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MASTER

FLEXIBLE ROOF DRILL FOR LOW COAL

Vol. 2, Phase III and Phase IV

DOE/ET/12579--T1 Vol 2

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by

NORMAN H. SHOUP

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FOREWORD

This report was prepared by Washington State University, Department of Mechanical Engineering, Pullman, Washington, under USBM Contract No. H0242027. The contract was initiated under the Coal Mine Health and Safety Program. It was administered under the technical direction of SMRC with Mr. Robert Burnham and later Mr. John Bevan acting as Technical Project Officer. Mr. R. J. Simonich was the contract administrator for the Bureau of Mines.

This report is a summary of the work recently completed as part of this contract during the period June 15, 1974 to October 7, 1977. This report was submitted by the authors on September 8, 1977, and consists of two volumes. Volume 1 includes Phase I and Phase II project work done under the direction of Principal Investigator Dr. L. Alden Kendall. Volume 2 covers Phase III and Phase IV, during which time Professor Norman H. Shoup was serving as Principal Investigator following Dr. Kendall's transfer to another university in August 1975. Phase V, mine test of the flexible drill system, was deleted from the contract under terms agreed on in Modification No. 5 effective April 1, 1977.

The author wishes to acknowledge the cooperation and generous assistance rendered by the United States Steel Corporation. At a most nominal cost, they have made their mine operations, equipment, and test facilities available during project field test studies and have provided valuable technical assistance throughout the design development phases. Particular thanks are due Mr. R. C. Miles, Associate Research Consultant at the U.S. Steel Research Laboratory, for his diligent efforts in coordinating the field work and vari-

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A project of this type and long duration involved the services of many members of the Mechanical Engineering faculty working as a team. The authors gratefully acknowledge the extensive and valued contributions made by Dr. Gary L. Sheldon, Dr. Arthur L. Jones, Dr. John A. Seevers, Professor Jack T. Kimbrell and Professor James S. Englund. Graduate research assistants participated extensively throughout the project. We are indebted particularly to Mr. William A. McIlvanie whose graduate thesis work, supported by the Bureau of Mines, provided the basis for the initial project proposal on development of the segmented shaft drill. Other graduate research assistants making important contributions to the project and design work were Sash Eirich, Gordon H. Beeman, Allan G. Olson, Anwar U. H. Mirza, John L. Stevens, Lowell R. McDaniels, Tony Chung, Rodger Cox, Arun K. Das, and Ray L. Davis. The construction of the prototype model and preparation of all plans and working drawings would not have been possible without the able assistance of our drafting technician Mrs. Karla Kalasz and all of the effort put in by Instrument Makers Marion L. Johnson, Jon B. Grimes, and others of the Engineering Shops most capably supervised by Mr. Tommy Hellesto.

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1.0.0 SUMMARY

Phase III of the flexible drill project includes refinement of design and production of working drawings for construction of a prototype machine selected from design concepts developed in Phases I and II. Phase IV involves construction, assembly, and initial laboratory testing of a working model prototype. It is characteristic that design modification and developmental testing of components extend into and continue through the construction phase. Consequently, no attempt is made to separate Phase III and Phase IV work into two different sections of this report.

Design specifications were developed for a flexible drilling and bolting system. The system configuration is based on the use of the Galis dual-boom roof drill as a vehicle for the drilling/bolting system. The WSU Flex-drill drivehead is mounted on the Galis drill boom and the Galis parts are modified to accommodate the revised system. The flexible drillhead supports a bolt bender/insertor designed by Bendix Corporation and is integrated into the system operation. A supplemental bolt thruster was designed to complete insertion of the roof bolt following operation and removal of the Bendix bender from the bolt shank. The complete cycle of drilling a 1-3/8-in. diameter bolt hole, bending a roof bolt into the hole, thrusting the bolt head and washer to the roof surface, and final torquing of the bolt is manually controlled by the operator located in a new position behind the bolting line. The new operating position is beneath newly bolted roof in a safer location in back of the stab jack and roof jacks positioning the drill boom and drivehead.

The Flex-drill/bolting system prototype was constructed from both purchased components and parts specially fabricated in the shops for this design. This unit was assembled and test-operated with appropriate support equipment in a laboratory test stand. Numerous test holes were drilled in blocks of concrete at feed rates of 5 ft/min with drill rotation speeds of 360 rpm. The drill feeds uniformly and cuts smoothly with no difficulty in collection of dust or clogging of the drill bit. The holes drilled were straight, as

evidenced by passage of a 1-1/4-in. diameter bar full depth into the hole with no binding or evidence of curvature. The flexible drill is capable of drilling 8-ft-deep roof bolt holes in low coal 36 in. in height.

2.0.0 INTRODUCTION

2.1.0 Phase II Design Status

At the completion of Phase II, the vertebrate or segmented drill design was definitely established and tested. The segments were hexagonal, jointed at 1-1/2-in. intervals, of three-toothed configuration, and retained by internal snap-ring connectors. The drivehead design was narrowed to five alternate concepts which differed mainly in the means employed to apply thrust to the drill string. Several variations of pawl-and-ratchet drives, yokes powered by reciprocating hydraulic or cam mechanisms, and both reciprocating and continuous friction drives were developed for consideration. A design decision on the drill thrusting mechanism became the critical path at this stage of development of the flexible drill. Test evaluation of the friction drive concept was then of high priority in the beginning of Phase III work. Once the friction drive system had been verified as feasible, the drill segment configuration no longer required shoulders or slots for the thrust mechanism. Also, this established the basic specification for the drivehead design configuration.

2.2.0 Drill System Concept

Before undertaking Phase III design development prototype specifications were presented for review. The drill system was to be designed for the Galis 3500 dual-boom roof drill as a test vehicle. The prototype flexible drill would include a boom-mounted drivehead powering a segmented flexible drill shaft with internal vacuum chip-removal tube and auxiliary dust collection system. In addition, the prototype would specifically incorporate operation of the Bendix bolt bender/insertor as an integral part of the drilling/bolting system. A device or mechanism was also required to complete insertion of the roof bolt into the drilled hole and to torque the bolt to the required tightness. The prototype was designed as a complete system to drill the hole, bend and insert the bolt, complete insertion and torque the bolt with all operations controlled by the operator from a safe working position.

2.3.0 Final Design, Construction, and Testing

Phase III project work consisted of preparation of the prototype design. Construction and laboratory testing of the flexible drill and bolting system is covered under Phase IV requirements. In the report that follows, prototype design specifications are given, steps in design refinement to meet those specifications are described, and operational tests of the equipment in the laboratory are evaluated.

The general performance of the WSU Flex-drill design was found to be quite good. The drill is capable of meeting specifications for drill thrust, torque load, feed rate, and retraction speed. The Flex-drill/bolter system, however, does not meet the original specification for operation in 30-in. coal. With the bolt bender and bolt thruster units attached a minimum coal height of 36 in. is required. The machine is capable of drilling and bolting holes within 31.5 in. of the face (30 in. specified) and within 18 in. of the rib as required. Bolting components of the system functioned as designed but were not completely operation-tested as a combined system in the laboratory. With the limitations in time and simulation equipment, laboratory tests were primarily directed toward evaluation of the flexible drill and its successful operation.

2.4.0 Summary Report of Inventions and Subcontracts

As part of this report, a copy of the Summary Report on Inventions, Department of the Interior Form D1-1216, is attached as Appendix T. This report includes the initial disclosure of invention--"Flexible Shaft Drilling System," by William McIlvanie, and subsequent related disclosures since this work was initiated in 1974.

3.0.0 PROTOTYPE DESIGN SPECIFICATIONS

3.1.0 System Concept

Phase III design work was enlarged to include the concept of a complete flexible drilling and bolting system. This system concept requires integration of the Bendix bolt bender/insertter with the flexible drill design developed during Phases I and II of this contract. In addition, an unspecified device is required to complete insertion of the bolt and torque the head to proper bolt tension. These components are all designed for use on a model 3500 dual-boom roof drill manufactured by the FMC Corporation, Galis Division (now Mining Equipment Division).

3.2.0 General System Configuration

Configuration of the system is closely associated with the vehicle chosen as host to the developmental flex-drill and bolter system. Selection of the Galis dual-boom roof drill was made by U.S. Steel as the equipment most readily available for mine tests required during Phase V of this project. This is also the same model drill (illustrated in Appendix C, Phase II report) used during underground tests of the segmented drill string.

3.2.1 Basic Components

Drawings and performance specifications of major components of the flexible drill/bolter system are contained in Appendix Q. For reference, these basic components are listed as follows:

- (1) Segmental flexible shaft, Drwg #7528-1, Fig. Q-1
- (2) Drivehead assembly, Drwg #7525-0, Fig. Q-2
- (3) Bendix bolt bender/insertter, Drwg #7524-0, Fig. Q-3
- (4) Galis Roof Drill
 - Boom assembly, Drwg #3508-D1, Fig. Q-4
 - Swing frame assembly, Drwg #3508-D2, Fig. Q-5
- (5) Bolt thruster, Drwg #7529-0, Fig. Q-6
- (6) Galis hydraulic system adapted for Flex-drill, Drwg #7527, Fig. Q-7

(7) WSU flexible shaft drill/bolter-inserter system, Drwg #7526-0,
Fig. Q-8

Basic and other components parts will be described in detail, and factors influencing particular design developments will be noted in the following discussion.

3.2.2 Galis 3500 Roof Drill

Selection of the Galis machine as a vehicle for the flexible drill system presented some advantages, but it also imposed restrictions on the design. This model, illustrated in Figure 42, is a well designed low-coal machine. In addition, the boom assembly is mounted on cam rollers and may be "sumped" forward 10 in. This sump motion provides means of adjusting drill hole location. Galis roof drills observed at a U.S. Steel coal mine had the sump mechanism disabled. When questioned, a maintenance supervisor indicated this refinement was not considered essential in their mine operation and its use had been discontinued to effect savings in maintenance costs.

This circumstance prompted early consideration of using this existing sump mechanism to obtain indexing motion between drilling cycle and bolt insertion cycle. The most feasible arrangement of drillhead and bolt bender was determined to be that shown in Figure Q-8 (Appendix Q). This places the bolt insertion centerline 22 in. forward of the drivehead drilling axis. Therefore, all that was required was a remodel of the sumping mechanism to increase travel from 10 in. to 22 in. This modification turned out to be more extensive than first anticipated. The pedestal assembly (Fig. Q-4) required rebuilding to accommodate a longer sump cylinder. The swing frame shown in Figure Q-5 must be cut in two and made 14 in. longer by welding in extension sections; and longer roller track guides for the boom assembly were needed. Figure 43 shows the boom in both extended sump position and retracted position.

With this general concept in mind, FMC Corporation engineers were consulted at the factory in Fairmount, WV. It was found that the boom and swing frame assembly could be purchased as a separate unit. As such, it would provide an integral structure for attachment of the WSU Flex-drill/bolter system. This was the basis for the detached module concept employed throughout the remainder of the project.

With hydraulic controls attached, the boom and swing frame assemblies make up one boom of the dual-boom roof bolter, as illustrated in Figures 44

Figure 42 Galis Model 3500 Roof Drill

Figure 43 Sump Movement, Pedestal Retracted (Left) and Pedestal Extended (Right)

Figure 44 Galis 3500 Roof Drill at Factory (Front Left Boom)

Figure 45 Galis Roof Drill at Factory (Dual Booms, Machine Center Looking Forward)

and 45 showing a vehicle during factory assembly. In this case the right boom (vehicle operator right) was selected because of the asymmetry of the Bendix bolter component. When fitted with flexible drill, bolter components, dust collector, and hydraulic control system, this equipment forms a self-contained module. The basic unit as set up in the laboratory is shown in Figure 46 with boom in the lowest and highest positions of travel. The unit lacks only support items normally provided by other elements of the vehicle: hydraulic pump supply, vacuum blower, and physical mounting support for swing frame actuation. The module is thus readily adapted for mock-up and laboratory test of the developmental drill system. Conversely, it may be retrofit to the vehicle with minimum difficulty. A single pivot pin attaches the module to the Model 3500 drill via two vertical hinge plates mounted on the vehicle. Retrofit is essentially complete when hydraulic lines have been connected to the hydraulic supply system located on the main structure of the Galis roof drill.

3.2.3 Drivehead Assembly

The drivehead, Figure 47, is trunnion-mounted on the Galis boom arms in the same manner as was the original drill head. The mounted unit is shown from beneath with boom arms elevated to roof level (Figure 48). A leveling linkage on the Galis boom maintains the drill in vertical alignment as the boom is elevated to any position. This linkage was rebuilt to accommodate the larger and heavier Flex-drill drivehead. The drivehead is essentially the basic unit of the drilling/bolting system. Brackets (Figure 49A) mounted on the forward side of the drivehead and gear case carry the Bendix bolt bender, the bolt thruster (Figure 49B) and associated indexing linkage. This provides a package unit with fixed spacing between elements that is always level and that may be positioned vertically or horizontally in a fully controlled manner.

The Flex-drill system has distinctive features which provide improved drilling characteristics. The drivehead assembly is pre-positioned against the roof with a force of 8,000 lb, a load exceeding thrust forces common to most drilling operations. Thus positioned, the drivehead provides a rigid and stable drilling platform.

Standard drill practice involves thrust load application to the drill by a boom or jack column actuated by a hydraulic ram under pressure. In contrast, the flexible drill string is advanced at a positive feed rate by an internal chain drive and gear system powered by a hydraulic motor. In this

Figure 46 Boom and Swing Frame Assembly
Side View - Boom Raised and Down

Figure 47 Drivehead Gear Case

Figure 48 Drivehead Gearcase - Boom in Elevated Position

A

B

Figure 49 Indexing Brackets (A) and Bendix and Thruster Mounted on Drivehead (B)

case, the thrust developed is a function of feed rate and drilling conditions at any particular time. If the thrust reaches excessive peak values, the hydraulic drive motor will stall at the pre-set relief pressure. If a soft spot is encountered or the drill breaks through a hard layer of rock, the thrust force automatically reduces until the feed advances the drill into an area of increased opposition. Under similar circumstances, "springy" booms and hydraulic actuators cause a sudden and too rapid rate of penetration--causing the drill to stick.

Drill feed and retraction are accomplished by a system of chain-mounted friction pads bearing against opposite sides of the segmented drill string. A normal (perpendicular to the surface) load of 16,000 lb is applied to the friction pads through roller bearings contained in a cantilevered beam loading system. Friction pads mounted on one of the drive chains are shown in Figure 50. Section 4 covers design of the friction drive system in more detail.

The hexagonal drill segments pass through a guide of matching geometry in the center of the drivehead. Torque is applied to the flex-shaft through this guide which is affixed to the friction drive housing and ring gear. The ring gear and housing are caused to rotate at drill speed by a hydraulic motor suitably connected through a reduction gear. The drill thrust or feed mechanism is not operative unless the thrust motor is started, even though the mechanism is "in toto" rotating at drill speed. Inputs of the drill rotation and drill thrust motors are combined through a set of differential gears so that power is applied to the thrust system only when there is a difference in the rotational speed of two matching ring gears (parts 1 and 4). This differential drive, shown in schematic form in Figure 51, permits the operator independent control of drill rotation and drill feed or retraction. Adjustable, but preset and lockable, flow control valves provide means of changing the drill speed and feed rate to accommodate drilling conditions. Drill speed is adjustable up to 400 rpm and feed rates are variable up to 8 ft/min.

3.2.4 Flexible Drill String

Final design of the drill segments is similar to that shown in Figure 52. The two-cable connector design was changed to a three-cable configuration to insure dynamic stability of the rotating shaft. Extensive refinement has taken place between original concept and present design. Both design and method of manufacture were optimized where possible. The basic segment is

Figure 50 Friction Pads Mounted on Drive Chains

Figure 51 Drivehead Gear Schematic

Figure 52 Segmented Drill String with Cable Connectors

machined from 1-1/4 hex stock of AISI 4140 steel, and then heat-treated to R_c 40 to improve wear characteristics. The corners of the hexagonal segment are machined to 1.375 in. diameter, forming six longitudinal lands or contact surfaces on a diameter equal to the nominal size of the drill hole. The carbide drill bit is sharpened 0.030 to 0.050 in. oversize to provide clearance for the starter section and drill segments. This clearance is ample to prevent sticking the drill, yet is within the tolerance for drilling straight holes when using starter sections as short as 9 in.

For specific dimensions of drill segments, reference is made to drawing No. 7528-1, found in Appendix Q, Figure Q-1. The segments have three teeth at either end capable of transmitting 3,000 in.-lb of torque. Designed clearance and tooth corner radii permit operation of the drill string around bends of 12-in. radius. Successive segments are connected by three stainless steel cables terminated at each end with silver-soldered sleeves. Tensile strength of the drill string with cable connectors exceeds 1500 lb. Rotation of the segments in curved alignment does not cause the wire cable connectors to bend or flex. The hole through which the cable passes has been made cone-shaped, thus precluding flexure of the connector wire and avoiding problems of metal fatigue.

A polyurethane chip-return tube was selected for flexibility, surface lubricity, abrasion resistance and ability to withstand vacuum pressures. This tube is fastened to the first segment behind the starter section and extends down the center of the complete drill string. While the tube does rotate with the drill segments, it must be free to slide longitudinally within the segments. A rotary union, Figure 53, connects the vacuum dust collector hose to the flexible drill string.

3.2.5 Bendix Bolt Bender/Inserter

The roof bolt bender was developed by Bendix Corporation under U.S. Bureau of Mines Contract No. HO230036 - Ref: Installation and Maintenance Manual U.S.B.M. - RBI-203, October 1974, Bendix Corp., Guidance System Division, Denver, Colorado. This unit receives an operator-loaded bolt, and continuously bends and inserts it into a predrilled hole in the mine roof. The roof bolt bender/inserter is intended for low coal operations where the roof bolt length exceeds the seam height. Any length bolt is continuously bent approximately 90 degrees, restraightened and fed into the predrilled bolt hole. The prototype roof bolt bender, Figure Q-3, consists of a housing/drive

Figure 53 Rotary Union for Dust Collector Hose

assembly, hydraulic motor, hydraulic control circuit, input-output hydraulic connections, and a deployment assembly (not shown). A unit without the deployment assembly was furnished by the Bureau of Mines for system development work on this contract.

The turret type of concept was proposed at several points during development of the flexible drill/bolter system. The WSU Flex-Drill design with extending drill string and containment tube is a component not well suited to a turret-positioned system. Consequently, a decision to utilize sump movement positioning between drilling and bolting operations established the general configuration of the system. The operating space envelope of the system is further refined and restricted by the Contract specification requiring capability to drill holes within 30 in. of the coal face and as close as 24 in. to the rib. The Bendix bolt bender was finally located in the system as shown in Figure Q-8. The prototype model of the Bendix machine was designed for attachment to the Fletcher LTDO with the operating position at "machine left" of the single boom. A more compact arrangement was obtained with the Bendix angled to the opposite side with the operator on the right. For this reason, the WSU system was designed for the right-hand boom of the dual-boom Galis 3500 roof drill. The operator's position and hydraulic controls were moved from the front to the rear outer side of the right-hand boom.

3.2.6 Bolt Thruster and Indexing Mechanism

The bolt thruster is a mechanism devised to complete insertion of the bolt following operation of the Bendix bolt bender. As illustrated in Figure Q-6, the bolt thruster is essentially a miniature drill boom mechanism. The thruster plate travels some 25 in. in a vertical path and is maintained in horizontal attitude by a leveling or pantograph linkage. The thrust arm is powered by a rotary actuator connected to the linkage chain and sprockets. As shown in Figure 54, with full hydraulic pressure, the actuator provides a thrusting force of 150 lb, pushing the bolt head and washer flush with the roof surface.

The bolt bending and insertion operation involves a coordinated sequence of events. To reduce opportunity for operator error, the bender and bolt thruster units are positioned hydraulically by an indexing linkage controlled by sequence valves. After placing a bolt in the bolt bender, the operator activates the bending process by manual hydraulic control. He stops the bending and insertion cycle as the bolt head nears the nip between the main roller

Figure 54 Bolt Thruster (A) Top of Stroke (B) Retracted

and the first straightening roll. At this point the bolt head is some 15 or more inches below the roof line. To release the bolt shank, the bender control valve is positioned in the reverse direction, retracting the straightening rolls. This completes the bolt bender operation. It is now required to disengage the bender from the bolt shank and complete insertion of the bolt.

The indexing system and attachment of parts is shown in Figure Q-8. Both the bolt bender and bolt thruster are pivotally mounted. They also have a common connection to a hydraulic actuating cylinder through two pivot linkages. The operating sequence is as follows:

1. Operator raises indexing valve handle.
 - a. Hydraulic cylinder extends, withdrawing bender from shank of bolt and bringing bolt thruster plate under bolt head.
 - b. Hydraulic cylinder reaches end of stroke--rise in hydraulic pressure activates sequence valve #1 switching flow to rotary actuator on bolt thruster.
 - c. Bolt thruster arm raises until bolt is firmly against roof. Relief valve "squeal" signals operator to return valve handle to neutral position.
2. Operator pushes indexing valve handle down--a second sequence valve is actuated in the reverse cycle, thus moving the thruster to the "down" position and restoring both units to the starting position. Figure 55 shows the bolt bender and thruster in "end-of-cycle" position with the boom lowered to the level of the bumper guard. This would be a tramming position, advancing the vehicle to the next row of bolt holes.

3.2.7 Hydraulic Control System

Reference is made to Figure Q-7, Appendix Q, for detailed layout of the hydraulic system. The Flex-Drill/bolter system is entirely hydraulic and is designed to operate from the 30-hp Galis electric motor-driven oil pump supply. This pump is a tandem-mounted gear pump with three independent sections supplying 18, 18, and 9 gal/min (gpm) for a total flow of 45 gpm. Flow from the two 18-gpm sections power the tram motors on either side when the machine is being moved. During the drilling operation these two pump outputs are combined to supply 36 gpm to the drill motor with some excess allowed for flow control bypass, providing variable drill speed. The 9-gpm pump is used independently to operate the various jacks or hydraulic cylinders.

Figure 55 Front End Drill, Low Position with Bumper

With the added equipment, a new set of flow requirements had to be met for the Flex-Drill system. The drill motor requires 29 gpm; the thrust motor takes 7 gpm at the maximum drilling feed rate and 17 gpm for rapid retraction; and the Bendix bender requires 14 gpm. To satisfy these diverse requirements, a flow divider (lower left corner of Fig. 46) is employed to repropotion the combined flow (36 gpm) of the two larger pump units. This 36-gpm stream is divided into two streams of 29 and 7 gpm. The 9-gpm pump flow and the 7-gpm stream are controlled by a pilot-operated flow combiner valve supplying either 7, 9 or 16 gpm to the appropriate hydraulic components in each mode of operation. The pilot system also serves to release a caliper brake on the drill thrust motor shaft. Pilot pressure actuates the brake when the motor is stopped and is released when the motor is powered in either the feed or the retraction mode. This serves to lock the thrust drive shaft and prevent unwanted "creep" or slow movement of the drill string during idle periods of Flex-Drill operation.

The operator position is just forward of the dust box at the hydraulic control panel shown in the foreground of Figure 56. Control valves (lower left of panel), boom and sump controls, and flow control valves (upper right part of panel, Fig. 57) operate the components mounted on the boom (from right to left--drill-rotate valve, bolt bender, drill thrust, sequence valve controlling bolt thruster and insertion cycle). Actual steps in the system operation are described in the following section.

3.2.8 Operation Sequence

The operation sequence for drilling and inserting a roof bolt using this prototype system is as follows:

1. A Model 3500 roof bolter is trammed into position--centered in the mine heading 4 ft beyond the last row of bolts set. If the mine heading is 16 ft wide, the dual-boom machine will be able to reach the four required holes without being relocated.
2. The swing frame is moved sideways to position the first hole (2 ft off centerline for 4-ft bolt spacing).
3. The two stab jacks are set against the floor, causing the swing frame to lift slightly free of the roof bolter chassis. These jacks stabilize and maintain the swing frame in a fixed location during drilling and bolt-insertion cycles.

Figure 56 Assembled Drill/Bolter - Side View

Figure 57 Assembly - Side View Boom - Drill/Bolter

4. Dual roof jacks are extended and set to maintain preload in opposition to the stab jacks.
5. If not already extended, the sump cylinder is actuated, positioning boom and drivehead beneath intended drill hole location.
6. The boom is raised to set the drivehead against the roof with a vertical preload of 5,000 lb force.
7. Drivehead torque and thrust controls are independent of each other. Normal sequence is to start drill rotation and then initiate thrusting or drill feed. The rotation speed and thrust feed rate are preset via flow-control valves. Once drilling is underway, the operator monitors a visual display indicating drilling depth.
8. When the visual display indicates that required depth has been reached, the operator stops the thrusting motion and reverses rotation of the thrust motor. The differential drive system causes the drill to be withdrawn from the hole at a rate of approximately 36 ft/min.
9. At a preset point the descending drill actuates a deceleration valve reducing flow to the retracting motor. As the flexible-shaft drill nears the retracted position, a limit switch stops oil flow to the motor and actuates a brake. Diversion of this oil flow through the relief valve signals the operator to restore the retract valve lever to the neutral position.
10. The drivehead is lowered from the roof about three in. and the sumpping action is activated to withdraw the drivehead and bring the bolt-bender into position.
11. The operator places a roof bolt into the bolt-bending unit and starts its operation.
12. When the bolt has been inserted into the hole, the bolt bender/insserter is actuated to release the bolt shank. An indexing cylinder is actuated to move the bender off the bolt. Pressure rise at completion of this action causes a sequence valve to activate the bolt thruster mechanism, pushing the bolt and washer plate on up to the roof.
13. The operator now sumps the drivehead forward again. A centering device incorporated onto the bolt causes it to be centered accurately when it is inserted during step 12. The drivehead is raised against the bolt to engage a square socket built into the drivehead.

Torquing action is continued until the required torque has been reached as determined by a preset relief valve switched into the hydraulic system during this particular operation.

14. The drivehead is now lowered away from the roof approximately 3 in.
15. The roof jacks are lowered and the stab jacks released so that the boom and swing frame can be moved sideways another 4 ft to position for the next bolt.
16. The cycle, steps 2 through 15, is now repeated for the second hole.
17. After the second hole has been drilled and the bolt inserted, the mining vehicle can be trammed forward to the next row of holes.

4.0.0 DESIGN REFINEMENT AND ANALYSIS

The prototype design has been described in the previous section. The various major components of the system were covered in some detail. However, the design analysis and decisions that delineated the final configurations of several important subsystems has not been covered. The following sections deal with the analysis and refinement of design in specific component systems.

4.1.0 Friction Drive for Drill Thrust System

4.1.1 Design Alternatives for Drivehead

Some five different systems for applying thrust to the segmented drill string were presented as viable alternatives at the inception of Phase III design work (Ref. Appendix P, Phase I and II: Figs. P-1,2,3,4).

1. Pawl-and-ratchet drivehead
2. Reciprocating hydraulic drivehead
3. Reciprocating cam drivehead
4. Reciprocating friction-pad drivehead
5. Continuous friction thrust drivehead

At this point, the drill segments had been designed to accommodate some sort of tooth, pawl or yoke thruster: viz., formed with a reduced center section and shouldered at either end. Friction drive was considered a somewhat "outside" possibility--attractive but with little precedence or reliable data on feasibility. In any event, the drill segment geometry could not be finalized until a decision was made concerning the thrust system. Each of the systems suggested may be categorized as either "mechanical engagement" thrusters or "friction" thrusters. "Mechanical engagement" implies the interlocking of teeth, pawls, yokes, serrations or other protruberances with a hole, pocket, slot, recess, or other depression in the segment to effect application of force and positive advancement of the drill segment some specified distance. In contrast, the "friction" system denotes movement of the segment by shoes, clamping pads, split collars, collets, or expandable sleeves deriving a driving contact (friction) force from application of forces "normal" or perpendicular to the direction of motion. It was determined that a fairly simple

series of tests would provide sufficient information to eliminate conjecture about the friction drive systems. If it were decided the friction system was not practical, selective investigation of the myriad technical variations would be initiated.

4.1.2 Verifying Tests

Two primary factors needed verification: 1) What maximum normal (transverse) load could the drill segments sustain without plastic deformation?; and 2) What was the actual coefficient of friction between drill segment and friction pad surfaces? To evaluate these factors, an orthogonal loading system was devised in which the drivehead thrusting conditions could be simulated. This test setup is illustrated in Figure 58. The main structure is a Young, 120,000-lb Universal, testing machine. A large "C" frame bending fixture was affixed to the testing machine platen. The bender is powered by a hand-pumped hydraulic ram capable of developing loads up to 20,000 lb. A hexagonal test section of drill segment stock, flanked by two test samples (as shown in Figure 59) is clamped in the C-frame (Figure 60) under a preset normal load. The testing machine platens are then brought together until contact is made with the load disk (part 1) atop the hex drill segment. Load is applied at a uniform rate until the static coefficient of friction is exceeded and slippage occurs between the clamped surfaces. This point is readily noted by the abrupt change in loading rate indicated on the testing machine scale. At times the transition from static to sliding friction is also accompanied by audible sounds. This simple procedure was used to evaluate a variety of friction surfaces, materials and conditions of surface contamination.

4.1.3 Coefficient of Friction

An attempt was made to evaluate coefficient of friction for combinations of four different variables: 1) composition and hardness of metallic pairs, 2) metal surface preparation, 3) two different geometries (Figure 61)--a vee block and a flat pad bearing on opposing faces of the hex stock, and 4) external contamination or materials between friction surfaces. Conditions of contamination included both dry and damp shale dust from actual mine drill cuttings in addition to an application of a water slurry of shale dust. One series of tests was made with a slurry of oil and shale dust. Figure 62 shows the effect of water slurry and dust on ground surfaces (3A) and (4), where contaminant matter was literally "brinelled" into the metal surface leaving

Figure 58 Laboratory Setup for Friction Tests

Figure 59 Friction Test Specimens (1) Load Disk
(2B) Hex Drill Stock (3C) Serrated Pads

Figure 60 Friction Test Specimens in Load Position

Figure 61 Vee Block Friction Specimens

Figure 62 Flat Block Friction Specimens

obvious depressions. The results of these tests are shown in Figure 63. It is of interest to note that drilling dust increases the coefficient of friction in all cases. Demarcation lines have been superimposed on the graph to show the normal load required to achieve design thrust for three minimum levels of friction coefficient. For example, a normal load of 12,000 lb would be satisfactory if the coefficient of friction encountered under all conditions were in excess of 0.25. It is obvious that this normal load is only adequate for roughly one-half the cases shown. A normal loading of 16,000 lb was selected as a more optimum value. To avoid the low friction values obtained for the metal-to-metal conditions, the friction pads were made from hardened tool steel and the surfaces were machine-serrated. In an additional test it was found that the oxidized surface of heat-treated drill segments had a lower coefficient of friction than machined surfaces on the same segment. In subsequent production of drill strings a sandblasting step was added to remove this oxide coating. Sandblasting reduces resistance to rusting and gives a less attractive finish, but it is not considered detrimental to life or serviceability of the drill string.

Friction test results confirmed the feasibility of applying continuous thrusting force to the drill string via a series of chain-mounted friction pads. This led to re-evaluation of a means to apply normal loads of this magnitude and verification of the ability of the currently designed drill segments to withstand this crushing transverse load. Transverse load capacity was evaluated using a "reduced-waist" segment as shown in Figure R-1, Appendix R. The load-deflection curve shows a proportional limit of about 30,000 lb load. With a redesigned flat-sided segment, there would appear to be more than ample load-bearing capacity at stresses well within the region of elastic deformation.

4.1.4 Normal Loading System

The normal load concept comes from the original friction loading systems proposed in Drawings 1331-74-08-1 and 2 (Appendix P, Figures P-1 and P-2, Vol. 1). These were friction drive variations operated in one case by an eccentric lifting mechanism and the other powered by the current friction pad and chain concept. In each instance the normal load is applied through rollers or a wedging action elastically deforming a load ring. The load ring described in Appendix J (Vol. 1) is an ideal means of storing elastic energy, but the load developed is dependent on deflection. With such a simple spring element

Figure 63 Coefficient of Friction for Selected Variables

the deflection is limited to small values on the order of 0.120 in. A relatively small variation in this deflection gives a considerable change in load, when one considers that 0.030 in. wear on the segments would reduce the normal load by half. The inability to compensate effectively for this factor, and the fact that there seems to be no easy way to adjust the load range of the ring, short of changing the ring dimensions, seemed a severe design limitation.

Hydraulic rams, rotating seals, etc., were reconsidered as a means of developing these relatively large loads. The design was finally modified to the present form consisting of a set of cantilevered arms (part -69) loaded at the ends by connecting rods and ball screws (part -235), Figure S-5. The ball screws are each loaded by a special torsion spring (part -78). The torsion springs are preloaded and set at the desired torque by turning with a spanner wrench and inserting a locking pin. Figure 64 shows preloaded springs with dial gage. Figure 65 shows normal load beam and friction pads with dial gage removed. Calculations for required torque efficiency and for load relations are shown in Appendix R, Figure R-2.

This system has several advantages over the classic load ring or proving ring type design:

1. Normal load can be preset to any controlled value.
2. Load can be removed from system without withdrawing drill string or inserting substitute spacer.
3. There is little or no need to compensate for wear. The low spring constant of the torsion springs provides ample "take-up" with only slight reduction in load.
4. In the event of severe wear, the torsion springs can be reset without great difficulty. This operation does require partial disassembly of the drill head, but could be done in the mine.

4.1.5 Friction Pad Design

The friction pad is designed as an integral part of the double drive chain and sprocket system, a partial section of which is illustrated in Figure S-4, Appendix S. The normal load is applied to the back of the friction pads (schematic diagram, Figure 66) by two clevis-mounted aircraft needle bearings. The bearings (part -211) impose high contact stresses on the friction pads (part -65); thus the pads were made from Air-Kool-S tool steel which is an air-hardening, low-distortion, alloy steel with excellent hardenability characteristics. Since the pad design (Figure R-5) is not ideal from the stand-

Figure 64 Preloading Torsion Springs

Figure 65 Drivehead Rotor Showing Normal Load Beam and Friction Pads

Figure 66 Friction Drive System - Current Design

point of notch sensitivity, this was an added reason for using this premium-quality tool steel.

Each pad is affixed between two parallel chains (section G-G, Figure S-6) with type D-3 attachment links having two extended (long) pins in each pin link. The drilled holes in the pads were carefully matched to the 5/8-in. chain pitch (which, incidentally, is not necessarily exactly the same on connector links and pin links). This is a characteristic of roller chain manufacture and needs to be considered in an application such as this. In any event the proper clearance (0.010 in.) between friction pads was achieved by some machine filing and fitting. This clearance is evident in Figure 65, as seen on the slack or return side of the chain. This is necessary to insure clearance as the pads separate and remesh as they pass around the sprockets. The roller contact surface is at the pitch centerline of the chain, and all non-rounded clearance gaps between pads are outside the perimeter of this pitch line. This indicates that theoretically the chain bends in only one direction. However, in practice a slight sag in the slack side of the chain can cause the pads to bind in an offset position. Thus, a slack chain may cause erratic or noisy operation and should be kept in tight adjustment.

As indicated best by the friction tests, the contact surfaces of the friction pads are serrated. The reverse side of the friction pads as assembled on the chain, serve as a cam roller track for the two normal load bearings. The matching "zig-zag" edges of the pads serve to maintain continuity of contact line between bearing rollers and friction pads. This feature insures that the high-pressure contact line between roller and pad surface is always maintained across at least 80 to 90% of the path width.

4.2.0 Differential Gear System for Drill Torque and Thrust

As originally proposed, the friction chain drive illustrated in Figure K-2, Appendix K (Vol. 1), was based upon a differential speed concept to obtain drill thrusting motion. This design utilized two separate hydraulic gear motors--one to provide drill rotation and one to provide thrust through the action of a set of bevel gears. The use of bevel or spiral bevel gears is common practice in most drill heads to obtain an efficient, simple and compact gear drive. In this case, however, a second set of bevel gears is employed to activate the chain drive and produce thrust or feed the drill through the drivehead. If the two main bevel gears are driven at the same speed, there is no relative motion between gears and no thrust motion--only drill

rotation. If the thrusting bevel gear is driven at a different or differential speed, the drill string is fed either in or out depending on the relative direction of gear rotation. It is important at this point to realize that the drill feed control is dependent upon the relative rotational speed of two independent hydraulic motors. In other words, control of the feed rate is external to the mechanical system and an integral part of the hydraulic system.

During the early stages of design concept, the significance of this factor was perhaps not fully appreciated. Subsequent analysis of this concept brought out possible operational difficulties that were at best difficult to cope with. For example, if these two motors were directly controlled by the mine operator, the simple action of turning on the rotate valve without activating the thrust valve would induce maximum drill feed rate. Or, if the drill were reversed, a rapid and uncontrolled retraction would result. Also, if the thrusting motor were free to rotate (open center flow), the torque reaction from the rotate motor would result in some intermediate thrust rate. Thus, several catastrophic combinations of valve operation can readily be foreseen. It is evident that operator control of this function would be unsafe and most difficult to control manually.

4.2.1 Hydraulic Control of Differential Speed

The next alternative considered was hydraulic fluid proportioning control to provide a more foolproof system. It soon was apparent that a rather complex hydraulic system would be required to do the job. Questions of reversibility, reflected load pressures, shock loading, and control problems all brought out the fact that many parameters were involved in such a system. It was decided that a more reliable system was potentially available in a mechanical design and that effort should be directed toward this end.

4.2.2 Interlocking Differential Gear Drive

A process of design refinement was undertaken along this line. Minimum height of the chain drive sprockets severely limited the space available for bevel gear drives if the goal of a 14- to 16-in. maximum drivehead depth were to be obtained. Also, the inherent large diameter of the chain drive and normal load system led to consideration of a spur gear or helical gear drive system. This involved use of three additional idler gears, but considerably simplified the differential gear box. Figure 51 shows a schematic of the gear system developed, and Figures 67 and 68 show the differential gear

Figure 67 Differential Gear Assembly

Figure 68 Differential Gear Train - (Lower Center)

assembly as a unit prior to insertion in the gear box. The drivehead rotor assembly may be considered as a separate unit which interfaces with the differential gear assembly at the simultaneous pitch lines between gears 1 and 5, and between gears 4 and 18 (Figure S-2, Appendix S).

4.2.3 Helical Gear Train Design

This gear system contrasts with the original concept in that there is automatically no feed or drill thrust when the drill is merely rotating. With differential motor off and locked, the drill motor drives gear no. 1 (and the drill and rotor directly) at 400 rpm. Through rotation of idler gear 5 and the differential gear train: gears 20, 21, 22, 6 and 18, the lower gear no. 4 that is concentric with the drill is also driven at 400 rpm. As a result, there is no relative motion between gears 1 and 4, thus no rotation of the planetary gear set no. 2 (17) engaged with gear 3 which is affixed to gear 4, again referring to Figure 51.

The operator may start drill rotation and control it independently of the feed rate. To feed the drill he merely starts the differential thrust motor. This rotates the cage of the differential housing and speeds up gear no. 4 which drives the planetary gears, which in turn drives worm gear no. 11 and turns the sprockets driving the friction pads. This action will feed the drill at a controlled rate up to 8 ft/min.

To retract the drill the operator stops the thrust motor (continuing rotation) and reverses its direction of rotation, automatically switching to a higher oil flow. This produces the reverse planetary gear action and retracts the drill at a rate of 36 ft/min. The system effectively eliminated the opportunity for damage from human error in operating the controls. Equally important, the torque from drilling effort is not imposed on the thrust motor, but is reflected directly back through the rotate gear train through coupling with the differential and drill motor idler gear. This avoids need for a thrust motor capable of balancing the drive motor torque, allowing use of a smaller-capacity motor of size sufficient only to handle the thrust horsepower.

The gear train configuration was expanded in the horizontal direction to save vertical space. The original design contemplated use of about 15 in. depth for the gear head, and 15 in. would be needed for curvature of the drill string around a 90° bend to direct it toward the stowage area. As may be seen from Figure S-4 (Appendix S), the radius of curvature of the drill string was started at approximately 4 in. up inside the hollow portion of the rotor gear

system. This allows more vertical room for the gear drive and particularly allows both drive motors to be vertically mounted with direct spline shaft drives without exceeding the overall height of the drivehead as seen in Figure S-1.

There are actually only two primary axes of gear rotation: 1) the drill axis, and 2) the axis of the differential gear system shaft as shown in Figure S-2, Section H-H. Gears (-8), (-18), and (-38) are all idler or input drive shafts. Spur gears could have been designed for this system, but it was found that helical gears could be custom-made at little or no greater expense and would operate more quietly and could be designed with less face width to save vertical space. All of the helical gears were designed with a diametral pitch of 6 or a normal pitch of 6.518 in. The gears were machined from AISI 4340 steel heat-treated to a hardness range of R_c 33 to 37. AGMA class 9 machined gears were specified with a backlash of 0.008 to 0.012 in. Gear teeth have a 20° pressure angle and are full-depth stub form with a 23° helix angle.

4.3.0 Drivehead Rotor Configuration

4.3.1 Rotor Construction

If one refers to Figure S-4 and S-5, one may see that the rotor is essentially a large disk with a bolt-on ring gear and two vertical bearing plates attached at right angles to the disk. Between the two plates are mounted the hexagonal drill guide and the friction pad chain assemblies. All other attachments bolt onto these two central bearing plates, as is illustrated in Figure 69. On the left is shown the basic structure with the gears installed that drive the sprockets and chains. The two lower gears are mounted directly on the sprocket drive shafts. The upper gear is an idler gear which engages the drive pinion. The plate in the foreground is the inner half of the worm gear drive housing. In the photo on the right this plate is shown in position and the worm gear and outer housing are shown in the foreground ready for assembly. The planetary gear has been added and the helical gear pair that drives the worm is apparent in both views.

Both of the gear housings in the rotor were designed to be sealed and packed with lubricating grease. The upper drivehead area is a dry area since it would be undesirable to put any lubricant on the friction pads. This area is sealed from the gear case oil by a plate (-117), Figure S-4, and felt seal which is in contact with the large gear (-1) on the lower part of the rotor.

Figure 69 Drivehead Rotor in Two Stages of Assembly

All gears in the lower gear case are sprayed with oil piped from a circulating gear pump.

Figure 70 shows the assembled rotor with starter section and carbide drill bit extending out of the drive socket. The friction pads and chain of the friction drive are visible just below the round cover plate that rotates with the drill. The rotor as assembled in this view is not rotationally balanced. Dynamic balance is achieved by adding a large steel plate with lead weights attached. This plate is shown bolted in place in Figure 71. The plate covers the two torsion springs and ball-nut assemblies used to provide the normal load on the friction pad rollers, and it is easily removed in case the torsion spring preload needs adjusting.

4.3.2 Magnetic Collector

The drill string segments are free to separate (up to 5/32 in.) when not under axial thrust loading. This allows flexibility in the rotating shaft as it passes through the curved sections of the containment tube. The most highly curved section is immediately below the drivehead, as may be noted in Figure S-4. The drill segments enter the drivehead rotor through a tapered bronze hexagonal guide (part -84) and pass between the loaded friction pads immediately above. Drawing S-4 shows rotated (90°) cross-sections of the drivehead rotor for convenience in illustrating all components. In this case we are interested in the central area, and a portion of this is shown full-size in Figure 72. The drill string segments are shown as they are drawn upward between the friction pads-with gaps between segments as they leave the containment tube. It was determined that separation between segments could cause discontinuity in thrusting and perhaps excessive wear if the segments slipped as they emerged from the friction pad area while under thrust load.

In the laboratory it was demonstrated that the links could be collected as they passed through the center of a DC-energized coil of wire. By mounting such a coil around a low-permeability section of the drill string housing, the segments could be brought together as they entered the drivehead--much as a magnetic core is drawn into the centered position of high flux in a solenoid.

A coil was designed for this purpose. This coil would be 2 in. to 3 in. long, about 6 in. in diameter with several thousand turns of #20 or 25 wire. Such a coil would give adequate field strength while meeting MESA temperature standards. A finned, high-permeability case would surround the coil to complete the magnetic circuit, act as a heat sink, and provide any explosion-

Figure 70 Top View Drivehead Rotor Assembly

Figure 71 Drivehead Rotor with Dynamic Balance Weights Installed

Figure 72 Magnetic Collector System

proofing deemed necessary. The required 250 VDC supply could be obtained by rectifying the 440 VAC line power supplied the vehicle, or if the vehicle were DC-operated, the coil could be redesigned for this voltage. The feasibility of this approach seemed valid. However, counting on meeting test requirements and obtaining mine safety approval for this electrical equipment seemed at some risk to the project schedule.

For this reason, commitment to the electrical design was postponed until the feasibility of a permanent magnet system could be more thoroughly investigated. A permanent magnet design did not at first seem very promising. Most high-strength magnets are susceptible to loss of strength from jarring or impact under certain conditions, and the lower-strength Alnico alloys would have to be quite large to obtain the necessary flux densities. More recent information was then found on the development of a new Cobalt-rare earth alloy magnet material that has high magnetic flux density and a high degree of permanence under relatively adverse conditions of temperature and shock. These materials were of sufficient promise that the design effort was redirected toward development of a permanent magnet design.

The only shape of Co-Sm magnets available at a not unreasonable cost was in 1-in. squares roughly 0.5 in. thick. They could be obtained polarized in the direction of the 0.5-in. thickness with the faces ground to quite close tolerances. Returning to Figure 72, one may see that the magnetic collector was designed around the shape of the available magnets. Four magnets (-241) were positioned as shown on the machined magnetic base (-81) on each side of the drill string. Parts -82 and -83 serve as vee-shaped magnetic pole pieces. Separating the gap in this rather flattened "horseshoe" magnet is a nonmagnetic bronze vee-block filler piece (-85). The spacer (-86) and the bronze entry guide (-84) serve to isolate the pole pieces from the adjacent magnetic steel structure. The primary flux pattern is illustrated by curved field lines. When introduced into the field, the magnetic flux crosses a minimum gap at the pole pieces and flows through the segments. As separated segments enter the pole region, they are drawn together with considerable force in spite of the tooth geometry which provides an alternate low gap path through the parallel sides of adjacent teeth.

After construction of this collector design, it was trial-tested in the shop for effectiveness. The initial tests were somewhat disappointing. It was found that the drill string tended to rotate in the hexagonal guide and make contact on the corners of the hex. This produced nonuniform or tapered

flux gap on both sides, resulting in relatively high contact force accompanied by the ever-present friction forces resisting movement in the longitudinal direction. The net result was that the friction forces reduced the "collector" force to 5 or 10 lb, which is not sufficient to provide reliable operation. The remedy for this problem was to apply a 0.010 to 0.015-in. thickness of nonmagnetic material to the pole pieces and guides. This effectively maintains a low-permeability gap--preventing rotation of the segments and the resultant redistribution and high concentration of flux at the corner contact areas previously obtained. Attempt was made to "hard chrome" plate the vee guides and pole pieces. The vee geometry made it difficult to obtain uniform coverage into the bottom of the vee. It was proposed that the vee sides be masked off and plated one side at a time, but the plating company could not handle this special order at a reasonable cost nor within a reasonable time. The alternate solution was to cast an epoxy coating on these surfaces and machine to proper thickness. This does not provide the wear-resistance and durability desired, but it will suffice for the purpose of initial drill testing.

4.4.0 Hydraulic System Development

The hydraulic system has been briefly described in Section 3.2.7 outlining the general specifications and operation of the Flex-Drill/bolter system. It should be explained that some features of the hydraulic system are not in accordance with preferences expressed by U.S. Steel mine maintenance and operations personnel with whom conferences were held on several occasions. The primary difference of opinion arises from the use of pilot oil-operated valves as compared to manual control. There is a general feeling that pilot-operated systems are not satisfactory because there is a supposed lag in reaction time, and the operator loses a certain amount of "feel" or sensitivity in controlling the operation when he is operating a pilot valve as compared to direct manual valve. This advice was not ignored but was carefully evaluated in terms of overall system requirements. It was decided that the design of a mechanical system for simultaneous activation of multiple valves and sequencing of hydraulic operations would be quite cumbersome. In addition, the limit and deceleration control valves controlling pilot oil pressure are small and may be actuated by forces of a few ounces. Full-size valves would require much more actuating force than could reasonably be imposed upon the drill string in this type of system to limit travel of the drill.

Thus, Figure 73 illustrates a schematic representation of the system developed to control drilling and bolting power functions. This design is a combination of manually operated valves and pilot-operated systems. Whenever possible, manual valves were used--for the rotate motor (valve 302), bolt torquing control (valve 300), operation of the bolt bender-thruster indexing cycle (valve 307), boom jack and sump cylinder (valves (332). The additional stab jacks and roof jacks would also retain manual operation. The Bendix bolt bender and the drill thrust motor are the only pilot-operated valve systems. The Bendix is pilot-operated to obtain combined flow (16 gpm) from the flow-divider system. The drill thrust motor is reversible and operates at two different flow rates for drilling (7 gpm) and retraction (16 gpm). Additionally, when the drill thrust motor is operated, the caliper brake (318) must be released and the related pilot valve on the stack valve (304) must be pressurized to permit flow to the motor. The need for the pilot oil system is evident when one considers the three diverse simultaneous functions that must occur each time the thrust motor is operated in either direction.

Some modifications in design of the hydraulic system have been made during construction. It was found that the source of pilot oil pressure designed as a bleeder tap (-313) from the 9-gpm oil supply line was not satisfactory. Fluctuations in this supply pressure were not entirely controllable by the pressure regulator (323). After some difficulty with the system, this problem was bypassed and pilot pressure was obtained from a separate gear pump built into the main oil supply sump used in the laboratory. This pump would of course not be available on the Galis machine as now designed for mine use. It is possible a tap source or a separate pump could be added to the drill system. This problem was set aside for later consideration as not essential to the more pressing work of completing the drill construction.

Another trouble spot in the hydraulic design was recognized and rectified. The cylinder actuating the master brake cylinder (316) would not properly return under spring force, preventing release of the caliper brake. A pilot-operated unloading valve (333) was designed and constructed in the Engineering Shop. Pilot pressure is continuously applied to the right side of the cylinder (315) causing the master brake cylinder to lock the caliper brake. When the drill feed motor is operated in either direction, the caliper brake is released by application of pilot pressure to the rod end of cylinder 315. The pilot valve provides positive unloading of pilot pressure and venting of the right side of the cylinder to the sump. Operation of the master brake

Figure 73 Schematic Diagram of Flex-Drill Hydraulic System

cylinder thus does not depend upon a counterbalanced oil pressure and spring return.

During preliminary operation of the indexing system for bolting, it was found that the small indexing cylinder (-325) and the rotary actuator (-326) pictured in Figure 54 were operating much too quickly because of the large stream flow and the small volume of the actuators. Restrictive orifices 1/16 in. in diameter (-334) were placed in the lines to avoid problems with the high inertia loadings encountered. After reducing the relief valve settings on these lines to 1200 psi and adjusting the orifice size, the desired operating velocities were obtained.

Some difficulty has been experienced in making the flow combiner valve section operate reliably. Part of the difficulty was due to back pressure on the sump return lines caused by downstream line losses. In most cases the pilot system return lines were plumbed into the most convenient return line available. When measured, these back pressures varied but ran as high as 40 or 50 psi depending upon what system was being operated at the time. This pressure is not significant in the 1500- to 2000-psi systems but is quite appreciable in the 150-psi pilot pressure system. As a consequence, return lines for the entire pilot pressure system were re-plumbed and piped directly back to the sump through separate lines. This change also bypasses the return line filter which has a 10- or 15-psi pressure drop. The present laboratory pilot pump system has its own filter so this is not a problem in the present operation. However, a modification will be needed on the roof drill vehicle to accommodate this situation for retrofit and testing in the mine.

4.5.0 Modifications of Galis Drill

As previously stated, the selection of the Galis drill was primarily based upon availability for mine testing. However, once selected, it was found to have several features that were quite readily adapted to the Flex-Drill design. The Galis Model 3500 is a compact, well designed machine used in low-coal operations. As illustrated in Figure 74, the vertical motion of the boom at the drill head axis approximately follows a straight line. The sump movement is shown to be 10 in., and the sump cylinder (original equipment) is described by a dotted outline on the extreme left. The whole swing frame and boom assembly is pivotally mounted on a guided roller assembly, and similar to the boom, it is moved laterally by a "straight-line" linkage and hydraulic jack.

Figure 74 Galis Boom Straight Line Motion

4.5.1 Sump Positioning of Flex-Drill and Bolter

The original concept for modification of the Galis equipment primarily involved substitution of a longer sump cylinder and extension of the guide tracks along the side of the swing frame. The distance between drill axis and bolter centerline determined the added length required for indexing, a minimum of 22 in. Addition of a 24-in.-stroke sump cylinder required extensive alteration of the pedestal assembly (Galis 3508-D1-C1) as shown in Figure 75. These modifications made it more difficult to accommodate the hydraulic hoses that must pass through this area in addition to placement of the new hose lines not previously required.

The extended sump movement was not obtainable simply by extending the original guides. A new guide system was designed, and 14-in. sections were welded into the swing frame as indicated in Figure 76. This modification resulted in a satisfactory working arrangement. However, when more detailed plans were being made for retrofit and testing on the mine vehicle, it was realized that the dual-boom roof drill would not be symmetrical. The drill boom modified for operation with the Flex-Drill system would have its drill center 22 in. forward of the comparable drill on the left boom. If the left boom were advanced to the full stroke of its sump motion (10 in.), this difference would be reduced to 12 in. It was felt that about six inches could be removed from the boom length, which might also improve the "overhung" load situation with Flex-Drill and bolter being much heavier than the original Galis drillhead. As a consequence, the drill boom arms were removed, flame-cut, and the outer trunnion plates rewelded 6 in. shorter. All these changes resulted in the Flex-Drill's centerline being only 6 in. ahead of the standard drill on the opposite arm of the machine.

Consultation with U.S. Steel officials indicated that this difference would not prevent use of a machine so equipped during the mine tests. This question was not pursued to the point of obtaining written approval from mine safety officials for this variance in location of roof bolt holes. If necessary, for limited test purposes the machine could be operated as a single-boom machine and sumped forward between operation of right and left booms, but not without considerable loss of efficiency.

As machine assembly progressed, it was found that additional minor modifications were required. Mechanical stops (adjustable) were installed (part 7540-10, Fig. 75) to limit the downward travel of the boom. This in turn avoids damage to the bolt bender and thruster from contact with the forward

Figure 75 Modification of Drill Boom and Swing Frame Assembly

Figure 76 Galis Swing Frame Modification

bumper guard and/or contact with the mine floor. With the addition of a large drivehead and bolt-bender equipment, the leveling mechanism was almost completely rebuilt, and more closely fitted bearing sleeves were installed to reduce backlash in the linkages.

Modification was made in the location of the canopy jack--originally tandem-mounted (Figure 42) with the stab jack--it was relocated about a foot rearward (Figure Q-8) just behind the hydraulic panel and is labeled "roof jack." This change in concept arose from two factors: 1) there was insufficient clearance for the rock bolt to be placed in the Bendix bolt bender with the canopy jack mounted on the stab jack; and 2) with the operator relocated behind the drill line, his position is beneath bolt-supported roof. In this case a full canopy should not be required--only a roof support jack that at the same time stabilizes the machine. This modification is noted for consideration at a later time during evaluation of safety compliance for the Mine Test phase.

It was anticipated that retrofit of the modified Galis equipment would be covered in some detail in this section of the report. However, since the Mine Test phase is not included in this work, a duplication of effort will be avoided by including detailed coverage of retrofit procedure in the installation and operations manual. A general discussion of major features in this area has already been covered in the section on design specifications for the prototype machine.

4.6.0 Flexible Drill String

Much has been described in Phases I and II of the development of the drill segment design. This work continued in Phases III and IV, and additional modifications and improvements of the segments and connectors were made. At the beginning of Phase III it may be fair to summarize that the design of segments then was quite workable but had certain deficiencies. The dimensions, strength, tooth design, shape and length had all been optimized to a fair degree. The primary deficiencies were in strength of the snap-ring retainer, slightly less flexibility than desired, and a lack of immediate prospect for a good containment tube. These and other problems were addressed by a subgroup of the project team.

4.6.1 Redefined Design Criteria

The design criteria for the drill string were re-examined in the light of the current state of development. The design criteria were as follows:

1. Torque capacity: 3000 in.-lb.
2. Thrust capacity: 6000 lb force
3. Tensile capacity (connectors): 1000 lb force
4. Rotational speed: up to 450 rpm
5. Sustained radius of curvature while rotating: 12 in. minimum
6. Transverse load capacity per segment (normal load): 16,000 lb minimum
7. Surface coefficient of friction with hardened friction pads: 0.16 or more
8. Fatigue life: 10^6 revolutions
9. Internal bore: not less than 0.70 in.
10. Material hardness for wear and strength: not less than R_c 33 to 37
11. Operable in contaminated environment: shale dust and drill cuttings
12. Chip-tube characteristics: abrasion-resistant, smooth, flexible, able to sustain 10 psi vacuum without collapsing
13. Tensile or compressive loading of free end sustainable by rotating drill string without adverse effects: not quantitatively established
14. Torsional load capacity of free end of rotating shaft without induced buckling or excessive wear: not established for all connector designs.

4.6.2 Improved Connector Design

It may be seen from the design criteria that certain areas of the flexible drill-string design needed improvement and that other areas lacked definitive information on performance characteristics. The connector design was of primary concern. A fresh look was taken at the various ideas for connector designs. The following concepts and modifications were considered (see Appendix U):

- a. Dowel pin connector: 1 1/16-in. pin pressed into a drill hole perpendicular to the tooth face, engaging a milled slot in the opposing tooth (Figure U-1)

- b. Roll pin connector: a variation of design "a"
- c. Spring-loaded steel ball detent connector (Figure U-2, another variation of design "a")
- d. Elastomeric connectors similar to those considered in Phase I work
- e. Retainer rings placed in grooves on the outer diameter of the segments
- f. Solid internal retainer ring: swaged into the groove
- g. Cable connector: longitudinal piece of cable with retainer collars silver-soldered at each end (Figure U-3).

Each of the above design concepts is described in detail and evaluated in Appendix U. The design analysis that led to development of the cable connection system for drill segments is also presented in detail. This development led to initial trial tests of a two-cable connector system. For ease of fabrication, some of the earlier-designed "reduced-waist" segments previously used with snap rings were converted to the cable design. A 2-ft length of this drill string was rotated at 474 rpm in the lathe-driven test fixture with the guide tube curved around a 12-in. radius. These drill strings would run for no longer than 10 min before failures occurred in the cables. It became obvious that the connecting cables were not flexible enough to accommodate this much bending and were experiencing accelerated fatigue failure.

To alleviate this condition the drilled holes were taper-reamed to provide clearance so that the cable did not flex but pivoted at the contact surface of the collar. The clearance angle required for any radius of curvature is the same as the half-angle between radial perpendiculars of adjacent segments passing through the center of curvature. The cone angle for the tapered hole is thus the same as the full angle and is approximately 7° for a 12-in. radius of curvature. As may be noted in Figure U-4, this cone angle was increased to 10° to provide ample clearance.

A 5-ft-long drill string was assembled with segments having the new cone design. No cable failures were experienced after running for 300,000 cycles. However, evidence of considerable wear was found, and the segments were tested for hardness. This check disclosed that many of the segments were from a batch that had not been heat-treated and were therefore in a low range of hardness (about $R_c 20$).

FIGURE 77 MULTIPLE RADII TEST CONFIGURATION

4.6.3 Development of Three-Cable Design

Segments for an 8-ft drill string were hardened and assembled in the two-cable configuration. This test was intended to include both multiple-radius configuration and the million-cycle endurance evaluation. This was the first time a full-length cable had been tested, and perhaps more importantly, was the first test of more than 90° of curvature. In addition, one may note in Figure 77 that there is a total of 360° of curvature with reverse 90° bends. With this preface, the test may be described as immediately unstable--the containment tube fasteners were broken and the drill string separated into two pieces.

Initially, it was felt that a sturdier test setup might eliminate the problem. The drill string was fastened to a sheet of reinforced plywood with metal straps placed every 8 in. Continued tests reverified that the drill string was unstable in the multiple-radius configuration. More specifically, the bends created sufficient resistance to rotation to cause a localized buckling at various points along the drill string. This buckling mode was further verified by applying torque to the short 3- and 4-ft lengths of drill string held by hand. In the two-cable connector design, the wire connectors were placed in a spiral fashion--advancing one tooth each segment and producing a recurrent pattern every third segment (3 tooth segments). Unfortunately, this design led to buckling of the segments under even small torsional loads and resultant high rubbing loads and corresponding wear at periodic points within the containment tube. Initially it was hoped that a minimum of cable connectors would be required to reduce the cost of machining and assembly labor. This was found not to be the case.

The immediate solution to the problem was to add the third connector cable and achieve a balanced symmetrical design. Component parts of the three-cable connector drill string are shown in Figure 78. The drill segments are all identically machined, although not identical end to end. The cable slots and holes were milled in a tape-programmed numerically controlled milling machine. Presoldered cables and sleeves are shown as prepared for assembly. The second sleeve is soldered "in situ" with a spacer of proper length between segments. Figure 79 shows a complete drill string with starter section and drill bit. The connectors are not alternated, but placed in line. This results in use of one long slot for the in-line cables instead of alternating short slots. The possible loss of wall strength and bearing surface on the hexagonal corners was of some concern. It was conjectured that the

Figure 78 Component Parts for 3-Cable Design Drill String

Figure 79 Assembled Drill String with Drill Bit and Starter Section

slot might tend to scoop up and fill with cuttings and dust from the sidewalls of the bored hole. Later field tests showed this not to be a problem.

The three-cable connector design was fabricated and run in the laboratory test setup, but with a reduced amount of total curvature. It had been shown that the two-cable design would operate with 180° of curvature although it was still sensitive to the buckling mode if too much torque or end loading were applied. With 180° of curvature the three-cable design appeared to operate without any buckling mode or excessive localized wear. The 180° curvature is roughly equivalent to the planned drill string configuration since the curvature following the first 90° bend immediately below the drivehead has much greater radius of curvature. This area of the containment tube flexes as the boom moves up and down but is a rather flat reverse curve in all positions.

4.6.4 Torsion Test of Cable Connector Segment Design

Introduction of a cable connector slot in the segment geometry made it desirable to recheck the torsional strength of the segment. A variation in heat treatment was also used in these tests to confirm the previous specification of hardness in the R_c 33 to 38 range. It was of particular interest to determine what effect, if any, location of the slot directly below the tooth would have on the torsional strength. Test results showed that hardness controlled the type of failure--from brittle to a ductile shearing type failure.

A typical failure at R_c 35 hardness is shown in Figure 80. The tooth experiences gross plastic deformation, and incipient failure separation occurs at the base of the tooth, while shearing displacement takes place across the top of the slot. In Figure 81 the harder specimen (R_c 40) shows a tendency to fail along a longitudinal line adjacent to the slot. This is characteristic of some of the shouldered specimens tested in previous phases. Specimens hardened to R_c 55 failed in a brittle manner at low loads--about 5,000 in.-lb torque. The R_c 35 hardness gave the best strength (22,800 in.-lb torque) and confirmed the optimum hardness range already specified. The R_c 40 specimen failed at 19,000 in.-lb, which is still more than six times the expected torsional loading encountered during drilling. The average torsional strength of the previously tested three-tooth segments was about 28,000 in.-lb. This would indicate that perhaps some reduction in torsional strength is suffered by using the slotted design. However, the margin of safety is still quite great and in no way precludes satisfactory performance of the cable slot segments.

Figure 80 R_c 35 Torsion Specimen - Failed at 22,850 in-lbs

Figure 81 R_c 40 Torsion Specimen - Failed at 19,000 in-lbs

4.6.5 Field Test of Three-Cable Design

At the time the segment connector design was changed from two cables to three cables, an evaluation of the starter section design was also needed. It was planned to make these simulated field drilling tests at the U.S. Steel facility at Monroeville, PA. Figure 82 shows this test facility shortly after initial construction but before installation of the drill equipment and instrumentation. Details of the design, instrumentation, and operation of this facility may be obtained by contacting the U.S. Steel Research Laboratory at Monroeville, PA. A general description of the facility is that it includes means for simulated roof drilling in actual roof material such as shale, sandstone, limestone, etc. The drill components are instrumented to provide data input for computer readout of drilling horsepower on a real-time basis. At the time tests were conducted, the installation was being tried for the first time, and some damage was sustained to the load cell system measuring torque and thrust. Therefore, the tests were conducted utilizing measurement of operating hydraulic pressures proportional to these torque and thrust loads.

A sample of the abbreviated data recording is shown in Figure 83. This particular test was made with an 18-in. flexible drill length attached to a starter section having 0.5-in.-diameter chip-removal holes. At the left of the chart the line marked "time" shows one-second "tick" marks. The Channel 1 record is of hydraulic oil pressure developed by the drill motor. The initial displacement in this curve is the "no-load" pressure required to rotate the drill. Channel 2 indicates the thrust pressure on the drill, which may be seen to relate quite closely with the torque pressure developed at a particular time. Shortly after start, the operator sensed a change in drill loading and removed the thrust loading. Limestone rock was being drilled, and there seemed to be a tendency for the drill to advance too rapidly and clog the drill with cuttings. The average density of limestone samples taken was about the same as grey shale (150 lb/ft^3). One may note a hump in the displacement curve (channel 3) at the time a high torque was developed, and the operator momentarily backed off on the thrust loading. The average drilling rate was 1.43 ft/min in this test. Sample test results are presented in Table 1.

Additional data were desired to evaluate two different starter section designs during these tests. An airflow orifice meter was placed on the vacuum blower line, and vacuum pressure gages were placed both upstream and downstream of the dust box. Ambient and outlet air temperatures were measured. No significant pressure drop or loss was evident across the dust box. The

Figure 82. U.S. Steel Drill Test Facility

FIGURE 83 TYPICAL DRILL TEST CHART RECORDING

Table 1
Typical Drilling Data, Tests at U.S. Steel Factory

No.	Drill Type	Torque Motor (psi)	Thrust Motor (psi)	Drilling Rate (ft/min)	Air Flow (cfm)	Vacuum (in. Hg)	Remarks
2	Standard	660	660	1.47	---	---	Yellow drill steel, hex bit holder with 7/16-in. holes
3	Flex-Drill	300	300	0.76	---	---	Starter section with 1/2-in. air holes
4	Standard	240	690	1.01	32.8	9.5	Same as test #2
5	Flex-Drill	390	780	0.94	29.83	11	Same as test #3
6a	Standard	450	750	1.25	31.25	10	Same as test #2
6b	Flex-Drill	300	600	0.94	31.33	10	Same as test #3
7	Standard	300	690	0.85	35.81	8	Same as test #2
8	Flex-Drill	510	750	1.43	29.61	10	Same as test #3
10	Standard	360	750	1.20	30.95	10	Same as test #2
11	Flex-Drill	450	930	2.40	26.86	12	Starter section with 7/16-in. holes

limestone drill cuttings did not appear to develop an excessive amount of fines that tend to clog the filters. Measured airflows are listed for the test runs, and there is a general indication that airflow is reduced somewhat at the higher drilling rates. The drilling rates, however, are all in a rather low range, which perhaps may be attributed to the drilling characteristics of limestone as compared to shale. There appears to be no large differences between drilling with a standard drill and the Flex-Drill string. Some variation is no doubt due to the difficulty of obtaining representative average values during each test. Since the procedure involves drilling a pilot hole with a standard bit and then completing the hole by inserting the flexible drill, each test pair should be comparable. However, the average thrust applied to test pairs of standard and Flex-Drill units varied widely in some cases. This is a good indication of the value of data read out in instantaneous values for all parameters on a comparable basis.

The test results indicate that the Flex-Drill has a somewhat restricted chip-return line and develops higher vacuum pressures and lower airflows than the standard bit. Either of the starter sections tested (1/2-in. chip ports or 9/16-in. chip ports) will operate satisfactorily. The smaller holes were used to increase air velocity at the entrance to the chip-return tube. This is compared to 5/8-in. chip ports in standard drill bit adaptors. The three-cable design did not pick up dust nor clog in the cable slots. The entire drill string assembly appeared to stay relatively clean and free of drill cuttings.

5.0.0 FABRICATION OF FLEX-DRILL AND BOLTER SYSTEM

Most of the machine work, fabrication of parts, and assembly of the Flex-Drill have been accomplished in the College of Engineering Shops. The basic support equipment composed of the Galis boom and swing-frame assembly was, of course, purchased directly from FMC Corporation. The extensive alteration and modification required to adapt the drill system to this equipment was done in WSU shops.

All of the hydraulic equipment--valves, motors, cylinders, hoses and fittings--were purchased from distributors of this equipment. Exceptions to the above were a pilot-operated valve and two modified valve spools fabricated by the shops in order to meet operational deadlines. Some special hose fittings were fabricated by the shops, but most of the work in this area involved cutting, fitting and installation of hydraulic lines.

Most of the drivehead was fabricated in the shop, with the exception of such items as gears, bearings and permanent magnets. The gears were cut by Portland Gear Works to design specifications. Parts of the normal load beam assembly were machined by other local shops under purchase order agreement. This work was all done according to specified drawings and let on the basis of low bid through the WSU Purchasing Office.

The design development of the drill string segments led to the fabrication of a considerable number and variety of drill segments. Production of final design segments was done in part with tape-programmed numerically controlled milling apparatus. These operations were confined to the drilling and slotting for the cable attachments. This machine capability was not acquired until after the final three-tooth design was established and the base stock of segment blanks had been machined. A tape program could have been set up to machine the tooth profile and easily meet the machining tolerances required on a production basis. It has been suggested that these segments could perhaps be made by investment casting--achieving a final product of correct hardness and dimensional tolerance adequate for the job and with a minimum of finish grinding.

This aspect of production design was not fully investigated. It is certainly a subject warranting consideration and evaluation if production of this type of drill segment is undertaken.

6.0.0 LABORATORY TEST EVALUATION

6.1 Test Facility

The laboratory facility provides simulation of the machine in the mine geometry. A test frame shown in Figure 84 provides a base for the drill boom and a crosshead that supports a column of concrete blocks. The blocks are secured by two 1-in. tie rods (part -10) that are affixed to a box-like structure that slides longitudinally within guides bolted to I-beams (part -11). This arrangement allows the stack of solid concrete blocks to be repositioned above the center of the drill, making it possible to drill three parallel holes in each stack of 4-x-8-x-16-in. blocks. The crosshead beams are adjustable in height. The simulated roof height in Figure 84 is 40 in. The nominal drilling depth of the block stack in this position is approximately 6 ft. To drill deeper holes, the top of the framework would need to be removed and the stack extended. The overhead bridge crane barely clears the framework, and it was deemed unsafe to operate with this type of potential hazard. Testing at this site was, therefore, limited to holes not more than 6 ft in length. The photograph of the overall test facility (Figure 85) also shows the support equipment for the drill system. The pump and motor unit is a hydraulic supply package. The unit consists of a 40-hp electric motor driving an Oilgear variable-displacement radial-piston pump. The pump flow is adjustable up to approximately 55 gpm. The unit has a 150-gal reservoir, but most of the oil is recirculated directly through the pump and only makeup oil and pilot pressure pump oil circulates through the tank. This arrangement does not provide the cooling capacity normally present in the Galis 200-gallon oil tank system. Excessive heating has not been encountered during laboratory tests thus far, and the system has been satisfactory for laboratory tests. The pump pressure capacity is 1300 psi, which is somewhat less than desired. Thus far, however, the drill torque required has not greatly exceeded 800 or 900 psi.

The drill rotate motor is the only motor supplied directly through the large (2-in.) wide gear in the flow divider. Higher pressures on the 1/2-in. and 1-in. gear sections of the flow divider are readily obtained because of the pressure intensification inherent in certain modes of operation of the

1	-25	LIFT PLATE (B)	9" x 8" x 1/4" HRS	-7	3,5
2	-24	RIB	4" x 18" x 1/2" HRS	-9	3
2	-23	SIDE PLATE	4" x 8" x 1/2" HRS	-1	2,3
2	-22	BOSS	1 1/2" RD x 1/2" HRS	-3	3
1	-21 ASSY	BED ASSY		-0	5
8	-20	ANCHOR BOLT	5/8" UNC x 2 1/2"	-21	5
4	-19	ANCHOR	1/2" x 2 1/2" x 1/2" x 8 1/2"	-21	5
2	-18	SHIM PLATE	1/2" x 3" x 5 1/2" HRS	-21	5
2	-17	SPACER BEAM	[8" x 13.75" x 2 1/2"	-21	5
2	-16	BED	W10 x 40 I x 8 PL	-21	5
4	-15	1" UNC NUT		-0	2
22	-14	1/2" UNC x 1" BOLT		-1	2
4	-13	MID PLATE	14" x 4" x 1/2" HRS	-0	4
8	-12	CHANNEL	6" E x 15"	-11	4
4	-11 ASSY	SPACER	7/8" 6" E	-0	4
2	-10	TIE RODS	1" RD x 78" HRS	-0	3
1	-9	LIFT PLATE (A)	18" x 9" x 1/2" HRS	-0	3
2	-8	BEAM PLATE	36" x 14 1/4" x 1/2" HRS	-1	3
2	-7	CAP PLATE	36" x 2 1/2" x 1/2" HRS	-1	2
2	-6	ANCHOR PLATE	36" x 1" x 1/2" HRS	-1	2
2	-5	TOP PLATE	18" x 12 1/2" x 1/2" HRS	-2	2
2	-4	SIDE PLATE	18" x 14 1/2" x 1/2" HRS	-2	2
1	-3	BASE PLATE	18" x 8" x 1/2" HRS	-2	3
1	-2 ASSY	CRADLE		-1	2
1	-1 ASSY	SUPPORT ASSY		-0	2
1	-0 ASSY	LOAD FRAME		-0	1
QTY	PART NO	DESCRIPTION	MATERIAL	USED ON	SH
PROJ REC	NO	QTY	DATE	REVISION	
WASHINGTON STATE UNIVERSITY COLLEGE OF ENGINEERING RESEARCH MECHANICAL ENGINEERING DESIGN					
LOAD FRAME MODIFICATION FOR DRILLING TEST FACILITY FLEXIBLE DRILL SYSTEM U.S.B.M.					
DATE 12-19-79, DRAW BY RLO					DRAWING NO
SCALE 0.1" = 1.0"					7532-0
PROJ NO 11N-8625-1331					TH L OF 3

Figure 84. Load Frame for Drill Test Facility

Figure 85. Laboratory Test Set Up

flow divider. Relief valves on the thrust motor and other equipment control valves have been individually set to provide protection from overpressure.

The other major support equipment is the blower and vacuum dust-collection system. The Roots-type blower and drive motor are located just behind the hydraulic unit and the swing frame. This equipment is barely visible in the picture (Figure 85), but consists of a positive-displacement blower and 7-1/2-hp electric motor. The blower is somewhat oversized for this application. However, it was available at surplus costs and was therefore adapted for use. At present the motor is being overloaded about 20% to obtain vacuum pressure of 10 in. Hg. This is marginally adequate for drilling tests with the long, small-diameter chip-removal tube used with the flexible drill.

It is thought, however, that better chip removal and drilling characteristics could be obtained with a blower properly sized and capable of 12-14 in. Hg vacuum pressures or higher. The dust collector box was constructed in the shop after an older Galis design. This filter element is a standard cloth unit used with this type of dust box. There are other dust collector units that are no doubt more efficient and may be more suitable for use with this drill system. The unit represents standard practice and leaves room for improvement if there is a need to enhance the operation of the dust collector.

6.2 Operational Evaluation

The drill was first operationally assembled and checked for correct hydraulic operation. Two shallow (1-1/2-ft-deep) holes were drilled to test initial operation. On the first hole a considerable amount of drill cuttings were falling down the drill hole outside the flexible drill. These drill cuttings appeared to be more than the normal amount resulting from starting the drill bit and drilling the first two or three inches. It was found that the vacuum pressure on the dust box was only 5 in. Hg. This was much too low. The vacuum lines were checked and leaky fittings tightened, and the bypass control was adjusted to provide maximum load on the motor. This effort increased the vacuum on the dust-collection system to 10 in. Hg. A second short hole was drilled and almost immediately upon completion the drill rotor "froze" and would not turn. The drill had rapidly advanced, cutting a clear, straight hole without excess "fall-out" down the hole before the drill stopped. This gave evidence that the cuttings could be collected through this long small hose when adequate vacuum pressure was used.

The more pressing problem at the moment was to determine what caused the malfunction in the drillhead. The unit was disassembled, broken pieces of needle bearing were found lodged between the drill rotor and the oil seal plate which separates the lower gear case from the upper rotor area. These pieces were immediately identified as parts of the aircraft bearings that provide normal loading on the back of the friction pads. To replace these bearings requires considerable disassembly of the rotor, so the entire rotor assembly was removed to the shop. The bearing failure was not caused by overload, but rather by misalignment resulting from insufficient side support. Shims restricting sideways movement of these needle bearings had been installed originally. However, it was found that the shims were slightly large in diameter and were being bent over by the corners of the friction pads. Removal of the shims solved one problem but created another. This slight clearance allowed the bearing to shift sideways far enough to catch under a friction pad corner and break out a piece of the end plate. From this point on, as the needles worked out of the bearing, they were promptly broken off in short lengths by the passing friction pads.

New bearings were ordered and installed properly. The end cover plates on the bearings were also machined with a 45° bevel to preclude being caught under the edge of a friction pad. This malfunction resulted in postponement of the test drilling until the second week in September.

Demonstration drill tests were finally rescheduled and run for U.S.B.M. personnel on September 15, 1977. The project Technical Officer, Mr. John Bevan, and Mr. Ken Derby of the Spokane Mining Research Center observed the tests. Mr. Larry Paulsen, also of the SMRC, recorded the drill operation on both still and movie film. Two 5-ft-deep holes were drilled at a feed rate of about 5 ft/min. The first two holes had been inadvertently drilled at a feed rate of 7-1/2 to 8 ft/min, which may have contributed to some overloading of the dust collector system, particularly with the initially low vacuum pressure available. The drill feeds evenly and cuts well. No new operational problems were encountered in the drilling of several more holes. The drill speed is set at 360 rpm and the feed rate currently used is 5 ft/min.

A limited amount of time was available for laboratory test evaluation of the drill prior to project termination. Instrumentation consisted primarily of pressure gages installed in the hydraulic and vacuum dust collection system. Little more could be installed within the time and remaining budget restric-

tions. These pressure measurements do give a rough indication of operational characteristics.

The drilling rate was determined by stop-watch measuring the average time to drill to a 5-ft depth. With the flow control set at mid-range, the drill feed rate was almost exactly 5 ft/min. Flow rate of the supply pump was adjusted to give a drill rotation speed of 360 rpm. At this setting there is little or no oil bypassed through the drill rotation motor flow control. The vacuum blower unit was set to provide the maximum vacuum (10 in. Hg) obtainable without severely overloading the blower drive motor. Several test holes were drilled at this setting..

The change in pressure for the rotate motor and the thrust motor was then noted during drilling operation. The thrust motor pressure required for "no-load" operation was 1000 psi. During drilling at 5 ft/min this pressure increased to 1200 psi and remained quite steady. This corresponds roughly to a thrust motor output of 0.59 hp. Allowing for the power loss in the worm-gear reduction, this translates into a thrusting force of about 2700 lb on the drill.

The rotational torque on the drill may be approximated somewhat more directly. In this case the "no-load" pressure required to rotate the drill rotor was about 450 psi. During drilling this pressure increased to 800 psi. However, the pressure gage indicated a rapid cyclic variation in pressure similar to the torque recording shown in Figure 83. Since the frequency response of the gage mechanism is unknown, the true amplitude can only be guessed. Nevertheless, an approximation can be made of the load and no-load fluid horsepower indicated. The no-load horsepower was about 5.5 hp and the drilling horsepower approximately 9.8 hp. This would indicate an average drilling torque of about 1600 in-lb required to drill the concrete blocks. These are admittedly only approximate values, but they do serve to illustrate the general operating conditions.

The segmented drill string appeared to stay relatively clean. There was some indication of drill cuttings or dust being compacted in the space between the segments, but there was no evidence of an increase or buildup during continued cycling of the drill in and out of the drilled holes. More extensive tests would be needed to evaluate this factor. There is a small accumulation of drill cutting on top of the drivehead occurring primarily during the start of the hole. Some particles do continue to fall from the drill hole, particularly during retraction of the drill, and are thrown out away from the drill

by the "slinging" action of the rotating cover plate. It is possible that an auxiliary vacuum collector shroud of annular form could be used to collect this material. It would, however, not be satisfactory if the vacuum collector restricted air flow up the bore hole, thus impairing operation of the primary dust collector system.

In regard to the operation of the hydraulic control system, most parts work quite well. The pilot system operates as anticipated. There is some lag in control response but it is not deemed excessive nor a detracting feature from effective manual control. Operation of this drill is somewhat different from the standard method in which close coordination between control actuation and physical handling and guiding of the drill steel is required to start the drill. This is a remote operation, and a new set of tactile, sound, and visual signals must be responded to in order to monitor the machine's operation.

The overall performance of the machine, even with this limited testing, is considered to be quite successful. The drill is capable of meeting the design specifications of thrust load, torque, feed rate, and retraction rate (with elevated flow rates). It does not meet the original specification of operation in 30-in. coal. A minimum roof height of 36 in. would be required. It will drill holes within 32 in. of the face (30 in. specified) and within 24 in. of the rib as specified. Operation of the combined flex-drill/bolter system was only partially tested in the laboratory. Since this system was not quite complete nor fully implemented, it remains to be tested.

APPENDIX Q

SPECIFICATIONS AND DRAWINGS OF MAJOR DRILL/BOLTER SYSTEM COMPONENTS

Specifications

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WSU Flex-Drill	Q-2
Bendix Bolt Bender/Inserter	Q-6
Galis Model 3500 Roof Drill	Q-8
WSU Bolt Thruster	Q-11
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WSU Drilling/Bolting System	Q-15

Drawings

<u>Figure</u>	<u>Title</u>	<u>Drawing No.</u>	<u>Page</u>
Q-1	Segmental Flexible Shaft (Three-Cable)	7528-1	Q-4
Q-2	Drivehead-Flexible Drill System	7525-0	Q-5
Q-3	Bendix Bolt Bender/Inserter	7524-0	Q-7
Q-4	Galis Boom Assembly	FMC 3508-D1	Q-9
Q-5	Galis Swing-Frame Assembly	FMC 3508-D2	Q-10
Q-6	Bolt Thruster	7529-0	Q-12
Q-7	Galis Hydraulic System Adapted for Flex-Drill	7527	Q-14
Q-8	WSU Flexible Shaft Drill/Bolter- Inserter System	7526-0	Q-16

WSU Flex-Drill Specifications

Parameter	Specification
1. <u>General Specifications</u>	
a. Vehicle for prototype design	Galis 3500 dual-boom roof drill
b. Mounting	Replaces right boom and swing-frame unit
c. Length	Adds approx. 29 in. to vehicle length
d. Road clearance	7 in. (bumper guard)
e. Controls	Independent hydraulic control valves mounted on each boom
2. <u>Drill and Bolter</u>	
a. Flex-Drill unit	Right boom module (wt: 6300 lb)
b. Bolter unit	Bendix bolt bender/insertor
c. Bolt thruster	150-lb capacity
d. Spindle torque	3500 in.-lb (at working pressure)
e. Drill speed	Variable to 450 rpm
f. Drill feed rate	Variable to 8 ft/min
g. Drill retraction rate	Variable to 37 ft/min
h. Drill thrust reaction	Varies with drilling conditions
i. Boom set against roof	8000 lb (on demand)
j. Boom swing from center-line of vehicle	6 ft 0 in.
k. Boom sump (indexing)	22 in.
l. Bolt torquing	Torque limit valve on drill spindle provided
3. <u>Hydraulic Specifications</u>	
a. Supply pump capacity	45 gpm total @ 2000 psi (tandem 9, 18, 18 gpm)
b. Combined and redivided flows	9, 29, 16 and 7 gpm
c. Hydraulic hose	Parker 305, flame-resistant MESA
d. Hydraulic fittings	No-Skive, pipe and str. thd. adapters
e. Fluid	Union Special Hydraulic No. 315 AW or Equivalent

WSU Flex-Drill Specifications (cont.)

Parameter	Specification
4. <u>Dust-Collection System</u>	
a. Blower	Roots Type AF 33 or equivalent
b. Vacuum capacity	22 CFM @ 5 psi
c. Collector	Galis 3517-D1, cloth filter and dust box
d. Drill configuration	Hollow-bore drill, 1/2-in. ID chip collector tube in Flex-Drill, 1-in. hose to dust box

NOTE: HEAT TREAT TO RC 38-40
MATERIAL - 4140

Figure Q-1 Three-Cable Drill Segment Design

Figure Q-2 Drivehead - Flexible Drill System

Specifications--Bendix Bolt Bender/Inserter

Parameter	Specification
1. Input power	Hydraulic, 14 gpm @ 1300 psi
2. Space required	ENVELOPE, 16 x 15 x 11 in. in height, width and depth
3. Weight	250 lb
4. Bolt size	5/8-in.-diameter, any length
5. Bolt end configuration	With heads, with/without expansion shell attached
6. Bolt washer	With/without roof plates; 8 x 8 in. maximum
7. Controls (required external)	Hydraulic valve, manual 4-way, 3-position: operate, open neutral, and release bolt

Figure Q-3 Bendix Bolt Bender/Insertter

Specifications--Galis Model 3500 Roof Drilling Machine
Dual-Boom, Straight Rotary, U.S. Bureau of Mines Approved

Parameter	Specification
<u>1. Approximate General Specifications</u>	
a. Length	22 ft 11 in. over booms
b. Width	9 ft 0 in. over tires
c. Height	27 in.
d. Weight	22,000 lb
e. Road clearance	7 in.
f. Tram (4-wheel drive)	Variable to 150 ft/min = 1.7 mph
g. Number of wheels	Four (4)
h. Size of tires	9.00 x 10, 16-ply
i. Steering	Orbitrol Steering (2-wheel)
<u>2. Drill</u>	
a. Drill units	Two (2)
b. Spindle torque	225 ft-lb (more on demand)
c. Feed length	61 in. (with no part change)
d. Drill speed	Variable to 475 rpm
e. Feed speed	Variable to 430 in./min = 35.8 ft/min
f. Thrust	8,000 lb (on demand)
g. Boom swing from center-line of unit	6 ft 0 in. each boom (straight-line motion)
h. Boom swing (total)	12 ft 0 in. maximum drilling width
i. Boom sump	10 in.
j. Torquing valve	included
<u>3. Electrical Specifications</u>	
a. Number of motors	Two (2)
b. Speed	1750 rpm
c. Continuous hp rating	30 hp AC
d. Control	Full voltage magnetic
e. Overload relay	Ambient compensated thermal
f. Voltage	440-V AC

Figure Q-4 Galis 3500 Roof Drill - Boom Assembly

Figure Q-5 Galis 3500 Roof Drill - Swing Frame Assembly

Specifications--Bolt Thruster

Parameter	Specification
1. Power	Hydraulic rotary actuator
2. Actuator output	1200 in.-lb torque @ 1500 psi
3. Thrust (at 1500 psi)	150 lb @ 1500 psi
4. Travel	25 in.
5. Controls required	4-way, 3-position manual valve, 2000-psi rating
6. Bearings	All links--porous bronze sleeve sprocket shaft--ball bearings
7. Height adjustment	4 in. vertically on pivot shaft
8. Angular position	Adjustable clamp on pivot shaft
9. Auxiliary controls	Sequence valves initiate operational steps

Figure Q-6 Bolt Thruster

Hydraulic Specifications

Parameter	Specification
1. Supply pumps required	9, 18 and 18 gpm Total: 45 gpm @ 2000 psi
2. Flow divider	Divides combined 18 plus 18 gpm into 29 gpm and 7 gpm streams
3. Divided and recombined flows available for all operating modes	9, 7, 16 and 29 gpm
4. Pilot pressure supply	Aux gear pump, 1 gpm @ 200 psi; alt. bleeder supply--untested
5. Hydraulic hose	Parker 305, flame-resistant MESA
6. Hydraulic fittings	Parker, pipe and str. thd. adaptors; Reusable hose end connectors
7. Hydraulic fluid	Union Special Hydraulic No. 315 AW
8. Filters	Schroeder or equivalent

Figure Q-7 Galis Hydraulic System Adapted for Flex-Drill

Specifications--WSU Drilling/Bolting System

Parameter	Specification
1. Hole depth	8 ft
2. Distance from face	31.5 in.
3. Distance from rib	24 in.
4. Swing from centerline	6 ft
5. Span bolt holes	12 ft (on 4-ft centers)
6. Drill hole diameter	1 3/8 in.
7. Bolt diameter	5/8 in.

Figure Q-8. W.S.U. Flexible Shaft Drill/Bolter-Insertion System

APPENDIX R

FRICTION DRIVE AND NORMAL LOADING SYSTEM

The basic supposition for workability of this design is that the drill segments can be clamped with sufficient force to apply the required thrust load encountered in drilling. Assuming a common relationship, one may say that for a given normal contact force (N) between two surfaces, there is some proportional friction force (F) that will resist or be in opposition to forces tending to cause relative motion between the two surfaces. This constant of proportionality (for any given surface conditions) may be called the coefficient of friction, μ , such that

$$F = \mu N \quad (R-1)$$

The value of the static coefficient of friction is generally less than 1.0 for metal surface pairs without lubrication; and for steels it may vary from 0.15 to 0.40 or more. With a low coefficient of friction the drill segment must be subjected to a transverse loading several times larger than the thrust load developed. It was then of importance to initially determine the transverse load-bearing capacity of the drill segments. Figure R-1 shows the load-deflection curve for a typical drill segment. The proportional limit appears to be at about 30,000 lb load, which allows use of normal loads up to 15,000 lb with a safety factor of 2.0, based on the elastic yield point.

The maximum thrust required for the drill was previously determined to be 5,000 lb. Since it was anticipated that torque would be applied to the drill segments as they passed up through the drivehead, some torque collar or socket would be required. This method of applying torque involves forcing the drill string up through a hex sleeve while this sleeve is applying torsional loading. Friction forces in torquing sleeve oppose the drill movement and add to the thrust required for drilling. By experiment this added thrust force was determined to be in the order of 1000 lb. This design thrust load was then increased to 6000 lb.

Figure R-1 Transverse Load-Deflection Curve for Typical Flexible Shaft Segment

Figure R-2 shows a force diagram of the drill segments and friction pad system. The normal load beams apply a force F_N to a pair of rollers on opposite sides of the drill string. Each roller then applies $N/2$ lb force to the back of the friction pad over which it is rolling. The net effect is the application of the normal load N to both sides of the segment and the development of friction forces F_{f1} and F_{f2} .

Summing vertical forces,

$$F_D + F_{t1} + F_{t2} = F_{f1} + F_{f2} \quad (R-2)$$

From Eq. (R-1),

$$F_{f1} = F_{f2} = \mu F_N \quad (R-3)$$

Substituting this relation for F_{f1} and F_{f2} into Eq. (R-2),

$$F_D + F_{t1} + F_{t2} = 2\mu F_N \quad (R-4)$$

If total design force is 6,000 lb ($F_D + F_{t1} + F_{t2}$), then

$$F_N = \frac{3000}{\mu} \quad (R-5)$$

This leaves only evaluation of the coefficient of friction (μ) to establish the required normal loading for the system. The friction tests showed that with reasonable precautions, a coefficient of friction greater than 0.20 was nearly always attainable. This corresponds to a normal load requirement of 15,000 lb. A design load of 16,000 lb was selected to allow for friction in the rollers and loading system not accounted for in the simplified analysis given above.

The friction pads are loaded by two normal load arms pivoted such that they form a "nut-cracker" with cantilever ratio of approximately 2 to 1. The ends of the cantilever ends are loaded by two torsion-spring-activated ball screw units acting in tandem. The manufacturers indicate an efficiency of 90% can be expected for the ball screws. The amount of effort (torque) required to produce or move a given load is as follows:

$$T_D = \frac{P \times \ell}{2\pi e} = 0.177 P \times \ell \text{ in.-lb} \quad (R-6)$$

where

T_D = direct torque (in.-lb)

P = load (lb)

ℓ = screw lead (in./turn) (1/2 in/turn)

e = ball screw efficiency (90%)

Figure R-2 Drill Thrust and Friction Force Relations

For this particular design it was found that a normal load of 16,000 lb should be adequate under most conditions of friction variables. The lever ratio on the load beam is 2.126 to 1. Therefore, a 7,523-lb force at the end of the load arm produces the required normal force. Each of the two ball screw units must then produce half this load, or 3,761 lb force. From Eq. (R-6), each torsion spring should apply 333 in.-lb torque to achieve the desired loading.

To provide a check on the torsion spring loading, the drill head rotor unit was placed in a testing machine and a measured load was applied to the ends of the load arms. The ball screw unit key pins were disengaged so that the torsion springs were not loaded in any way. Deflection of the load arms was measured with a dial gage attached between arms as shown in Fig. R-3. The load deflection relation is plotted in Figure R-4, showing the total deflection (both arms) present when the theoretical design load is attained. Torque wrenches are also employed during the preloading of the torsion springs as an additional verification of proper adjustment of the normal load system. The final configuration of friction pads used with this loading system is shown in Figure R-5.

Figure R-3 Load Arm Deflection Measurement

Figure R-4 Load Arm Deflection with Direct Loading in Testing Machine

Figure R-5 Friction Pad

APPENDIX S

CROSS-SECTION DRAWINGS OF MAJOR ELEMENTS OF THE DRIVEHEAD ASSEMBLY

<u>Figure</u>	<u>Drawing No.</u>	<u>Sheet No.</u>	<u>Section</u>	<u>Page</u>
S-1	7525-0	2 of 7	Drivehead Assembly	S2
S-2	7525-0	3 of 7	Section H-H	S3
S-3	7525-0	4 of 7	Sections B-B and C-C	S4
S-4	7525-0	5 of 7	Sections D-D and E-E	S5
S-5	7525-0	6 of 7	Sections A-A and I-I	S6
S-6	7525-0	7 of 7	Sections F-F and G-G	S7
Drivehead Parts Identification List				S8

Figure S-1 Drivehead Assembly

Figure S-2 Drivehead Section H-H

Figure S-3 Drivehead Section B-B and C-C

Figure S-4 Drivehead Sections D-D and E-E

Figure S-5 Drivehead Sections A-A and I-I

Figure S-6 Drivehead Sections F-F and G-G

Parts List--Drivehead

Part No.	Foreign Part No.	Description
7525-0		Drivehead assy
7525-1		Helical ring gear, DP 6 in., 87T, RH
7525-2		Helical idler, DP 6 in., 18T, RH
7525-3		Helical ring gear, DP 6 in., 40T, LH
7525-4		Helical pinion, DP 6 in., 87T, RH
7525-5		Helical drill idler, DP 6 in., 57T, LH
7525-6		Helical gear, CP 6 in., 57 teeth, RH
7525-7		Helical pinion, DP 6 in., 27T, RH
7525-8		Helical pinion, DP 6 in., 27T, RH
7525-9		Helical gear, DP 6 in., 12T, LH
7525-10	Browning W6A-2G	Worm, DP 6 in., RH (Steel)
7525-11	Browning BWG 636-2	Worm gear, DP 6 in., 36T, RH, Bronze
7525-12		Helical pinion, DP 6 in., 15T, RH
7525-13		Helical idler, DP 6 in., 24T, LH
7525-14		Helical gear, DP 6 in., 24T, RH
7525-15		Helical gear, DP 6 in., 24T, LH
7525-16a	Chain Belt 50B 12H	Sprocket, 5/8 dia., 12T, 1.25 bore
7525-16b	Chain Belt 50B 12H	Sprocket, 5/8 dia., 12T, 1-in. bore
7525-17		Helical idler, DP 6 in., 18T, RH
7525-18		Helical idler, DP 6 in., 30T, LH
7525-19		Helical ring gear, DP 6 in., 48T, LH
7525-20	Browning YSMS7F21 RHX1	Spiral miter gear, DP 7 in., 21T, RH
7525-21	Browning YSMS7F21 LH	Spiral miter gear, DP 7 in., 21T, LH
7525-22	Browning YSMS7F21 RHX1	Spiral miter gear, DP 7 in., 21T, RH
7525-23	Browning YSMS7F21 LHX1	Spiral miter gear, DP 7 in., 21T, LH
7525-24		Gear mounting plate w/flex shaft guide tube rotor assy
7525-25		Gear mounting plate/ -3 and -4
7525-26		Thrust-bearing ring
7525-27		Bearing housing
7525-28		Bearing spacer ring
7525-29		Spiral miter gear shaft
7525-30 assy		Thrust motor mount/Brake cover case
7525-31		Brake case end plate
7525-32		Brake case side plate
7525-33		Brake cover plate and motor mount
7525-34 assy		Differential gear cover
7525-35		Differential case bottom
7525-36		Tubing for differential casing
7525-37		Bolting flange/differential casing
7525-38		Rotate motor mounting plate
7525-39		Lower bearing plate assy
7525-40		Intermediate bearing plate
7525-41		Top bearing and cover plate
7525-42		Idler bearing plate, N 18
7525-43		Idler bearing plate spacer block
7525-44a		Spacer/used between plates -39 and -40
7525-44b		Spacer/used between plates -39 and -40
7525-44c		Spacer/used between plates -39 and -40

Parts List--Drivehead (Cont.)

Part No.	Foreign Part No.	Description
7525-45		Drivehead gearcase assy
7525-46		Lower plate
7525-47		Step plate of gearcase
7525-48a		Side plate
7525-48b		Bolting flange
7525-49a		End plate
7525-49b		Bolting flange
7525-50		Lower bearing plate
7525-51		Gear housing segment
7525-52		Bolting flange
7525-53		Trunnion shaft
7525-54		Caliper brake spacer and mounting block
7525-55a		Spacer blocks
7525-55b		Spacer blocks
7525-56		Normal load beam pin
7525-57		Retainer flange
7525-58a		Drivehead cover plate
7525-58b		Bearing and torque collar
7525-59		Drivehead housing assy
7525-60		Vacant (revised)
7525-61		Cover for gear -17
7525-62		Cylindrical housing
7525-63		Flex-tube mounting plate
7525-64		Vacant (revised)
7525-65		Friction pad
7525-66		Roller clevis
7525-67		Normal load beam assy
7525-68		Load beam ears
7525-69		Load beam
7525-70		Tee pin
7525-71		Torsion adjust ring
7525-72		Torsion lock ring
7525-73		Spring guide sleeve
7525-74		Pin carrier ring
7525-75		Rod end, female
7525-76		Rod end, male
7525-77		Spring bushing
7525-78		Torsion spring
7525-79		Worm gearcase assy/with plate -98
7525-80		Magnet assy
7525-81		Magnet base (lower)
7525-82		Upper pole piece
7525-83		Lower pole piece
7525-84		Lower segment guide
7525-85		Insert segment guide
7525-86		Spacer segment guide
7525-87		Torquing segment guide
7525-88		Torque drive plate
7525-89		Hex torque sleeve

Parts List--Drivehead (Cont.)

Part No.	Foreign Part No.	Description
7525-90		Drill shaft cover
7525-91		Friction drive bearing plate assy
7525-92		Gear -12 housing
7525-93a		Friction drive bearing plate
7525-93b		Plate for friction drive
7525-94a		Gear -14 housing
7525-94b		Gear -15 housing
7525-94c		Gear -13 housing
7525-95		Normal load arm support
7525-96a		Plate spacers, gear -14 housing
7525-96b		Plate spacers, gear -15 housing
7525-96c		Plate spacers, gear -16 housing
7525-97		Worm gear case assy
7525-98		Mounting plate - worm gear
7525-99		Bolting plate and worm gear housing
7525-100		Spacer (bearing race for worm gear)
7525-101		Bearing housing worm gear
7525-102		Worm bearing and gear housing
7525-103b		Split bolting flange
7525-104		Partial cover plate/worm gear
7525-105		Top cap for worm case
7525-106		Seal expander ring
7525-107		Eccentric sleeve for sprocket adjust
7525-108		Bearing retainer, cover for gear -9
7525-109		Spacer used on shaft -29
7525-110c		Spacer used on -123 idler shaft
7525-111		Washer (thrust drive shaft) gear -8
7525-112a		Sprocket spacer -121
7525-112b		Sprocket spacer - 119 and -120
7525-112c		Sprocket spacer -119 worm side
7525-113a		Sprocket spacer sleeve
7525-113b		Sprocket spacer sleeve
7525-114		Worm spacer, worm to double bearing
7525-115		Flanged bushing -126 spline shaft
7525-116		Worm gear shaft cover plate
7525-117		Oil keeper for gear -1
7525-118		Vacant
7525-119		Upper gear shaft
7525-120		Upper sprocket shaft
7525-121		Lower sprocket shaft
7525-122		Adjuster shaft
7525-123		Idler shaft
7525-124		Dual bearing shaft
7525-125		N -5 idler shaft
7525-126		Differential shaft
7525-127		Washer (N -5 gear) top
7525-128		Washer (used on miter gear -22)
7525-129		Washer (N -5 gear) bottom
7525-130		Washer/ -152, thrust drive shaft

Parts List--Drivehead (Cont.)

Part No.	Foreign Part No.	Description
7525-131		Miter gear bearing keeper
7525-132		Outer bearing keeper
7525-133		Spacer/thrust drive shaft
7525-133a		Spacer/thrust drive shaft
7525-134		Bearing housing/bottom of differential cage
7525-135		Washer-spacer for -126
7525-136		Spacer / n -2, shaft -124
7525-137		Spacer / N -17, shaft -124
7525-138		Oil seal ring/pressed into plate -98
7525-139		Mounting bolts for -38 and rotate motor
7525-140		Dual bearing cap /N -2, -17
7525-141		Dual bearing spacer/N -2, 17, shaft -124
7525-142		Upper N -18 spacer
7525-142a		Upper N -18 spacer
7525-143		Lower N -18 spacer
7525-143a		Lower N -18 spacer
7525-144		Gear spacer on shaft -119 for gear -14, -15
7525-145		Bolt on bearing retainer ring, -119
7525-146		Oil pump mounting plate
7575-147		Plug/pressed into gear -7
7525-148		Bearing spacer/torsion spring assy
7525-149		Spacer-washer/-9 gear shaft
7525-150	Stone HDI No. P43	Oil pump/mounted to gear -7
7525-151		Spacer/bearing to worm
7525-152		Differential drive shaft
7525-153		Caliper brake shaft
7525-154 assy		Cage assy
7525-155a		Cage top
7525-155b		Cage bottom
7525-156		Cage side
7525-157		Rotor plate, gear mounting plate
7525-158		Rotor - shaft.
7525-200a	MRC 206 SZZ	Ball bearing, MRC, double-seal
7525-200	MRC 305 SZZ	Ball bearing, MRC, double-seal
7525-201	MRC 306 SZZ	Ball bearing, MRC, double-seal
7525-202	Hoover 99303	Ball bearing, Hoover, double-seal
7525-203	N D 499605	Ball bearing, ND, double-seal
7525-204a	Torr M-16121	Needle bearing (one end closed)
7525-204	Torr B-1612	Needle bearing
7525-205	Torr IR -1312	Inner race
7525-206	Torr B -2012	Needle bearing
7525-207	Torr IR -1616	Inner race
7525-208	Torr 10NVK1628YZP	Sealed bearing
7525-209a	MRC 109 KSZZ	Ball bearing, MRC, double-seal
7525-209	MRC 209 SZZ	Ball bearing, MRC, double-seal
7525-210	MRC 210 SZZ	Ball bearing, MRC, double-seal
7525-211	Torr 10NBL2426YJ	Cam bearing
7525-212	Kaydon KD90AR	Ball bearing, Kaydon

Parts List--Drivehead (Cont.)

Part No.	Foreign Part No.	Description
7525-213	MRC 7220	Ball bearing, MRC
7525-214	MRC 7113KR	Ball bearing, MRC
7525-215	Hoover 77-R-12	Ball bearing, Hoover
7525-216	SKF 62082 RSJ	Ball bearing, SKF
7525-217	MRC 5210KF	Ball bearing, MRC
7525-218	MRC XLS-4 1/2	Ball bearing, MRC
7525-219	ND 455606	Ball bearing, New Departure
7525-220	SKF 6204-2RSS	Ball bearing, SKF, double-seal
7525-221	Torr IR-1612	Inner race
7525-222	Torr M20121	Needle bearing
7525-223	MRC 210R	Ball bearing
7525-224	Torr B1212	Needle bearing
7525-225	Torr B-1616	Needle bearing
7525-226	Torr IR-1216	Inner race
7525-227	MRC 5211KF	Double ball bearing
7525-228	Fafnir 9107K	Single ball bearing
7525-229	Waldes (5100-98), Spir-O-Lox (RS-98)	Retainer ring
7525-230	Spir-O-Lox RR-267	Retainer ring
7525-231	C/R 14960	Oil seal
7525-232	Oilite SSC-1102	Super oilite stock
7525-233	Spir-O-Lox RRN-185	Retainer ring
7525-234	Oilite TT 1701	Thrust washer
7525-235	Oilite TT 1303-2	Thrust washer
7525-236	Waldes 5100-118	Retainer ring
7525-237	Waldes 5100-100	Retainer ring
7525-238	Spir-O-Lox RR-244	Retainer ring
7525-239	Spir-O-Lox RS-900	Retainer ring
7525-240	Spir-O-Lox RR-393	Retainer ring
7525-241	Permag Sierra Corp 90S33A	Co-Sm magnets
7525-242	National 622712	"O" rings
7525-243	C/R 3869	Oil seal
7525-244a	C/R 27269	Oil seal
7525-244b	C/R 27278	Oil seal
7525-245	Ringfeder Corp RfN 7012-IN	Locking collar
7525-246		Nylon pad
7525-247		Torsion locking pin
7525-248		Torsion coupling pin
7525-249	Rex No. 50	Chain, roller, W/D3 attachments
7525-250	Tolomatic HP-10-D	Disk, caliper brake
7525-251	Tolomatic HP-10-D	Caliper assy
7525-252	Spir-O-Lox RR 352	Retainer ring
7525-253	Warner Elec R-0502	Ball nut assy

APPENDIX T

SUMMARY REPORTS OF INVENTIONS AND SUBCONTRACTS

U.S. Department of the Interior

Form DI-1216

(December 1973)

T-2
**UNITED STATES
DEPARTMENT OF THE INTERIOR**

Date December 1977

SUMMARY REPORT OF INVENTIONS AND SUBCONTRACTS

The following report must be submitted in *triplicate* as part of the interim or final report as provided for by the REPORTS and/or PATENT ARTICLE in the grant or contract.

CONTRACT DATA	
A. NAME OF CONTRACTOR Washington State University	B. ADDRESS Pullman, Washington, 99163
C. IDENTIFICATION OF CONTRACT OR GRANT <i>(Title, number, date, and Department agency involved)</i> "Flexible Roof Drill for Low Coal", U.S.B.M. Contract H0242027. Project initiated in June 15, 1974, under Coal Mine Health and Safety Program, administered by Spokane Mining Research Center.	
D. DESCRIPTION OF WORK Conceptual development, prototype design, construction and testing of a flexible segmented shaft roof drill and bolting system.	
E. CONTRACTOR'S PRINCIPAL INVESTIGATOR Norman H. Shoup	F. GOVERNMENT TECHNICAL PROJECT MONITOR R. J. Simonich, Contracting Officer John Bevan, Project Technical Officer

(Check appropriate boxes)

1. Type of Report:

☐ Interim { From June 15, 19 74
 To October 7, 19 77
☒ Final.

2. Interim Report Data:

A. Invention made ☒, not made ☐, during interval of (1).

B. If invention(s) made, provide the following information:

☐ Previously fully disclosed in Invention Disclosures. Give dates submitted, and Contractor's docket numbers.

.....

☐ Invention Disclosures attached herewith. Give Contractor's docket numbers.

.....

3. Final Report Data:

A. Invention(s) previously reported—

Date Submitted	Contractor's Docket Number
January 14, 1975	Case No. 2468
October 1, 1974	Case No. 2469

B. Invention(s) reported herewith—

WSU (Ref. 1331-3)
WSU (Ref. 1331-4)

C. Others (explain)—

.....

.....

.....

D. No inventions were made under the grant or contract. ☐

4. Patent application(s) filed and contemplated to be filed by Contractor under the terms of the grant or contract:

Application Serial No.	n/a				
Date of filing					
Contractor's Docket No.					

5. Subcontracts containing patent rights article:

☒ None. ☐ Listed below are subcontractors.

Name of Subcontractor	Address	Subcontract Number	Date Executed
(1)			
(2)			
(3)			
(4)			

6. Attach a copy of the patent rights article employed in each subcontract set forth in 5.

n/a.

7. Contractor or Grantee certification.

I certify that this Summary Report of Inventions and Subcontracts including any attachments is correct to the best of my knowledge and belief.

Date December 19, 1977

Signature

Norman K. Shoup

Title

Principal Investigator

APPENDIX U

DEVELOPMENT OF CABLE CONNECTOR DESIGN

Concepts for Improving Segment Connectors

- (1) Dowel pins would be installed in holes drilled perpendicularly through the tooth face. These pins would slide in slots milled in the opposing tooth face, Figure U-1. The milled slot length would limit relative motion to the minimum separation between segments allowing the desired curvature. With a pin in each tooth face, three pins would have to shear off during failure under uniform tensile loading. For example, if 1/16-in. pins were used, the load required to shear three SAE 1050 steel pins would be approximately 664 lb. This is somewhat less than the desired capacity of 1000 lb given in the design criteria. Larger-diameter pins could be used at the expense of tooth contact area and tooth strength.
- (2) A variation of design (1) would be to use roll pins instead of dowel pins. Since the roll pin is designed to be driven into a slightly undersized hole, there is much less precision required in the drilled hole tolerances. The roll pin has the disadvantage of having a lower shear strength. This can be overcome partially by insertion of piano spring wire into the roll pin. Properly sized, the piano wire does not restrict the spring action of the roll pin and considerably increases the shear strength.
- (3) A third modification of the pin-and-groove connector is shown in Figure U-2. Blind holes are drilled in the tooth face and a spring and steel ball inserted. Shims would be used to depress the balls during assembly and then removed, allowing the balls to ride in the milled slots. Disassembly is not possible with the blind holes, but if desired, the springs could be retained by setscrews. The effect of drilling dust buildup in the spring area is not predicted beyond possible restriction of movement in the segment joint.

Figure U-1 Dowel Pin Segment Connector

Figure U-2 Ball - Spring Segment Connector

- (4) Elastomeric connectors were reconsidered to determine if some improvement in bonding techniques could be found... Beyond what was reported in Phase I, no new technology was found that would greatly increase the feasibility of this concept.
- (5) Stronger snap rings were considered. The spiral snap ring placed in grooves on the external diameter of the segments would be larger and stronger because the shear loading would be applied at a greater radius in the failure mode due to bending. However, the restriction of air passage along the flat sides of the segments would be severe. Sufficient area between drill segments and the drill hole is required to allow an adequate volume of air intake for chip removal at the cutting bit.
- (6) Modification of the internal snap ring currently being used was considered as a viable solution. This concept involved use of a solid ring that would be "rolled" or expanded into the snap ring groove. This would involve a special swaging tool but would give a firmly positioned ring that could not pop out or work its way out as the overlapped spiral retainer rings were found to do at times. The solid ring could be increased in thickness to give adequate shear strength at the tooth interface. The primary difficulty encountered in increasing retainer ring thickness is that a corresponding increase in groove width must also be made to retain sufficient flexibility to meet the minimum drill string curvature radius. The resulting wide grooves considerably reduce the tooth profile and load-bearing capacity.
- (7) Figure U-3 shows a connector concept employing a cable with collars attached at either end. The cable passes through longitudinal holes drilled in each segment. This idea worked quite well with the original segments having the reduced-diameter "waists." The cable collars restrain separation of the segments when they came into contact with the machined shoulder just behind the teeth. In the case of the smooth-sided segment, a pocket or groove was milled in the side of the segment as shown in Figure U-4 to two cable designs.

Of all the above concepts, the cable connector seemed to have the greatest potential for a high-strength, economical, readily manufactured segment

Figure U-3 Cable Connector for Flexible Shaft Segments

Figure U-4 Flexible Drill Segments - Two Cable Design

connector. This concept involved considerable change in segment geometry and the need for partial recheck of drill segment stress analysis. The first priority was evaluation of the cable connector in regard to methods of fabrication and strength of the cable and collar attachment.

Testing and Development of Cable Connector

The first connectors tested were made from piano wire since this material has an indicated ultimate strength of over 300,000 psi, particularly in the smaller wire sizes. The corresponding ultimate tensile load on 14-gage wire (0.062 dia) should be in the order of 930 lb. Piano wire of this size was tested with various end connectors. The methods of fastening the sleeve to the wire end were (1) swaging, (2) plasma arc welding, (3) silver solder, and (4) epoxy-resin adhesive. Table U-1 shows the results of tensile tests of these 14-gage connectors.

Table U-1
Tensile Test of 14-Gage Piano Wire Connectors

<u>Method of Fastening Ends</u>	<u>Load at Failure (lb)</u>
Swage	350
Plasma arc	550
Plasma arc	750
Silver solder	800
Silver solder	850

The swaging was performed using a rotating hammer-type swaging machine to secure 1/4-in. OD by 0.30-in. long collars. In each case the plasma arc and silver solder tensile specimens failed in the piano wire. The plasma arc process appears to temper the wire more than silver-soldering as evidenced by the lower failure loads. An enlarged photograph of a fractured plasma arc-wire end is shown in Figure U-5. The test results on the piano wire were somewhat disappointing, even though the silver-solder method achieved almost 90% of the theoretical strength expected.

It was hoped that more reliability in the strength of end connectors could be achieved. For this reason, various end connection methods were evaluated using a flexible stainless steel cable strand. Various methods of

Figure U-5 Fractured Plasma Arc Wire End

Figure U-6 Cable with Collar Pre-Soldered on One End

end-fastening collars to two different sizes of cable were tried. The tensile test results on the stainless steel cable are listed in Table U-2. Figure U-6 illustrates a typical cable connector.

Table U-2
Tensile Tests on Stainless Steel Cable
With Various End Connectors

Cable Diameter (in.)	Fastening Method	Failure Load (lb)
3/32	Swaging	330
3/32	Epoxy	425
3/32	Soft solder	200
3/32	Silver solder	1,350
3/32	Silver solder	1,330
3/32	Silver solder	1,335
3/32	Silver solder	1,330
1/8	Swaging	325
1/8	Plasma arc	350
1/8	Plasma arc	500
1/8	Silver solder	700
1/8	Silver solder	700
1/8	Silver solder	970
1/8	Silver solder	1,050

These results are generally comparable to the piano wire tests as related to end-connecting method used. However, the silver-soldered specimens in the 3/32-in. cable were considerably higher (1,350 lb) and more consistent than any other previous results. By comparison the silver-soldered ends on 1/8-in. cable varied as much as 50% from the low value to the highest value. This indicated a lack of control of the procedure and technique of silver-soldering in the shop. Examination of some specimens showed inconsistency in "tinning" the stainless steel and obtaining a good flow of solder into the joint. Additional tests verified improvement in soldering technique, and the collar geometry was optimized to develop full shear strength in the cable joint,

thus producing failure in the cable itself. Figure U-7 illustrates this type of failure in a 7 x 7 cable with plasma arc welded ends. Figure U-8 shows similar specimen failure in stainless steel 1 x 7 cables with silver-soldered ends.

Figure U-7 Fractured 7 X 7 Cables with Plasma Arc Welded Ends

Figure U-8 Fractured 1 X 7 Stainless Steel Cables with Silver-Soldered Ends