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DESIGN, FABRICATE AND TEST A CONICAL BORER

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13. ABSTRACT

This report summarizes Foster-Miller Associates' effort completed during the first year of a proposed three-year effort to design, fabricate, and test a conical self-advancing and self-rotating boring machine. The conical borer will use a proven and economical 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 36 inch bore. The concept offers a substantial improvement over the present state-of-the-art, particularly in hard-rock boring applications and can be rapidly put to practical use using proven components.

The first year program was successfully concluded when the nose section of the borer - designed, fabricated and assembled during the program - was demonstrated at the test facilities of the Hughes Tool Company. This nose section reamed an 8-3/4 inch pilot hole in a block of granite to a final dimension of 14-1/8 inches with less than one-fifth the thrust required by a conventional 13-3/4 inch bit. It is expected that this performance, as well as the torque requirements, will be improved with further testing.

Preliminary design of the overall boring unit was also completed and presented. With its own hydraulic motor drive it will not require a rigid link to the surface for torque or thrust. It contacts the wall only through its

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13. Abstract (Cont.)

roller cutters, requiring neither torque nor thrust from wall jacks or other external mechanisms.

The prototype is expected to consume about 115 horsepower, rotate at 40 rpm, and advance in granite at about 9 feet per hour. The unit will weigh about 18,000 pounds in contrast to about 135,000 pounds of thrust necessary to drive a conventional bit at this rate of advance. In principle much higher penetration rates are feasible, and the prototype will be designed and powered to explore this area.

Conclusions and recommendations for the remaining phases of the proposed program are presented.

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Mine Rescue Equipment	4					
Roller Cutters	10					
Rock Mechanics	10					
Rapid Excavation	4					
Mechanical Rock Fragmentation	1					
Rock Fragmentation	2					
Reamer	4					

DESIGN, FABRICATE AND TEST
A CONICAL BORER

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1. Summary

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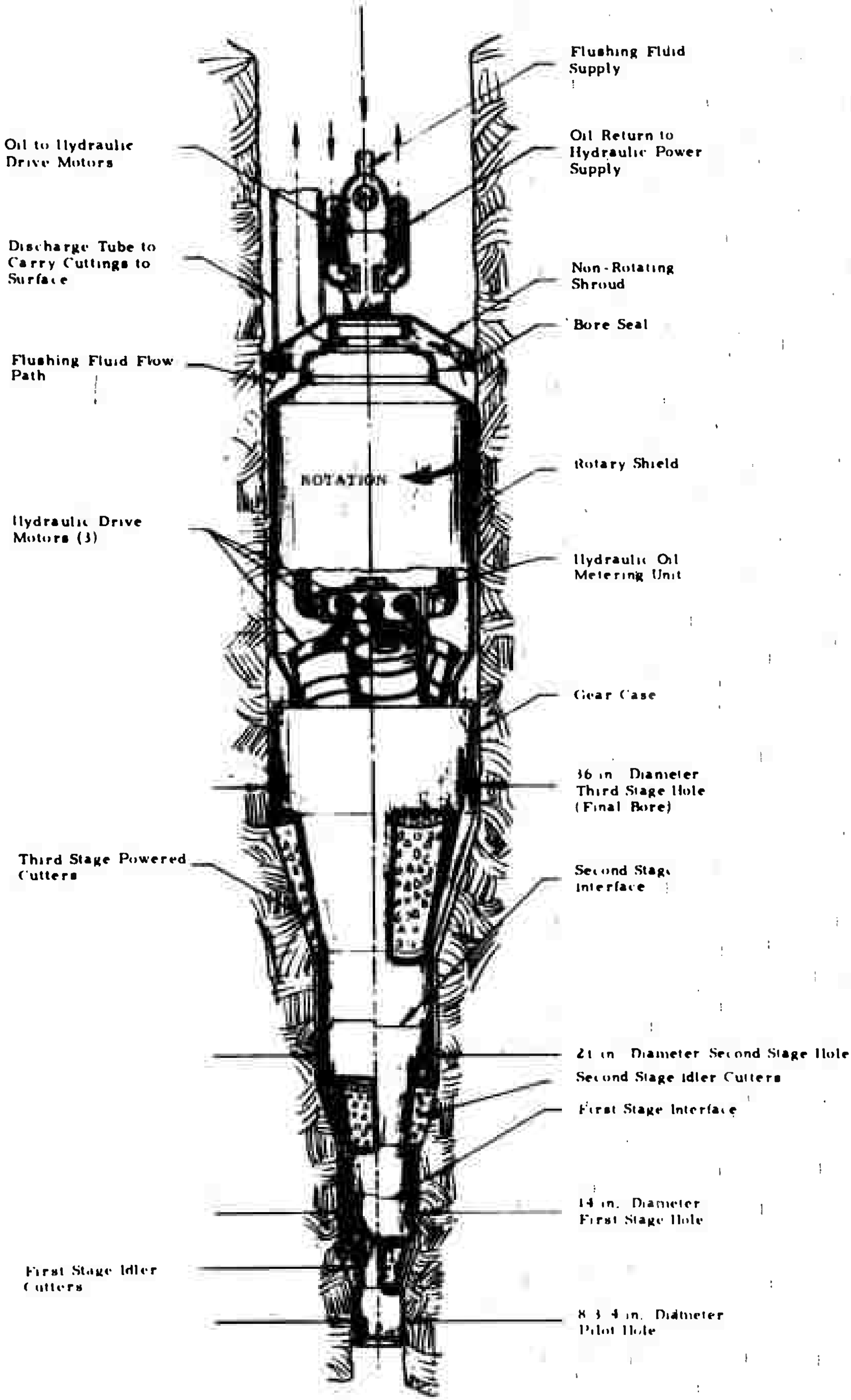
The first year program was successfully concluded when the nose section of the borer — designed, fabricated and assembled during the program — was demonstrated at the test facilities of the Hughes Tool Company. This nose section (Figure 1) reamed an 8-3/4 inch pilot hole in a block of granite to a final dimension of 14-1/8 inches with less than one-fifth the thrust required by a conventional 13-3/4 inch bit. It is even expected that this performance, as well as the torque requirements, can be improved with further testing.

Preliminary design of the overall boring unit was also completed (Figure 2). With its own hydraulic motor drive it will not require a rigid link to the surface for torque or thrust. It contacts the wall only through its roller cutters, requiring neither torque nor thrust from wall jacks or other external mechanisms.

The prototype is expected to consume about 115 horsepower, rotate at 40 rpm, and advance in granite at about 9 feet per hour.

Nose Section of the Conical Borer

Figure 1



Preliminary Overall Design

Figure 2

The unit will weigh about 18,000 pounds in contrast to about 135,000 pounds of thrust necessary to drive a conventional bit at this rate of advance. In principle much higher penetration rates are feasible, and the prototype will be designed and powered to explore this area.

The following sections describe in detail the first year's activities, including the background, the conical borer principle, the prototype design that was developed, and the test results. In addition, conclusions and recommendations for the remaining phases of the proposed program are presented.

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 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 proven fragmentation principle involving roller cutters. The overall machine concept is based on design studies and hardware developed by Dr. Carl R. Peterson while at Ingersoll Rand Research, Inc. and Foster-Miller Associates, Inc. The concept promises substantial and prompt improvements over present performance especially in hard rock where today's boring devices are particularly limited.

Conventional mechanical tunnel or shaft boring devices have been justifiably criticized for poor reliability, massiveness, and difficulty of maintenance. But is the mechanical disintegration process at fault, or isn't it simply the fault of the overall design of the conventional large boring machine? Mechanical disintegration can be quite reliable, as for example in everyday oil-well drilling practice. Indeed, deep well drilling would be virtually impossible without highly reliable roller bits.

Without a lengthy design review, it can be fairly said that most, but of course not all, of the problems with today's mechanical borers can be traced to the large thrust required by these devices. Obviously the massiveness of the machines is due largely to this thrust requirement. A good many mechanical problems are developed in the process of generating thrust at one end of the machine and applying it to moving cutters at the other end. Present machine manufacturers are inclined to discount this view, pointing out that their conventional wall jack and thrust components are just simple, reliable, hydraulic cylinders. This is true. But the overall machine is not simple, has not proven reliable, and <u>is</u> massive and costly. Furthermore, at least in hard rock, machine performance is limited by the inability to produce sufficient thrust. This has been pointed out as a shortcoming quite apart from whatever reliability problems may arise.*

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 described in the following sections, the proposed boring device generates all necessary excavation forces (including torque) <u>directly</u> from the rock surface being excavated,

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*Williamson, T. N., et al., "Scientific and Technical Applications Forecast-Excavation", 1964.

while contacting the rock only through the roller cutter elements. The machine is compact, simple, and rugged in comparison to a conventional wall jack machine. Furthermore, the conical cutter head accommodates more cutters than does a conventional flat cutter head of the same diameter. Hence, the proposed machine can be designed for higher penetration rates. In fact, if it proves economical, and if other system operations such as mucking are not limiting, the proposed machine could even be designed to advance at four or five times the present small hole roller bit penetration rate (which rate is already many times today's large hole performance).

This report describes the design, fabrication and testing of the nose section of a conical borer. In the proposed follow-on program, the nose section will be mated with a powered main frame and idler section resulting in a completely self-contained conical borer capable of enlarging an existing pilot hole of 8-3/4 inches to 36 inches.

The following section describes in detail the principal of the conical borer.

3. The Conical Borer Principal

3.1 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.

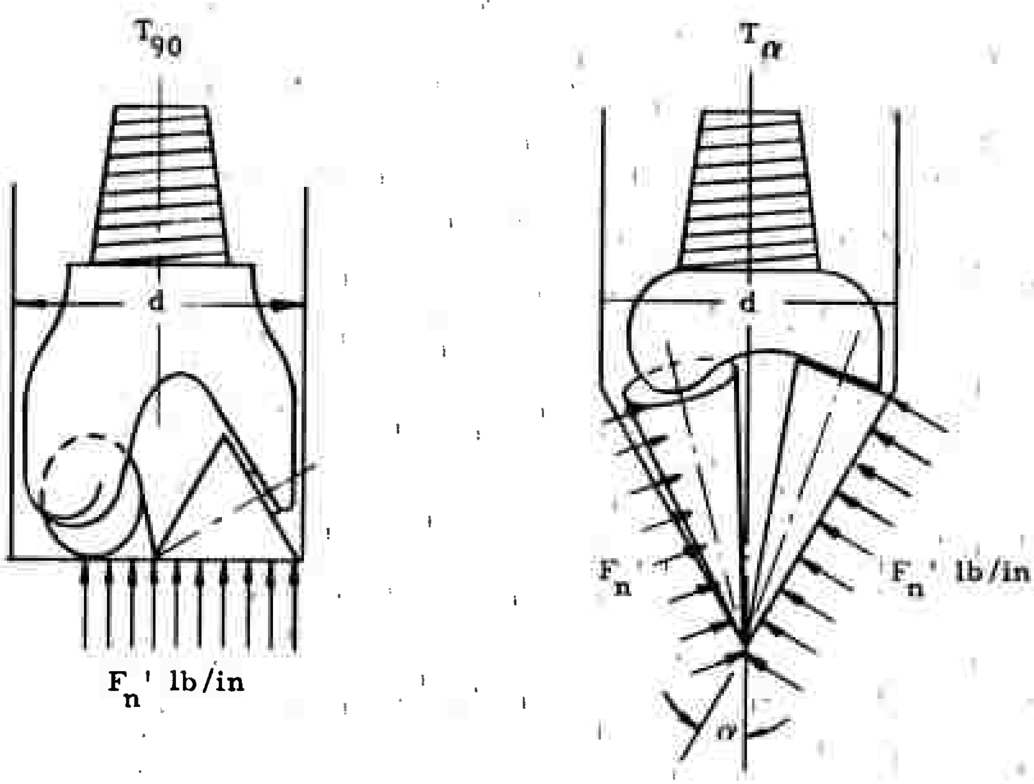
The conventional bit, shown in Figure 3(a), experiences a distributed line load, F_n' pounds per inch, along the rolling contact of each cone. Load distribution is assumed constant along the line length for convenience in illustration only. Since there are three rollers, the total roller length is $\frac{3}{2}$d,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

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

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

Consider next the three-roller conical bit of Figure 3(b), 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

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



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

Flat and Conical Bit Comparison

Figure 3

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 booth 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:

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

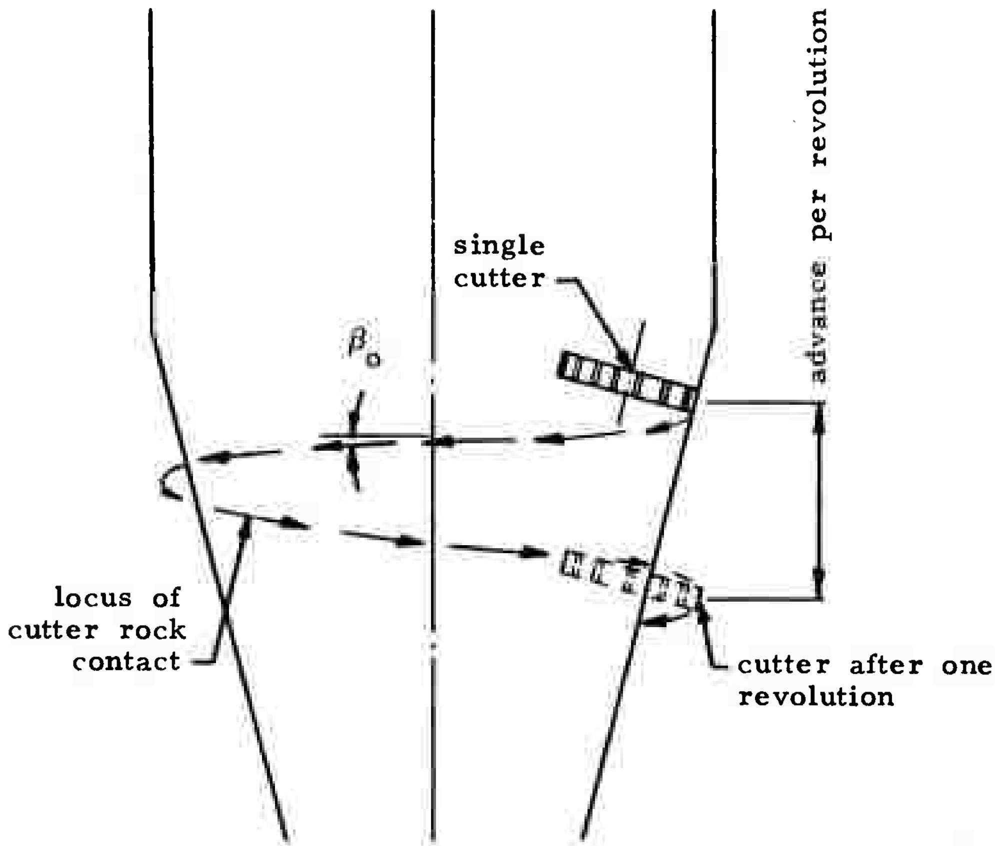
$$T_\alpha = \sin \alpha \; T_{90} \quad \text{at constant } R \tag{4}$$

In passing we note that Figure 3(b) 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 3.3), 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.

3.2 The Helical Cutter Path

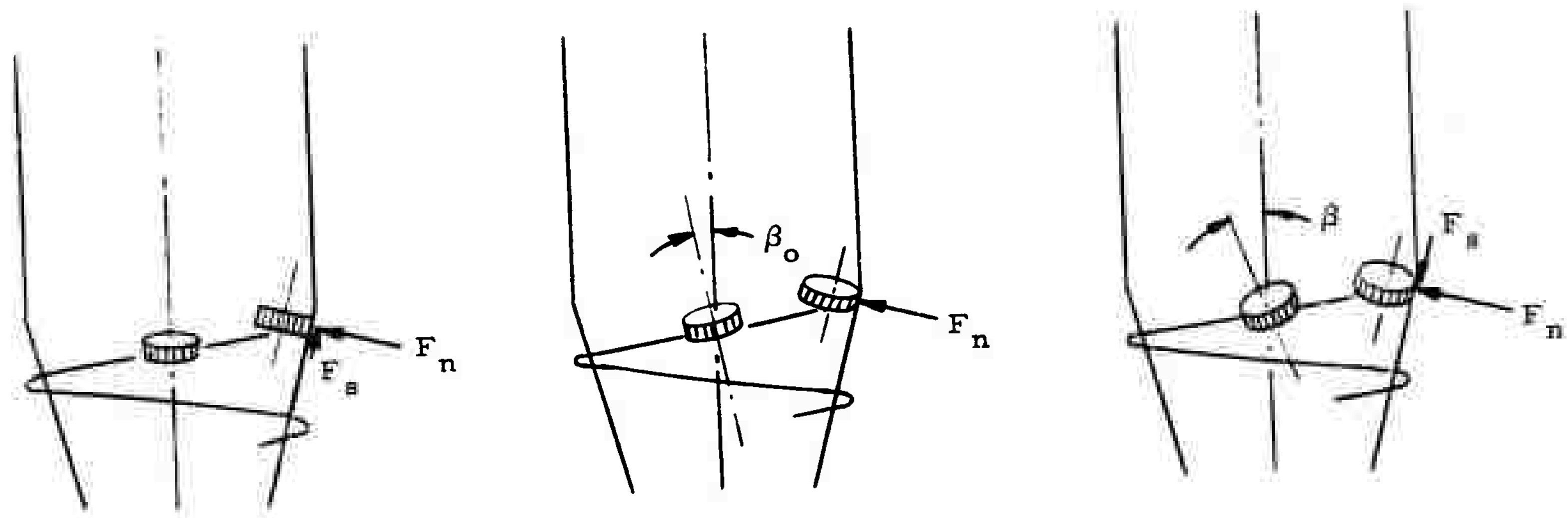
The simple description of Section 3.1 is based upon an examination of a normal force at the cutter-rock interface. There is, in addition, a rolling force which requires the application of torque to rotate the bit and through which energy is transmitted to fragment the rock. In addition to these well-known 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 4. 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 5(a). 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 4, shown again in Figure 5(b), would prevail. If, now, the cutter can be skewed beyond the advance helix angle β_o, as shown by Figure 5(c), the cutter teeth will experience a rearward skidding motion as they contact the rock, as described in



Helical Cutter Element Advance

Figure 4



(a) No Skew
$\beta = 0$

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

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

Skewed Cutter Geometries and Forces

Figure 5

Section 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.

3.3 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 6. 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,

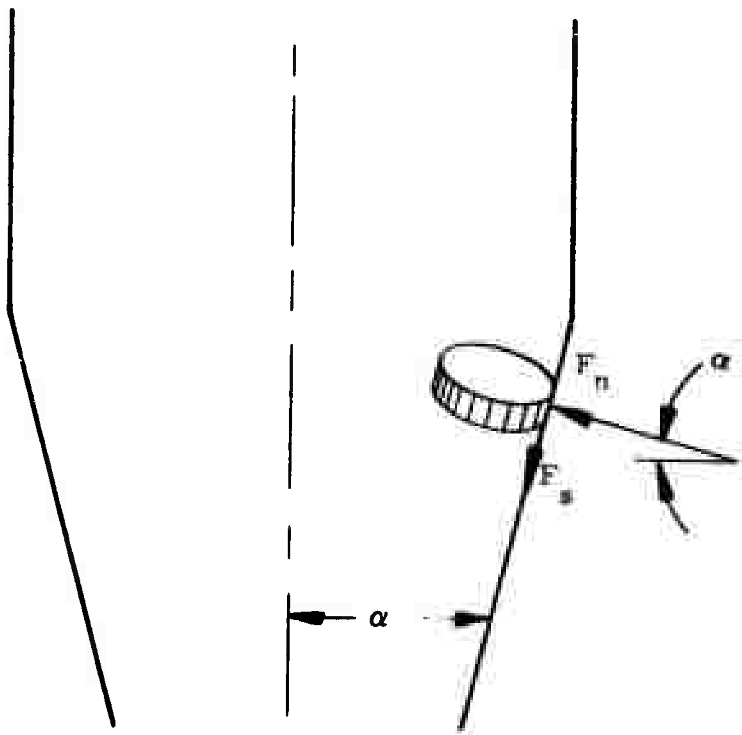
$$F_s \cos \alpha \geqslant F_n \sin \alpha \tag{5}$$

or

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

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 6

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 7. 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 3.4).

The kinematics of individual tooth penetration and the origin of the cutter side force are shown in Figure 8 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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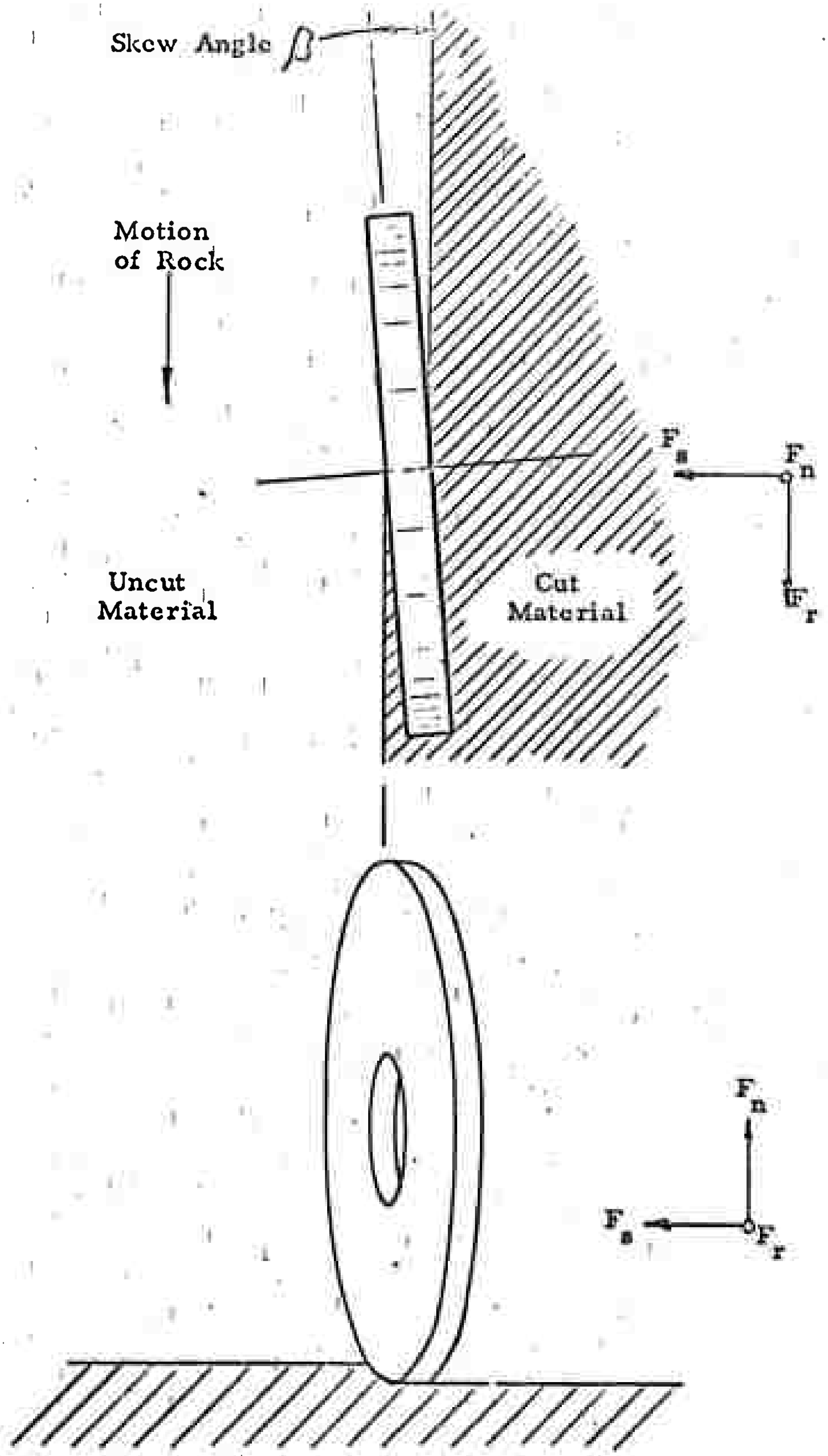
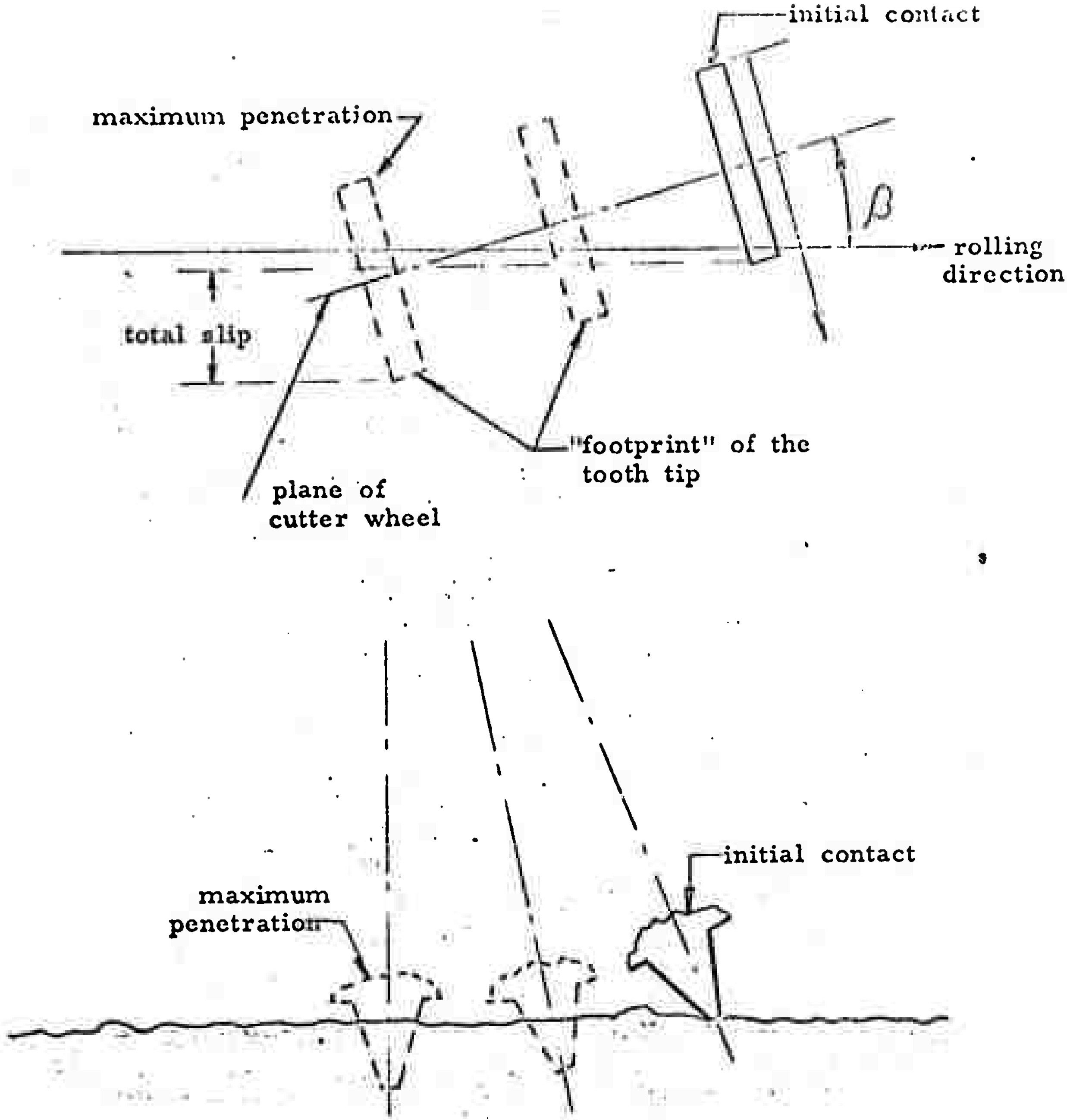
* Peterson, C. R., "Rolling Cutter Forces", Paper No. SPE 2393, Society of Petroleum Engineering of AIME, also published in Society of Petroleum Engineering Journal, March, 1970.



Plan View of Cutter on Rock
(Forces Defined are Forces on Cutter)
Figure 7



Skewed Cutter Penetration Geometry

Figure 8

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 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 8, 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.

Simple analytical work, similar to that of the cited reference, can be applied to the variation of F_s/F_n with cutter diameter and rock surface radius. The effect of borer weight can be considered in a simple modification of Equation (6). Still, a phased program is proposed in which these uncertainties can be eliminated before construction of the more costly components of the prototype is begun.

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.

3.4 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 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 dirst, 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.

3.5 Self-Rotating Drive

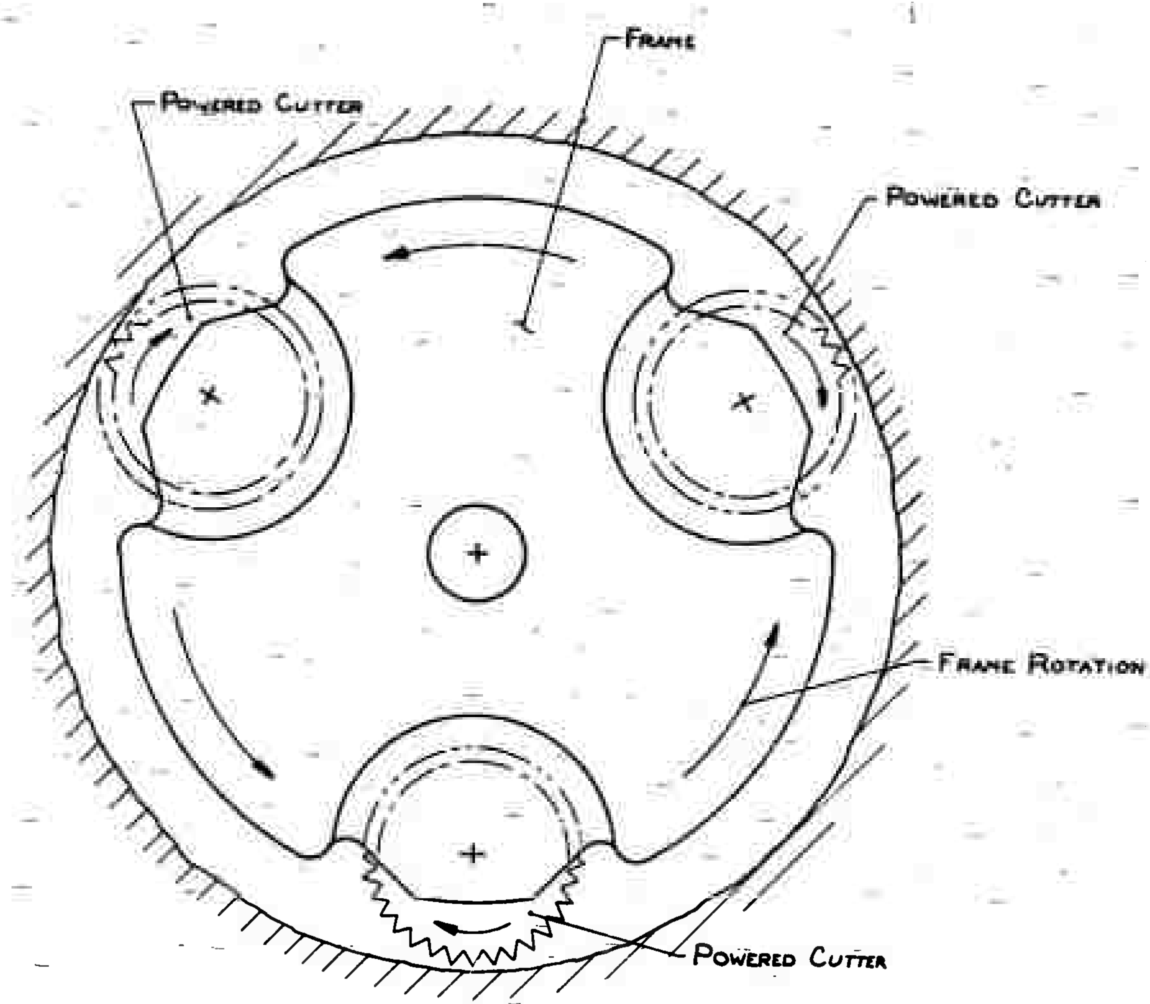
The conical borer was first conceived to reduce or eliminate the need for external thrust. As an added benefit, the much greater total roller length of the conical borer permits the simultaneous attack of a much greater quantity of rock and, hence, a proportionately greater advance rate. Since the basic 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 9. Though not as obvious, the action is identical if the power happens to come from a single frame-mounted motor, geared to the rollers, as on the proposed conical borer.

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, as illustrated in Figure 2 of this report,



Torqueless Rotary Drive from Frame-Mounted Motors

Figure 9

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.

3.6 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 <u>without</u> 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 <u>external</u> thrust required
(2) No <u>external</u> 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 simple, high-speed, lightweight, highly portable, economical boring system.

4. Design of the Prototype Borer System

In this section the sizing and preliminary design of the overall unit is discussed and performance estimates are made.

4.1 Overall Design Considerations

The subject contract calls for the design of a borer "in the range of from 2 to 5 feet", in diameter, to enlarge a pilot hole "in the range of from 6 to 12 inches", in diameter. In our proposal, a diameter of about three feet was suggested, with the further suggestion that a four-foot diameter machine should also be considered since the borer weight would then be sufficient to simultaneously drive a pilot bit in possible future programs. This ultimate capability is an attractive option but, in addition to larger diameter, it requires a smaller hole-bottom angle in order that machine weight need not be used to advance the conical borer proper. After preliminary design considerations, it was concluded that for fabrication economy the prototype would be three feet in diameter. Furthermore, the hole-bottom half-angle was selected as 18^{o} to keep the machine as short as possible while still permitting self-advancing performance in vertical boring where the weight of the unit assists advancement. Pilot hole diameter was selected as 8-3/4 inches, a standard bit size. Eventually smaller holes can be used, but the difficulties associated with smaller nose cutter bearings are better left to a later development.

Results of our preliminary design are shown in Figure 2 (Page 3). Details of this configuration and its estimated performance are discussed in the following sections.

4.2 Structure, Power Train, and Cutters

The boring machine consists of five major systems (Figure 2):

(1) Cutters;
(2) Supporting frame;
(3) Drive system;
(4) Flushing systems; and
(5) Support (surface) equipment.

The cutters, frame and drive system considered together constitute the basic mechanical elements of the borer. At present, these elements may be further subdivided to distinguish:

(1) A single stage of powered cutters at the rear of the machine;
(2) Two stages of idler cutters;
(3) A one-piece frame;
(4) Hydraulic drive motors;
(5) A transmission housing; and
(6) A hydraulic/pneumatic rotary union.

Hydraulic motors drive the three powered cutters of the third stage through a set of helical gears or Schmidt Couplings. Since the powered cutters are engaged or locked to the wall of the hole, the main frame will rotate in the manner of the rotating cage in a planetary gear system (Figure 9). The reaction torque generated by the powered rollers is transmitted to the frame and used to overcome the "rolling resistance" of a selected number of idler stages and idler cutters (Figure 2). Or, rephrasing it, the tangential forces on the frame generated by the powered cutters must provide the total torque required by all of the idlers. The powered cutters, therefore, must be large enough to provide this force without slippage, even in cases of non-uniformities which may require the idler cutters to engage hard material while the top cutters are in soft material.

It is conceivable under certain operating conditions that a local jam-up might stall the drive motors. If this condition should occur, then it is only necessary to reverse the motors and to back off the conical borer. This technique was followed on several occasions during the development of a six-inch demonstration model at Ingersoll Rand.

The hydraulic/pneumatic rotary coupling is a multi-ported device (Figure 2) through which high pressure hydraulic fluid is channeled to the drive motor, and flushing air is channeled to the nozzles adjacent to the cutters. It is combined with a pressurized housing which keeps all contaminants out of the motor transmission area. The coupling also serves as a swivel from which the boring machine is suspended while being hoisted from the bore. It is designed to contain 3,000 psi hydraulic pressure and moderate air pressure and to support an appreciable (500,000 lbs) static load.

4.3 Flushing Considerations

The flushing system can be treated subsequent to much of the mechanical design effort, but this does not mean that proper flushing is a matter of secondary concern. Indeed, in large hole boring, adequate flushing has been a major problem of conventional systems,* and it will be an even greater problem if we wish to take advantage of the conical borer's higher rate of penetration.

Reverse circulation of flushing fluid is commonly used in large hole boring. In conventional reverse circulation pneumatic flushing, the drilled hole is sealed at the surface. Pressurized air is forced down the annulus between the drill stem and the hole wall, across the face of the cutter, and out through the center of the drill stem. Loss of circulation due to faults or highly permeable soil is frequently encountered and is a major drawback of this technique.

The conical bit can utilize conventional reverse flushing techniques with perhaps some modifications to produce the desired flow pattern on the hole bottom, and it would attain conventional flushing performance in so doing. Fortunately, however, since the conical borer does not require a rotating drill stem, it can easily take advantage of a potentially much more effective double-line system.

This system is shown in Figure 2. Basically, it consists of the following elements:

(1) A relatively high-pressure line carrying air (or other flushing medium) to the borer;

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*Woodruff, C. R., "Large Diameter Holes -- Past, Present, Future", Paper No. 801-38b, American Petroleum Institute, 1962.

Dellinger, T. B. and Presley, C. K., "Large Diameter Shaft Drilling with Reverse Circulation Air, Casing and Cementing for AEC Emplacement Holes", Paper No. 851-38-H, Division of Production, American Petroleum Institute, Dallas, Texas, 1964.

(2) A duct and nozzle system built into the borer to discharge air upwardly adjacent to the bore wall in a pattern as desired around the cutters;

(3) A shield, rotating with the borer, to carry the cuttings-laden air over the power section;

(4) A non-rotating hood, sealed against the bore wall, to direct the air to the discharge line; and

(5) A discharge line carrying cuttings to the surface.

Both the inflow and discharge lines can be flexible and, since they do not rotate, they need not be coaxial.

This system offers several substantial advantages. By directing the flushing fluid as desired over the hole-bottom, flushing can be much more effective than the random "vacuum cleaner" effect of a conventional system.* More importantly, flushing fluid is exposed to the bore wall only in the cuttings area, and only <u>after</u> it has picked up cuttings. This substantially reduces lost circulation, of course, by the simple reduction of wall area. Furthermore, if circulation is lost, the fluid will carry cuttings with it (not a bad idea in itself if the cuttings can be carried away down hole) and these cuttings will eventually plug the faults causing lost circulation.

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*Note that a conventional reverse flush system has little or no directed flow, and it experiences minimum velocity at maximum diameter where most cuttings originate.

This approach has a design difficulty, but this does not seem insurmountable. If the borer region is at elevated pressure, necessitated by a pressure drop in the cuttings discharge line, this pressure acts upwardly against the non-rotating hood, tending to lift the borer from the hole bottom. This can be overcome by installing ejectors in the discharge line (fed directly from the adjacent inflow line) to permit discharge from a low-pressure borer region. Alternatively, the hole can be sealed at the surface in a conventional manner with the non-rotating hood still in place to direct cuttings to the discharge line.* The hood need not seal against the wall since there will be no fluid flow past the hood anyway. A third alternative is shown in Figure 10 in which the non-rotating hood is loaded against the wall to carry the pressure force. The hood would reciprocate through a fixed stroke relative to the cutter, with a temporary shut-off of flushing fluid during the (short) downstroke of the hood. In addition to eliminating the undesirable pressure force on the borer, this concept avoids a rubbing seal against the bore wall. However, this concept is beyond the scope of the presently proposed program. It is expected that, at least for some time, the prototype borer will be used in shallow holes where this pressure force will not be important.

The conical borer itself offers a favorable geometry for flushing. The additional space available allows room for the system of directed jet nozzles. Fluid flow is in a more nearly straight line pattern, without the abrupt turns inherent in a flat hole

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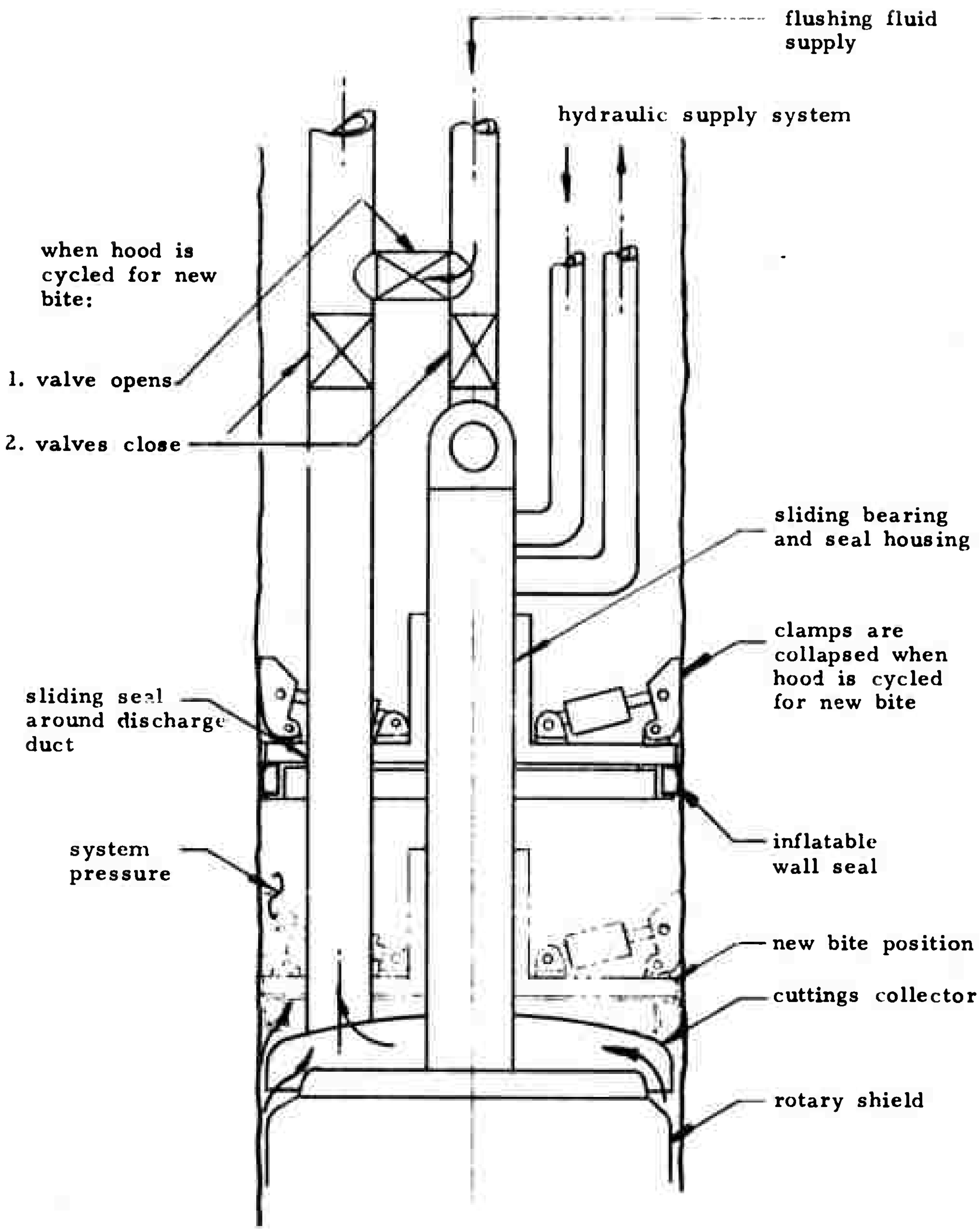
*This alternative would suffer conventional lost circulation problems.



Reciprocating Hood Concept

Figure 10

bottom flow. Such turns tend to separate heavy cuttings from the flushing fluid. The large hole bottom area means that, for any given penetration rate, the cuttings density on the hole bottom will be decreased. The proposed machine, for example, will experience only one-fourth the cuttings density of a conventional machine at the same penetration rate, even without improved flow patterns.

While the mucking air flow requirements will of course depend upon many factors relative to the design of the boring machine, the actual requirements will also be dependent upon local conditions relative to the particular drilling situation. During the final design of the conical boring machine, these limiting conditions will have to be considered in detail. As a starting point for designing of the flow visualization mock-up and the mucking investigation program, an understanding of the relative magnitudes of the potential air flow requirements is essential. These approximate minimum requirements can be determined based upon data and correlations readily available in the literature.*

A correlation proposed by Dallavalle as reported by Leva estimates the minimum carrying velocity in vertical pneumatic transport lines to be given by:

$$u_c = 910 \left(\frac{\rho_s}{\rho_s + 62.3} \right) D_p^{0.6} \qquad (7)$$

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*Leva, Max, "Fluidization", McGraw-Hill Book Company, Inc., New York, 1959, Chapter 6.

where u_c is the minimum carrying velocity (ft/sec),

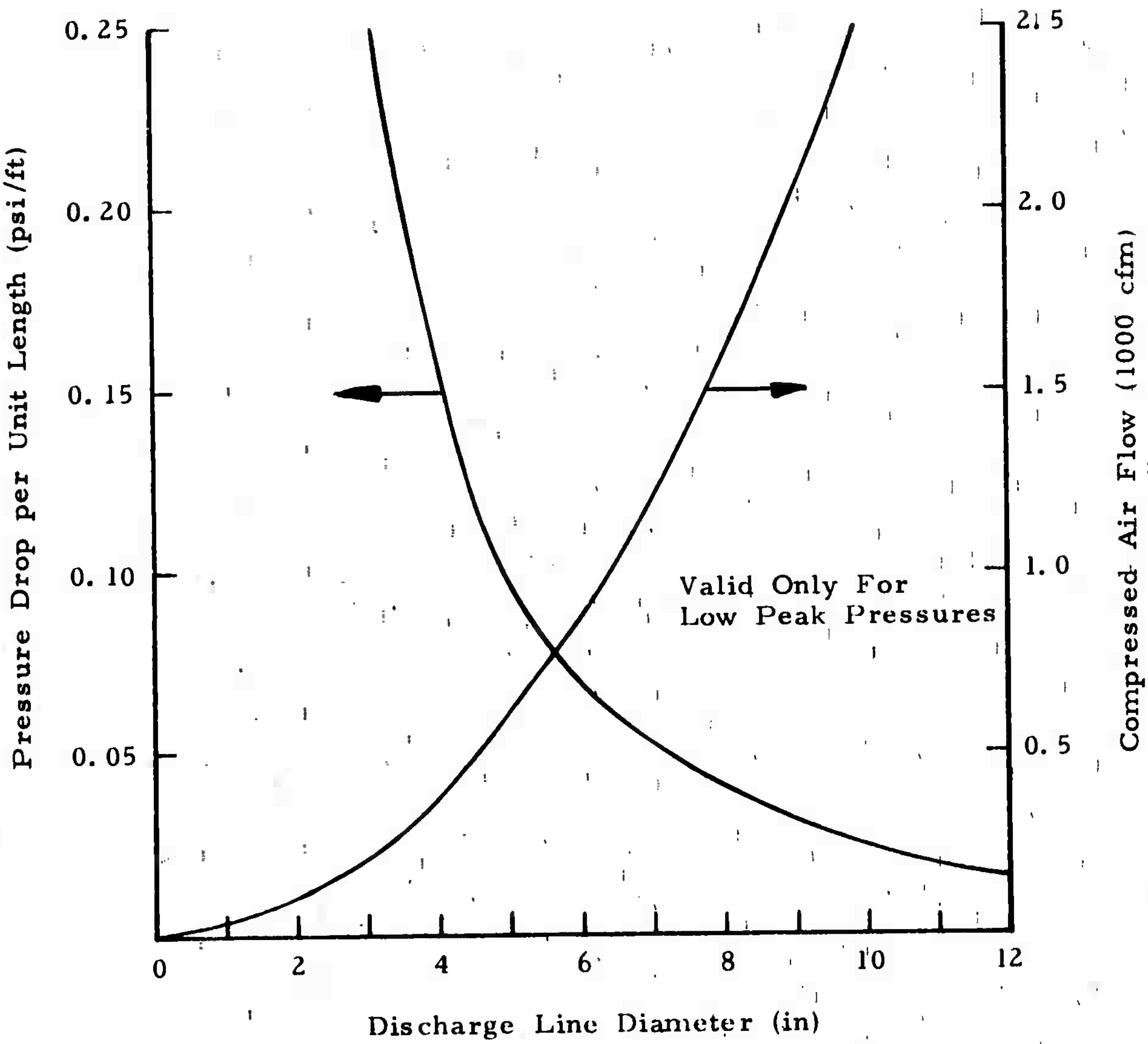
ρ_s is the solid density (lb/ft^3), and

D_p is the particle diameter (ft).

For drilling in granite (ρ_s = 165 lb/ft^3), the required velocity to transport 1/4-inch diameter chips is, by this correlation, 64.7 ft/sec. The correlation was developed from data taken at light solids loading conditions (no correlation exists for appreciable solids loads), so this must be considered no better than a ballpark approximation.

Leva, reporting on the work of Hinkle, presents a correlation for the total pressure drop due to combined gas and solids flow in vertical, cylindrical-conduit, pneumatic transport systems. Based on Hinkle's correlation, and using 80 ft/sec as the design air flow velocity, the pressure drop and volume air flow required to transport 4.7 ft^3/min of 1/4-inch diameter chips is given in Figure 11. (4.7 ft^3/min solids volume flow is the rate resulting from drilling a 36-inch diameter hole from an 8-inch pilot at 42 ft/hr penetration rate.) The correlations presented in this Figure are valid only for low peak pressures since all compressibility factors have been omitted. However, the relative values are an adequate guide for design of the flow visualization mock-up and test.

For those applications which permit downward flushing, the conical hole bottom may permit simple gravity flow of cuttings, particularly if a large pilot hole (12 inches or greater) is available. However, for small pilot holes, it may be necessary to provide fluid assist to insure high cuttings velocity, hence low density in the tip region. Without such assistance, the tip cutters might become overloaded with cuttings from the larger portions of the borer.



Compressed Air Requirements for Reverse Circulation Mucking
of 36-Inch Diameter Hole, Boring at
42 Feet Per Hour Penetration Rate

Figure 11

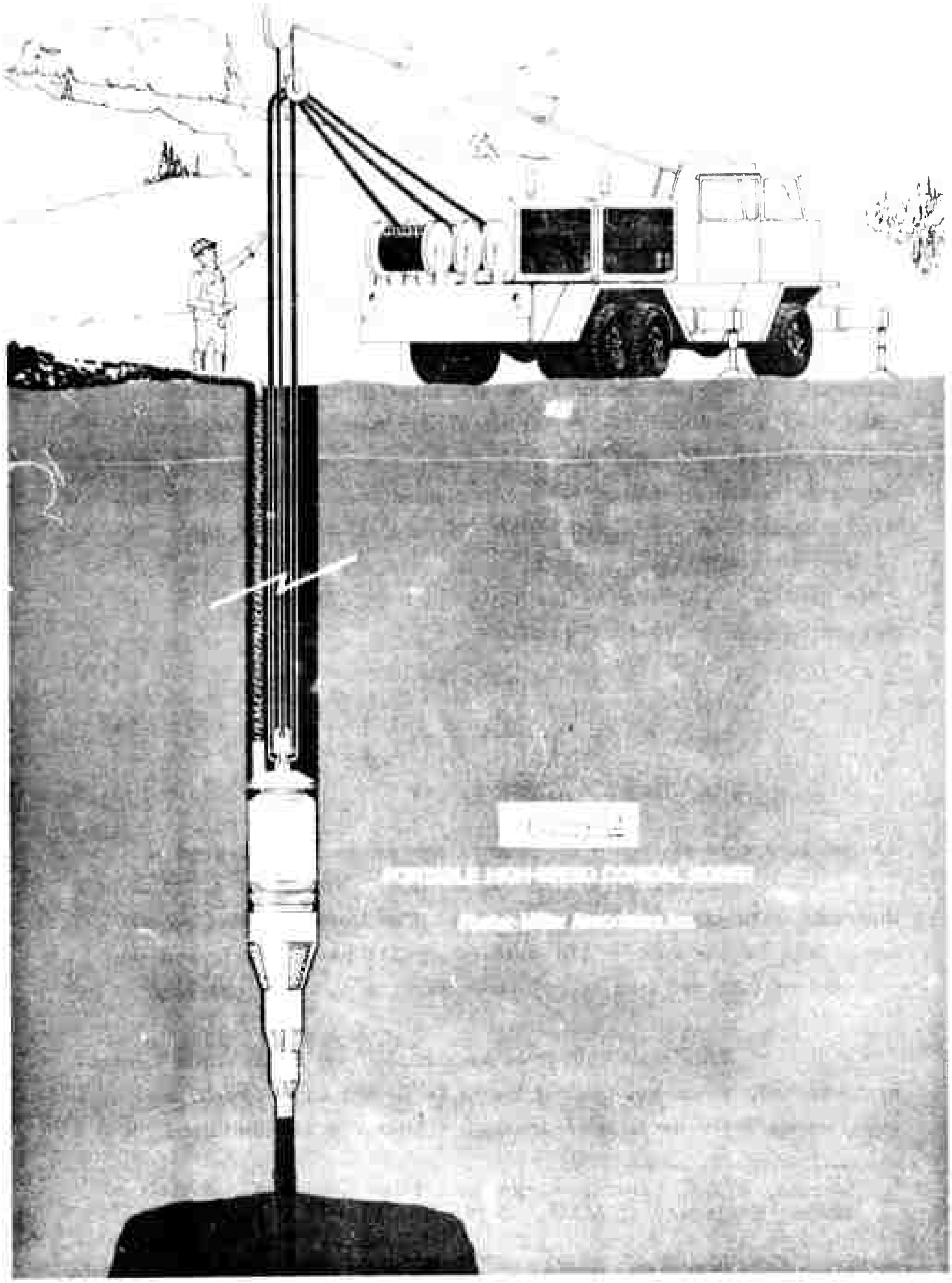
4.4 Surface Equipment

Surface equipment, shown in Figure 12, is also greatly simplified by the proposed borer. Clearly the conical borer will require substantially less hoisting capacity than would conventional equipment. Moreover, the proposed borer requires no thrust or torque from the surface and, hence, no rigid drill pipe. The borer would simply hang from a carrier cable with flexible power, air, and cuttings return lines as needed. The entire suspension system can be reeled in and out of the hole, although it might be more convenient to store the flexible cuttings return line in segments of modest length.

Surface equipment would then consist of a power unit, probably diesel-powered hydraulic with associated controls, oil filters, coolers, and reservoir, a hoist of modest capacity and height, storage reels for hydraulic lines and air line, a system to store and handle the cuttings return line, and a compressor of conventional type. For boring to an open mining area, downward cuttings removal would eliminate the cuttings return line and the seals necessary for reverse flushing, thereby substantially reducing (or eliminating) the required compressed air capacity. For portable or any other applications, therefore, the proposed conical borer offers both an enormous saving in down-hole weight and a comparable saving in surface equipment size, weight, complexity, and expense.

4.5 Performance Estimates and Comparisons

The power consumption and penetration rates both for the conventional Tricone drill and the conical borer were predicted



on the basis of Morlan's data* for bits of about 10-inch diameter, scaling the power requirements from this to the desired diameter in proportion to the projected hole-bottom area. This in effect assumes the same specific energy for all hole diameters. Assume a bit loading of 5,000 pounds per inch of diameter (a loading reached by some pull-type raise borers, but not often reached by large hole gravity-fed borers). Since the conical borer advantages are most evident in hard rock, let us consider boring in gray granite, a relatively hard material (31,000 psi compressive strength) of northern Texas, and often used by roller bit manufacturers for hard rock performance tests. Morlan indicates that at 60 rpm, a 10-inch roller bit would penetrate this material at 14 feet per hour and require 14 horsepower. At the same load per inch of diameter and at the same speed, a 36-inch bit would advance at the same rate and require power in the ratio $(36/10)^2$. Accounting for the pilot hole difference then, the required power at 60 rpm would be:

$$hp_{60} = 14 \left(\frac{36^2 - 8.75^2}{10^2} \right) = 170 \qquad (8)$$

An industry rule of thumb suggests a maximum rotary speed of 120/d where d is the bore diameter in feet. Thus, for a 3-foot hole this rule suggests a speed of 40 rpm. The corresponding power would then be reduced to 170 x 40/60 or 113 horsepower, and the penetration rate reduced in the same ratio to 9.3 feet per hour.

This speed limit is presumably set by dynamic considerations and, since the conical borer is locked to the rock, it experiences little or no such loading. Thus, in all likelihood, it

* Morlan, E. A., "Boring Large Hole Mine Openings", Society of Mining Engineers of AIME, Paper No. 61 AU 27.

could run at the higher speed to provide the higher penetration rate. Alternatively, the conical bit could be designed (by proper cone and skew angle selections) to provide the equivalent of a higher bit loading to attain the higher penetration rate at 40 rpm. Assuming penetration per revolution proportional to load per inch of diameter (the usual assumption), an equivalent load of 5,000 x 60/40 or 7,500 pounds per inch would be required to penetrate gray granite at 14 feet per hour and 40 rpm. This is easily done with the conical bit, but it can be seen that a substantial increase in bit loading would be required for a conventional bit.

Penetration rate is highly dependent upon rock properties, and compressive strength is not the only important variable. For example, Morlan indicates that at 5,000 pounds per inch and 60 rpm, Virginia dolomite (31,000 psi compressive strength) is penetrated at 42 feet per hour as compared to 14 feet per hour in gray granite of the same strength. Presumably the power requirement would be about the same in both cases, but the data are insufficient to determine this.

In summary, if we consider two typical hard rocks, the relationships between power, speed, bit load, and penetration rate are estimated to be as shown in Table I.

This Table dramatizes several features of the conical bit. First, a bit weighing perhaps 15-20,000 pounds is seen to replace roughly ten times that in dead weight on a conventional bit. Second, it suggests that the conical bit can operate at higher speeds, although this is not proven as yet. And third, it indicates that if sufficient power is available, the conical bit is capable of very high penetration rates in hard rock. Clearly, detailed testing of the nose

Table I

36-Inch Bit Power and Penetration Rate Estimates*

Bit	rpm	Load, lbs	Power, hp	Penetration Rate, ft/hr	
		Load	Power	Gray Granite	Virginia Dolomite
Conventional	40	135,000	113	9.3	28
		202,000	170	14	42
	60	(not advisable)			
Conical	40	0	113	9.3	28
		0	170	14	42
	60	0	170	14	42
		0	255	21	63

* Includes influence of 8 3/4-inch pilot.

section of the borer will be necessary to establish more precise power requirements (See Section 6), but this estimate is adequate to establish a tentative design around existing motors. For the present, a Vickers model MHT 500, yielding about 200 horsepower at a convenient speed, has been assumed in preliminary design and pricing.

5. Fabrication and Testing of the Nose Section

The first year's effort called for the complete design, fabrication, and testing of the nose section of the borer. Available data appeared sufficient to determine proper self advancing design parameters, but they are not sufficient to predict cutter power or torque requirements in the selected rock. Therefore the nose section was tested to establish overall power requirements.

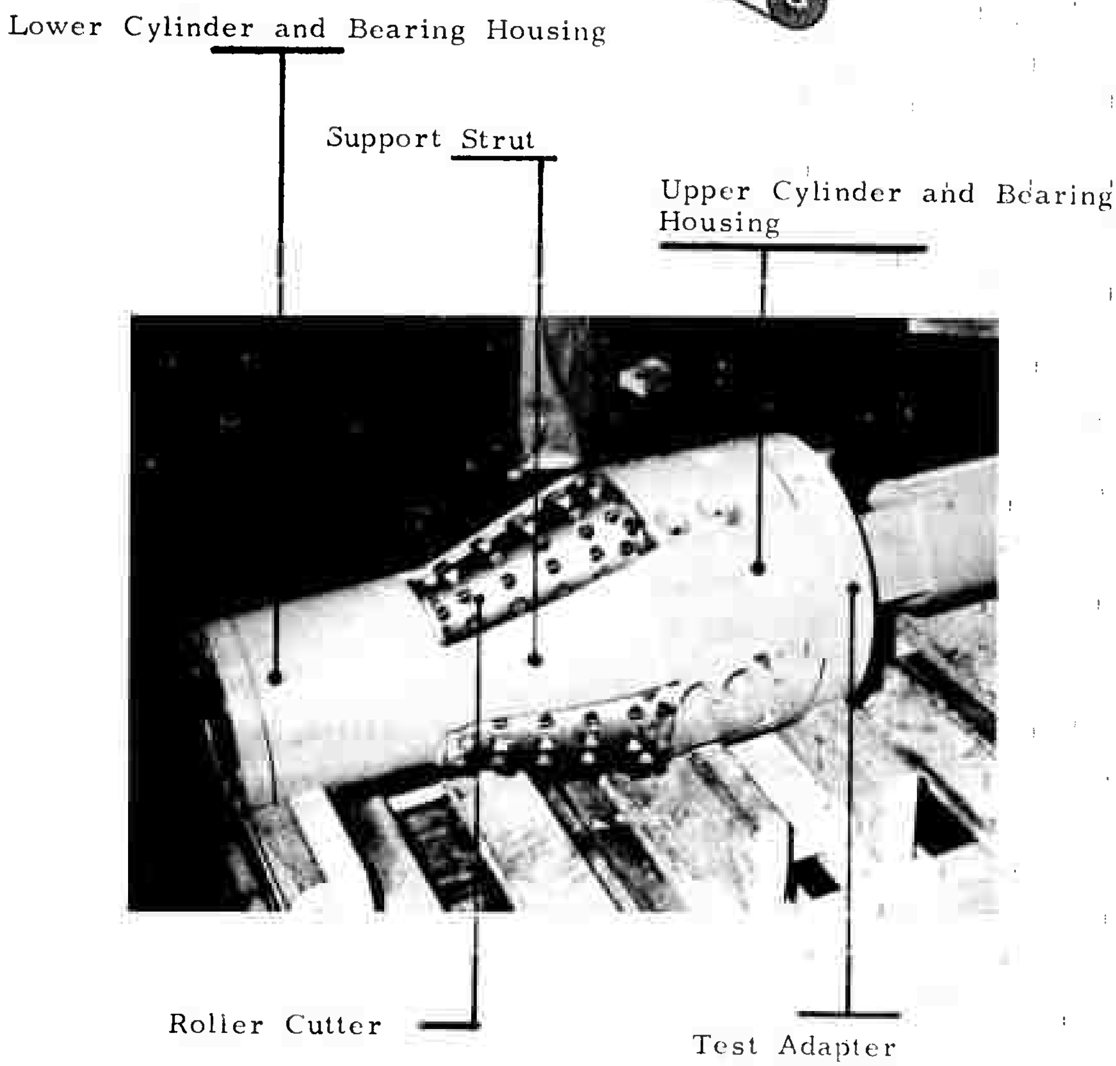
In the following subsections the design and fabrication details of the nose section, the test facilities and procedure, and the test results are presented and discussed.

5.1 Design and Fabrication Details of the Nose Section

Figure 13 is a photograph of the assembled nose test section. It is designed to ream an 8 3/4 inch hole to 14 inches. The hole-bottom half-angle is 18° and the cutter skew angle is 4°.

The nose frame is a weldment of five pieces: the lower cylinder or bearing housing; the three struts; and the upper cylinder or bearing housing. The upper portion of the upper cylinder shown in the photograph is an adapter (not a separate piece in the mock-up) that will connect to the test facility for testing of the nose section. This adapter is bolted to the nose frame through a face spline. The same bolting arrangement and face spline will serve to connect the nose section to the complete borer main frame.

In order to provide maximum bearing capacity in the



Photograph of Nose Section

Figure 13

cramped lower cylinder, a unique bearing assembly method has been designed. Lower end bearings, both radial and thrust, are contained in three separate cartridges, with the cartridges serving as outer races for the three radial bearings. These bearing cartridge assemblies are inserted axially into the frame from the lower end, thus avoiding split bearing caps in the lower cylinder.

Upper bearings are also contained in fully enclosed (i.e., not split) blocks which fit within essentially rectangular seats in the upper cylinder. The four retaining bolts which can be seen in Figure 13 are for retention only; they carry no loads imposed by boring.

Assembly of cutters is a three-step procedure:

(a) Each upper bearing assembly and block is slipped over the stub shaft extending from the upper cutter end;

(b) With lower bearing cartridges not in place each cutter and upper bearing combination is inserted in the frame, with the lower cutter stub shaft fitting easily within the large lower bearing cartridge bore; and

(c) Each lower bearing cartridge is then pushed over the lower stub shaft within its bore in the lower cylinder.

The three lower bearing cartridges are retained axially by a single cover bolted within the end of the lower cylinder.

The three struts connecting upper and lower cylindrical frame sections are roughly triangular in cross section to fully utilize the space available between cutters. A one-inch hole extends through each strut to carry flushing fluid to the lower end of the borer. This fluid will issue upwardly from the sides of the lower cylinder.

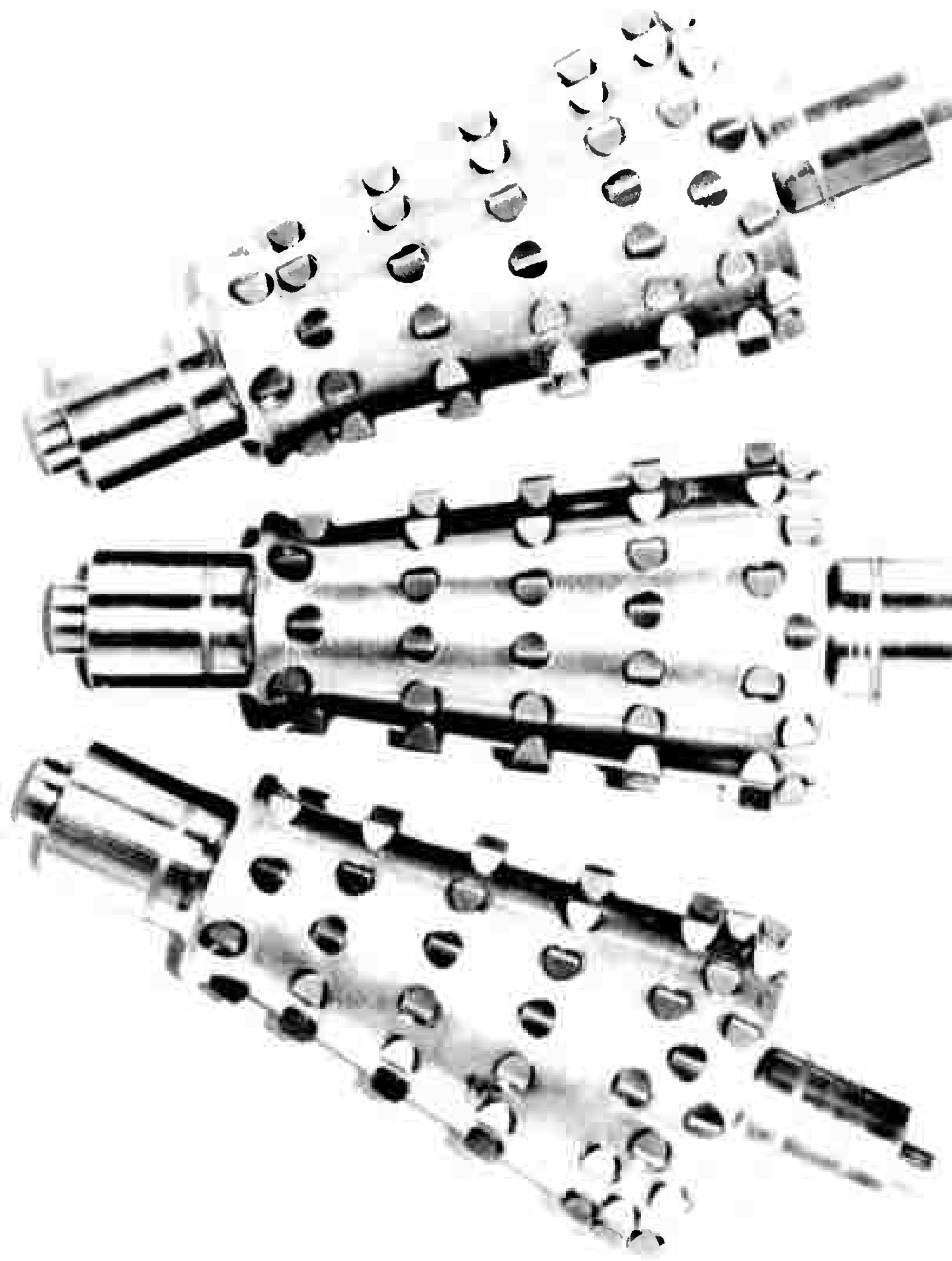
The frame also contains a lubricant reservoir connected to all of the bearings to provide long-term lubricant capacity. This reservoir is provided with an external fitting for convenient replenishment, and it is pressurized by the flushing fluid (acting through a rolling diaphram) to provide positive lubricant pressure. Long-term lubricant capacity was of course not necessary for the nose section tests, but it will be necessary in anticipated testing of the complete borer.

Figure 14 shows the three completed cutters with their carbide cutter teeth. They were designed with excess teeth in anticipation of a downward adjustment. Tooth density has a major effect on required power. It was expected that after detailed testing the tooth density will be decreased.

5.2 Test Facilities, Equipment and Procedure

The tests were run at the laboratory facilities of the Hughes Tool Company in Houston, Texas. The rock drilled was pink granite with approximately 30,000 psi compressive strength.

First, an 8 3/4 inch pilot hole was drilled in the granite with a conventional Hughes W7R rock bit. After the pilot



Completed Roller Cutters

Figure 14

hole was drilled it was reamed with the conical borer to an approximate diameter of 14 1/8". All tests were run at 40 rpm and water was used to flush the cuttings out of the hole. The load was varied from 2,000 pounds to 10,000 pounds and the drill rate and torque were measured.

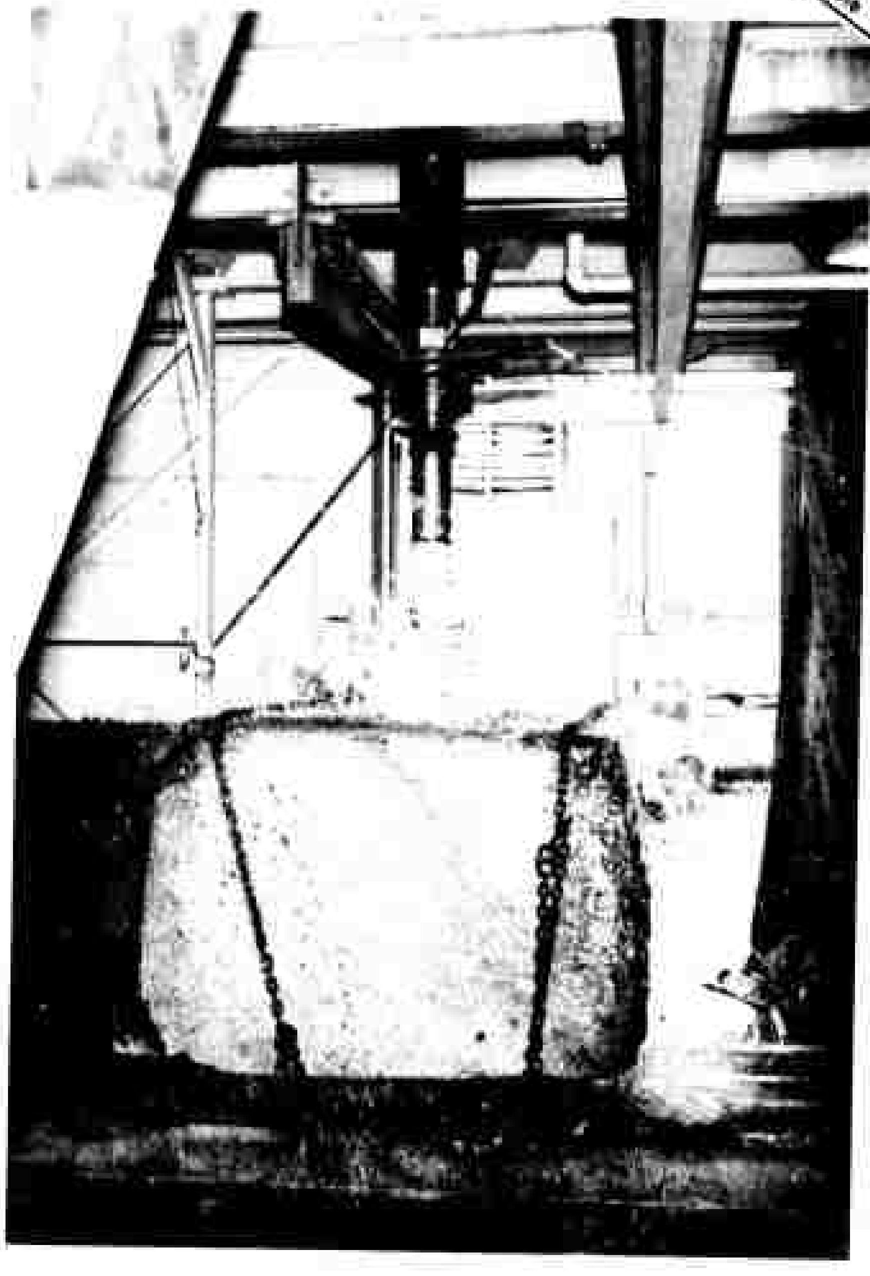
Figure 15 is a photograph of the conical borer, the test equipment, and the granite specimen. Figure 16 shows the actual tests in progress.

For comparison, a 13 3/4 inch W7R rock bit was used to ream the same size hole in the same rock. The drill rates of this bit were held constant at 3, 6, 9 and 12 feet per hour while the torque and load required to maintain these rates were measured.

5.3 Presentation and Discussion of Experimental Results

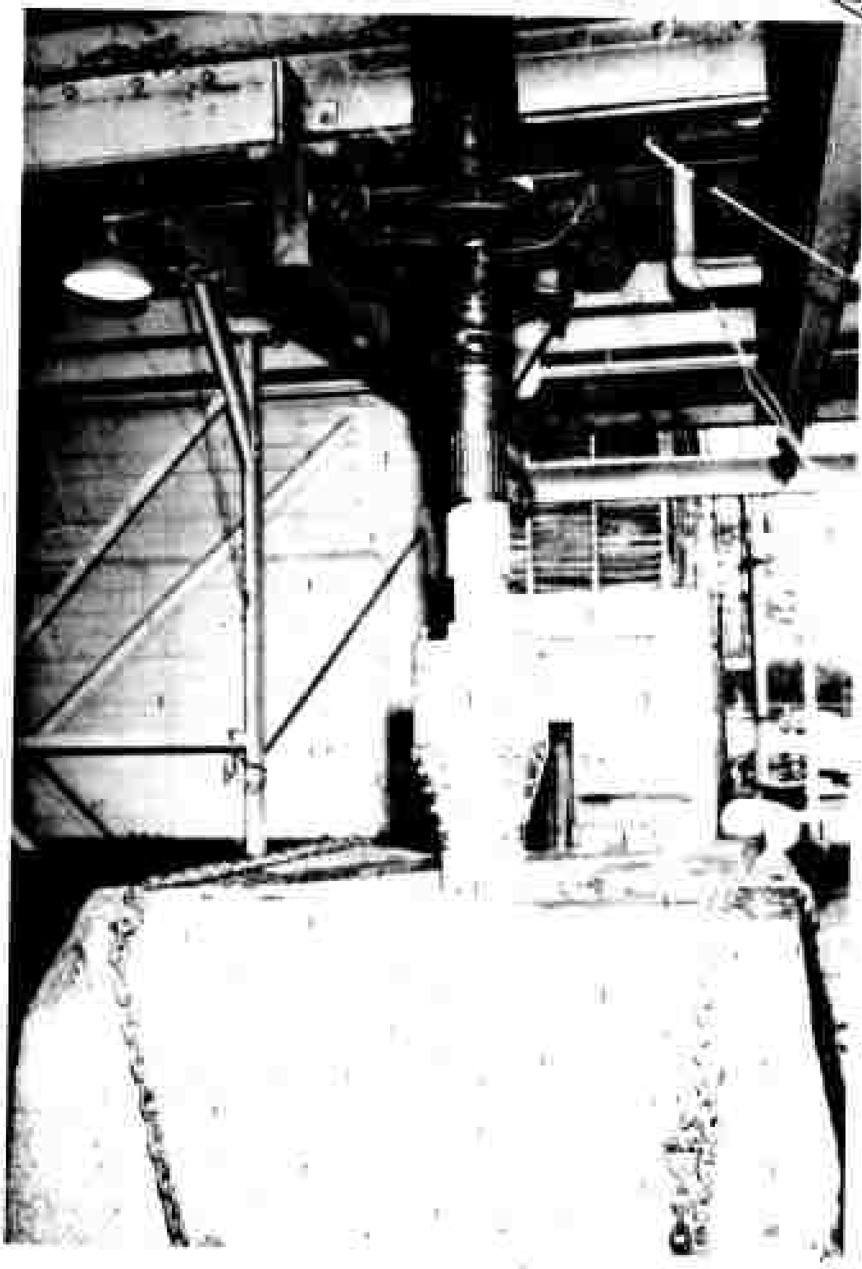
Figures 17 and 18 present the experimental thrust and torque requirements of the conical reamer as compared with a conventional Hughes W7R bit. The following conclusions may be drawn:

(a) As predicted the conical reamer significantly reduces the required thrust, to less than 20 percent of that of a conventional bit. Note also that the conventional bit is smaller in diameter. Furthermore, these results were achieved with a tooth density greater than optimum for the conical borer.

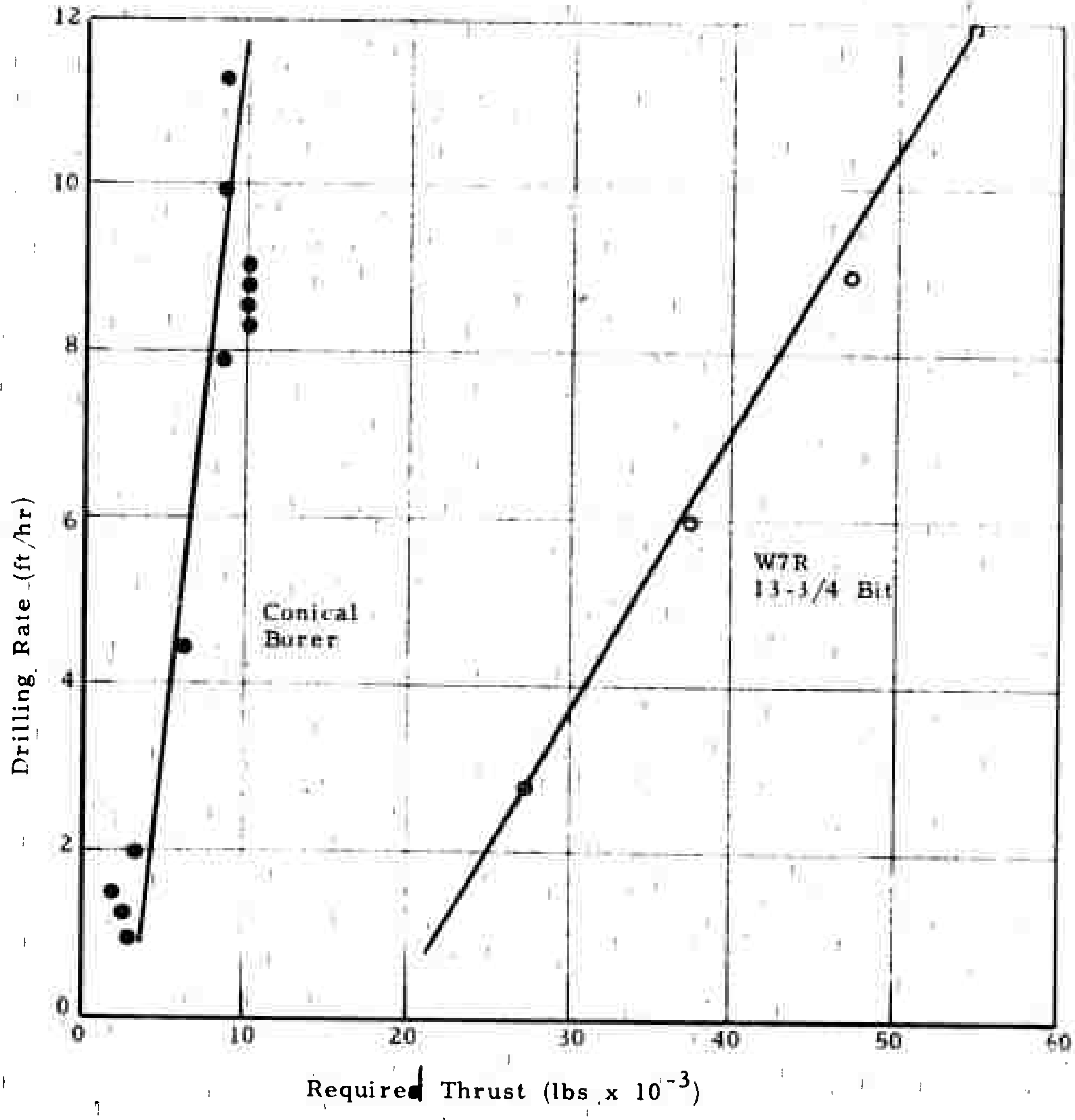


The Conical Borer and Test Equipment

Figure 15

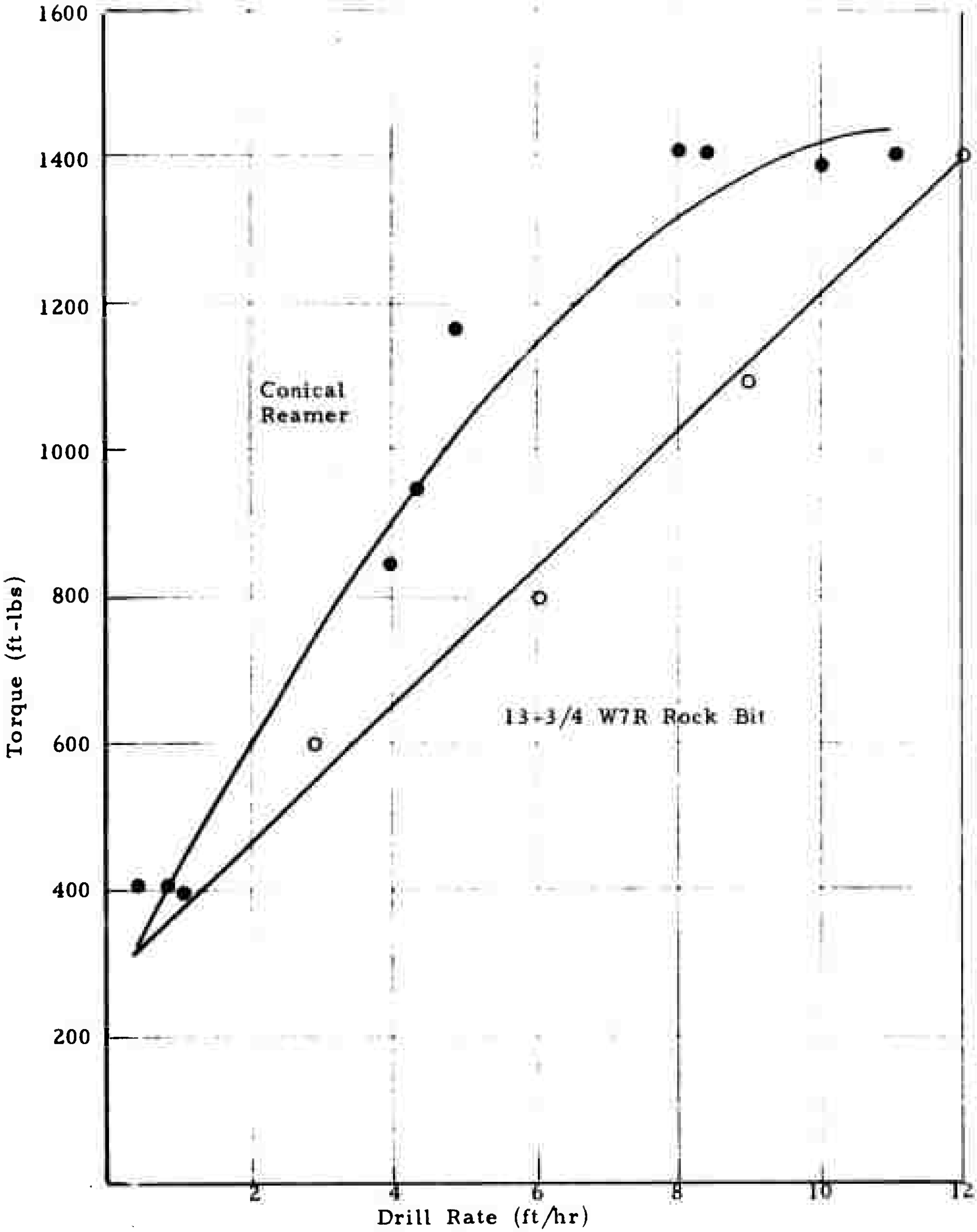


Drilling with the Conical Borer
Figure 16



Experimental Thrust Results

Figure 17



Experimental Torque Results

Figure 18

(2) The nose section, as was planned, has some self-advancing capability (it required about 33 percent less thrust than a simple conical reamer) but still requires some thrust force for advancement. In the completed borer this force will of course be furnished by the weight of the borer structure. It was designed in this manner to reduce the chances of it stalling out and locking itself in the hole.

(3) The current tooth density requires 13 percent more power than the conventional bit. This was expected as the tooth density in this unit was purposely made greater than the optimum. It can be expected that decreasing the density will reduce the power requirements to that of the conventional bit. Indeed, it can be seen that at the higher drill rates (and at the higher tooth loadings) that the conical data levels off and approaches that of conventional bits. Reducing the tooth density on the conical reamer should bring the data together.

In summary then the experimental results confirm the analytical predictions, prove out the design, and back up our confidence in the ultimate performance of the complete borer system. Further experimentation is desirable to improve cutter performance.

6. Conclusions and Recommendations

The following conclusions may be drawn:

(a) The test results demonstrate the validity and advantages of the conical borer principle. Thrust requirements were reduced by a factor of five over conventional bits while, even with a recognized high tooth density, the required torque was within 13 percent of the predicted values.

(b) Additional testing and tooth distribution and density modification will further improve performance.

The following recommendations are made:

(a) That additional testing be undertaken to verify these results and to improve the tooth distribution and density.

(b) That detail design fabrication and assembly of the remaining borer stages be initiated.