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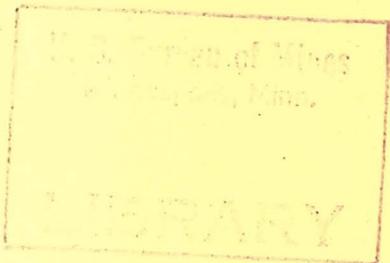
DEVELOPMENT OF HIGH-PRESSURE
WATER JET EQUIPMENT FOR UNDERGROUND
APPLICATION OF EXCAVATING ENERGY MINERALS

Prepared for

UNITED STATES DEPARTMENT OF THE INTERIOR
BUREAU OF MINES

by

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FINAL REPORT

on

Contract No. JO155133

July 15, 1976



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| 16. Abstract <p>A high-pressure intensifier system was designed, constructed and field tested as part of a program to determine the feasibility of applying high-pressure water jets on conventional mechanical rock excavation equipment to increase advance rates. Also developed were a high-volume swivel, capable of transferring the high-pressure water to a rotating cutterhead and a manifold system for locating and protecting the jets on the cutterhead.</p> <p>Field tests were conducted in a granite quarry near Skykomish, WA, approximately 90 miles east of Seattle. The concept of improving excavation rates by using water jets to assist mechanical cutting was proven to be successful, with an average increase in excavation rate of 50 to 60%. For some tests, the excavation rate was more than doubled.</p> <p>Machine torque requirements were reduced approximately 25% when the water jets were used. It was suggested this was due to reduced cutterhead friction in the presence of the water jets.</p> <p>A comprehensive economic analysis was undertaken as part of a concurrent study to determine the degree of cost benefits attainable from increased advance rates. Very significant savings were seen to be possible. It was concluded that the basic concept of combining a water jet capability with a mechanical excavating machine is definitely possible.</p> | | | |
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- FOREWORD -

This report was prepared by the Colorado School of Mines, Excavation Engineering and Earth Mechanics Institute, Golden, Colorado under USBM Contract Number JO155133. The contract was initiated under the Advanced Coal Mining Technology Program. It was administered under the technical direction of Twin Cities Mining Research Center with Mr. John Chester acting as the Technical Project Officer. Mr. William Meyers 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 4, 1975 to July 4, 1976. This report was submitted by the authors on July 15, 1976.

ABSTRACT

A high-pressure intensifier system was designed, constructed and field tested as part of a program to determine the feasibility of applying high-pressure water jets on conventional mechanical rock excavation equipment to increase advance rates. Also developed were a high-volume swivel, capable of transferring the high-pressure water to a rotating cutterhead and a manifold system for locating and protecting the jets on the cutterhead.

Field tests were conducted in a granite quarry near Skykomish, Washington, approximately 90 miles east of Seattle. The concept of improving excavation rates by using water jets to assist mechanical cutting was proven to be successful, with an average increase in excavation rate of 50 to 60 percent. For some tests, the excavation rate was more than doubled.

Three different jet patterns were investigated, with only the configuration of jets between cutters being successful.

Machine torque requirements were reduced approximately 25 percent when the water jets were used. It was suggested this was due to reduced cutterhead friction in the presence of the water jets.

A comprehensive economic analysis was undertaken as part of a concurrent study to determine the degree of cost benefits attainable from increased advance rates. Very

significant savings were seen to be possible, depending primarily on the percentage improvement assumed. A break-even increase in advance rate of 10 percent was found, with savings increasing rapidly for greater improvements in advance rate. Using the 50 percent increase attained in the field tests, cost savings of 14 and 24 percent were projected for 10 foot and 20 foot diameter tunnels respectively. Also, these benefits were seen to be relatively insensitive to water jet equipment costs.

The problems with the equipment are described, and although numerous, they were not unexpected for a prototype development. Most of the problems were solved or could be relatively easily solved and it is, therefore, concluded that the basic concept of combining a water jet capability with a mechanical tunnel boring machine is definitely possible.

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1.0 INTRODUCTION

As our demand for energy and minerals resources grows and these resources become less surficial and/or lower grade, more effective rock excavation techniques are required to provide the resources while maintaining reasonable cost. A current limitation to improving excavation rates of mechanical equipment is simply a structural one, the amount of energy deliverable to the face being limited by the size and metallurgy of the cutting tools and supporting structure. High-pressure water jets offer a potential means of delivering significant additional power to the cutting face and thereby increasing excavation rates. It was the purpose of the work described in this final report to design, construct, and test a high-pressure generation and delivery system for use on rock excavation equipment.

The primary objective of the project was the development and testing of a high-pressure, high-flow intensifier, suitable for use in underground excavation applications. Due to the hostile environment and limited space associated with most underground excavations, the intensifier had to be designed as compact and rugged as possible.

Also to be developed was a high-pressure swivel capable of transmitting the flow from the intensifiers to the rotating head(s) usually associated with mechanical rock excavation equipment.

The investigation essentially consisted of two parts,

initial development of the intensifier and swivel, and the field testing of the equipment in conjunction with a mechanical tunnel boring machine. This report summarizes the work and results of that investigation.

2.0 EQUIPMENT DEVELOPMENT

2.1 Intensifier

To provide the high pressure water for slotting of the rock, Flow Research Inc. designed and fabricated a 1,000 hp, 60,000 psi pumping system for this program. The intensifier is the heart of the pumping system. Its construction is illustrated in Figure 1a, 1b and 1c. Functionally, the intensifier is an oil-driven reciprocating plunger pump. A large piston in the center is cycled back and forth by oil. A small piston on each end pumps water. The area ratio between the large piston and the small piston is $11 \frac{1}{4}$ to 1 so that 5,000 psi oil pressure results in a 56,000 psi water pressure. As the piston is stroking, one end is on the intake stroke and the other on the exhaust stroke. Thus, the pump delivers water on both strokes.

The pumping action requires an inlet and outlet check valve on each end of the intensifier. Usually, these check valves are joined by a tee-connection and fed into the cylinder. This construction results in high stress at the tee-joint. The high stress cycles with each stroke of the pump causing a fatigue failure. In the design shown in Figure 1a, this tee-connection is avoided through the use of coaxial check valves.

Sealing of the plunger is accomplished with a four piece elastomer type seal for positive sealing. The main water seal is accomplished with an L-shaped filled teflon ring (see inset Figure 1c). This ring is energized to squeeze against the plunger with an O-ring. The O-ring and L-shaped teflon piece provide the seal against leakage of water. Behind this seal

Intensifier Principle of Operation

When the piston reaches the end of its travel, the switching circuit reverses the oil flow direction. The dual small cylinders allow the intensifier to pump on both leftward and rightward strokes.

FIGURE 1 A

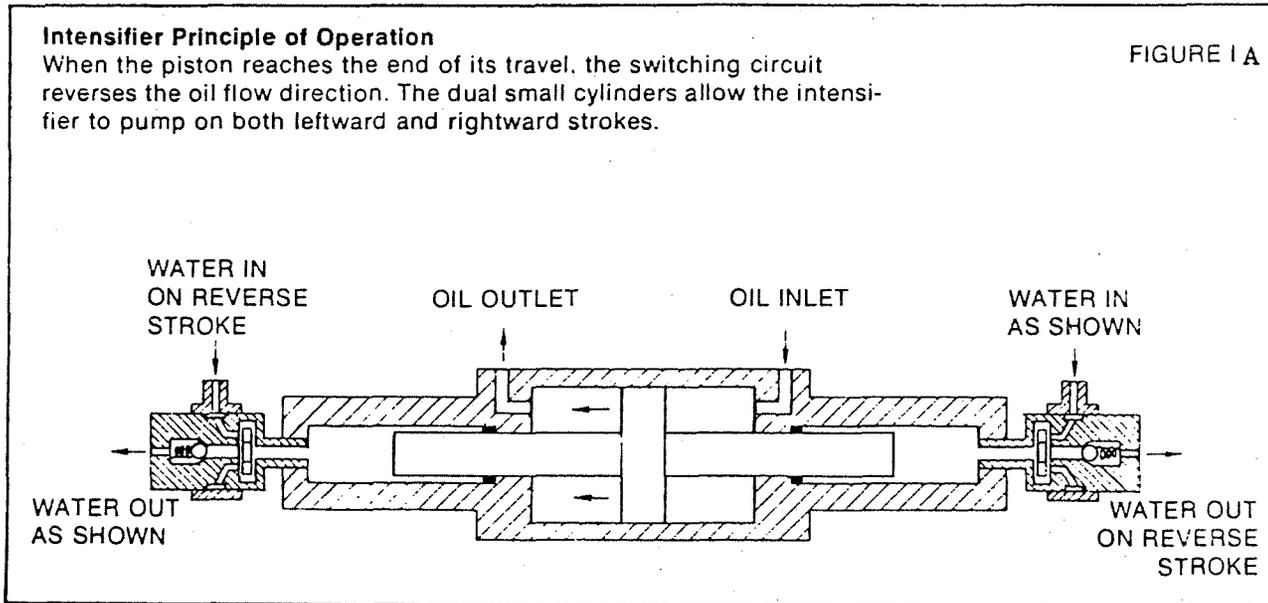


Figure 1 - Intensifier Principle of Operation

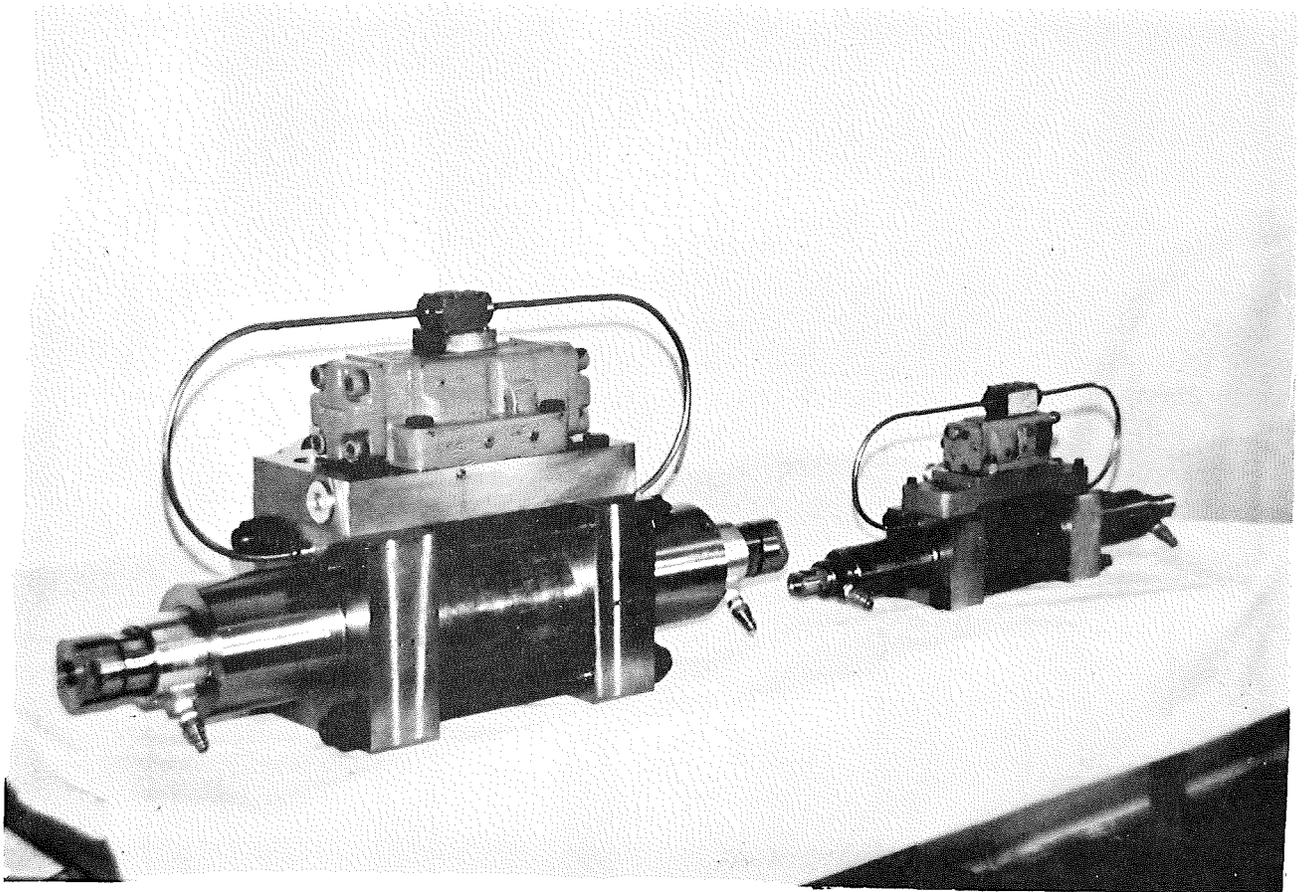
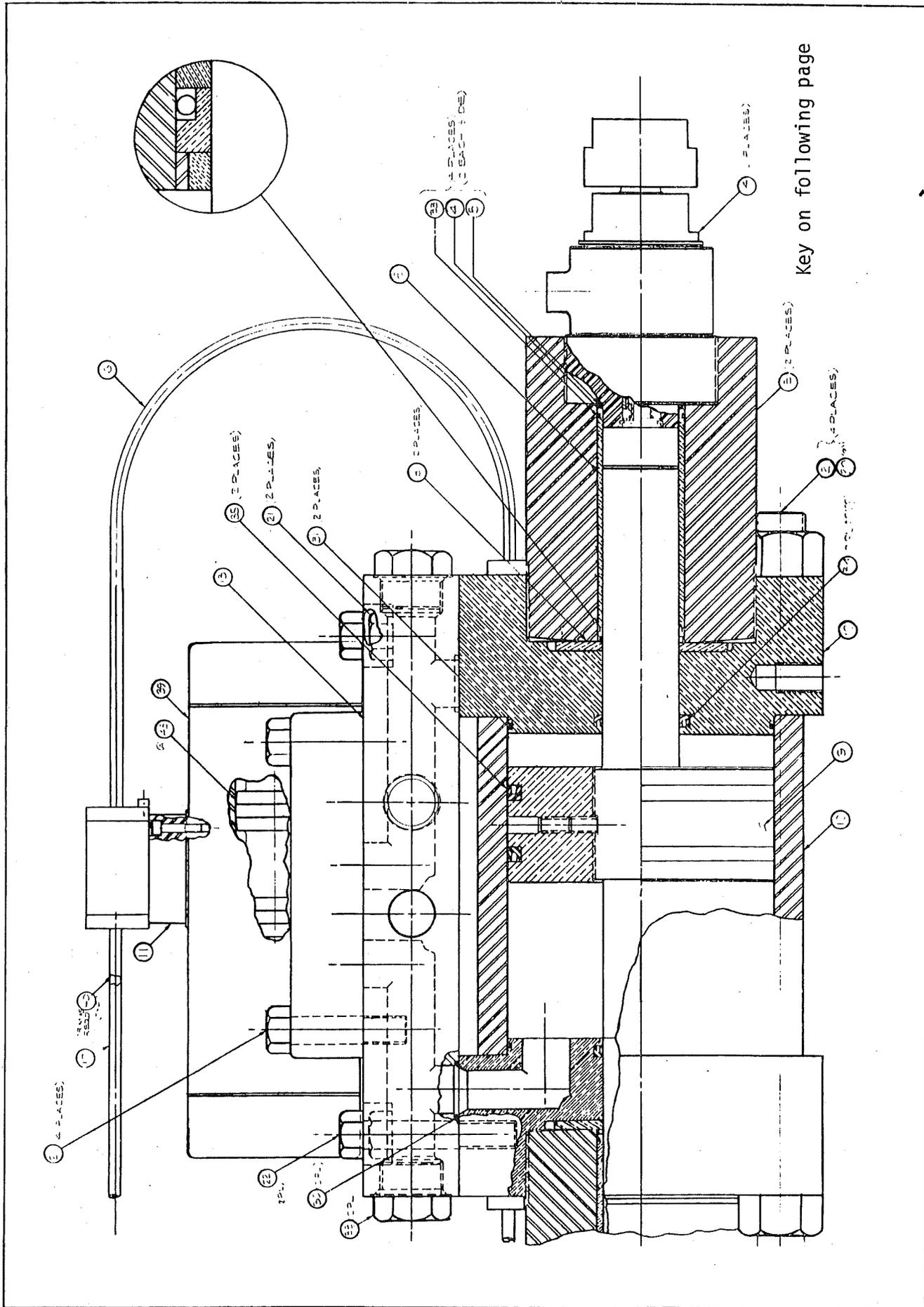


Figure 1b - The Large 250 HP Intensifier Designed and Built for This Investigation and the Smaller Original High-Pressure Intensifier



key on following page

Figure 1c

Parts List

| Qty. | Item No. | Description | Specification |
|------|----------|---------------------------|------------------------|
| -1 | | | |
| - | 1 | Intensifier Assy. | -- |
| 4 | 2 | Tie Rod | B1096-1 |
| 2 | 3 | Seal Ring | B1097-1 |
| 4 | 4 | High Press Seal | B1105-1 |
| 4 | 5 | Seal Ring Assy. | B1106-1 |
| 2 | 6 | Seal Back-up Sleeve | B1107-1 |
| 2 | 7 | Actuating Pin | B1108-1 |
| 2 | 8 | Actuating Plunger | B1109-1 |
| 1 | 9 | Piston/Rod Assy. | C1043-1 |
| 2 | 10 | Cylinder | C1044-1 |
| 1 | 11 | Manifold-shift Valve | C1047-1 |
| 1 | 12 | End Bell | D1037-1 |
| 1 | 13 | Manifold | D1038-1 |
| 2 | 14 | End Valve Assy. | D1040-1 |
| 2 | 15 | Hi Press End Cyl. | C1040-1 |
| 1 | 16 | Cable Guide | C1048-1 |
| 1 | 17 | Cable Guide | C1052-1 |
| 1 | 18 | Pilot Valve | B1003-1 |
| 1 | 19 | Spool Shaft | B1004-1 |
| 8 | 20 | Nut | 1½-12 UN-2B GR 8 |
| 6 | 21 | Soc HD Cap Screw | 3/4-16 UNF-2Ax3LG GR 8 |
| 2 | 22 | Soc HD Cap Screw | 3/4-16 UNF-2Ax4LG GR 8 |
| 4 | 23 | Soc HD Cap Screw | ¼-20UNC-2Ax1LG |
| 8 | 24 | Soc HD Cap Screw | #10-24UNC-2Ax5/8LG |
| 4 | 25 | Soc HD Cap Screw | #10-24UNC-2Ax1 3/4LG |
| 4 | 26 | O-Ring | A1025 |
| 1 | 27 | O-Ring | A1008 |
| 1 | 28 | O-Ring | A1010 |
| 4 | 29 | O-Ring | A1343 |
| 2 | 30 | O-Ring | A1342 |
| 2 | 31 | O-Ring, with back-up ring | A1341 |
| 2 | 32 | O-Ring | A1016 |
| 4 | 33 | O-Ring | A1344 |
| 2 | 34 | Seal | A1346 |
| 2 | 35 | Seal | A1345 |
| 1 | 36 | Shifter Valve | A1024 |
| 2 | 37 | Snap Ring | A1009 |
| 2 | 38 | Plug | A1347 |
| 1 | 39 | Valve | A1330 |
| 2 | 40 | Cable | A1539 |
| 2 | 41 | Spring | A1541 |
| 2 | 42 | Spring | A1348 |
| 1 | 43 | Spool Modification | B1122 |
| 1 | 44 | Nut | ½-13UNC-2B GR 8 |
| 6 | 45 | Bolt, Soc Head | 5/8-11UNC-2Ax3½LH |
| 1 | 46 | Soc HD Cap Screw | ½-13UNC-2Ax2 5/8LG |
| 1 | 47 | O-Ring | A1545 |
| 1 | 48 | Connector Plate | D1108-1 |
| 8 | 49 | Jam Nut | 1½-12UN-2B GR 8 |
| 8 | 50 | Washer | 1½ Hardened |
| 6 | 51 | Hi-Collar Lock Washer | 3/4 |
| 1 | 52 | Opposite-12 | D1037-2 |

is a hard polymer back-up ring with a steel hoop around its outer diameter. The hoop prevents extrusion of the back-up material between the high-pressure cylinder and the metal back-up ring. The metal back-up ring floats on the piston shaft and is clamped in place by the cylinder at final assembly. In this way, a close fit can be achieved on the piston shaft without alignment problems.

This intensifier was a scaled-up version of the smaller intensifier that has been in commercial service for about 2 years. It was, however, the first large intensifier ever built to this design, and several problems were encountered. At full flow, there was a large hydraulic shock initiated by the shifting of the oil to reverse the piston direction. This shock was reduced by slowing the rate of shifting and starting the shifting action earlier in the stroke to prevent the piston from hitting the end. It was later further reduced by installing an air-filled stand pipe in the oil outlet line from the intensifier.

The second problem encountered was the fracture of the piston shaft in a plane perpendicular to the centerline of the piston. Metallographic analysis showed that the fracture was caused by metal fatigue. Since the piston is loaded only in compression, this result was at first a mystery. It was later learned that some of the pistons had warped during heat treatment and were straightened afterwards, before final grinding. Such a procedure would result in a residual tension in one

side which would be relieved periodically by the stroking of the piston thereby allowing a fatigue crack to grow. New pistons were manufactured with an improved heat treatment technique and with extra allowance made for grinding so that no straightening was necessary. These pistons performed satisfactorily for the remainder of the test and no further problems are anticipated.

Yet another problem that occurred early in the start-up was deterioration of the seal between the low-pressure cylinder and the endblock of the intensifier (see 31 on figure 1). This seal was an O-ring with a backup ring and tolerances as recommended in the manufacturers handbook. However, apparently the high-pressure combined with the cycling allowed the back-up ring to be continuously eaten away causing an eventual failure of the O-ring. This problem was stopped by replacing the hard rubber back-up rings with fabricated back-up rings made from Lexan sheet. These rings lasted well for the duration of the experiments but inspection afterward revealed slow erosion. This seal should be redesigned for long-life.

2.2 Hydraulic Power Package

The purpose of the hydraulic power package was to provide oil at pressures up to 5,000 psi for driving the intensifiers. The power package was mounted on two separate identical skids. Each skid had two 250 HP motors. Each motor had two variable volume pumps capable of 45 gpm at 5,000 psi. Most of the

problems with the pumping units involved these power packages. Figure 2 shows the intensifier and hydraulic power packages mounted on the skids in the shop, and figure 3 shows the electrical controls for the power packages.

The first hydraulic circuit used is illustrated in figure 4. The De Levan pumps draw oil from the reservoir through a strainer. The pumps are variable volume with the adjustment being made through a nob on the pump case. The case drain from the pump is returned directly to the reservoir. Two pumps are plumbed together to run one intensifier. A pressure gauge is installed immediately upstream of the intensifier. A relief valve to protect pumps against over pressure and to prevent over pressurizing the high-pressure lines is installed in the high-pressure line. The relief valve is vented by means of a push-button valve to unload the pumps and stop the motion of the intensifiers. The relief returns to the tank in a common line with the return from the intensifiers. The return line passes through a filter on one side of the skid and through a filter and a cooler on the other.

This circuit was plagued with many problems, most of which were associated with the pulses generated by the shifting of the intensifier. The high-pressure lines leading from the pump to the intensifier were plumbed with pipe threads which were a continual source of annoyance due to leakage. The pulses from the intensifier damaged both the filter and the oil cooler. The cooling capacity required of the cooler

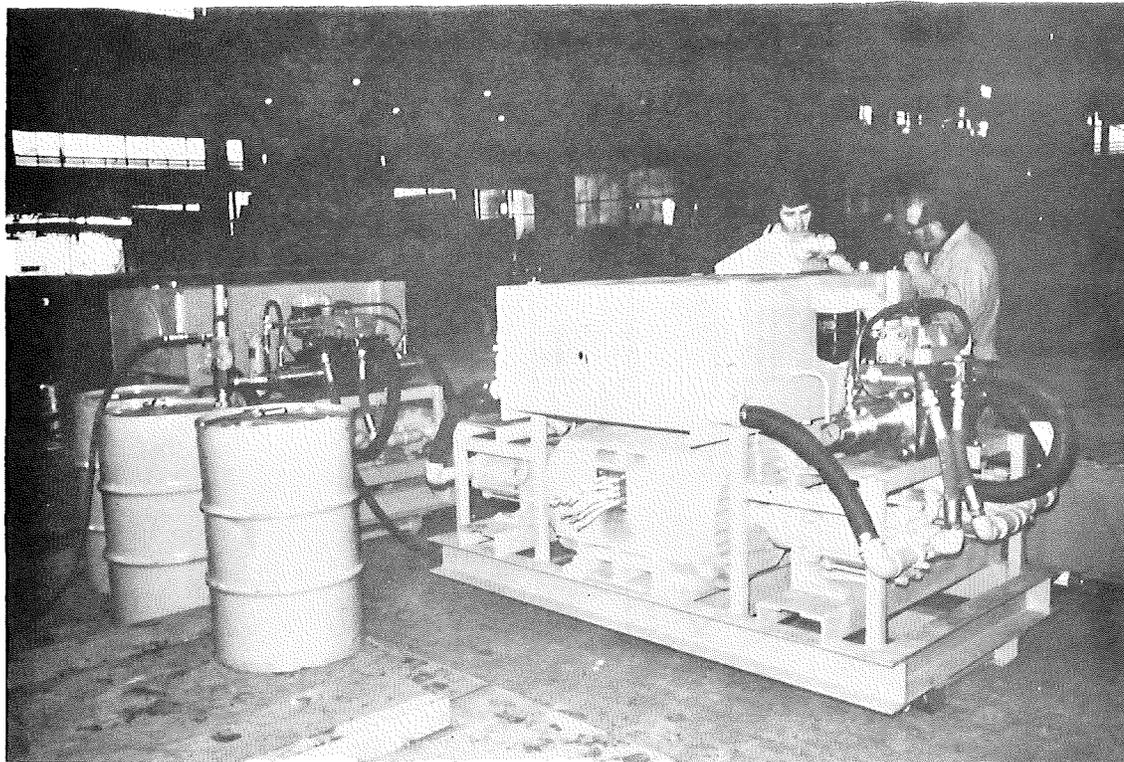


Figure 2 - Intensifier in the Shop

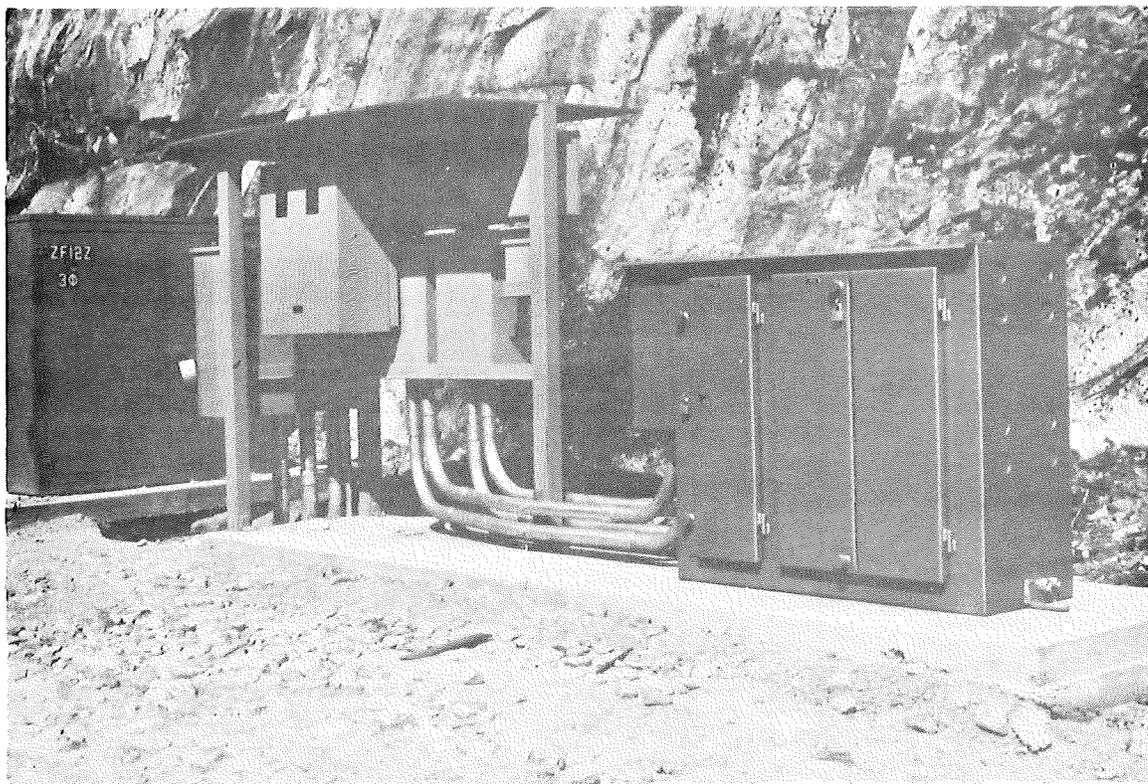


Figure 3 - Electrical Controls for Hydraulic Power Packs

had been underestimated and the units tended to overheat. The pressure pulses from the intensifier were greatly reduced by the methods described earlier and the oil circuit was changed to eliminate the other problems.

The second circuit, shown in figure 5, was essentially the same as the first except that the cooler and filters were removed from the return line and placed in a separate loop. A larger cooler was installed. A return line from the intensifier was plumbed with solid piping. This was perhaps a mistake, since it was later found that adding compliance to the return line by means of a stand pipe with air in it helped to reduce the water hammer shock upon shifting.

With these changes the operational problems were greatly reduced. One skid ran quite well, but the oil in the other skid still tended to overheat during long runs. It was suspected that the pulses in the return line circuit affected the performance of the relief valve so that oil was spilled over the relief causing excess heating. The only remaining problem was the continual leakage from the pipe thread joints.

Any remaining problems could probably be eliminated, and the operation of the units made more simple by making the following additional changes:

- 1) plumb the pressure lines with rigid pipe using no pipe thread joints.
- 2) plumb the return line with a flexible hose to reduce any remaining water hammer shock.

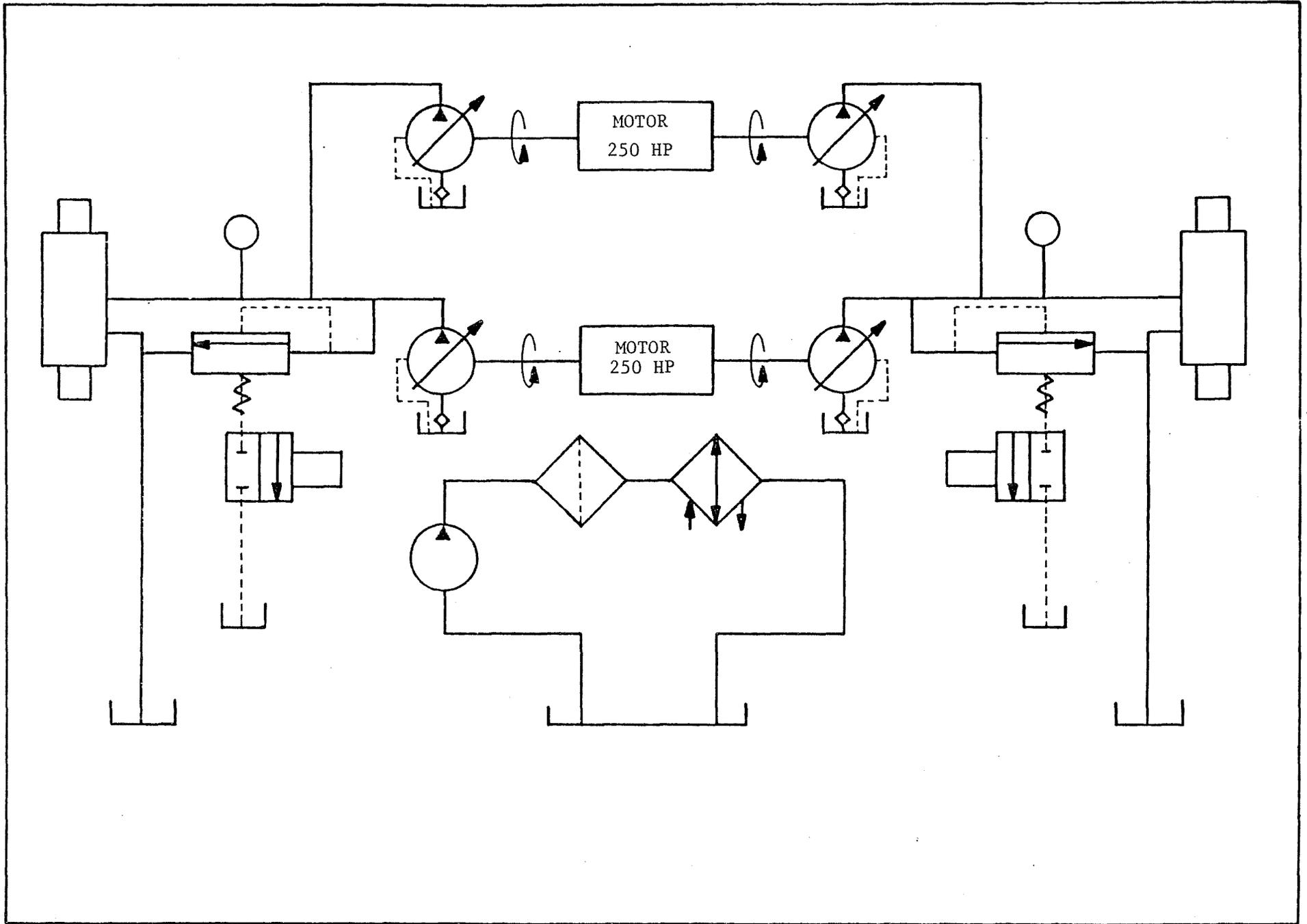


Figure 5 - Circuit 2

- 3) relieve the pressure lines directly to the tank and not into the return line.
- 4) plumb both pumps from one motor to one intensifier so that each intensifier can be run separately.
- 5) add additional oil cooling capacity.

2.3 Operation

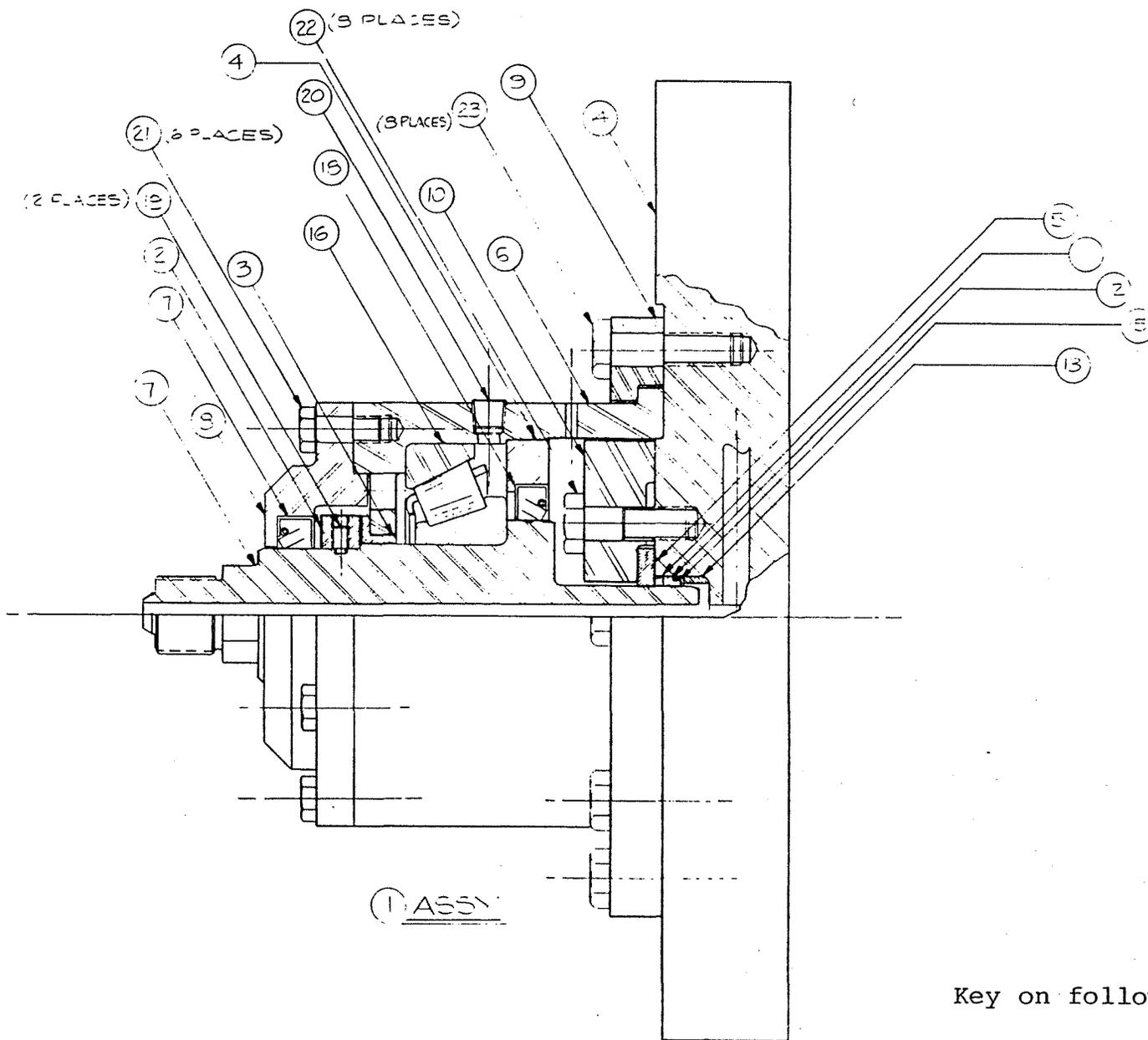
The water pumped by the intensifiers was drawn from a nearby river and pumped to a storage tank near the pumping equipment. It was drawn from the storage tank and pumped through a 20 micron filter to the intensifiers where it was delivered at a pressure of about 60 psi. The outputs from the intensifiers were ganged together into a 160 cubic inch volume that served as an accumulator to keep the jets running steadily during the shifting of the intensifiers. From the accumulator, four separate pressure lines of 9/16 outside diameter, 3/16 inside diameter and of 303 stainless ran approximately 30 feet to the centerline of the tunnel. At this point, they were joined with elbows to lengths of tubing running along the tunnel to the machine. As the machine moved forward, the 30 ft. lengths of tubing bent in the direction of motion. After the machine had advanced 10 ft the elbows were disconnected, the 30 ft lengths bent back and a 10 ft length of tubing installed between the elbows and the machine. In this way, the machine could advance 10 ft at a time with a steady supply of high-pressure water.

All the electrical switches for the control of the pump motors were mounted at the boring machine operator's station. During start-up of the pumps the following procedures were followed:

First, the relief valves on the 5,000 psi oil were vented by depressing the vent buttons to unload the pumps. The motors were then started and the reliefs closed allowing the pressure to increase. If the pressure was not at the desired level, the reliefs were vented again and the pumps strokes adjusted to increase or decrease the flow rate. At the end of the run, the pumps could be shut off from the boring machine operator's control or from an emergency stop button located near the pumping equipment.

2.4 High Pressure Swivel

The water from the pumping system is carried by four tubes along the machine to the swivel. The function of the swivel is to carry the water from the stationary tubes to the rotating head. Figure 6 shows a cross section of the high pressure swivel. Part 7 is stationary and most other parts are rotating. The four input lines from the pumps are first manifolded together and then piped through the swivel. On the rotating side, the water again enters a manifold for distribution to the nozzles. The seal, as shown in parts 11, 12, and 15 is the same type of seal as the rod seal in the intensifier and is in fact the same seal used in the small 50 hp industrial intensifiers mentioned earlier.



Key on following page

Figure 6 - High-Pressure Swivel

| Qty. | Item No. | Description | Specification |
|------|----------|----------------------|---|
| - | 1 | Swivel Assy. | |
| 1 | 2 | Bearing Plate | B1125-1 |
| 1 | 3 | Bearing | B1126-1 |
| 1 | 4 | Seal Retainer-Inner | B1127-1 |
| 1 | 5 | Seal Ring | B1128-1 |
| 1 | 6 | Housing | C1061-1 |
| 1 | 7 | Shaft | C1062-1 |
| 1 | 8 | Seal Retainer | C1063-1 |
| 1 | 9 | Retaining Plate | C1064-1 |
| 1 | 10 | Spacer Bar | C1065-1 |
| 1 | 11 | Seal Ring Assy. | B1005-1 |
| 1 | 12 | High Pressure Seal | B1006-1 |
| 1 | 13 | Spacer | B1133-1 |
| 1 | 14 | Manifold | D1056-1 |
| 1 | 15 | O-Ring | Parker 2-022 (A1015) |
| 1 | 16 | Bearing | Timkin 72200 Cone, Timkin 72487 Cone |
| 1 | 17 | Seal | C/R 18699 |
| 1 | 18 | Seal | C/R 27427 |
| 2 | 19 | Set Screw, Dog Point | 1/4-20UNC-2Ax 3/8 LG |
| 1 | 20 | Pipe Plug | 1/8 NPT, STL |
| 6 | 21 | Bolt, Hex Head | 3/8-16UNC-2Ax 1EG GRS |
| 8 | 22 | Bolt, Hex Head | 1/2-13UNC-2Ax 1 1/2 LG Stainless Steel 18-8 |
| 8 | 23 | Bolt, Hex Head | 1/2-13UNC-2Ax 1 3/4 LG GRS |

The shaft that rotates within the seal is made from 15-5 PH heat treated to the H900 condition. In assembly, parts 5, 10, 11, 12, 15, are aligned with an alignment tool before bolts 22 are tightened. The remainder of the swivel is then bolted in place as an assembly. The 20 ton thrust load produced by the pressure acting on the end of shaft 7 is taken by a tapered roller bearing. This bearing is sealed in an oil bath for lubrication. A vent-hole is provided to prevent leakage from the high pressure water seal forcing its way past the oil seal into the oil chamber. No problems whatsoever were encountered with this swivel. In fact the seal was never changed during the whole experimental program. The swivel is shown mounted on the rear of the cutterhead in Figure 7.

2.5 Manifold

The problem of locating and plumbing to nozzles on an existing machine where no provisions had been made for such an installation was a difficult one. Ideally, one would wish to place the manifold behind the cutter head and plumb to the nozzles by passing through the cutter head. On this machine the manifold had to be placed on the front face of the cutter head. The nozzle locations were chosen so as to place one nozzle on the same path as each cutter and one nozzle midway between the paths of adjacent cutters. Plumbing from the swivel to these locations was accomplished by means of three-manifolds. The first manifold was in fact the base on which

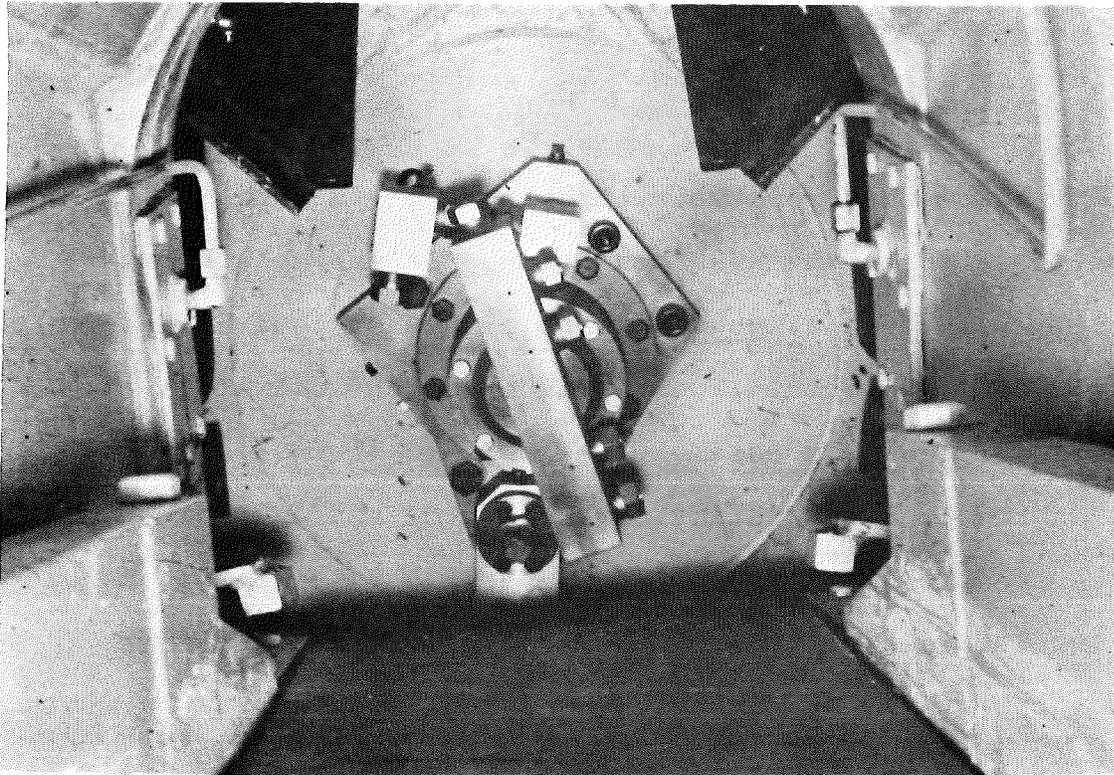


Figure 7 - High-Pressure Swivel Mounted
to the Back of the Cutterhead

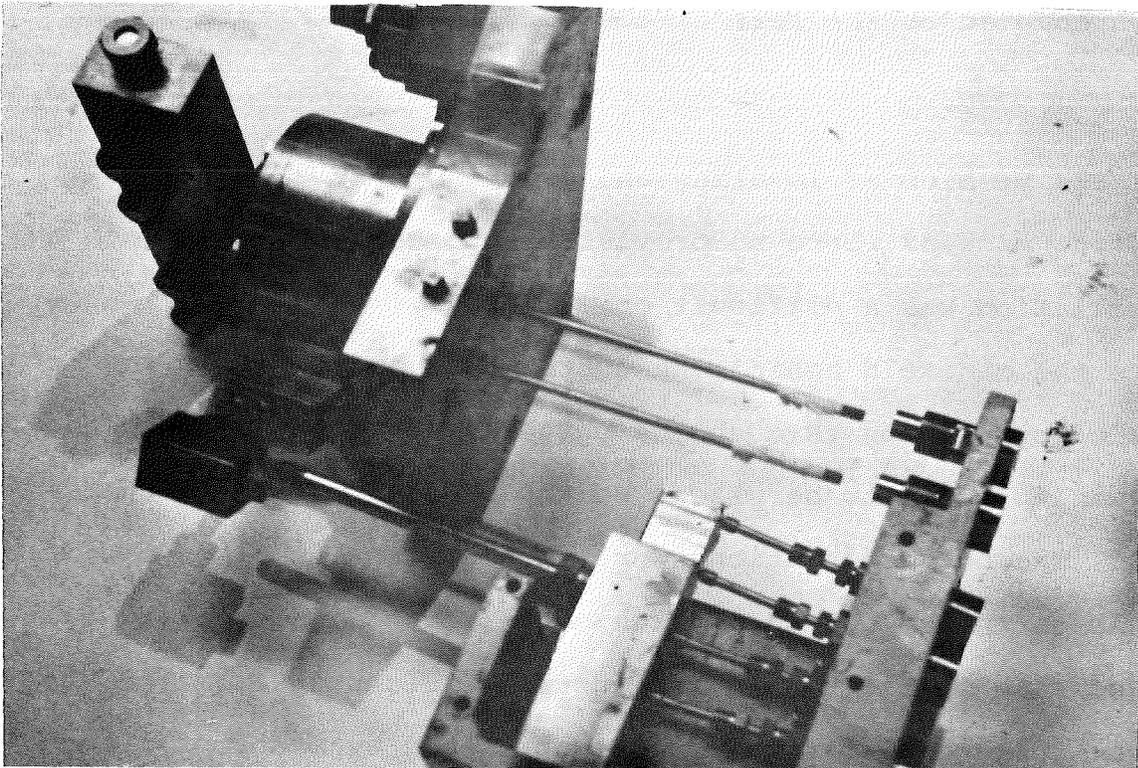


Figure 8 - Relationship of Swivel to Small Manifold

the swivel was mounted. The two center nozzles were plumbed through holes drilled in the cutter housings and tied directly to this main manifold. Going out along a radius the next five nozzles were plumbed in a separate manifold on the front face of the cutter head. This manifold was tied to the main swivel manifold with a standard high pressure tubing connection using 9/16 diameter tube. A view showing the matching of this small manifold with the swivel is given in Figure 8. The remaining nozzles were plumbed to a large manifold that ran along the radius for a distance of 34.5". The connection between this manifold and the main manifold at the swivel was made very difficult because of space restrictions. The flow rate required was too large to carry in a single 9/16 tube and no space was available for multiple tubes. No commercial fittings for this large size were available. In addition, there was some question as to the alignment accuracy required to connect the two manifolds using short lengths of large diameter tube. The connection was accomplished with the floating link arrangement shown in figure 9. Large pins were fabricated from 17-4 PH and sealed to the two manifolds with a cone and thread type of seal. The pins were then joined by a length which was sealed by O-rings backed up with chamfered metal rings. There was a leakage problem with these seals, and they had to be replaced several times during the course of the experiments. If this link design is required in another installation, it is recommended that the seals used

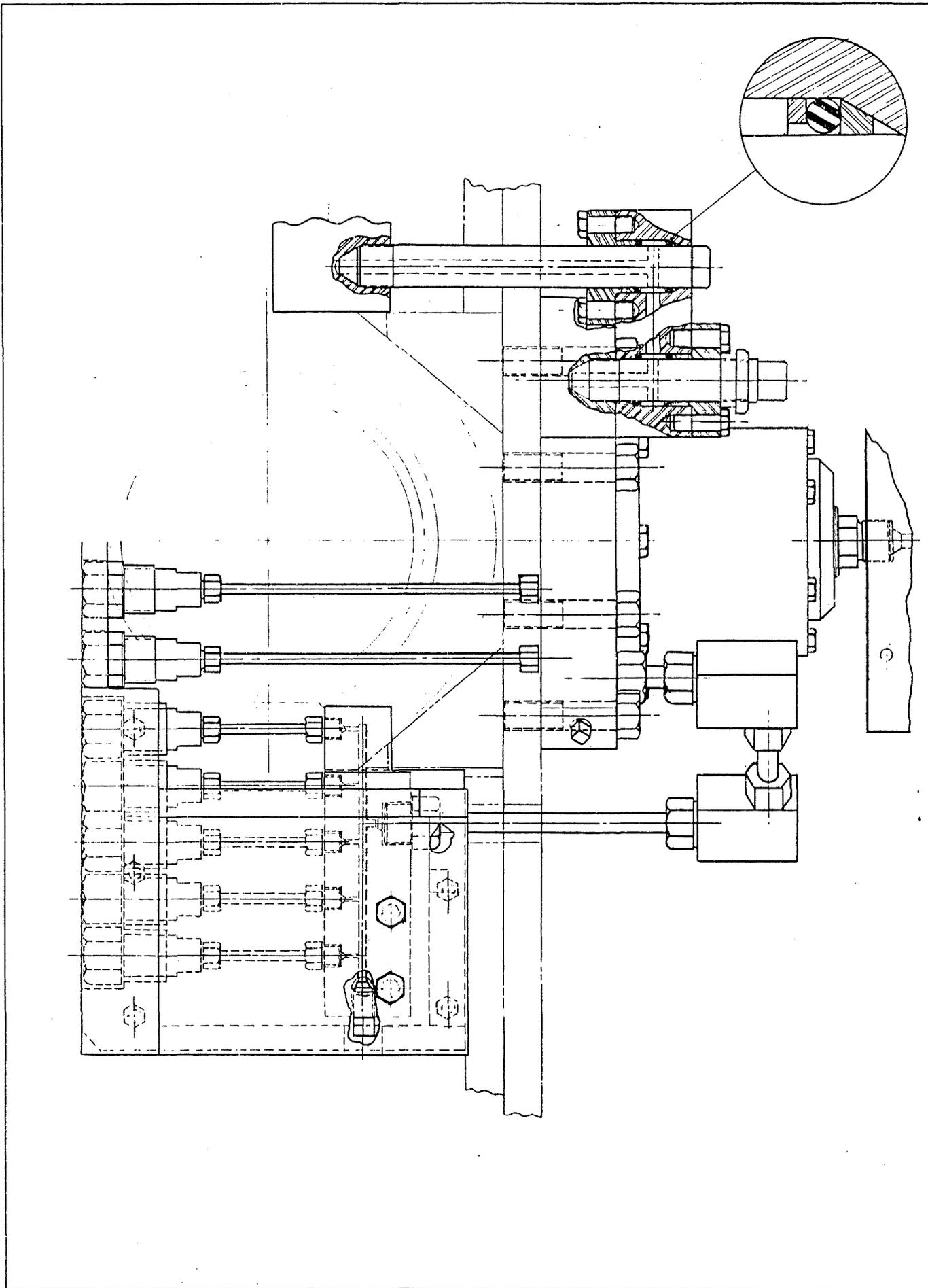


Figure 9 - Assembly of Swivel, Inner Manifold, and Link to Outer Manifold

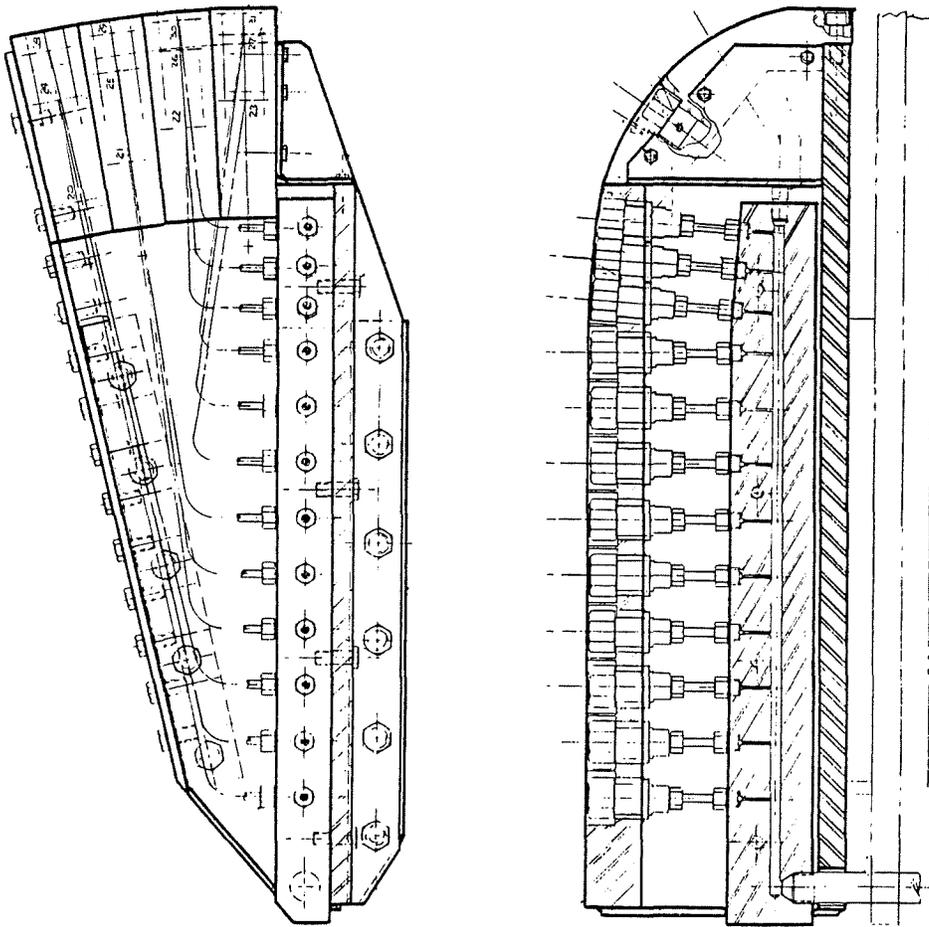
Scale 1:3.6

be the same as those shown for the swivel and the intensifier rod seal.

Initially it was not known how close the nozzles could be placed to the rock face without rubbing. Accordingly the manifolds were mounted on 1" thick shims so that they could be adjusted by changing the shim thickness.

The manifold on the outer radius is a rectangular block of 17-4 PH with a long through hole gun drilled into it. All the outer nozzles were then plumbed into this manifold using 1/4" tubing and standard cone and thread tube fittings. Figure 10 shows the connection between the nozzles and the manifold as well as the protective shield covering the manifold. The nozzles could be moved slightly by bending the quarter inch tube. In this manner, the nozzles were adjusted to their final exact position and locked in place with set screws and shims driven between the nozzles and the protective shield. Views of the assembled manifolds are provided in Figures 11 through 15.

The problems encountered with the manifolds were relatively minor. Occasionally a tube fitting would vibrate loose resulting in a leak. If this leak were allowed to continue for a long period of time, it eventually would erode the sealing surface making it difficult to reseal the joint. This type of problem could probably be greatly reduced by assembling the units with a thread locking compound. If the problem does occur, the seats and tube end can be refinished with hand tools which should be kept in the field.



Scale 1:6.5

Figure 10 - Outer Manifold

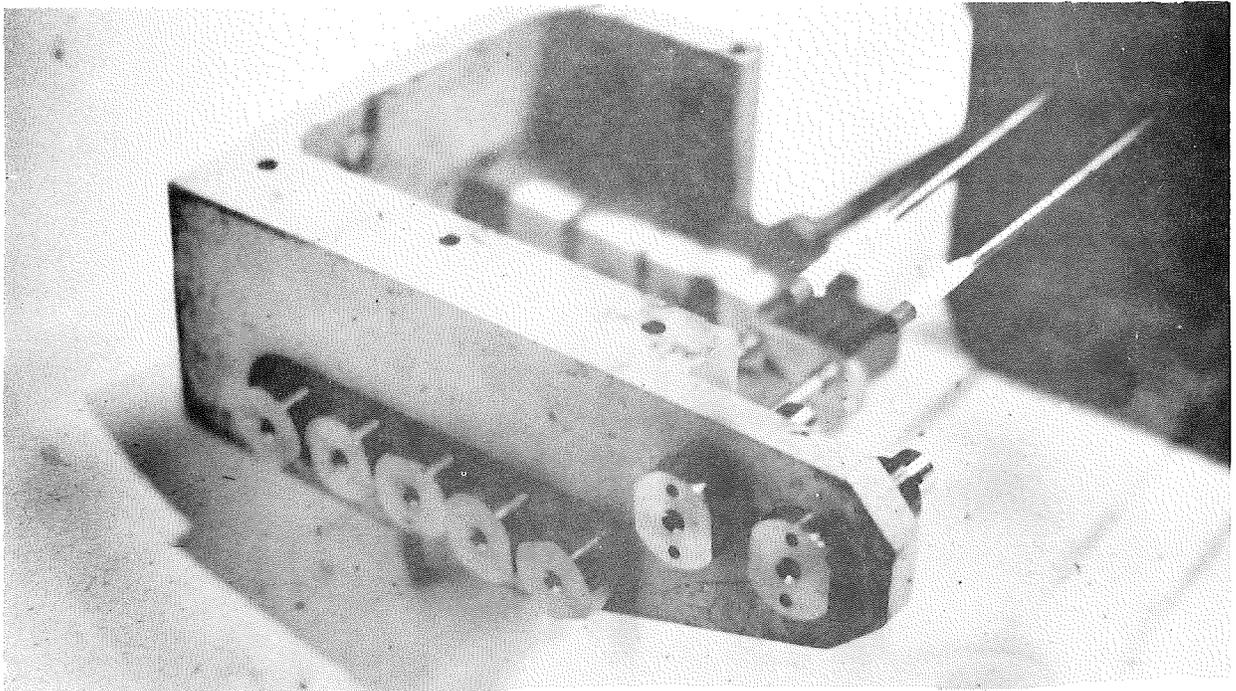


Figure 11 - Front View of Small Manifold approx. $\frac{1}{4}$ scale

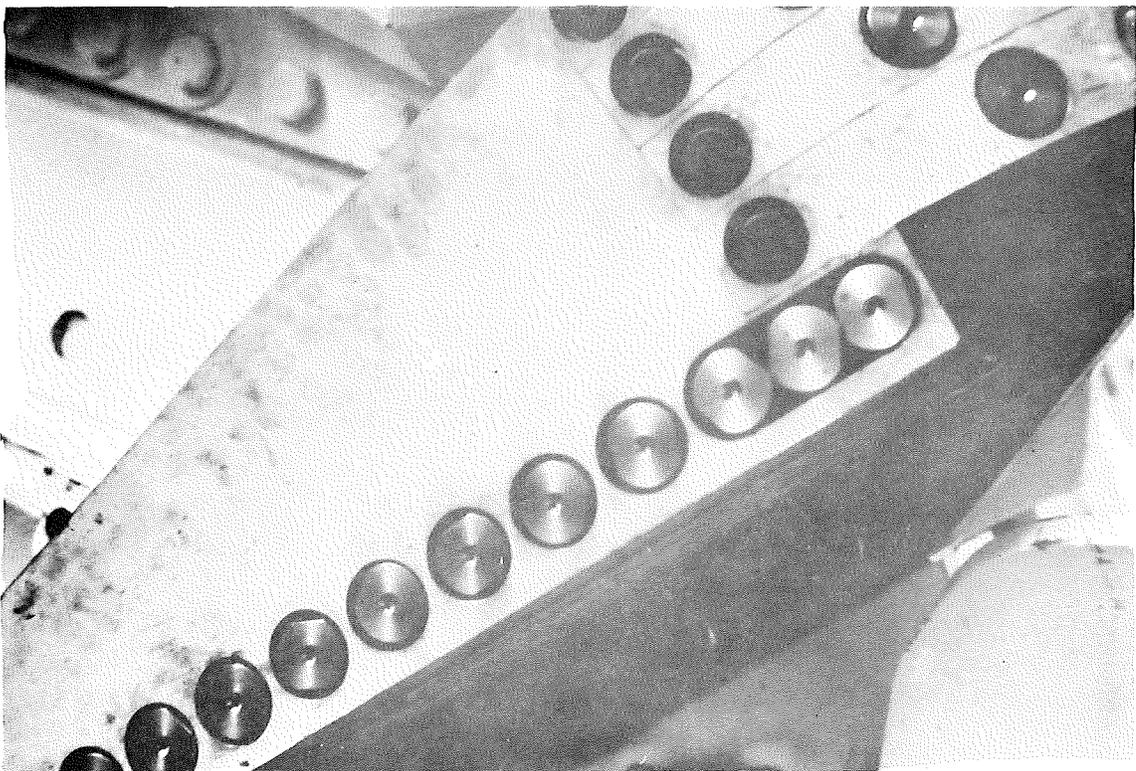


Figure 12 - View Showing Arrangement of Nozzles 8 through 19 on Large Manifold approx. $\frac{1}{4}$ scale

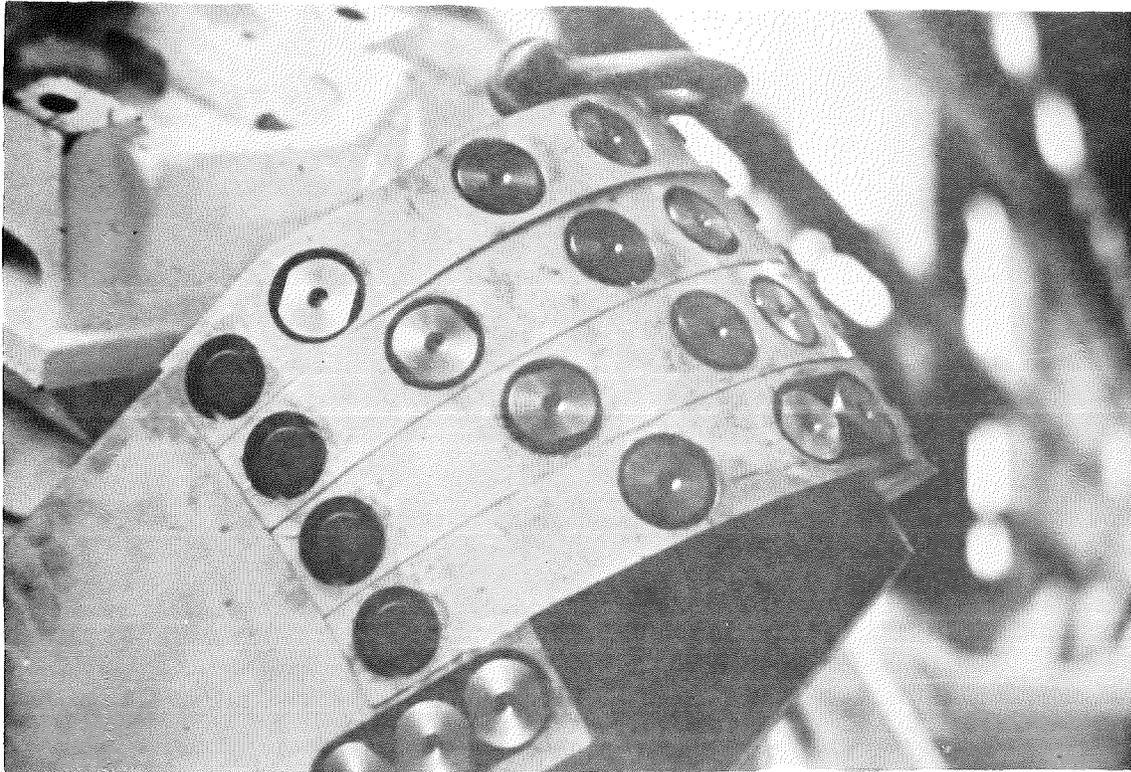


Figure 13 - View Showing Nozzles for the Curved Section of Tunnel Face at the Gauge approx. $\frac{1}{5}$ scale

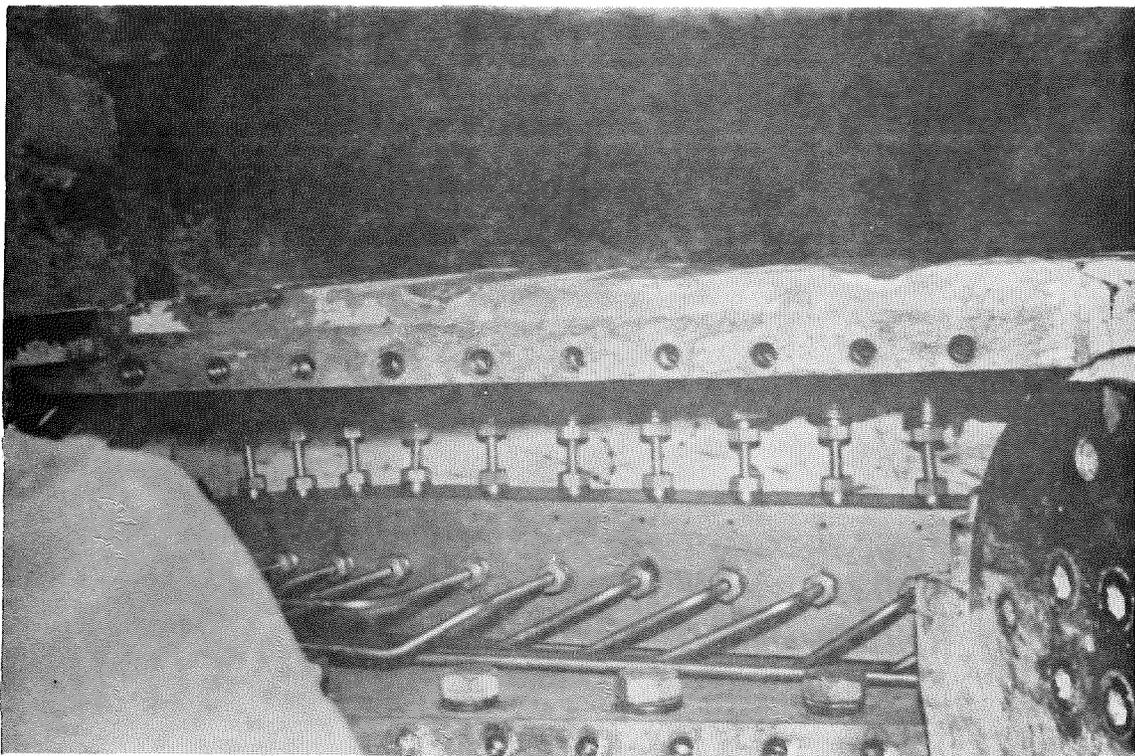
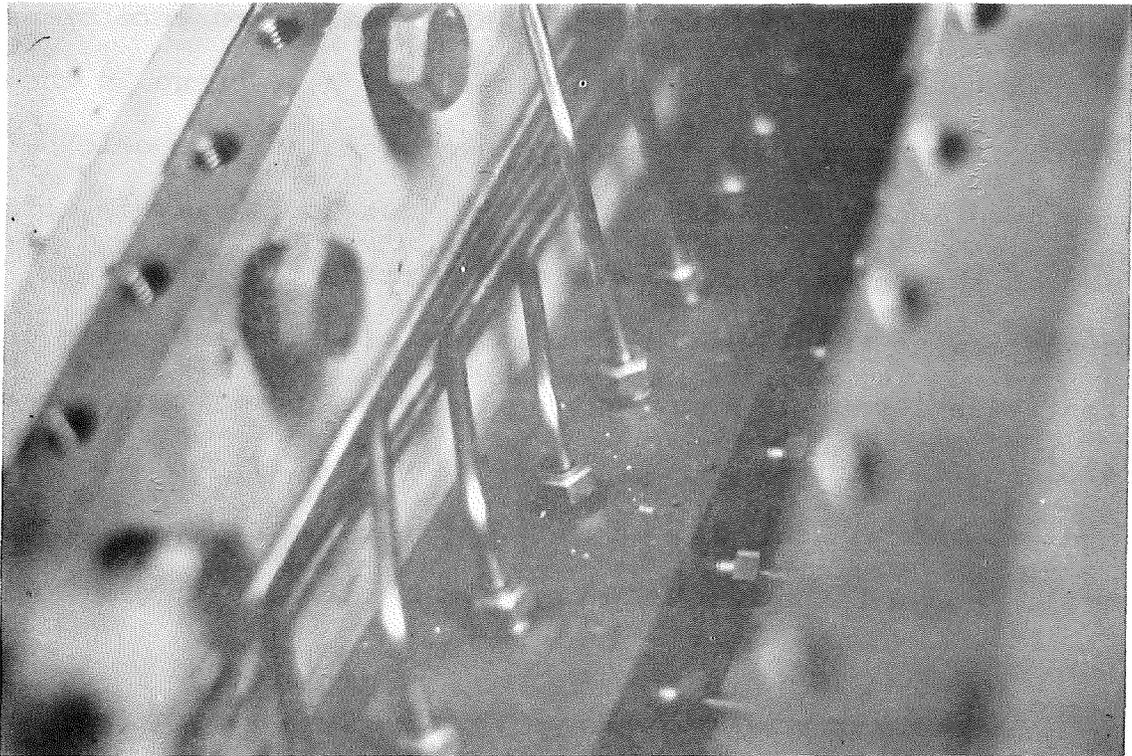


Figure 14 - View Inside Manifold Showing $\frac{1}{4}$ " Tubing Supplying Individual Nozzles approx. $\frac{1}{5}$ scale



approx. $\frac{1}{2}$ scale

Figure 15 - Close-up of Connections to
Main Block Inside Manifold

3.0 FIELD TESTING

3.1 Test Site

After the portal preparation and installation of the required transformer and electrical controls, the tunnel boring machine (TBM) and the water jet intensifiers were transported to the site by truck and off loaded by crane to their respective positions. The water jet packages were originally designed to be towed behind the TBM, but to facilitate working on the units, they were placed outside the tunnel and connected to the TBM by high pressure lines. The system and test site layout is represented in Figure 16. There was sufficient flexibility in the high-pressure water jet lines so that as the machine moved forward in the tunnel, the lines could bend until a new section of high pressure tubing could be added.

The excavated material was carried by conveyor belt from the rear of the TBM to outside the portal where it was emptied into the bucket of a large rubber-tired loader. The rock chips were then either spread as fill or stockpiled for later use.

Other facilities on the site included an existing office railroad car and a well-stocked parts car, a valuable selection of tools, heavy equipment handling capability, and a large well-equipped shop provided in cooperation with M-K. Water was supplied by a large centrifugal pump taking water

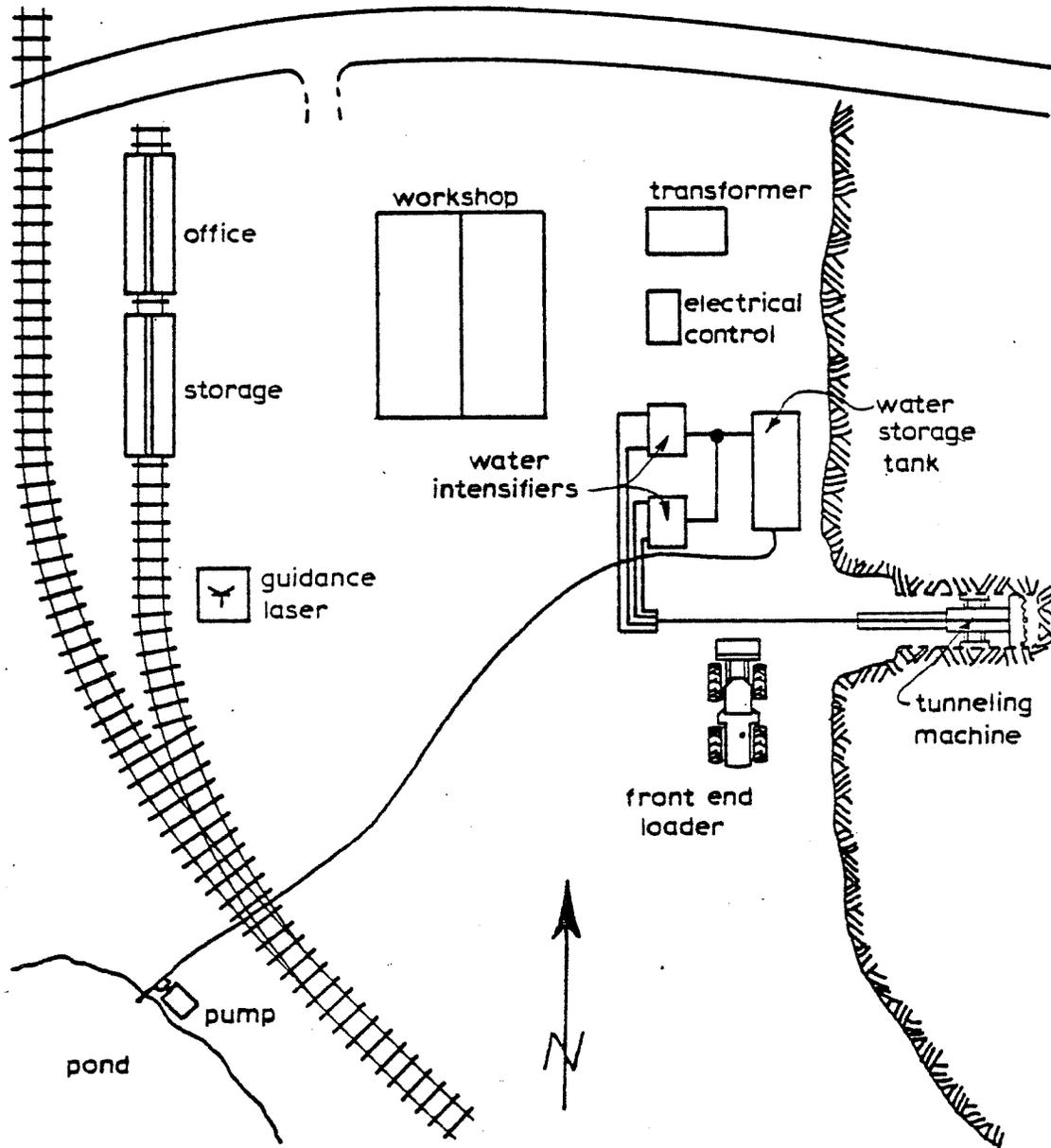


Figure 16 - Representation of Field Test Site Layout

from a holding pond and filling a 500 gallon tank near the intensifiers. The water was then run through a two-stage filtering process before supplying the intensifiers. Boring machine alignment was controlled by a tripod mounted laser approximately 150 feet behind the TBM.

A number of photographs of the site and equipment are presented in Figures 17 through 20.

3.2 Testing Procedure

The boring machine was placed inside the blasted section of tunnel and the grippers thrust against the sides of tunnel using timber spacers to distribute the load and improve the grip. The cutterhead was then advanced into the virgin tunnel face. There was some jumping of the machine as it started to cut the uneven face, and it was difficult to maintain an adequate grip. Some of the consequences of this early boring are evidenced in Figure 21. Finally, things quieted down as the cutterhead began cutting a full face. There were still problems gripping the uneven walls of the blasted section of tunnel. Therefore, boring was continued until the grippers were in machine bored tunnel. Figure 22 shows the boring machine in operation early in the bore. Note the muck coming off the trailing conveyor.

Since, at that time, the water jet equipment was still being assembled and tested, it was decided to proceed with a number of unassisted mechanical cutting tests to provide some baseline data. Tests were run at 4,000 psi, 3,500 psi and

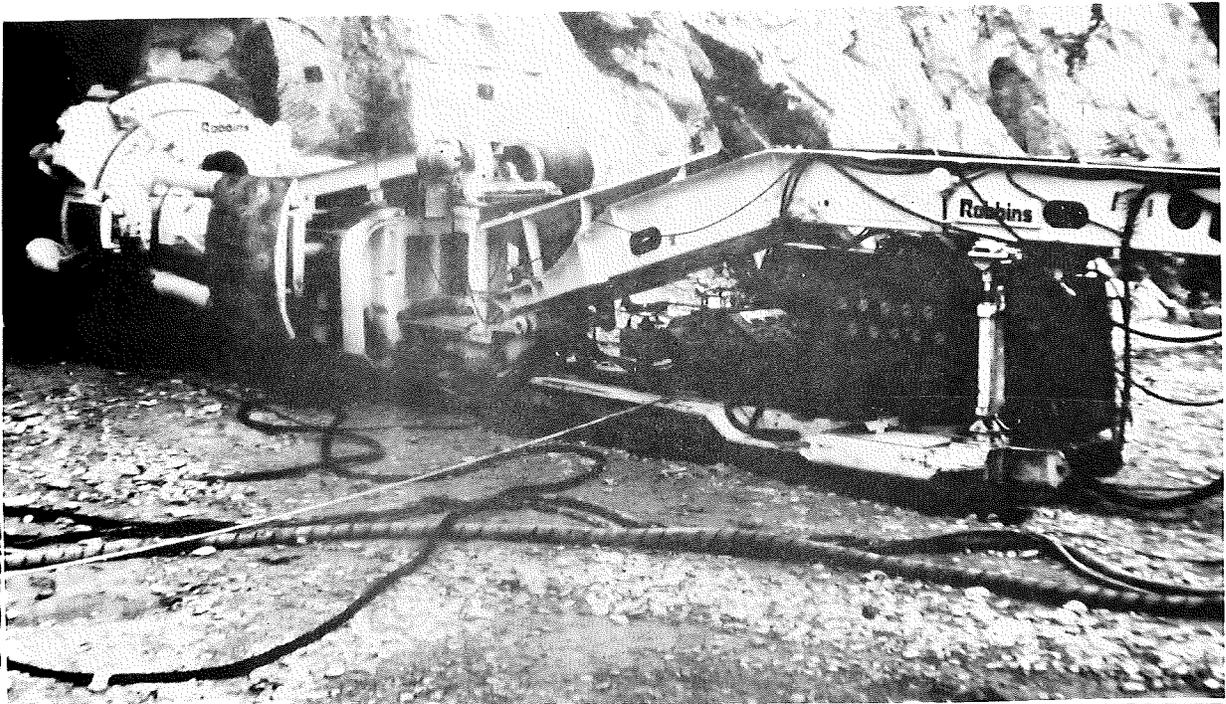


Figure 17 - The Tunnel Boring Machine Outside
the Portal at the Field Test Site

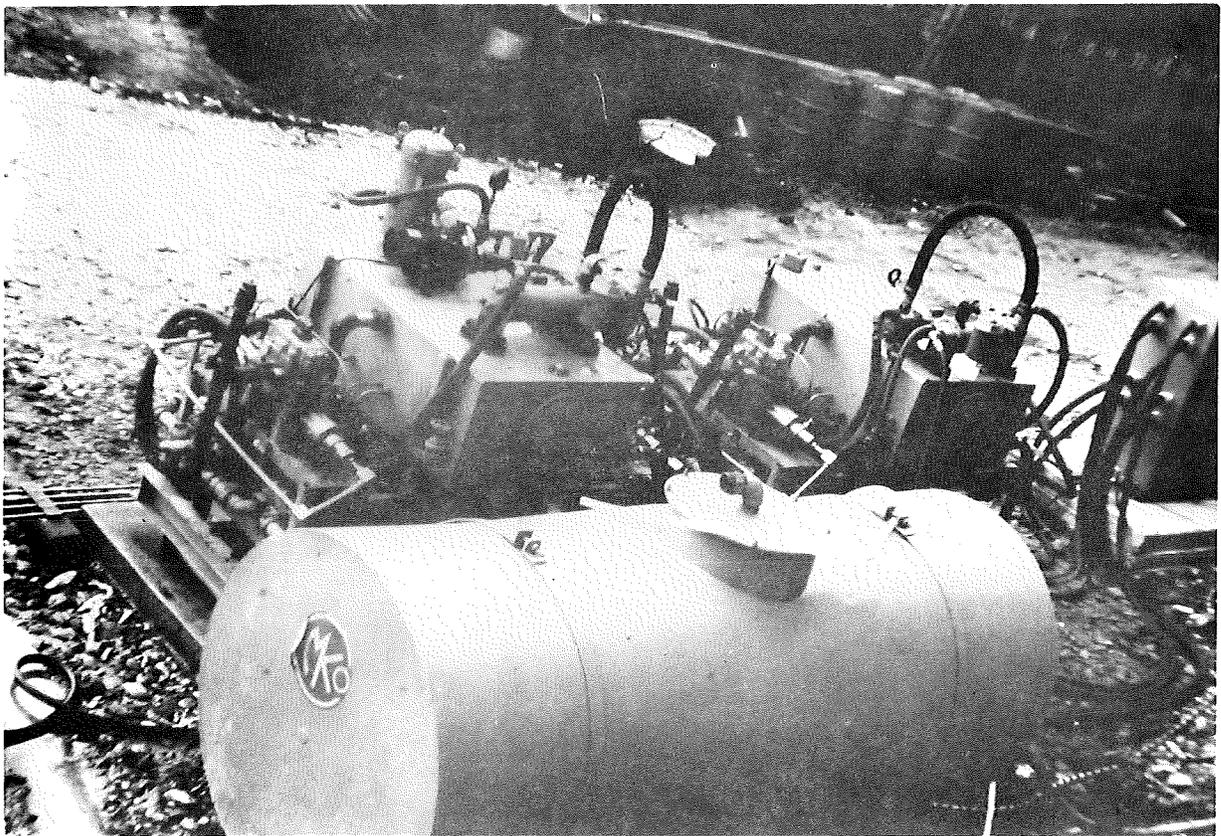


Figure 18 - View of the Intensifier Packages
and Water Storage Tank

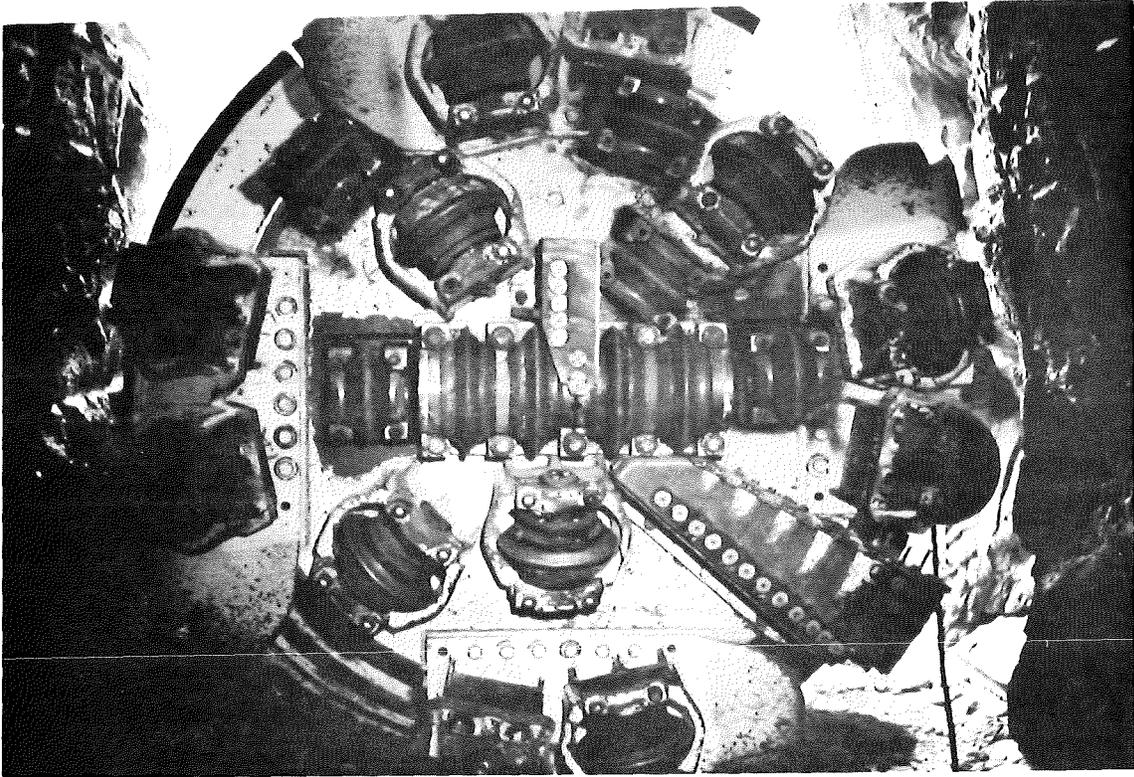


Figure 19 - The Cutterhead as Seen From Inside the Bore

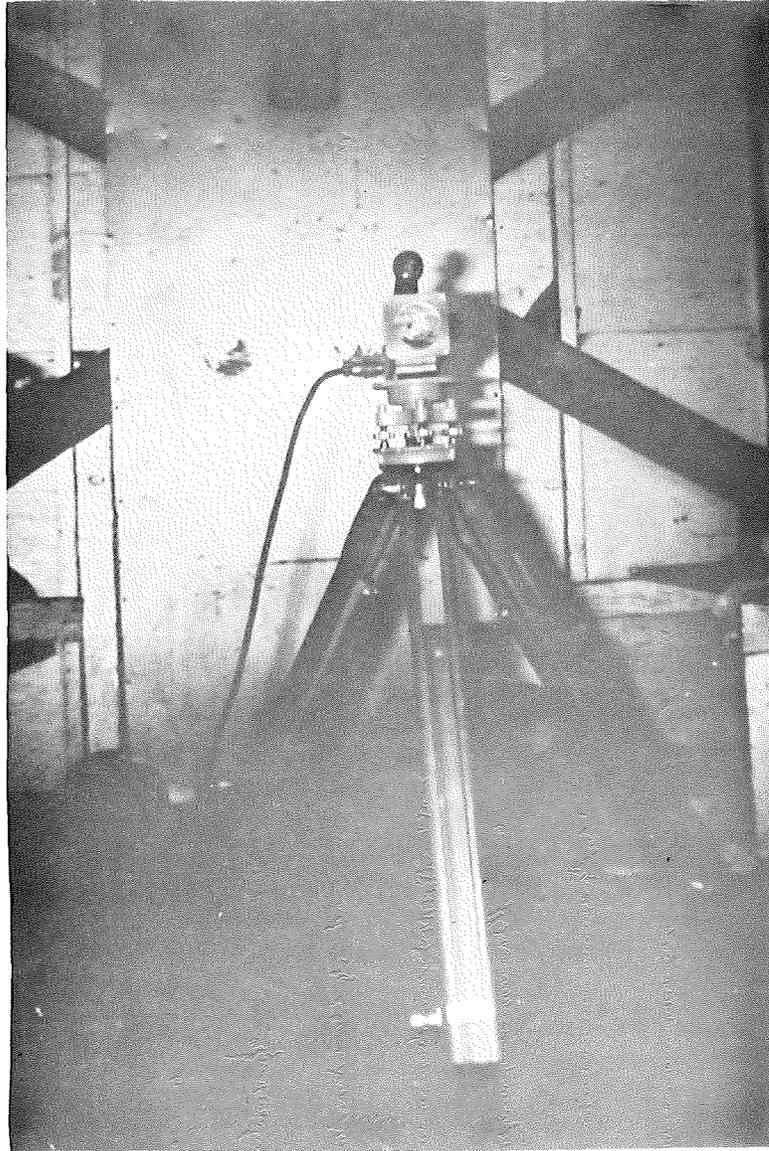


Figure 20 - The Tripod Mounted Laser Used
for Maintaining Line and Grade



Figure 21 - Cutter Damaged in Early Boring



note operator for scale

Figure 22 - Starting the Machine Bored Tunnel

3,000 psi thrust pressures on the cutterhead. The results were plotted as advance rate vs. thrust, and an expected performance curve developed. Two more tests were conducted at thrusts of 3,250 psi and 3,750 psi resulting in the advance rates for these two thrusts falling almost exactly on the expected advance rate-thrust curve from the first set of tests. Therefore, it was felt the system of testing was valid and reliable.

Due to the probability of variations in rock condition along the bore, and the possibility of variations in machine performance, it was determined that each test with the water jet should be immediately followed by an unassisted mechanical test at the same thrust. Of course, a standard duration of test is necessary to provide a uniform basis for comparison of results. Time was chosen as the basis for determining length of test because, for a given period of time, the number of passes by the cutters over the rock face would be the same. Generally, the duration of test was fifteen minutes. However, when for any reason it was not possible to obtain that length of test, or it was decided to extend the run, the corresponding comparison test was performed for an equivalent length of time.

Before each night's testing, certain preliminary operations were performed. The 500 gallon holding tank was filled, the water filter elements were changed, and the water jet system brought up to pressure. The system was then visually inspected for leaks or plugged nozzles. The expected flow

for a given nozzle configuration and pressure could then be easily checked using the installed pressure and flow meter. If conditions appeared satisfactory, the TBM was moved forward to the face.

As mentioned, the normal sequence of testing consisted of running a particular water jet assisted test followed by an unassisted mechanical test. When everything appears to be ready, the TBM and the water jet intensifiers are started and the cutterhead is advanced into the tunnel face at the appropriate thrust. To attain steady state conditions, a 0.50 inch advance is taken before data is recorded. The correct operation of all components is also ascertained during this preparatory boring. Next, a reference mark is placed on the tunnel wall and a stop watch and tape measure used to record advance versus time for the duration of the test. Eventually, the advance for each minute of the test was recorded to insure the observation of any performance changes during a particular test. Also continuously recorded during each test were thrust, torque, and water jet pressure, with muck samples being taken for appropriate runs. At the end of the specified length of test, the water jet was turned off but the TBM was allowed to bore an additional 0.50 inch to eliminate the effect of the water jet on the face. The recording of the data for the corresponding unassisted mechanical testing was then begun. Again, thrust and torque along with advance rate, are recorded and muck samples obtained when required.

After completion of the unassisted mechanical testing, the TBM was moved back from the face to allow access ahead of the cutterhead. The condition of the face (presence of planes of weakness, inter-cutter ridges, etc.) was noted, cutters, nozzles, and manifolds were inspected for any visual damage, and the water jet system tested for any leaks or plugged nozzles. If any minor repairs were required, they were performed and the TBM moved back to the face for the next test. Major modifications or repairs were made during the day while testing was not permitted due to high electrical power demand.

Tests were performed at various combinations of thrust, water jet pressure, nozzle sizes, and jet configuration. Variations in rock quality did occur along the bore and equipment problems often interrupted the tests. Some of these problems are elaborated in the next section.

3.3 Problems Encountered

Problems are, of course, expected in any field scale experiment. However, this program experienced the additional adverse situations of having to use prototype equipment for which time did not permit extensive preliminary testing, and where the required operating hours of midnight to 5:00 a.m. limited the ability to recover from equipment failures the same day. We were very fortunate in having experienced, highly motivated personnel from all three participating organizations who could generally remedy the many small

problems which occurred almost nightly, thus avoiding major delays in the program.

The major source of problems was the hydraulic power packages supplying high-pressure oil to the intensifiers. Small but continuing problems with the inadequately designed hydraulic power packs repeatedly interrupted testing. Hydraulic hammer effects caused many leaks to develop and also the failure of several components including the return line filters and oil cooler. In addition, there were burst hoses, damaged O-rings, faulty relief valves, overheating of oil, fluctuations in pressure, and fatiguing of connections. Generally, all these problems were simply a result of undersized hydraulic pumps and an oil reservoir that was too small. The hydraulic pumps were originally for operation at 3,000 psi and were modified to operate up to 5,000 psi. The 150 gallon reservoirs on each skid proved to be too small to allow effective cooling of the oil and to allow stabilization of the oil.

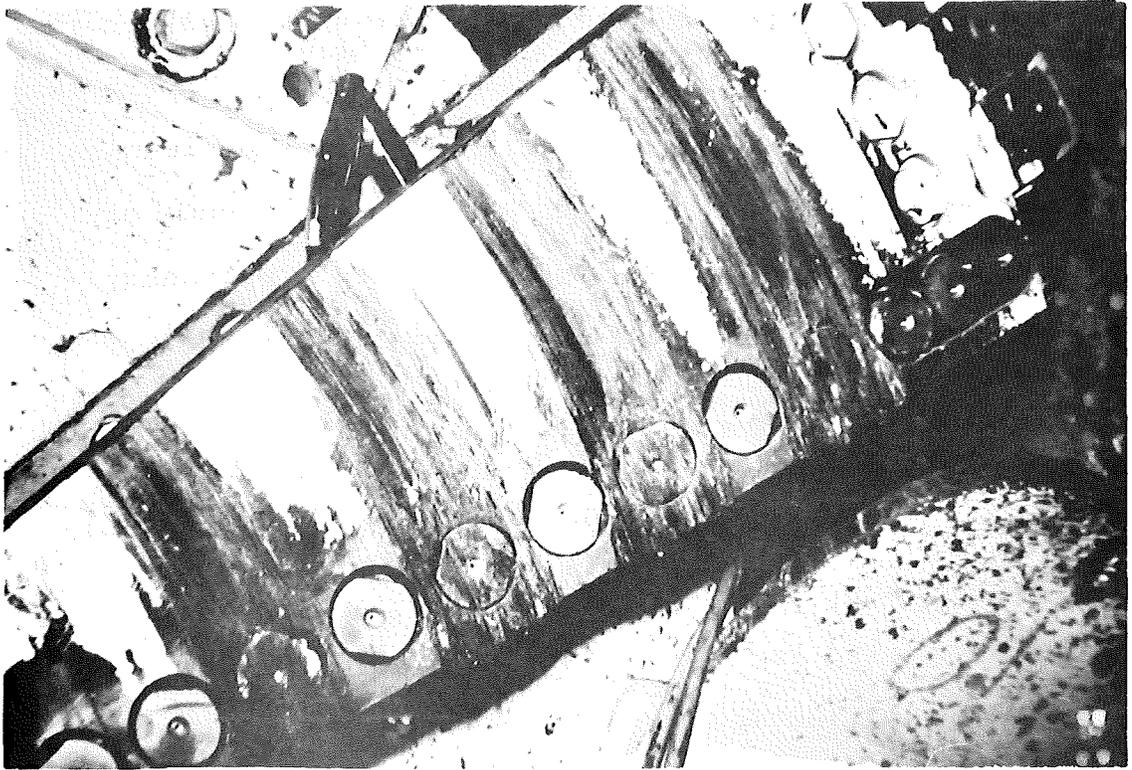
Many of the problems were reduced by strengthening components of the system and installing a standpipe to dampen the hydraulic hammer effects. The overheating problem was eventually controlled by installing larger heat exchangers on a separate cooling circuit.

The anticipated problem with the intensifiers, namely the seals, turned out not to be of major significance. The most consistently occurring problems were for an intensifier

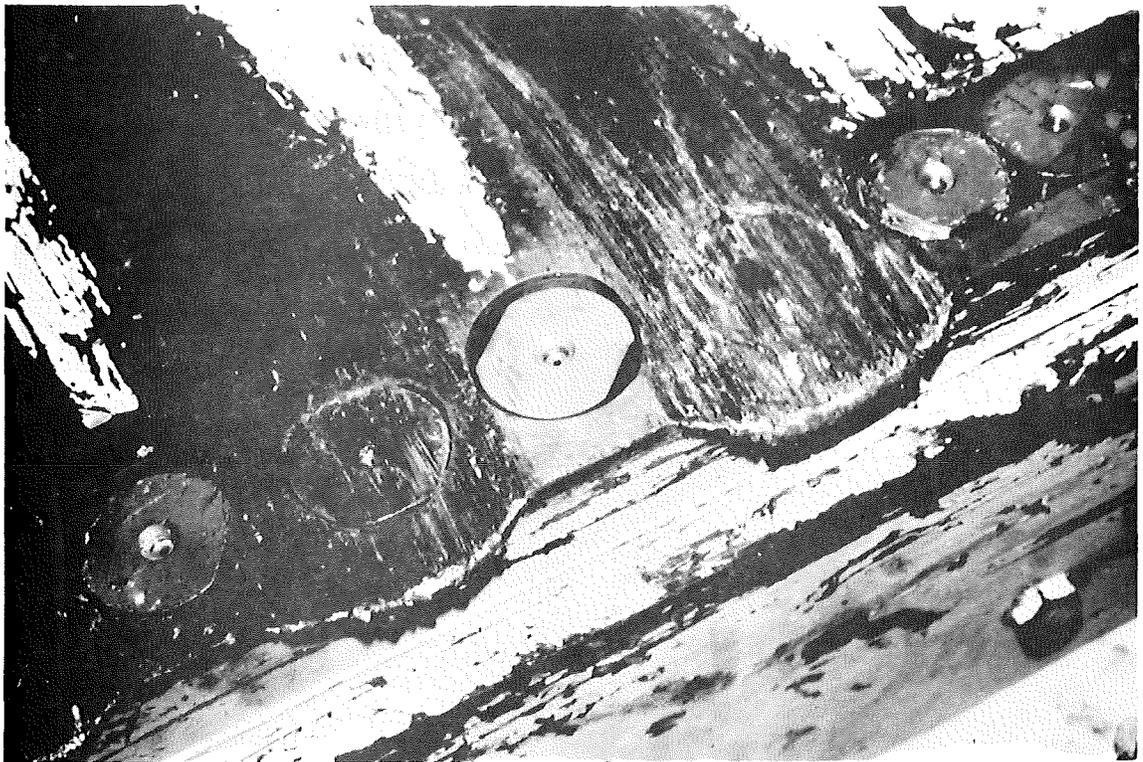
to stop cycling during a test, or for a check valve to become faulty. The failure to cycle was caused by improper lengths of shifting cable and was eliminated. The check valves were easy to replace, but design improvements should be pursued.

The problems that caused major delays were two broken high-pressure pistons and a stretched low-pressure cylinder. It was determined the piston failures were caused by improperly supporting the piece during heat treating. Two replacement pistons had to be fabricated, with a spare being ordered should any of the remaining pistons fail. Analysis of the stretched cylinder showed the wall thickness to be below specifications; therefore, a new cylinder and spare were ordered. While the replacement parts were being fabricated, some testing was able to be conducted by testing with fewer nozzles or lower pressures. Toward the end of the field work, leakage in the connection from the swivel to the large manifold began to be a problem. This would sometimes take one or two days to repair, but it was felt the problem could be eliminated in the future.

The manifolds survived remarkably well considering their exposed position on the cutterhead. One-inch standoff of the jets was found to be too close (see Figure 23) due to the high ridges between cutter paths resulting from the wide cutter spacing used. When smaller protective cutters were installed to operate in the area of wide spacings, the manifold was successfully returned to one-inch standoff.



approx. 1/3 scale



approx. 1/2 scale

Figure 23 - Damage Caused by High Ridges Between Cutters

Occasionally the nozzle connections would work loose in the manifolds, necessitating occasional tightening. In the future, some means of locking the connections would be desirable.

The nozzles performed well, with the occurrence of plugging diminishing as the job progressed. Apparently, slivers of metal entered the high-pressure lines during assembly of various connections. Through careful assembly and by making connections with low-pressure water running in the lines, the incidence of plugging was greatly reduced. Discussions later indicated a high-pressure filter could be easily fabricated for use right at the cutterhead, thus effectively eliminating jet plugging problems.

In summary, although a large number of problems occurred, it was an untried prototype system where numerous problems are expected. The problems identified here either have been or can be effectively mitigated to provide a usable production system. It would appear then that the basic concept of combining a water jet capability with conventional mechanical excavation equipment is definitely possible.

4.0 RESULTS

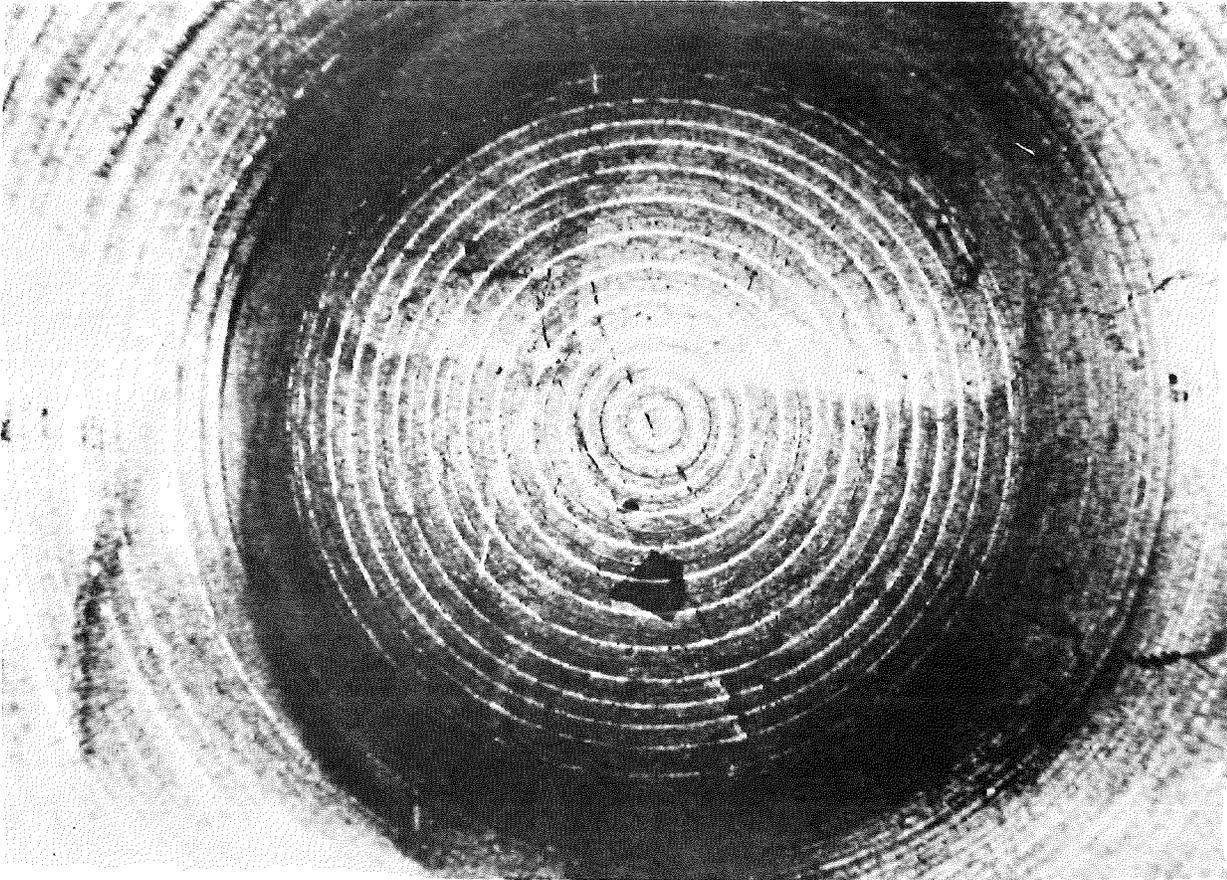
Field tests were performed of water jet assisted rock boring. Tested were effects of jet pattern, number of jets, jet location, jet size, jet pressure, and machine thrust.

Only the configuration of jets between cutter paths was found to be effective in the field. The configurations of jets under the cutters and under and between cutters actually reduced advance rates. After determining that jets between cutters was the only successful pattern, the majority of testing was restricted to that configuration.

Thrust was found to have a significant effect on penetration rate both for the mechanical and water jet assisted boring. Wide variations in boring performance were, however, observed at any given thrust. These variations are expected since such factors as rock strength, presence of planes of weakness, and variations in machine performance will affect the penetration rate for each test. Some of the changing rock conditions are illustrated in Figures 24 and 25.

To determine the benefit of water jet assisted rock cutting, while reducing the effect of some of the test-to-test variation, the differences between each pair of water jet and unassisted mechanical tests were averaged for each thrust level. This resulted in estimated average improvements in advance rates of 54%, 50% and 60% at thrust levels of 3,000 psi, 3,500 psi, and 4,000 psi respectively.

Larger nozzles were tried in an attempt to further



approx. 1:21 scale

Figure 24 - Tunnel Face Showing Planes of Weakness
and a Large Inclusion at Lower Center



APPROX 1/2 SCALE

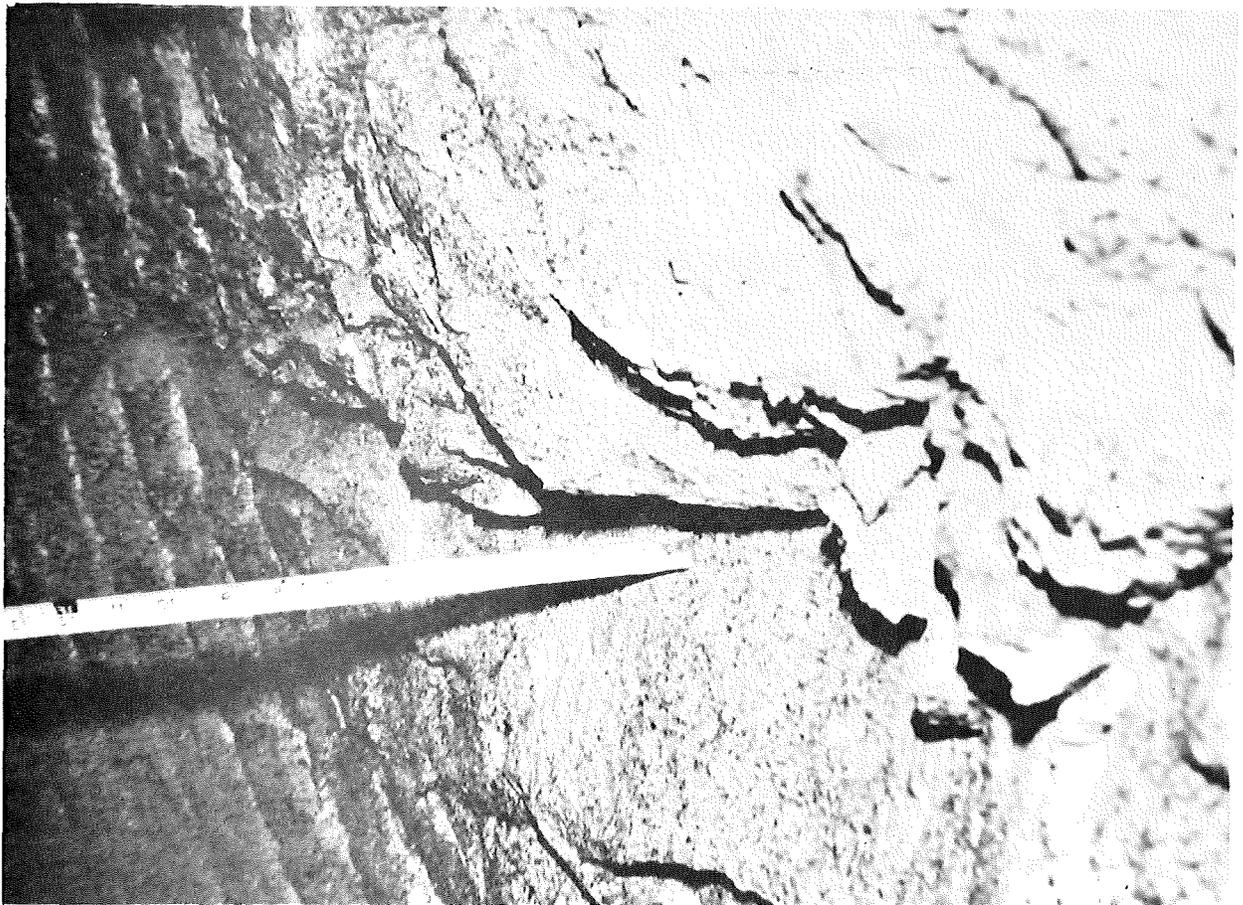


Figure 25 - One of the More Severe Zones of Fractured Rock Encountered

increase rock excavation rate, but limitations resulting from nozzle holder geometry apparently prevented the expected increase in jet slot depth. Double nozzles were also tested, but time limitations prevented any definitive conclusions to be drawn on their effectiveness.

Laboratory tests had indicated a definite variation in jet performance for variations in jet pressure. In the field, however, performance improvement was observed to be much less affected by changes in water jet pressure. Again, scatter in the data may have effectively masked any effect.

A surprising result of the water jet testing was a significant reduction in machine torque as can be seen in Figure 26. Since the torque required to actually cut the rock should not significantly change, the reduction in torque is believed due to reduced friction on the cutterhead.

The field data and a more detailed analysis of the data can be found in the Final Report to the National Science Foundation (1) on Contract No. APR-7421784 A02.

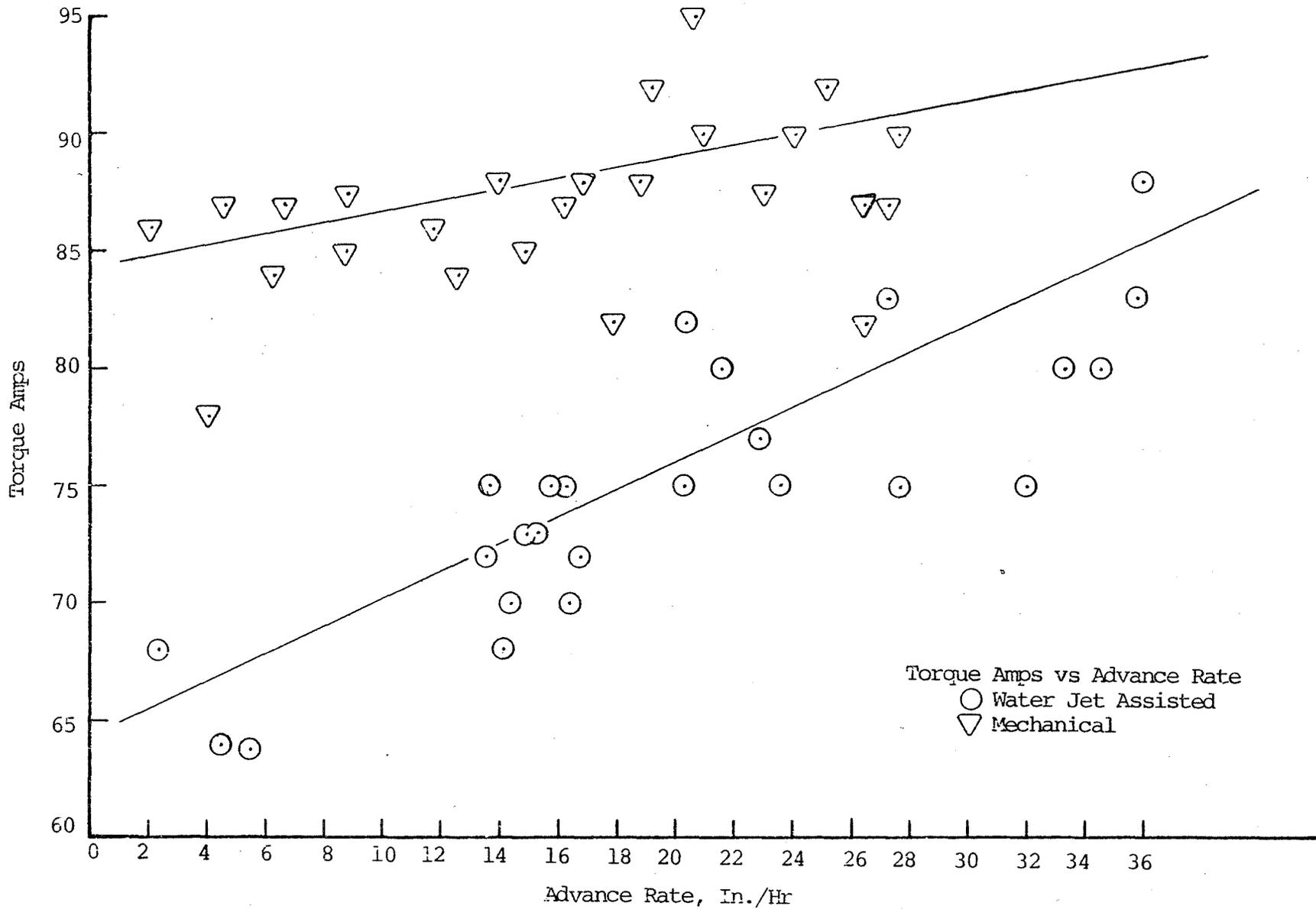


Figure 26 - Torque Amps Vs. Advance Rate, Water Jet and Mechanical

5.0 CONCLUSIONS

The effective application of a team effort was successful in accomplishing the objectives of the research program. High-pressure equipment suitable for use in a rock excavation environment was designed, constructed, and tested. This included high-pressure intensifiers for developing up to 56,000 psi water pressure, a large flow swivel to transfer the high-pressure water to a rotating cutterhead, and a manifold system for distributing the high-pressure water along the cutterhead, location of the nozzles, and protection of the nozzles.

Prototype development problems occurred, but there were no problems that would prevent the successful application of water jet assisted excavation of rock. The average increase in advance rate obtained during field testing was from 50 to 60 percent. For some tests, advance rates were more than doubled.

The rock of the test site was a hard, tough granite. However, water jet kerfing tests in other rock types indicate the technique of water jet assisted mechanical cutting should be equally applicable to softer rocks.

An interesting observation during the field tests was an approximate 25 percent reduction in required torque when the water jets were used. It was proposed this was a result of reduced friction on the cutterhead.

Three different jet patterns were tested, all of which

proved effective in assisting mechanical cutting in the laboratory. In the field tests, only the configuration of jets between cutters was effective. The configurations of jets under the cutters and jets under and between cutters actually reduced machine advance rate.

Various nozzle sizes were used in the field. The 0.012 inch nozzle was found to cut the best. Larger sizes were tried but found to be less effective, possibly due to hydraulic characteristics of the nozzle holders used. Multiple nozzles in a single nozzle holder were tested in the laboratory and to a limited extent in the field. Although cutting deeper than a single nozzle, interference caused by the close proximity of the jets caused the double nozzle to be less effective than two separate passes of a single nozzle in the same slot. Since multiple nozzles appear to be the best way to apply larger amounts of power to a particular location, studies are needed to determine minimum separation distances required between jets of a multiple nozzle and how to design and place this multiple nozzle on a boring machine.

Variations in performance with changes in water pressure were fairly pronounced in the laboratory tests, but the test to test variations encountered in the field tended to mask any detectable effect for the field tests. Since the level of pressure chosen affects the power required, an understanding of the effect of pressure combined with other variables of jet operation would be very valuable in selecting operating pressures in an optimally designed system.

Sieve analysis of the cuttings from both the water jet and unassisted mechanical tests showed very good correlation to expected size distribution. The presence of jet slots between cutters should effectively reduce the size of the chips on the order of the cutter spacings in half. This was indeed observed for the samples from tests with jets between cutters. Also, the occasional chips coming out with visible jet slots on them indicated that for wider spacings of cutters, deeper slots may be necessary than are required for smaller spacings. The relative distributions of larger chips was seen to vary with penetration rate, there being more chips over 2 inches and more fines for the lower penetrations.

Finally, to determine if the improvements offered by water jet assisted tunneling warranted the additional expense of such a system, a comprehensive economic analysis was undertaken (1). Very significant savings were seen to be possible, depending primarily on the percentage improvement assumed. A breakeven increase of approximately 10 percent was found, with savings increasing rapidly for greater improvements in advance rate. Leveling off of cost benefits occurred for large increases in advance rate. Using a 50 percent increase in advance rate cost savings of 14 and 24 percent were projected for 10-foot and 20-foot diameter tunnels respectively. With the optimization of equipment and machine design, even higher increases in advance rate than obtained in the prototype test should be possible, with corresponding increases in cost savings.

6.0 REFERENCES

1. Wang, F. D., R. J. Robbins, and J. H. Olsen, "Water Jet Assisted Tunnel Boring," Final Report to the National Science Foundation, Contract No. APR 7421784 A02, February, 1976, 178 pp.

APPENDIX A
TUNNEL BORING MACHINE SPECIFICATIONS

MACHINE GENERAL SPECIFICATIONS

MODEL 73-114 TUNNELING MACHINE

General Description

The Robbins' tunneling machine used on this project was a Robbins' rotary rock tunneling machine. The machine consists of three main structural components; the cutterhead, the cutterhead support and main beam, and the gripper and thrust assembly. Other components include electric drive motors with gear reducers, bearing and ring gear assembly, side supports, operator's controls, muck buckets, conveyors and electrical and hydraulic equipment. The cutterhead is fitted with housings for Robbins' disc cutters.

Cutterhead

The cutterhead is a heavy steel plate weldment. It is supported on anti-friction main bearings. The cutterhead is driven by two gear motors through a bull gear. Mounted on the periphery of the cutterhead are the buckets which scoop up the muck and deposit it on the machine conveyor inside the cutterhead support.

Disc cutter assemblies mount in individual housings welded to the cutterhead. The cutter housings accept the cutter assemblies from the front. Cutters are secured in the housings with four cap screws.

Cutterhead Support and Main Beam

The cutterhead support and main beam form the structural support for the cutterhead and propulsion system. The cutterhead support is a heavy weldment which mounts the main bearings, gear motors and front side supports, and dust shield.

Main Bearing, Bull Gear and Gear Reducers

The machine uses a large ball type main bearing with integral

bull gear. The bearing and bull gear chamber is completely sealed and pressure lubricated.

The cutterhead is driven electrically through two planetary gear reducers. These are in line reducers with a drive pinion on the output shaft. The reducers mount in machined openings in the cutterhead support.

Front Side Supports

Two front side supports are provided to stabilize the machine. The position of each support is controlled by a hydraulic cylinder and a mechanical link.

Muck Conveyor

The tunnel machine is provided with a troughed belt conveyor which transfers the muck from inside the cutting head back to the rear of the main beam. Here the muck is transferred to the secondary transfer conveyor which moves it over the top of the operator's skid and delivers it to the back up muck handling system.

Gripper and Thrust Assembly

The gripper and thrust assembly consists of structural carrier, a gripper assembly and the thrust cylinders. The carrier mounts on top of the main beam on machined ways. The wall gripper assembly is connected to the carrier by a trunnion. Cutterhead torque is reacted by the wall gripper through the carrier and the main beam and cutterhead support. The machine is allowed to move forward in the carrier on the main beam ways.

The gripper mechanism, when hydraulically energized, forms a beam spanning the tunnel at springline. The anchoring force provides sufficient reaction to resist the torque, thrust and steering forces. It also supports the dead weight of the rear of the machine when the rear support is raised during the boring cycle. The thrust cylinders

are anchored to the gripper shoes and thrust on the cutterhead support.

Steering

While providing rigid support for both torque and thrust reaction the grippers also provide the primary steering forces. The gripper cylinder floats allowing steering while boring. With this steering system line and grade changes can be made during the boring cycle in small increments for smooth curves without overloading gage cutters, providing line and grade correction at any time.

The gripper cylinder is connected to the main beam through a trunnion. Steering is accomplished by changing the position of the rear of the machine. The pivot point is at the center of the cutterhead support and the rear of the machine is moved either sideways or up or down to change the heading of the machine as desired. This move is accomplished by transferring oil from one half of the gripper cylinder to the other (unbalancing) for lateral steering or by lifting or dropping the rear of the machine with the vertical steering cylinders for vertical movement.

Electrical

The cutterhead drive motors were 460 volt, single speed motors. Current transformers in each motor feed are connected to ammeters on the operator's console.

Motors are fed from a panelboard containing circuit breakers and magnetic starters for each motor.

Controls are operated at 120 volts and are mounted at the operator's console. Lighting fixtures are 120 volts and are mounted to give good visibility for operation and servicing.

Hydraulics

Industrial class hydraulic equipment is used. Maximum pressure is

4,000 psi but normal operating pressure is 3,000 psi. This was modified to 5,000 psi maximum.

A low pressure - high volume system provides high volume oil for a fast regripping cycle.

MODEL 73-114ROCK TUNNEL BORING MACHINESPECIFICATIONS, FEATURES, AND OPERATING CYCLE

| | |
|-------------------------------|---|
| Machine Diameter | 84 inches |
| Cutters | Robbins Disc Cutters |
| | 12 - 11" |
| | 4 - 12" |
| | 5 - 8½" |
| Cutter Edge Angle | 105 ⁰ , 90 ⁰ |
| Cutter Bearing Capacity | 25,000 lbs. + |
| Rotation Power | Two 75 hp A-C Motors and Gear Reducers |
| Torque | 86,568 ft. lb. max. |
| Hydraulic System | Pumps - 15 hp, 5000 psi max. |
| Propel Cylinders | Four at 5" dia. |
| Propel Force | |
| @ 5000 | 400,000 lbs. |
| @ 4000 | 320,000 lbs. |
| @ 3000 | 240,000 lbs. |
| Stroke | 24 inches |
| Length of Machine | 28' 7½" |
| Length of Drive Head Assembly | 4' 4 3/4" |
| Front Length | 15' 9¼" |
| Rear Section | 12' 10" |
| Belt Width | |
| Machine Conveyor | 14" |
| Belt Width | 18" |

| | |
|------------------------|---|
| Belt Speed | 200 FPM |
| Conveyor Drives | 5 hp |
| Conveyor Capacities | 13 cu. yd./hr. |
| Gripper Cylinders | 8" Bore, 8" Stroke |
| Gripper Force Max. | 500,000 lbs. |
| Gripper Shoe | 30" wide x 39½" high |
| Gripper Pressure | 60,000 lbs./sq. ft. |
| Side Support Cylinders | 4" Bore x 5" Stroke |
| Electric System | Motor Circuit - 460 volt, 3 phase, 60 hp, Controls and Lighting 120 volt, 60 hp |
| Machine Weight | Approximately 23 tons |