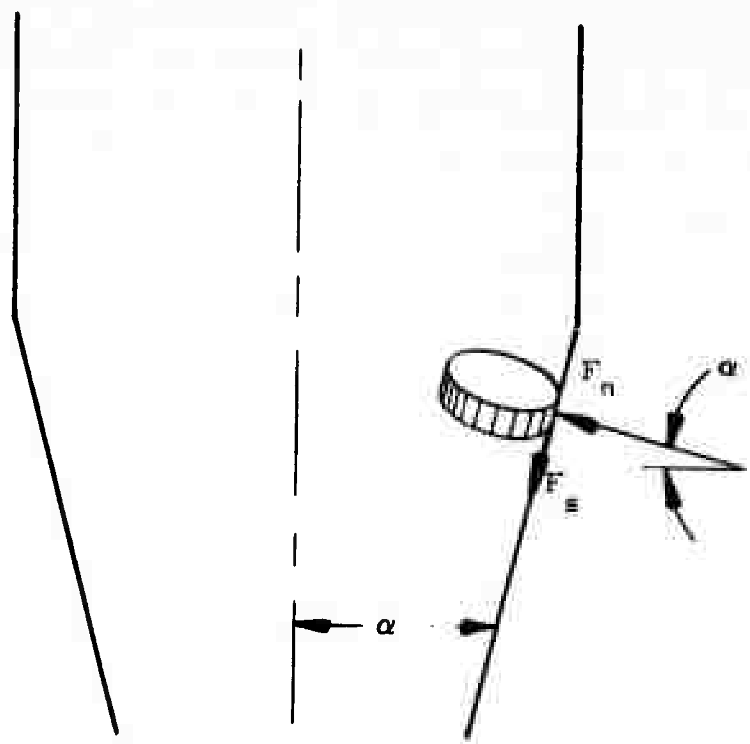
$$F_s \cos \alpha \geqslant F_n \sin \alpha \tag{A-5}$$

or

$$\frac{F_s}{F_n} \geqslant \tan \alpha \tag{A-6}$$

In this formulation, as in actual behavior, the action of the self-advancing borer is analogous to that of more conventional devices wherein α is the "friction angle" below which a member will not slip, and F_s/F_n is the coefficient of friction. The conical borer is, however, self-advancing rather than simply self-locking, because the rollers permit rotation even though the borer is axially locked by the side forces on the cutting teeth.

The kinematics of skewed cutter motion and the accompanying forces are best seen in terms of linear rolling over a plane rock



Rock-forces on Skewed Cutter for Self-Advancing Action

Figure A-4

sample. Tests of this type have been performed* and the available data are sufficient to permit the design of a prototype, self-advancing conical borer. Single row cutters of conventional tooth size and shape were rolled over a variety of rock specimens as shown in Figure A-5. Normal force, F_n, side force, F_s, and rolling force, F_r, were measured separately and correlated for a variety of skew angles, cutter penetrations, cutter diameters, and rock types. "Sharp" and "dull" teeth were included in the study. In terms of the important force ratio, F_s/F_n, the data indicate useful side force ratios at moderate (4°) skew angles, with little or no variation with rock type or the "dullness" of the teeth. A slight decrease of F_s/F_n with increasing penetration is noted, which is of benefit in the stable operation of the conical borer (see Section A. 4).

The kinematics of individual tooth penetration and the origin of the cutter side force are shown in Figure A-6 for a plane surface. The "footprint" of the tooth is shown for several subsequent positions as seen by an observer riding with the cutter. It can be seen that the tooth initially contacts the rock surface at a position laterally displaced in the direction of the skew from that when the tooth is at maximum penetration. In tests, the side-slip of the tooth appeared to be approximately parallel to the long dimension of the footprint. This motion generates a side force on the tooth, tending to resist the side-slip in a manner which is believed to be frictional. Side-slip continues beyond maximum penetration, of course, but the tooth is unloaded in this region and little side force will result. In transferring these sketches to the

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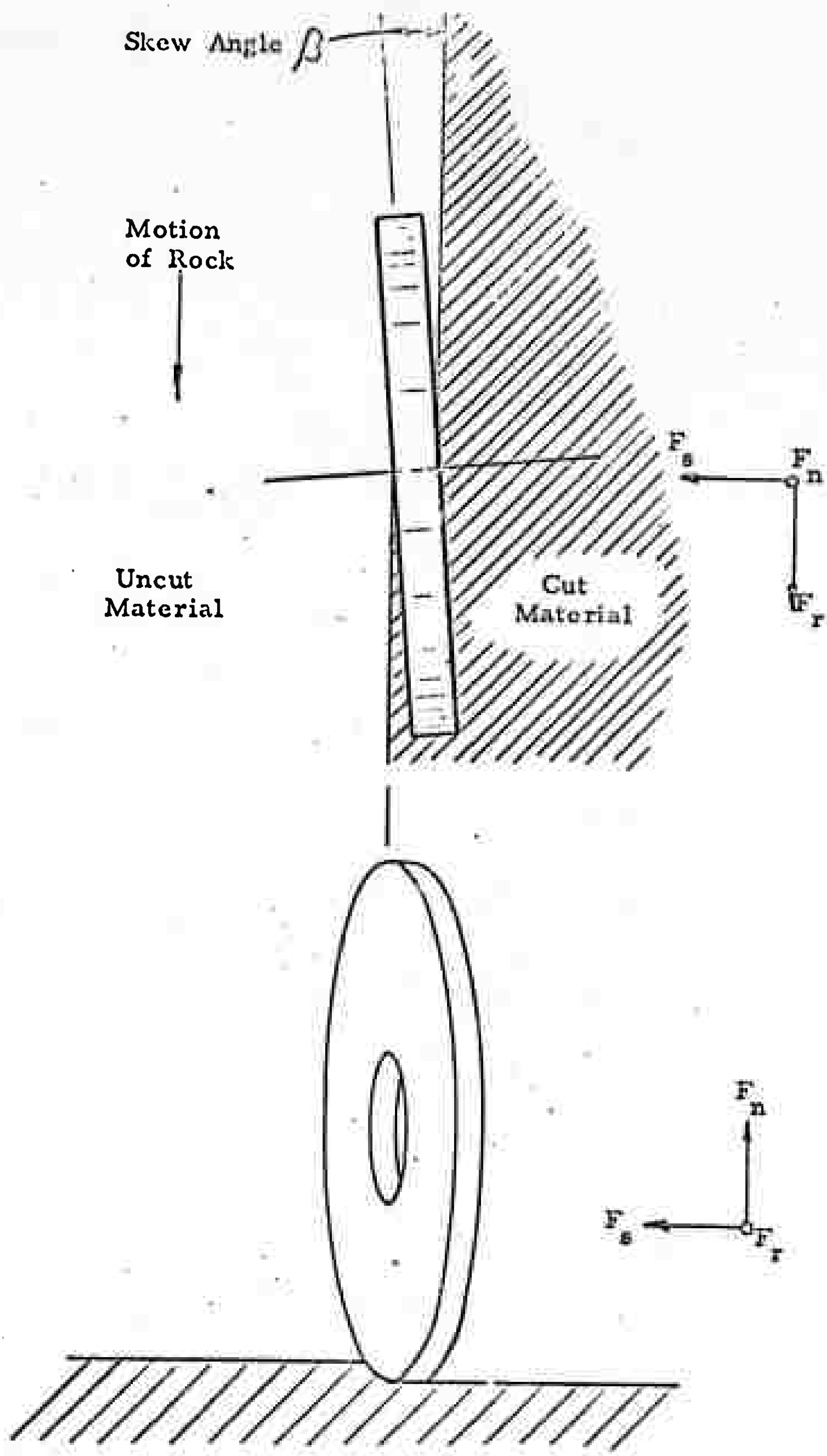
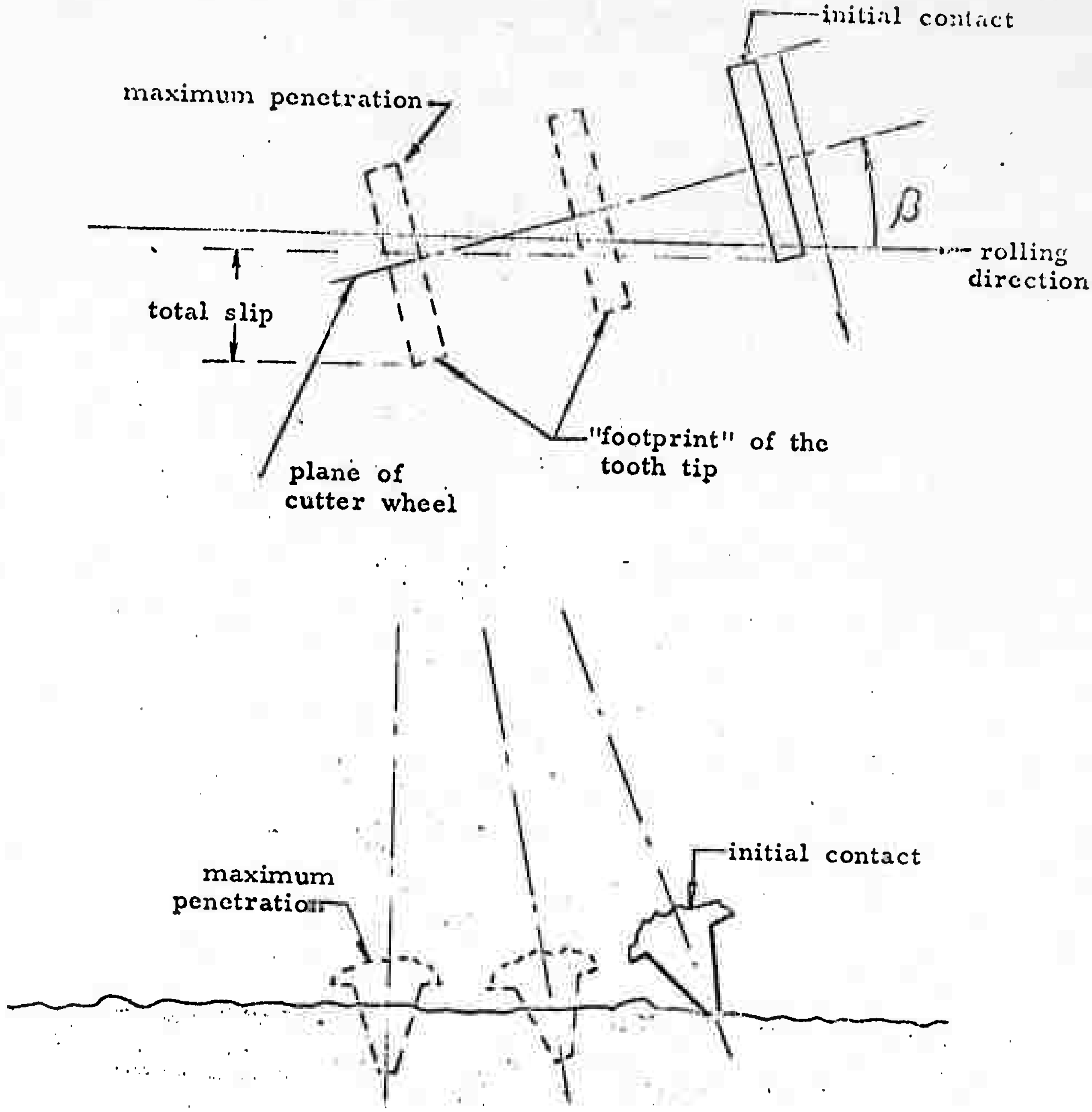
* Peterson, C. R., "Rolling Cutter Forces", Paper No. SPE 2393, Society of Petroleum Engineering of AIME, also published in Society of Petroleum Engineering Journal, March, 1970.



Plan View of Cutter on Rock
(Forces Defined are Forces on Cutter)

Figure A-5



Skewed Cutter Penetration Geometry

Figure A-6

conical geometry, the top of the page corresponds to the apex of the cone; hence the tooth slip is toward the base of the cone. It is important to note that in the performance of these tests and in the operation of a self-advancing borer, the side-slip is actually present at all times. The action is not dependent upon the precarious maintenance of an impending slip.

Typical data indicate a side-to-normal force ratio of approximately 0.2 at a skew of 4°. From Equation A-6, then a self-advancing borer of this skew would require a hole-bottom angle, α, of about 11°.

Greater skew angles generate greater F_s/F_n. This would permit greater hole-bottom angles and, hence, shorter borers. On the other hand, greater skew also contributes to more rapid tooth wear. Commercial applications of the conical borer would necessitate an optimization of the relationship between skew angle and tooth life, probably through rather extensive field testing and design evolution. However, for mine rescue service, considerable tooth life can be expected at 4° skew and the prototype would be constructed at approximately that angle.

While the cited data provide a preliminary design choice, there remain some uncertainties which suggest that the proposed borer development program should include some testing of simple components before the more complex components are constructed. For example, the ratio F_s/F_n is somewhat dependent on the ratio of cutter diameter to penetration. As can be seen from Figure A-6, a tooth on a larger diameter roller, at a given penetration and skew angle, would experience a greater arc length of contact and distance of side-slip, and would generate a larger side force. Similarly, if the rock surface were concave, as it would be in the conical borer, each tooth would again experience a greater side-slip. That is, self-advancing conditions in a conical hole will be somewhat easier to realize than the flat surface data would indicate. Finally, in a vertically downward boring application, one can use the weight of the borer to assist penetration.

The self-advancing condition depends on the ratio of side-to-normal force, and not on the magnitude of the side force. Tests to date have shown that this ratio is quite independent of the rock hardness. Hence, each row of cutter teeth, if skewed properly, will provide its own necessary side force regardless of local rock conditions. Therefore, the complete borer will be insensitive to extreme inhomogeneity in rock properties over the length of the borer.

A.4 Stability of Self-Advancement

The observed behavior that F_s/F_n decreases with increasing penetration contributes to the stable operation of the conical borer. Once the borer geometry is is fixed in terms of cone angle, skew angle, and tooth geometry, its advance behavior is fixed. Suppose that the actual F_s/F_n is greater than the designer anticipated at the design penetration (as it may well be on a concave surface). The resultant force on the cutting teeth will then have a forward component, and the borer will pull itself in to greater penetrations. Greater penetration will, of course, generate greater forces and require greater cutter torque. However, provided the borer does not stall or break first, as it cuts deeper the ratio F_s/F_n will decrease. Furthermore, the helix advance angle, β_o, will increase so that the effective skew, $\beta - \beta_o$, will decrease.

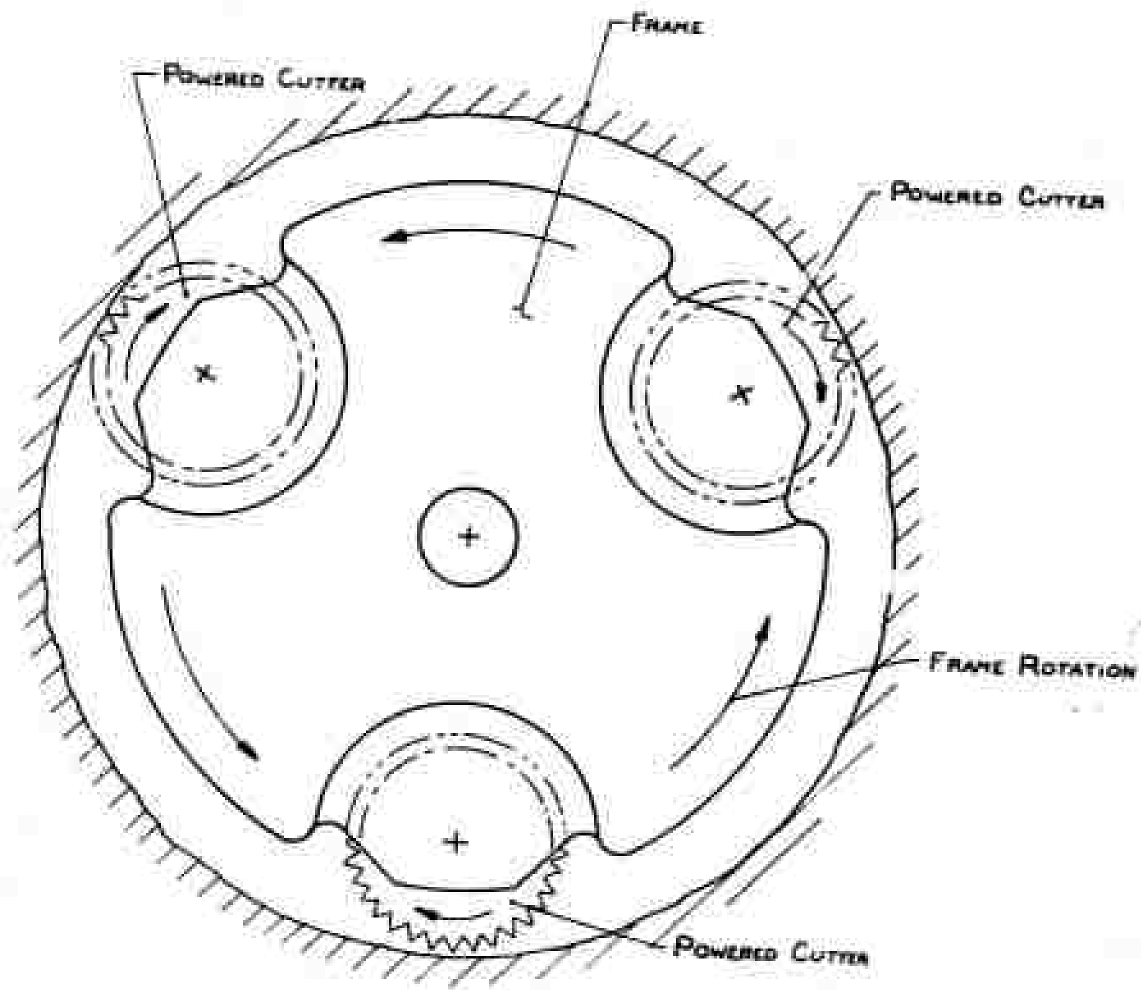
Thus, stable operation will be found at some greater penetration. The major design task is clearly to avoid a geometry which stalls or breaks before this stable operation is reached. Design variations to achieve the desired performance are clear cut, but, at present, preliminary testing of borer components is indicated to assure proper performance.

A.5 Self-Rotating Drive

The conical borer was first conceived to reduce or eliminate the need for external thrust. As an added benefit, the much greater total roller length of the conical borer permits the simultaneous attack of a much greater quantity of rock and, hence, a proportionately greater advance rate. Since the basic rock fragmentation mechanism has not been changed, the power input and torque would also be proportionately greater. The permitted advance rate increase, therefore, would require a torque increase which might be inconvenient or impossible.

An analogy might help to illustrate this point. Consider an ordinary twist drill, as used for metal. The tip of such drills is slightly conical in shape. A more acute tip would permit metal cutting over a larger face and would result in a greater advance rate (thrust and speed being equal). But, even if tip stress problems could be overcome, such operation would not be desirable simply because the drill body cannot transmit the required torque.

Excess torque requirements can be eliminated by simply driving individual rollers. In fact, the need for external torque is completely eliminated by driving rollers from frame-mounted motors. This self-rotating drive is no different from that of any conventional, self-propelled wheeled vehicle in linear motion. The concept is most easily visualized in terms of small, individually-driven rollers on a large frame, such as shown in Figure A-7.



Torqueless Rotary Drive from Frame-Mounted Motors

Figure A-7

The concept of driven rollers is not new, although their use on a conical borer might be. The conical hole shape, in fact, lends itself to driven-roller drive because of the relatively small angle between roller and hole axes.

In addition to avoiding a possibly excessive torque requirement, a self-rotating drive has other advantages. Since neither external thrust nor torque need be provided, the boring unit can be simply suspended in the hole on a cable. Power and cleaning air or liquid can be supplied via flexible lines connected through suitable slip rings or rotary unions. Cuttings return can also be done in a non-rotating flexible line or by a number of other methods. Thus, the entire suspension and communication system can be simply reeled in and out of the hole, with no need for rigid drill pipe, massive surface rotary drive, or excessively high hoist equipment. This, of course, is ideally suited to applications requiring lightweight, highly portable equipment. But beyond this special circumstance, the concept of a boring system without drill pipe has been the dream of practically every deep-hole borer.

A. 6 Summary of the Conical Bit Concept

The conical borer operates somewhat like a screw as it rotates and advances into the rock, although this analogy is not entirely correct. The borer uses a "wedging" action to replace the very large axial force requirement of conventional machines by an array of equally large (or even larger) radial forces. Individual roller cutter forces, normal to the conical rock surface, are nearly radial. Considered together the radial components of these forces are self-canceling and, in this sense, "free". The relatively small axial component can be canceled if individual cutters are skewed to develop a frictional side force. This action, which has been successfully demonstrated, is essentially independent of rock properties so that each individual cutter provides its own axial force.

Overall borer behavior is the same for all rock types (all that can be bored by roller cutters) and the device would tolerate extreme variations in rock properties over the length of the borer head.

The most notable, or perhaps even spectacular, characteristic of the conical borer is of course self-advancement. However, further advantages are gained largely because the conical shape provides more room adjacent to the hole bottom. This means more room for cutter teeth, more room for bearings, and more room for cuttings removal. More cutter teeth permit higher power inputs and greater penetration rates (still without thrust) or, conversely, if cutter teeth are not increased, conventional penetration rates are attained at decreased cutter loading. This plus more bearing room means greater bearing life. In fact, the overall machine configuration permits the designer to insert virtually as much bearing capacity as he desires by simply using more rollers of shorter individual length. More room permits freedom to direct cuttings flow as desired or, even without this benefit, decreased cuttings density on bottom. For example, the proposed machine at a given penetration rate and without improved cuttings flow would have only one-quarter the hole-bottom cuttings density of a conventional bit.

In fundamental terms the proposed conical borer promises to eliminate the major obstacles to present mechanical borer advance rate: that is, limited power input as limited in turn by excessive thrust requirement and/or excessive cutter bearing loads; and a limitation imposed by poor cuttings removal.

In specific terms, the following advantages can be cited for the conical borer in comparison to conventional borers:

(1) No external thrust required

(2) No external torque required

(3) High power density, hence high penetration rate. Longer Bearing Life

(4) Heavy loads confined to one compact, rugged element

(5) Good cuttings removal. Low total weight

(6) Overall system simplicity

(7) Negligible vertical load on rock

These features add up to a simply, high-speed, lightweight, highly portable, economical boring system.