**Fabrication, Assembly, Bench and Drilling Tests
of
Two Prototype Downhole Pneumatic Turbine Motors**

Final Technical Report

**Principal Investigator: William C. Lyons
Pneumatic Turbine Partnership
Santa Fe, New Mexico**

Authors

**Robert Bookwalter
Patricia D. Duettra
Philip Johnson
William C. Lyons
Stefan Miska**

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TABLE OF CONTENTS

List of Figures	vi
List of Tables	viii
Acknowledgments	ix
Executive Summary	xi
1. Introduction	1
2. Fabrication and Assembly of the First Prototype Downhole Motor	19
2.1 Thrust Assembly	19
2.2 Turbine Motor and Gearbox Assembly	21
2.3 Bypass Assembly	22
2.4 Assembly of the First Prototype Motor	24
3. Bench Tests of the First Prototype Downhole Motor	26
3.1 Gearbox Run-in	30
3.2 Temperature Tests	32
3.3 Downhole Motor Test	33
4. Shallow Hole Drilling Test with the First Prototype Downhole Motor	36
4.1 Location and Equipment	36
4.2 Baseline Drilling Data	38
4.3 First Day Testing	42

4.4	Second Day Testing	45
5.	Fabrication and Assembly of the Second Prototype	
	Downhole Motor	48
5.1	Thrust Assembly	48
5.2	Turbine Motor and Gearbox Assembly	49
5.3	Bypass Valve Assembly	50
5.4	Assembly of the Second Prototype Motor	51
6.	Bench Tests of the Second Prototype Downhole Motor	52
6.1	Gearbox Run-In	52
6.2	Temperature Tests	54
6.3	Downhole Motor Test	56
7.	Bench Test Results	58
7.1	First Prototype	63
7.2	Second Prototype	70
7.3	Comparison of Bench Test Results	77
8.	Drilling Test Results	80
8.1	Conventional Rotary Drilling Results	80
8.2	Downhole Motor Tests on the Rotary Table	85
8.3	Drilling Test Results	96
8.4	Comparison of Bench Test Results and Drilling	
	Test Results	105
9.	Proposed Design Changes for All Three Prototype	
	Downhole Motors	106
10.	References	109

Appendix A	Bench Test Data for First Prototype	110
A.1	Details of Compressor Calculations	111
A.2	Gearbox Run-In Data	112
A.3	Bench Test Data,	118
Appendix B	Bench Test Data for Second Prototype	124
B.1	Temperature Test Data	125
B.2	Lock-Rotor Torque Data	130
B.3	Bench Test Data	135
Appendix C	Drilling Test Data for First Prototype	165
Appendix D	Bench Test Programs	167
D.1	Program TTST	168
D.2	Program MAIN	175
Appendix E	Drilling Test Programs	190
E.1	Program SURF_VLV	191
E.2	Program FIELD_PR	196

LIST OF FIGURES

Figure 1	Typical Drill String with Downhole Pneumatic Turbine Motor	3
Figure 2(a)	Assembly Drawing of the Bit Sub	4
Figure 2(b)	Assembly Drawing of the Thrust Assembly	5
Figure 2(c)	Assembly Drawing of the Lower Main Housing with the Gearbox	6
Figure 2(d)	Assembly Drawing of the Upper Main Housing with the Turbine & 1st Stage of the Gearbox	7
Figure 2(e)	Assembly Drawing of the Bypass Assembly	8
Figure 3	Thrust Assembly	10
Figure 4	Turbine Motor and Gearbox Assembly	11
Figure 5	Bypass Valve Assembly	12
Figure 6	Turbine Motor Module	13
Figure 7	Turbine Motor Rotor and Vanes	14
Figure 8	Gearbox Module	16
Figure 9	Disassembled Gearbox with Components and Housing	17
Figure 10	Key-Way Thread Locks	25
Figure 11	Test Stand	27
Figure 12	Instrument Platform	28

Figure 13	Brake System (12" diameter) and Torque Measuring Arm	29
Figure 14	Assembly Drawing of Test Stand	31
Figure 15	Drilling Rig	37
Figure 16	Downhole Motor after Drilling	44
Figure 17	Brake System (20" diameter), Torque Tension Cable, and Load Cell Measuring Device	53
Figure 18	Expected Dimensionless Performance Curve	59
Figure 19	Expected Particular Performance Curve: 3600 SCFM, 50 psig	62
Figure 20	Experimental Dimensionless Performance Curve	65
Figure 21	Experimental Particular Performance Curve: 3600 SCFM, 50 psig	66
Figure 22	Dimensionless Torque Vs Utilization Factor	71
Figure 23	Expected Dimensionless Performance Curve	72
Figure 24	Experimental Dimensionless Performance Curve	74
Figure 25	Stall Torque vs Air Flow Rate	92
Figure 26	Peak Horsepower vs Air Flow Rate	93
Figure 27	Injection Pressure vs Air Flow Rate	94
Figure 28	Horsepower vs RPM, 1800 SCFM	100
Figure 29	Horsepower vs RPM, 2400 SCFM	101

LIST OF TABLES

Table 1	Predictions and Results for the First Downhole Motor Bench Tests	64
Table 2	Predictions and Results for the Second Downhole Motor Bench Tests	75
Table 3	Conventional Water Drilling Test Results	81
Table 4	Conventional Air Drilling Test Results	82
Table 5	Data from the Motor Test on the Rig Floor	86
Table 6	Results of Pressure Calculations on Rig Floor Test Data	89
Table 7	Predicted vs Calculated Values for Pressure Calculations	90
Table 8	Numerical Summary of Stall Torques, Horsepower, and Injection Pressures for Various SCFM	95

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solutions to the problems that are part of the fabrication, assembly and testing of any complicated mechanical system, such as the downhole pneumatic turbine motor.

EXECUTIVE SUMMARY

The first and second prototype downhole pneumatic turbine motors have been fabricated, assembled and tested. The first prototype downhole motor was bench tested at the Clay Groomer Machine Shop, Farmington, New Mexico. This bench test was performed in the test stand with 1200 SCFM to 3600 SCFM of compressed air supplied to the motor; the second prototype was tested with 1200 SCFM to 3600 SCFM of compressed air. All bench tests showed that the motor will produce horsepower and bit speeds approximating the predicted values. Specifically, the downhole pneumatic turbine motor produced approximately 50 horsepower at 100 rpm, while being supplied with about 3600 SCFM of compressed air.

The first prototype was used in a drilling test at the Bayless Drilling Company yard on La Plata Highway, west of Farmington, New Mexico; this test was performed using a 10 ⁵/₈ inch bit. The motor drilled from a depth of 389 feet to a depth of 789 feet in the Kirtland formation. This first prototype motor drilled at a rate exceeding 180 ft/hr, utilizing only 3000 SCFM of compressed air. This drilling rate is over three times the rate of penetration recorded earlier in this hole while drilling conventionally (rotating from the surface) with water as the

circulating fluid, and about twice the rate recorded in this hole while drilling conventionally with air as the circulating fluid.

High temperature tests (at approximately 460°F) were carried out on the thrust assembly and the gearboxes for the two prototypes. These components operated successfully at these temperatures.

Although the bench and drilling tests were successful, the tests revealed design changes that should be made before drilling tests are carried out in geothermal boreholes at the Geysers area, near Santa Rosa, California. These recommended design changes are:

- The torque transfer connection between the housing of the turbine and the main housing should be strengthened.
- The torque transfer connection between the gearbox housing and the main housing should be strengthened.
- A ratcheted system or a clutch system should be placed in the thrust assembly to prevent or restrict backward rotation of the main shaft when the drill string is rotated from the surface.
- The bypass valve should be modified to prevent water and cuttings from entering the supply air stream to the turbine motor during connection operations.
- The air flow passage around the gearbox should be modified to allow more aerodynamic air flow and thus increase the

horsepower output of the turbine motor.

In general, the first two prototype motors operated well under simulated maximum torque loads during both the bench tests and during the drilling test. The first prototype survived totally intact, so that with the design changes recommended above, the first prototype can join the second and third prototypes in the follow-on field testing at the Geysers area (or elsewhere).

1. Introduction

The objective of this research and development project was the fabrication, assembly and testing of a downhole pneumatic turbine motor. This technology is patented through U.S. Patent Nos. 4,434,862 and 4,553,611.

The initial intended application for this technology is the drilling of directional geothermal boreholes at the Geysers area near Santa Rosa, California. The bottom 2000 foot section (approximately from 6000 feet to 8000 feet) of these boreholes must be drilled with air in order to eliminate formation damage in the steam producing formations. In such boreholes, the downhole pneumatic turbine motor must operate successfully in a temperature environment of up to 460°F. The downhole pneumatic turbine motor also has applications in hot borehole drilling at the Nevada Test Site, and in specific oil and gas drilling operations.

The downhole pneumatic turbine motor has been developed to operate on surface supplied compressed air with total volumetric flow rates from 2400 SCFM to 3600 SCFM. The motor will deliver from 25 to 60 horsepower to the bit, at bit rotation speeds of 60

to 120 rpm. This motor is specifically designed to drill in the hard metamorphic rock of the Geysers area, where the weight on bit could be as high as 50,000 lbs.

The downhole pneumatic turbine motor is designed to be used with conventional drill string equipment. Figure 1 shows the typical placement of the downhole motor in the drill string.

The downhole pneumatic turbine motor is composed of three basic assemblies (from the bit up): the thrust assembly with the bit sub, the turbine motor and gearbox assembly, and the bypass valve assembly (see Figures 2(a) through 2(e)).

In general, the downhole pneumatic turbine motor is driven by a small, single-stage, edge-entry pneumatic impulse turbine motor, which is located in the turbine motor and gearbox assembly. This motor is actuated by compressed air supplied by surface compressors via the drill string (i.e., the drill pipe and collars) to the upper end of the downhole motor.

The turbine motor converts the expansion of the compressed air into mechanical energy. This mechanical power from the output shaft of the turbine motor is carried to the input shaft of the gearbox via the upper dogbone connector. The mechanical power is

Figure 1
Typical Drill String with Downhole Pneumatic Turbine Motor

Figure 2(a)
Assembly Drawing of the Bit Sub

Figure 2(b)
Assembly Drawing of the Thrust Assembly

Figure 2(c)
Assembly Drawing of the Lower Main Housing with the Gearbox

Figure 2(d)
Assembly Drawing of the Upper Main Housing with the Turbine
and the 1st Stage of the Gearbox

Figure 2(e)
Assembly Drawing of the Bypass Assembly

transferred through the gearbox to its output shaft, then via the lower dogbone connector to the main shaft in the thrust assembly. The main shaft through the thrust assembly transfers this power directly to the drill bit.

The expanded air, after passing through the turbine motor, continues inside the main housing, past the upper dogbone, the gearbox, and the lower dogbone, and then enters the inside of the main shaft; it exits through the bit nozzles where it picks up and entrains the bit cuttings, carrying them to the surface through the annulus of the borehole.

The thrust assembly is shown in Figure 3, the turbine motor and gearbox assembly in Figure 4, and the bypass valve assembly in Figure 5.

Figure 6 shows the turbine module with the upper dogbone. The turbine motor module is placed into the main housing of the assembly with the upper dogbone connector and the gearbox module. Figure 7 shows the details of the rotor and vanes of the turbine motor module. This turbine motor is a unique edge-entry design that operates at peak power at about 20,000 rpm. Because of the edge-entry design, the turbine motor is able to operate successfully with significant amounts of

Figure 3
Thrust Assembly

Figure 4
Turbine Motor and Gearbox Assembly

Figure 5
Bypass Valve Assembly

Figure 6
Turbine Motor Module

Figure 7
Turbine Motor Rotor and Vanes

solid particles (drill pipe scale) and water particles (mist drilling) entrained in the compressed air flow through the single rotor.

Figure 8 shows the gearbox module. The gearbox is composed of four stages of planetary gear reductions, the total reduction in speed through the gearbox being 168:1. Thus at peak output power at the turbine rotor (i.e., about 20,000 rpm), the output shaft of the gearbox, and therefore the bit, speed will be about 120 rpm. The gearbox module is a sealed unit and contains its own oil lubrication system. Figure 9 shows the disassembled gearbox with components and housing.

The turbine motor module was designed and fabricated by Bear Engineering Company, West Roxbury, Massachusetts. The gear transmission train was designed by Arthur Williams, Burbank, California, and the gears fabricated by the Advance Gear and Machine Shop, Incorporated, Gardena, California. The gearbox module housing and gear packaging were designed by the Pneumatic Turbine Partnership, and the final fabrication and assembly of the gearbox was carried out at the Clay Groomer Machine Shop, Incorporated, Farmington, New Mexico.

Figure 8
Gearbox Module

Figure 9

Disassembled Gearbox with Components and Housing

The thrust assembly and the bypass valve assembly were designed by the Pneumatic Turbine Partnership; fabrication and assembly of these components also took place at the Clay Groomer Machine Shop.

This report describes the details of fabrication, assembly and testing of two prototype downhole pneumatic turbine motors. The fabrication and assembly of these two prototypes began in early summer 1986; the bench tests of the first prototype started in January 1987, and the drilling test was performed in March 1987. The bench tests of the second prototype started in early April 1987.

2. Fabrication and Assembly of the First Prototype Downhole Motor

The fabrication and assembly of the first prototype downhole turbine motor began in early summer 1986. Most of the components of this motor were completed by early fall 1986. The major delay in the completion of the prototype was due to the fact that the gearbox was not finished on schedule. In early January 1987, the Pneumatic Turbine Partnership took control of the remaining fabrication and assembly operations for the gearbox. The first prototype gearbox was completed and ready for bench testing in late January 1987.

2.1 Thrust Assembly

The thrust assembly contains the main shaft which rotates the bit and transfers the weight on bit from the drill collars through the housing of the motor to the rotating shaft. The thrust assembly has a large radial bearing near the bit to keep the lower end of the shaft rotating about its center line. Above this radial bearing is the main thrust bearing which carries the entire weight on bit load when the motor is in operation and

drilling. Above the main thrust bearing is a series of five radial bearings to ensure the concentricity and stability of the rotating main shaft. The top bearing in the assembly is a combined radial thrust bearing which allows the main shaft to be rotated while pulling out of the borehole under load.

Lubrication for these bearings is provided by a high temperature grease that is packed around all bearing surfaces. The grease is held in place by two sets of seals located below and above the bearings. The main shaft and its bearing are held in position by a large nut which is threaded onto the top of the main shaft and locked in place.

During the initial assembly of the thrust, difficulties were encountered in the installation of the lower seals where the seals slip over the lower inner bearing race. Spacer rings were heat fitted above and below the race, and the upper portion of the thrust ring was tapered.

The main shaft was then installed by cooling it in liquid nitrogen to a temperature of approximately -300°F , allowing it to slip through the inner diameters of the bearings and seals of the thrust assembly. After the temperatures had normalized, the system rotated relatively easily, and upon disassembly no damage

was found to either the seals or the bearings.

2.2 Turbine Motor and Gearbox Assembly

The components of the turbine motor module were fabricated in the Boston, Massachusetts, area. The module was assembled by Bear Engineering Company and shipped to New Mexico for incorporation into the main housing of the turbine motor and gearbox assembly. The turbine motor module components are fabricated from 17-4 PH stainless steel. The bearings on the rotor shaft are lubricated with high temperature grease; seals are placed so as to contain the grease.

The components of the gearbox were fabricated by the Advance Gear and Machine Shop in Gardena, California, and by the Clay Groomer Machine Shop in Farmington, New Mexico. The gears and the planetary carrier shafts are fabricated from special steel and are capable of operating at very high temperatures when properly lubricated with the appropriate oil.

The housing of the first gearbox was heat treated and nitrided after the internal gears had been cut in the interior surface of the housing. The heat treating caused a slight warpage of the housing which made assembly of the first gearbox difficult. This

problem has been eliminated in succeeding gearbox housings by only nitriding the housings after the internal gear cutting. There were many small tolerance fit problems that had to be resolved as the gearbox was assembled. Despite the complexity of this module and the numerous small corrections that were made, the gearbox was assembled very successfully, indicating that no major problems existed.

The gearbox is sealed at both ends, and all bolt openings through to the housing are sealed, to prevent leakage of the lubricating oil.

The main housing of this assembly was fabricated at the Clay Groomer Machine Shop. Once the turbine motor module and the gearbox module were available, the assembly of this major component of the downhole motor was completed without problems.

2.3 Bypass Assembly

The bypass valve assembly was fabricated and assembled at the Clay Groomer Machine Shop. The bypass valve assembly of the pneumatic motor enables air to be vented from the interior of the motor to the annulus of the drilled hole. By venting air to the annulus, the performance of the turbine is greatly reduced, thus

slowing the rotational speed of the bit and preventing turbine motor runaway when there is no weight on the bit.

The bypass consists of a sliding piston inside a housing, a compression spring, and venting ports. When weight is applied to the downhole motor, the spring is compressed, causing the piston to slide up and close the venting ports. As the weight is released, the spring pushes the piston down, opening the ports.

The assembly of the bypass section was successful, but the sliding action of the piston would not travel the desired length. First, the "O"-ring grooves were deepened to allow more clearance in the sliding action of the rings against the piston wall. It was then found that a pocket of air was being trapped between the two sets of seals and being compressed to the point that no further travel was permitted. This problem was solved by drilling a small hole in the side of the piston between the two sets of seals to allow the compressed air to escape to the inner portion of the motor.

The bypass assembly was then tested; it required approximately 500 lbs of force (the design force in the spring) to fully position the seals. Under the relatively low pressure used (50-100 psi), the system sealed very well.

2.4 Assembly of the First Prototype Motor

There are two field connectors on the downhole motor, one at each end. These are 6 ⁵/₈ inch API Regular threaded connections (standard oil field tool joint connections). All 10 ⁵/₈ inch diameter bits through 13 ¹/₂ inch diameter bits have 6 ⁵/₈ inch API Regular threaded shanks, which can mate directly to the box connection at the bottom of the motor. The top connection on the motor is also a box to facilitate mating to the drill collars. These two field connections can be made up in the field on the rig floor, where power tongs can be used to tighten the threaded joints.

There are three threaded connections that join the various assemblies together; these are stub acme threads. These threads are of a finer pitch and are not field connections (i.e., they are made up in the shop). They are made up with about 10,000 ft-lbs of torque and are then locked with a key-way device; the key-way prevents the high torques in the drill string during drilling from causing further tightening of the joints. Figure 10 shows the key-way device for these threaded connectors.

Figure 10
Key-Way Thread Locks

3. Bench Tests of the First Prototype Downhole Motor

Figure 11 shows the test stand which was designed both for running in the gearbox and for stack testing the entire downhole motor under simulated downhole conditions. The test stand, located at the Clay Groomer Machine Shop, is approximately 25 feet high and is equipped with an instrument platform where the compressed air flow to the motor can be controlled and the various pressure gauges and rotation meters can be conveniently read (see Figure 12). The stand can accomodate the gearbox alone for run-in tests or the entire downhole motor for full stack testing.

When the gearbox is run in, an auxiliary air activated turbine motor is used to turn the gearbox input shaft. A brake system with a torque measuring arm is used to provide back torque to the gearbox and to allow this back torque to be measured. Figure 13 shows the brake system, with a 12 inch diameter brake disk, and the torque measuring arm.

When the entire downhole motor is tested, the downhole motor output shaft (i.e., the bit end of the motor) is connected to the brake system, and the torque is measured in the same way as for

Figure 11
Test Stand

Figure 12
Instrument Platform

Figure 13
Brake System (12" Diameter) and Torque Measuring Arm

the gearbox run-in. For this test, a 16 inch diameter, 65 lb/ft (15.25 inch inside diameter) casing is used to enclose the downhole motor. The casing allows the back pressure in the system to be varied while testing. This is necessary to simulate the variety of downhole annular conditions to which the motor is likely to be exposed. The test stand can accomodate tests both at atmospheric conditions and at elevated temperatures. Figure 14 shows a detailed drawing of the test stand with the downhole motor in place.

3.1 Gearbox Run-in

The purpose of the gearbox run-in is to burnish any asperities on the hard gear teeth, so that, at high temperature operation, there will be less possibility of scoring on the teeth. The first gearbox was run in for a total of about five hours at increasing speeds with increasing back torque. Since the gearbox was enclosed in a protective casing, the power loss created by the oil pump in the gearbox caused the temperature of the gearbox to increase as the run-in continued. Therefore, the run-in was stopped periodically to allow the system to cool.. The run-in continued until a steady rotation speed of 15,000 rpm with a backload of about 10 HP was reached (see Appendix A, Section A.2 for the data from this gearbox run-in).

3.2 Temperature Tests

The insulating effect of the protective casing allowed the gearbox to be tested at temperature. During the 15,000 rpm run, the thermocouples attached to the external housing of the gearbox showed the external temperature rising to approximately 330°F. This external temperature indicated that the internal temperature in the gearbox should have been at or above 365°F. After the high temperature test, the gearbox was disassembled; it was found that bearings on one of the planets on the first stage planetary system had failed. One of these bearings had turned blue, indicating overheating. It was strongly suspected that this set of bearings had a blocked oil passage, which had not allowed sufficient oil to reach the bearing. Further, all the bearings in the first and second stages were Grade 1 bearings, which are marginally rated to 18,000 rpm. The first stage bearings were replaced with Grade 7 bearings, rated to 60,000 rpm; the second stage bearings were replaced with Grade 5 bearings, also rated to 60,000 rpm.

The thrust assembly was tested to a temperature of 460°F to ensure that the main shaft would not bind due to thermal expansion of the shaft, housing or bearings. Thermocouples were placed on the outside of the thrust assembly, and it was wrapped

with high temperature heat tape; an insulation blanket was placed around the entire assembly. A sensitive torque wrench was used to test the torque necessary to turn the thrust assembly as the temperature was increased. Torque measurements were taken at atmospheric temperature and at temperatures of 200°F, 300°F, 400°F, and 460°F. The assembly was then allowed to cool to 100°F and a final torque reading noted. In general, the torque readings varied very little during the entire test; the low torque recorded was 5 ft-lbs, and the high was 7 ft-lbs.

3.3 Downhole Motor Test

The bench tests of the downhole motor were carried out at atmospheric conditions. The test stand allows the simulation of various downhole back pressures.

The entire downhole motor, except for the bypass valve assembly, was placed in the large diameter (15.25 inch inside diameter) casing, and the air flow from the compressors was set at 1200 SCFM (see Section 4 and Appendix A, Section A.1, for details on the compressors). The back pressure was adjusted to 50 psig and the torque recorded. The brake was released incrementally to allow the motor to increase speed gradually. At various rotation speeds, the back torque was measured.

With the brake locked again, the flow rate was increased to 2400 SCFM, the back pressure adjusted to 50 psig, and the torque recorded. The brake was released incrementally to allow the motor to slowly speed up, and the back torque was recorded at various speeds.

Maintaining the 2400 SCFM flow rate, the brake was again locked, the back pressure was increased to 100 psig, and the back torque at the brake recorded. Again the brake was released, the motor allowed to speed up, and the back torque recorded at various speeds.

The flow rate was then increased to 3600 SCFM and the back pressure adjusted to 50 psig, with the brake locked. However, on this test, the output torque exceeded the capacity of the brake system, causing the brake to deform and lock. The test was halted, and only limited readings were obtained for that flow rate. (See Appendix A, Section A.3, for plots of the bench test data.)

It was clear from this last test that the brake system which provided the back torque was inadequate to hold the full power of the downhole motor, i.e., when operating with 3600 SCFM.

The brake system was redesigned to handle greater torque loads, and the torque measuring arm was replaced with a tension measuring device (i.e., a power tong gauge) that could provide more reliable torque measurements at the higher torque loads anticipated.

4. Shallow Hole Drilling Test with the First Prototype Downhole Motor

The objectives of the drilling test of the first prototype downhole motor were:

- To ascertain whether the downhole motor would drill.
- To establish the performance of the downhole motor relative to conventional rotary drilling.
- To establish operational procedures for running the motor in field situations.
- To identify weaknesses in the downhole motor that were not apparent during the bench tests.

4.1 Location and Equipment

The field test of the first prototype motor took place at the Bayless Drilling, Incorporated, yard in Farmington, New Mexico. This yard is located on the La Plata Highway on the west side of Farmington. The downhole motor was to be used to drill a section of borehole within 1000 feet of the surface. The rock formation to be drilled was the Kirtland.

Figure 15 shows Rig No. 1 of Bayless Drilling, Incorporated, an

Figure 15
Drilling Rig

IDECO H-1000, which was the rig used during the drilling test. The compressors and boosters were supplied by Four Corners Air Drilling Service. The compressors were Quincy Rotating Screw, Model 850-200. Four Corners Air Drilling Service operates these compressors more slowly than their rated speed; therefore, the 750 SCFM compressors operate at about 715 Actual CFM. At the average operating atmospheric pressure and temperature at which the tests were carried out, each compressor was producing the equivalent of 600 SCFM. (See Appendix A, Section A.1, for further details on air volumetric rate of flow.)

The Bayless rig was used to drill a 17 $\frac{1}{2}$ inch diameter borehole to a depth of approximately 273 feet. This surface borehole was drilled by conventional rotary drilling with mud as the circulating fluid. The hole was cased with a 11 $\frac{3}{4}$ inch diameter, 42.00 lb/ft, J-55 casing. The casing was cemented with an expanding cement to aid in preventing water from seeping into the borehole from the near-surface formation.

4.2 Baseline Drilling Data

After the cement had set, conventional rotary drilling was used to drill a 10 $\frac{5}{8}$ inch diameter borehole through the casing shoe and into the Kirtland formation. This drilling was done with an

insert roller-cone bit, using water as the circulating fluid. (The drilling of the Kirtland formation with nearly the same bit as would be used later with the downhole motor was carried out to establish the drilling rate for this formation using conventional drilling with water as the circulating fluid.) Drilling in this manner progressed for approximately 50 feet. During this drilling sequence, various weights on bit were tried, and the instantaneous rate of penetration was recorded. The maximum conventional drilling rate with water was 48 ft/hr.

While still drilling conventionally, the circulating fluid was then switched to air. The borehole was cleared of water and air drilling established. Approximately another 50 feet were drilled with air as the circulating fluid. Here again, various weights on bit were tried and the instantaneous rate of penetration recorded. The maximum conventional drilling rate with air was 110 ft/hr.

At the end of these conventional drilling tests, the depth of the borehole was about 384 feet.

On March 16, 1987, the downhole motor was brought to the drill rig. The downhole motor was run into the borehole with a 10 ⁵/₈ inch diameter insert roller-cone bit. A stabilizer had been

placed above the motor. Three 8 inch by 2 ⁷/₈ inch drill collars plus seven 7 inch by 2 ¹/₄ inch drill collars were used to provide weight to the bit. The drill pipe was 4 ¹/₂ inch diameter pipe.

Because the bypass valve assembly was, at this point, not yet tested, an attempt was made to prevent the turbine from running away by putting the downhole motor on the bottom of the hole and placing about 30,000 lbs of weight on the bit. This weight on bit was intended to keep the motor stalled while the hole was blown clean of residual cuttings (fill) and water. Also, the drill string was rotated slowly.

After circulation was established with 1800 SCFM, and drill string rotation was established at about 20 rpm, the weight on bit was reduced to attempt to get the motor to drill ahead. Drilling began and continued for five feet, then stopped abruptly with a noticeable increase in rotary table speed.

The downhole motor was removed from the borehole and disassembled at the Clay Groomer Machine Shop.

It was found that the rotation of the rotary table had caused the main shaft of the motor to rotate backwards, relative to the

housing of the downhole motor, when weight was on the bit. This relative backwards rotation of the main shaft turns all other moving components in the motor backwards. The oil pump in the gearbox does not pump oil while turning backwards, and therefore the bearings in the high speed sections of the gearbox were not lubricated as they rotated at about 4000 rpm. The bearings on the input shaft of the gearbox seized, and it is most likely that the five feet of drilling was primarily the result of the locked-up system (i.e., locked-up bit, relative to the motor housing). At this point, the depth of the borehole was 389 feet.

On March 21, 1987, after replacing the bearings in the upper stages of the gearbox, the downhole motor was brought back to the drill rig. The downhole motor was run into the borehole with a 10 ⁵/₈ inch diameter insert roller-cone bit, but with no stabilizer placed above the motor. It had been decided not to rotate the drill string slowly while drilling with the downhole motor. Two 8 inch by 2 ⁷/₈ inch drill collars plus nine 7 inch by 2 ¹/₄ inch drill collars were used to provide weight to the bit.

The bit was allowed to touch bottom and was then raised five feet to ensure that the bypass valve ports were open. The hole was blown clear of residual cuttings and water with 1800 SCFM. The

motor was then lowered to the bottom of the hole and drilling attempted. The downhole motor did not drill; the air flow rate was increased to 2400 SCFM, but still the motor did not drill. The downhole motor was removed from the borehole and was disassembled at the Clay Groomer Machine Shop.

It was found that the small bolts which hold the lower bearing in place on the turbine had loosened. This had allowed the turbine rotor to lower and hit the stationary spool of the turbine motor frame. There was evidence of prolonged rubbing between the bottom of the rotor and the spool, indicating that, during other tests with this turbine motor, rubbing of the two components had taken place. This problem with the turbine motor was resolved and the downhole motor reassembled for drilling the next day. The depth of the borehole was still 389 feet.

4.3 First Day Testing

March 22, 1987 was the first day of successful testing of the downhole motor.

The downhole motor was again placed back in the borehole. The 10 ⁵/₈ inch diameter insert roller-cone bit was again used, with no stabilizer above the motor. Two 8 inch by 2 ⁷/₈ inch drill

collars plus nine 7 inch by 2 $\frac{1}{4}$ inch drill collars were used to provide weight to the bit.

The bit was allowed to touch bottom and then raised five feet to ensure that the bypass valve ports were open. The hole was blown clean of cuttings and water with 1800 SCFM. The motor was then lowered to the bottom of the hole and drilling commenced. After the motor had drilled for about 30 feet, the air volumetric rate of flow was increased to 2400 SCFM. Drilling continued through the remainder of the day, to a depth of about 505 feet. While drilling down the last joint of drill pipe, it was noticed that the hole was making more water than usual. Using only 2400 SCFM, the maximum drilling rate for the 118 feet drilled was 100 ft/hr. Figure 16 shows the downhole motor after its first successful day of drilling.

Figure 16
Downhole Motor after Drilling

4.4 Second Day Testing

March 23, 1987, was the second day of successful testing of the downhole motor.

The 10 ⁵/₈ inch diameter insert roller-cone bit was again used, with no stabilizer. Two 8 inch by 3 inch drill collars plus twelve 7 inch by 2 ¹/₄ inch drill collars were used to provide weight to the bit.

The downhole motor was again placed back into the borehole and the hole blown clear of residual cuttings and water. Drilling commenced at about 505 feet with 2400 SCFM. After drilling had become well established, additional air volume was added. This brought the air volumetric rate to 3000 SCFM. The hole was making more water than the previous day; therefore, a water and surfactant solution was injected with the air from the compressors. The mist injection rate was approximately 10 barrels of water per hour. This misting operation continued through the entire day of drilling.

As drilling progressed, it became apparent that the hole was making progressively more water. At a depth of 789 feet, after making a connection, the standpipe pressure rose sharply when the

process of re-establishing circulation, to clean the hole before drilling ahead, was attempted. The pressure rose to approximately 900 psig as the hole tried to unload residual cuttings and water. A large amount of cuttings and water finally emerged from the blowby line, but the standpipe pressure dropped only to about 580 psig, and the motor was not drilling. It was apparent that the hole was making too much water and that the downhole motor was drilling too fast for the air flow to clear the hole of both water and cuttings. The downhole motor was removed from the borehole.

The downhole motor had drilled to a depth of 789 feet, with a maximum drilling rate for the 284 feet drilled of 186 ft/hr. The total drilling time during the two successful days of drilling was about 12 hours, including trip and connection time.

Drilling at 3600 SCFM was not attempted primarily because the available weight on bit was marginally adequate to stall the downhole motor at 3000 SCFM. In addition, it was recognized that although 3600 SCFM would have provided an additional 600 SCFM for hole cleaning, it would also have increased the power of the motor, and thus the drilling rate, producing more cuttings. In fact, from the test data, if sufficient drill collar weight had been available, the drilling rate at 3600 SCFM extrapolates to

approximately 250 ft/hr.

The downhole motor was disassembled at the Clay Groomer Machine Shop. It was found that during the high pressure surge on the last attempted connection, residual cuttings and water had entered the bypass valve ports and packed off the top of the turbine motor. This occurred because the 900 psig in the drill string had been sufficient to lift weight above the valve and cause the ports to open briefly.

During this second day of drilling, Sandia National Laboratory personnel recorded the vibrations of the drill string while the downhole motor was drilling. This data was obtained by means of a triaxial set of accelerometers placed on the top of the kelly. The data will establish the acceleration loads experienced by the drill string and, therefore, the downhole motor. Also this data may be used to indicate the speed at which the motor is turning while operating. Such data, when reduced, could provide instrumentation that would allow future drillers using the downhole motor to know the exact rotational speed of the bit, thus reducing the chance of runaway and allowing for more efficient drilling.

5. Fabrication and Assembly of the Second Prototype Downhole Motor

The fabrication of the major components of the second prototype downhole motor began around September 1986. Because of the long lead time necessary to fabricate the rotor and the nozzle plate in the turbine motor module, and the gears in the gearbox module, these items had to be started early. Assembly of the turbine motor module and of the gearbox module (the most complicated mechanisms in the downhole motor) began in December 1986.

5.1 Thrust Assembly

During the bench and the drilling tests of the first prototype downhole motor, no problems were found with the thrust assembly. However, from an overall drilling operation standpoint, it was determined that it would be necessary to have a mechanism incorporated into the downhole motor that would prevent or restrict the entire rotating system (i.e., the main shaft, the lower dogbone, the transmission, the upper dogbone, and the rotor) from rotating backward. The most convenient position in the rotating system to place such a mechanism is in the thrust

assembly. This mechanism to restrict backward rotation of the main shaft will be designed and incorporated into the thrust assembly for the first, second and third prototype downhole motors (see Section 9). Because of the lead time necessary to carry out the design, fabrication and assembly of the new thrust assembly, and because of the lack of any problems with the first prototype thrust assembly, the first prototype thrust assembly will be used for the bench tests of the major components of the second prototype downhole motor, namely the turbine motor module and the gearbox module.

5.2 Turbine Motor and Gearbox Assembly

The turbine motor module for the second prototype downhole motor was supplied by Bear Engineering Company. There was no problem with the fabrication or the assembly of this module.

The remainder of the fabrication of the gearbox module for the second prototype downhole motor was carried out by the Clay Groomer Machine Shop, and by the Advance Gear and Machine Shop. The gearbox module was assembled at the Clay Groomer Machine Shop. Very few problems were encountered in the assembly of this module. Further, due to the lack of warpage in the outside housing of the gearbox, the resistive torque necessary to turn

the completed gearbox was only about one in-lb.

The present turbine motor and gearbox assembly design, as modified from the drilling test results, was adequate for use for the bench test of the second prototype downhole motor. There are plans to modify the torque connection between the gearbox module and the main housing of the assembly, and the torque connection between the turbine housing and the main housing. These modifications will be carried out later, in order to allow the first and second prototype downhole motors, in addition to the third prototype downhole motor, to be available for the drilling tests in the Geysers area.

5.3 Bypass Valve Assembly

The bypass valve assembly cannot be tested in the bench test stand.

This assembly will be modified to allow filters to be placed between the valve and the turbine motor. Thus, the bypass valve assemblies for both the first and the second prototype downhole motors, in addition to the third downhole motor, will be modified for future testing at the Geysers area.

5.4 Assembly of the Second Prototype Motor

No difficulties were experienced in assembling the second prototype downhole motor for bench testing.

6. Bench Tests of the Second Prototype Downhole Motor

The bench tests of the second prototype downhole motor were conducted using the new 20 in diameter brake assembly (see Figure 17). This assembly was expected to eliminate anomalies, such as those observed in the first prototype bench test, caused by the previously underdesigned brake assembly. In addition, the torque tension cable attached to the load cell provided more reliable measurement of the torque at the output shaft of the downhole motor, and the optical tachometer was replaced by a mechanical tachometer.

6.1 Gearbox Run-In

The gearbox for the second prototype motor was run in during the temperature test (see Section 6.2). Because of the improved quality of the second prototype gearbox, i.e., the lack of warpage in the outer housing, this gearbox was found to be quite easy to rotate, requiring less than one in-lb of torque. Thus it was felt that a prolonged run-in period at normal temperatures was unnecessary. The successful high temperature testing of the gearbox justified the elimination of the preliminary prolonged gearbox run-in.

Figure 17
Brake System (20" Diameter), Torque Tension Cable,
and Load Cell Measuring device

6.2 Temperature Test

The temperature test of the second prototype gearbox was carried out in almost the same manner as that of the first prototype gearbox. The gearbox was placed in its protective casing in the test stand with five thermocouples attached along the gearbox housing. The protective casing was then wrapped in insulation to aid heat retention.

Temperature measurements were taken from the five thermocouples at approximately one minute intervals. The rotational speed of the input turbine varied from 3900 rpm to 10000 rpm, with the speeds increased approximately every thirty minutes. The backload on the transmission varied from approximately 2300 ft-lbs to 2500 ft-lbs of torque throughout the test.

Starting at 50°F, it took the transmission approximately 155 minutes of continuous running to attain an exterior temperature of 330°F. This exterior housing temperature correlates to an internal temperature of approximately 365°F. (Plots of the data obtained from the temperature test are in Appendix B, Section B.1.)

At this time, the gearbox was removed from the test stand and

inspected. No leaks could be found from any seal location and the input stage rotated smoothly. The gearbox was allowed to cool and then disassembled for inspection of the internal components. No damage was found to any components of the system.

The transmission temperature rise versus time can be used to estimate the power absorbed by the transmission. If it is assumed that the average of the temperatures recorded at each time represent the average temperature of the transmission and the test casing, then the rate of temperature increase is proportional to the horsepower. Assuming that the test casing weighs 300 lbs, and that the transmission weighs 375 lbs, then the constant of proportionality between horsepower and the rate of temperature increase (in degrees F/min) is 1.2. That is, the horsepower is 1.2 times the temperature gradient. At 3940 rpm (input shaft speed) the transmission used 3 hp, at 6120 rpm it used 3.3 hp, at 7930 rpm it used 3.8 hp, at 8880 rpm it used 3.65 hp, and at 10000 rpm it used 3.1 hp.

The decrease in power consumption at higher speeds is unexpected. It may be due to lubrication effects. Based upon a careful consideration of the experimental data it is believed that the decrease really exists. In any case the power losses are small when compared to the turbine output at flow rates of 2400 SCFM or

greater.

Based on the measured external temperatures of the transmission and the estimated rate of power consumption the internal temperature can also be estimated. If steady state heat flow is assumed, then the temperature drop across the transmission wall that corresponds to a heat flow equivalent to 2.5 hp can be calculated. This estimate (5 degrees F) is conservative since the conditions are actually transient.

6.3 Downhole Motor Test

The entire downhole motor, except for the bypass valve assembly was mounted in the large diameter casing in the test stand. The brake was locked, and the back pressure valve was closed (>125 psig back pressure). The compressors were then started and set to deliver an air flow of about 1800 SCFM. The back pressure was set to 125 psig, and the torque and system pressures were recorded. The back pressure was then decreased in 25 psi increments from 125 to 25 psig, and at each new back pressure the system pressures and the torque were recorded.

The foregoing procedure is termed a lock-rotor torque test. It provides data on both the maximum torque delivered by the tool,

and the pressure losses in the system. Following the 1800 SCFM test, the test was repeated at flow rates of 2400, 3000 and 3600 SCFM. (See Appendix B, Section B.2, for the lock-rotor torque test data.)

In addition to the lock-rotor torque test, a rotating torque test was run. With a flow rate of 1800 SCFM and a back pressure of 50 psig, the brake was released incrementally to allow the motor to slowly accelerate, and at each new brake setting the speed and torque were recorded. The rotating torque test was repeated at 1800 SCFM - 75 psig, 2400 SCFM - 25 psig, 2400 SCFM - 75 psig, 3000 SCFM - 25 psig, 3000 SCFM - 50 psig, 3000 SCFM - 100 psig, 3600 SCFM - 25 psig, 3600 SCFM - 50 psig and 3600 SCFM - 100 psig, to give the full range of operating performance.

During the 1800 SCFM rotating torque test the tool ceased to deliver torque to the brake. The tool was disassembled, and a failure in the new lower turbine bearing was discovered. The cause of this failure has not been determined, but it does not appear to be a significant problem since the replacement bearing ran much longer at higher loads and higher speeds during the remainder of the tests.

7. Bench Tests Results

The bench tests of the downhole turbine motor yielded data on pressure losses within the tool, and data on the bit torque as a function of bit main shaft speed (see data plots in Appendix B, Section B.3). These data were then incorporated into a mathematical model to predict tool performance.

The system pressure losses were first calculated, and the results of the calculations were compared to the measured system pressure losses. [1,2] Once a good pressure match was obtained, the pressure calculations were used to predict the pressure losses across the turbine. From these predicted pressure losses, a maximum theoretical horsepower and a maximum theoretical turbine speed can be calculated. The turbine motor is an impulse turbine. Real impulse turbines are characterized by a dimensionless performance curve, such as that shown in Figure 18. The turbine efficiency, which is plotted on the abscissa, is the ratio of the actual horsepower to the maximum theoretical horsepower. The utilization factor, plotted on the ordinate, is the ratio of the rotor tip speed to the air speed. Figure 18 represents the expected dimensionless performance curve for the downhole turbine motor. The torque and rpm data taken

Figure 18
Expected Dimensionless Performance Curve

from the bench tests were used to develop an experimental performance curve. The experimental horsepower was calculated as a product of the measured bit main shaft speed and the measured torque, divided by a conversion factor

$$HP_e = \frac{TN}{5250} \quad (1)$$

where

T is the torque, (ft-lbs)

N is the bit speed, (rpm).

The experimental efficiency is simply the ratio of this measured horsepower to the computed maximum theoretical horsepower

$$e = \frac{HP_e}{HP_{pt}} \quad (2)$$

The experimental utilization factor is the ratio of the measured bit speed to the computed maximum theoretical bit main shaft speed, and is given by

$$U_e = \frac{N_e}{N_{mt}} \quad (3)$$

These calculated efficiencies and calculated utilization factors were used to develop an experimental dimensionless performance curve.

Once the pressure model and the experimental dimensionless performance curve are constructed, the tool performance at any

flow rate and back pressure can be predicted. The pressure model is used to compute the pressure drop across the turbine and, in turn, the maximum theoretical horsepower and the maximum theoretical bit main shaft speed. The experimental performance curve is then converted to a particular performance curve by multiplying the efficiencies by the maximum theoretical horsepower, and the utilization factors by the maximum theoretical bit main shaft speed. The result is a plot of bit horsepower versus bit main shaft rpm.

For the purposes of illustration, a particular performance curve has been generated from Figure 18. At 3600 SCFM and a back pressure of 50 psig, the maximum theoretical horsepower was computed to be 135 HP, and the maximum theoretical bit main shaft speed was computed to be 253 rpm. Under these conditions, the particular performance curve would be as shown in Figure 19.

The computations described above were performed using a FORTRAN program called TTST; a copy of TTST, along with a sample data file and its sample output, are presented in Appendix D, Section D.1. A FORTRAN program, MAIN, was developed to select the system parameter giving the best fit to the measured pressure data. MAIN, with a sample data file and its sample output, are given in Appendix D, Section D.2. The particular computations for the

Figure 19
Expected Particular Performance Curve: 3600 SCFM, 50 psig

first prototype downhole motor bench test are presented and discussed in Section 7.1, and for the second test in Section 7.2.

7.1 First Prototype

The data from the bench test of the first prototype downhole motor has been reduced and assembled in Table 1, along with the maximum theoretical values for a number of conditions not run in this bench test. For all cases, it was assumed that the air temperature was 40°F, and that the atmospheric pressure was 12 psia. From the three test conditions, the average peak efficiency is computed to be 0.18, and the average peak utilization factor is computed to be 0.21. Using these values, and assuming that the efficiency curve is an inverted parabola (which in theory it should be), the experimental dimensionless performance curve, Figure 20, was generated.[3] As was discussed in the preceeding section, this curve can then be used to predict tool performance at any of the untested conditions given in Table 1, by simply multiplying the efficiencies by the tabular value of the peak theoretical horsepower (HP_{pt}), and multiplying the utilization factors by the tabular value for the maximum theoretical rpm (N_{mt}). This has been done for the case of 50 psig back pressure and 3600 SCFM, and the particular performance curve is shown in Figure 21.

TABLE 1
PREDICTIONS AND RESULTS FOR THE FIRST DOWNHOLE MOTOR BENCH TESTS

Q	1200	1800	1800	1800	2400	2400	2400	3000	3000	3000	3600	3600	3600	3600	3600
P _b	50	20	50	100	20	50	100	20	50	100	20	50	100	150	200
P _{tc}	69	82	91	125	114	117	143	145	145	163	177	176	187	216	252
P _{te}	75					141	164					175			
HP _{pt}	12	66	33	13	102	62	28	134	98	50	161	135	76	45	29
HP _{pe}	1.6					11	6.6								
e	0.13						0.18	0.24							
T _{lt}	1930	5530	3940	2470	7950	6210	4190	10170	8730	6200	12240	11213	8410	6460	5170
T _{le}	540					2350	1250					5000*			
N _{nt}	4368	42168	29904	18648	45192	35280	23856	46368	16464	28224	46536	42504	31920	24528	19656
U _e	0.2						0.19	0.25							

64

Where:

- Q = The air flow rate (SCFM)
- P_b = The pressure downstream of the bit orifices (psig)
- P_{tc} = The pressure 20 ft upstream of the turbine, calculated (psig)
- P_{te} = The pressure 20 ft upstream of the turbine, measured (psig)
- HP_{pt} = The peak power delivered by a perfectly efficient impulse turbine, operating at the given conditions (HP)
- HP_{pe} = The measured peak power of the actual turbine (HP)
- e = The tool efficiency; the ratio of the measured horsepower to the maximum theoretical horsepower
- T_{lt} = The lock-rotor torque of a perfectly efficient impulse turbine operating at the given conditions (ft-lbs)
- T_{le} = The measured lock-rotor torque of the actual turbine (ft-lbs)
- N_{nt} = The maximum turbine speed for a perfectly efficient impulse turbine operating at the given conditions (rpm)
- U_e = The utilization factor; the ratio N/N_{nt} , where N is the speed at which HP_{pe} was measured

* Estimated from the stress required to deform the brake assembly

Figure 20
Experimental Dimensionless Performance Curve

Figure 21

Experimental Particular Performance Curve: 3600 SCFM, 50 psig

Comparing the expected with the experimental performance predictions for the 3600 SCFM, 50 psig back pressure case (Figures 19 and 21) shows a large discrepancy. The expected peak performance is 47 horsepower at 119 rpm, while the experimentally predicted performance is 24 horsepower at 53 rpm. This discrepancy was most probably caused by rubbing between the turbine rotor and the turbine housing in the turbine module. After completing the bench test and disassembling the tool, it was noted that the turbine produced a slight clicking sound when rotated by hand. This sound disappeared as the turbine warmed up, and its source could not immediately be explained. After the first attempt to drill, however, this turbine was disassembled, and it was discovered that a black discoloration had developed where the rotor was rubbing on the housing (see also Section 4). Such an assembly defect could easily have produced the noted loss in power in the bench tests.

There are several other factors that may also have contributed to the low horsepowers and speeds observed in the first downhole motor bench test.

1. The pressure losses in the test stand flow system were consistently higher (6% to 15%) than predicted. This is almost certainly due to a restriction in the flow path where the air leaves the tool. In preparation for the second

test, the test stand has been modified to reduce this restriction.

2. The optical tachometer system used to measure the bit main shaft speed proved extremely unreliable, and the actual speeds were often counted visually. A mechanical tachometer has been installed for the second prototype bench test.

3. As was stated in Section 3, the brake system failed when the 3600 SCFM test was attempted. It is possible that the bending of the brake system could have produced anomolous readings during portions of the test. A stronger brake system has been fabricated, as described in Section 6.

4. The air flow rate is not known precisely. The air company can only provide an estimate, and PTP does not currently have the facilities to measure the flow rate directly.

5. The bench test was performed on the entire tool, while the expected dimensionless performance curve is based on the assumption of no power losses in the transmission or in the thrust assembly. In actual service, both the transmission and the thrust must, of course, consume some power, and the rate of power consumption by those components will vary with the tool speed. Based on the temperature tests of the first transmission, it is believed that this power loss should not exceed 5 horsepower.

While the measured horsepower are lower than expected, and also probably lower than the true values, they are still sufficient to produce acceptable drilling rates. Furthermore, these horsepower are associated with excellent lock-rotor and operating torques. For some purposes, the torque performance of the tool is of interest, and, for the first prototype, that can be estimated as

$$T = 0.35T_{lt} \left(1 - \frac{N}{0.42N_{mt}} \right) \quad (4)$$

where

T is the predicted torque, (ft-lbs)

T_{lt} is the theoretical lock-rotor torque (Table 1), (ft-lbs)

N is the bit speed, (rpm)

N_{mt} is the theoretical maximum bit speed (Table 1), (rpm).

The factor 0.35 is computed by relating the observed lock-rotor torques (three values from the bench tests, four from the drilling test) to the predicted lock-rotor torques. This factor is actually one-half the ratio of the efficiency to the utilization factor. The factor 0.42 is the peak utilization factor, 0.21, multiplied by two. For the 3600 SCFM, 50 psig back pressure case discussed previously, this indicates a lock-rotor torque of about 4000 ft-lbs at a peak power of 2000 ft-lbs at 53 rpm.

7.2 Second Prototype

The data from the bench test of the second prototype downhole motor has been reduced and assembled in Table 2, along with the maximum theoretical values from a number of conditions not run in this bench test. For all cases it was assumed that the atmospheric pressure was 12 psia. The temperatures at which the actual tests were run may be found in Appendix B, Section B.2, along with the raw data. For the theoretical cases where no data were collected a temperature of 60 degrees F was used.

The torque and rpm data from all the tests are also presented in reduced form in Figure 22. The abscissa of the figure is the ratio of the measured torque to the theoretical lock-rotor torque, and the ordinate is the ratio of the measured turbine speed to the theoretical maximum turbine speed (utilization factor). This dimensionless treatment allows data from widely varying operating conditions to be compared.

In a like manner, the horsepower and rpm data from all ten tests are presented in Figure 23, as a plot of the efficiency (the ratio of the measured horsepower to the theoretical horsepower) versus the utilization factor.

Figure 22
Dimensionless Torque Vs Utilization Factor

Figure 23
Expected Dimensionless Performance Curve

Based on these data the peak efficiency of the tool is estimated to be .3, and the peak utilization factor is estimated to be .41. These values were used to develop the experimental dimensionless performance curve presented in Figure 24. This curve represents the current best estimate of the tool performance, and its application is as discussed in Sections 7.0 and 7.1.

The measured efficiency is a bit lower than the expected efficiency (.3 vs .35) and the measured utilization factor is lower than the expected utilization factor (.41 vs .47). This is not surprising. The expected values are based on turbine performance alone and thus assume a transmission that absorbs no power. In practice it takes 3-4 hp to run the transmission, and correcting for this power loss would increase the measured efficiencies and probably decrease the scatter in the data. This surely accounts for the low efficiencies measured at 1800 SCFM (see Table 2).

The slightly low efficiencies measured at 3600 SCFM cannot be explained by the transmission power losses. Instead they seem to indicate that the compressors were not delivering the full 3600 SCFM. The air company had had some trouble with their booster compressors (used only to increase the pressure of the air delivered by the screw compressors) and at the high pressures

Figure 24
Experimental Dimensionless Performance Curve

TABLE 2
PREDICTIONS AND RESULTS FOR THE SECOND DOWNHOLE MOTOR BENCH TESTS

Q	1800	1800	1800	2400	2400	2400	3000	3000	3000	3000	3600	3600	3600	3600
P _b	25	50	75	25	50	75	25	50	75	100	25	50	75	100
P _{tc}	55	71	92	70	83	97	87	96	113	132	106	113	127	143
P _{to}	62	79	99	70	85	100	85	93	113	139	90	100	120	143
HP _{pt}	39.9	27.8	18.6	59.5	46.5	35.2	77.3	64.9	53.3	41.8	93.9	84.9	73	59.3
HP _{pe}		4.8	2.2	17		10.4	23	19.2		12.1	25	22.4		14.7
T _{lt}	4348	3640	2971	6058	5421	4663	7730	7093	6490	5695	9309	8827	8313	7100
T _{le}		942	667	2340		1827	2950	2706		2054	3359	3045		2465
N _{nt}	32360	27100	22100	34700	30260	26630	35250	32280	28980	25872	35570	33920	30930	28380
U _e		0.32	0.27	0.38		0.39	0.41	0.4		0.41	0.38	0.39		0.37

Where:

- Q = The air flow rate (SCFM)
- P_b = The pressure downstream of the bit orifices (psig)
- P_{tc} = The pressure immediately upstream of the turbine, calculated (psig)
- P_{to} = The pressure immediately upstream of the turbine, measured (psig)
- HP_{pt} = The peak power delivered by a perfectly efficient impulse turbine, operating at the given conditions (HP)
- HP_{pe} = The measured peak power of the actual turbine (HP)
- T_{lt} = The lock-rotor torque of a perfectly efficient impulse turbine operating at the given conditions (ft-lbs)
- T_{le} = The measured lock-rotor torque of the actual turbine (ft-lbs)
- N_{nt} = The maximum turbine speed for a perfectly efficient impulse turbine operating at the given conditions (rpm)
- U_e = The utilization factor; the ratio N/N_{nt} , where N is the speed at which HP_{pe} was measured

involved in this test it is quite likely that the full flow rate was not achieved.

The discrepancy in the measured versus the expected peak utilization factor is probably due to inaccuracies in modeling the pressure losses within the system. While the pressure model was in error by an average of only 3.1 psi when compared to the experimentally measured pressures, the pressure loss across the turbine was consistently overestimated. Thus the calculated theoretical values for the maximum horsepower, maximum torque and maximum turbine speed are all somewhat high, giving conservative estimates of both the peak efficiency and the peak utilization factor.

Modeling the pressure losses within the system is not simple, but is an essential part of predicting and analyzing the tool performance. The flow path within the tool consists of at least two choke points, a section of annular flow, a section of pipe flow and several changes in flow direction. The complexity of the problem has been compounded by problems with the pressure measurement equipment and lack of tools to measure the flow rate. However, the results are encouraging. The latest series of pressure data were predicted reasonably well, and in the process it was determined that the most significant pressure losses occur

as the air flows around the transmission. Design changes to reduce this loss will certainly increase the output power at all operating conditions.

As with the first prototype the the torque performance can be predicted with a simple analytical relationship

$$T = 0.35(T_{1t})(1 - N/ (.82N_m)) \quad (5)$$

where

T is the predicted torque, (ft-lbs)

T_{1t} is the theoretical lock-rotor torque (Table 2), (ft-lbs)

N is the bit speed, (rpm)

N_m is the theoretical maximum turbine speed (Table 2), (rpm)

The factor of .35 is one-half the ratio of the peak efficiency to the peak utilization factor, and the factor .82 is twice the value of the peak utilization factor.

7.3 Comparison of Bench Test Results

There were some signifigant differences in performance between the first and second tool. The peak efficiency of the second tool was higher than that of the first (.3 vs .18), as was the peak utilization factor (.41 vs .21).

At least part of this difference can be attributed to better

measurement and analysis of the system pressures during the testing of the second prototype. During the first tests only three of the six pressure gauges functioned. The three gauges attached to the actual tool did not work, and all internal pressure drops were calculated based on pressures measured 20 ft upstream and 15 ft downstream of the tool. During the second test five of the six gauges worked reliably. The pressure data from the second test indicated large pressure losses around the transmission. These losses were not accounted for in analysis of the data from the first test. Consequently, the estimates of the maximum theoretical horsepower, torque and bit speed were probably too high and the estimates for peak efficiency and utilization factor were probably too low.

The test of the second prototype was free from serious operational problems with either the brake system or the tachometer, both of which may also have contributed to the low performance observed in the first test. However, all of these effects combined do not fully explain the discrepancies. It now seems highly likely that the first prototype turbine was not performing correctly. Both units developed similar lock-rotor torques, but the output torque declined far more rapidly with increasing speed in the first unit than in the second. Based on observations of scuff marks on the rotor of the first unit it

appears that the rotor was rubbing against the housing, a problem that could have caused the observed behavior. The design of the turbine will be modified slightly to assure that this cannot be a problem in future units.

8. Drilling Test Results

The first successful field tests of the first prototype downhole pneumatic turbine motor were conducted on March 22 and 23, 1987, in the Bayless Drilling Company yard in Farmington, NM (see Section 4).

8.1 Conventional Rotary Drilling Results

Prior to the downhole motor drilling test, two conventional rotary drilling tests were performed, one using water as the drilling fluid and the other using air as the drilling fluid. These conventional rotary drilling tests were conducted to give baseline rates of penetration that could be used to compare these tests with the drilling performance of the downhole motor.

Table 3 gives the conventional rotary drilling test results while using water as the drilling fluid.

Table 4 gives the conventional rotary drilling test results while using air as the drilling fluid.

The conventional rotary tests using water were conducted with a

TABLE 3
CONVENTIONAL WATER DRILLING TEST RESULTS

<u>Weight on Bit (lbs)</u>	<u>Rate of Penetration (ft/hr)</u>
10,000	30
20,000	48
28,000	56

TABLE 4
CONVENTIONAL AIR DRILLING TEST RESULTS

<u>Weight on Bit (lbs)</u>	<u>Rate of Penetration (ft/hr)</u>
10,000	85
15,000	90
20,000	96
25,000	112

constant rotary bit speed of 60 rpm. The conventional rotary test using air were conducted with a constant rotary bit speed of 100 rpm. The selection of these rotary speeds was based on the "driller's intuition" to accomplish "optimal" drilling rates.

The instantaneous drilling weight as a function of weight on bit and rotary speed is given by [4,5]

$$ROP = K \left[\frac{W}{4D_b} \right]^{a_1} \left[\frac{N}{100} \right]^{a_2} \quad (6)$$

where

K is the drillability factor, which depends on the bit type, drilling fluid type and rock formation properties, etc., (ft/hr)

W is the weight on bit, (10^3 lb)

N is the rotary speed, (rpm)

D_b is the rock bit diameter, (in)

a_1, a_2 are exponents depending upon drilling conditions, (dimensionless).

The above equation ignores the effects of cutting structure wear.

Since the tests were performed at constant rotary speeds, the exponent a_2 was chosen to be 0.75, which is consistent with a soft formation. Applying regression analysis techniques using the data in Tables 3 and 4, the following relationships were determined.

a) Using water as the drilling fluid:

$$ROP_w = 108.6 \left[\frac{W}{4D_b} \right]^{0.616} \left[\frac{N}{100} \right]^{0.75} \quad (7)$$

b) Using air as the drilling fluid:

$$ROP_a = 123.9 \left[\frac{W}{4D_b} \right]^{0.279} \left[\frac{N}{100} \right]^{0.75} \quad (8)$$

The drillability factor, K , in Equation (7), obtained for the conventional air drilling test (i.e., 123.9 ft/hr) is approximately 14% greater than the corresponding value for the water drilling test (i.e., 108.6 ft/hr). This result is explained by the effect of the differential pressure on the rate of penetration. The rather small increase in drillability is related to the shallow hole depth (approximately 350 ft) at which these tests were performed. It is also evident that the ratio of the drilling rates (ROP_a/ROP_w) is dependent on the magnitude of W (the weight on bit). If, for example, 25×10^3 lbs bit weight would be used, the air drilling rate would be about 36% higher than the corresponding value for water drilling.

In contrast to the consistent behavior of the drillability factors, the magnitude of the weight on bit exponent (a_1) for air drilling (0.279) seems much too low. Additional tests are needed to verify these results. Also further tests should be conducted to find the exponent of the effect of rotary speed (a_2) on the drilling rate.

8.2 Downhole Motor Tests at the Rotary Table

Before the downhole motor was run into the borehole, the motor was tested on the rig floor by locking the rotary table and measuring the reaction torque with the tongs. This rig floor testing was carried out for various volumetric flow rates, and for both open and closed bypass valve positions. These tests gave important stall torque data for the downhole motor and pressure injection tests.

These stall torques and injection pressures were measured for three volumetric air flows: 600 SCFM, 1200 SCFM and 1800 SCFM. The data obtained from these tests are given in Table 5.

A FORTRAN computer program, SURF_VLV, was developed to evaluate the results of the tests; a listing of this program, with a sample data file and sample output, is given in Appendix E, Section E.1. Using the ambient pressure (12 psia) and temperature (40°F) conditions, the program predicts the injection (stand pipe) pressure and the magnitude of the stall torque at the rotary table, for each air flow rate. The pressure loss equations used in this program are [6,7] :

TABLE 5

DATA FROM THE MOTOR TEST ON THE RIG FLOOR

<u>Air Flow Rate</u>	<u>Injection Pressure</u>			<u>Stall Torque</u>		
	(Bypass Open / Closed)			(Bypass Open / Closed)		
600	10	/	20	300	/	800
1200	30	/	55	800	/	1500
1200	35	/	55	800	/	1600
1800	not recd	/	85	1200	/	1800

$$P_{u(p)} = \left[P_{d(p)}^2 + \frac{0.0375(\dot{m}_a)^2(T_{av})}{d^{5.333}}(L) \right]^{0.5} \quad (9)$$

where

$P_{u(p)}$ is the upstream pressure in an element of the air flow stream, where the cross-sectional area of the element is pipe-shaped, (lb/ft²)

$P_{d(p)}$ is the downstream pressure in that pipe-shaped element, (lb/ft²)

$\dot{m}$ is the mass flow rate of the air, (lb_m/sec)

d is the inside diameter of the element, (ft)

L is the length of the element, (ft)

T_{av} is the average temperature in the flow stream, (°R)

and

$$P_{u(a)} = \left[P_{d(a)}^2 + \frac{0.0375(\dot{m}_a)^2(T_{av})}{(D - d)^{1.333}(D^2 - d^2)^2}(L) \right]^{0.5} \quad (10)$$

where

$P_{u(a)}$ is the upstream pressure in an element of the air flow stream, where the cross-sectional area of the element is an annulus, (lb/ft²)

$P_{d(a)}$ is the downstream pressure in that annular element, (lb/ft²)

D, d are the diameters of the annular element, (ft)

To determine pressure losses through the bit nozzles and the

turbine nozzles, the conventional choke flow equations for subsonic and for sonic flow were utilized.[11]

The results of the computations are shown in Table 6. All calculations were performed using 0.35 as the turbine efficiency and 0.46 as the utilization factor. After comparing the experimental results with the computed results, the following statements can be made:

1. The stall torque values compare satisfactorily, with the possible exception of those obtained at 1800 SCFM.
2. The magnitudes of the injection pressures correlate very favorably.

The relative differences between the recorded and the calculated values are presented in Table 7. To determine the relative differences for the 1200 SCFM flow rate, the arithmetic average of the values obtained from the two tests was used.

Unfortunately, neither the 600 SCFM nor the 1800 SCFM flow rate tests were repeated, so the results could not be verified.

However, the tests proved the soundness of the bypass valve operating principles, the correctness of the flow equations used in the flow predictions, and showed a higher than expected turbine performance, (i.e., stall torque).

TABLE 6

RESULTS OF PRESSURE CALCULATIONS ON RIG FLOOR TEST DATA

<u>Air Flow Rate</u>	<u>Injection Pressure</u>			<u>Stall Torque</u>		
	(Bypass Open / Closed)			(Bypass Open / Closed)		
600	9.6	/	21.0	287	/	716
1200	29.3	/	48.9	867	/	1550
1800	53.7	/	79.4	1537	/	2326

TABLE 7

PREDICTED VS CALCULATED VALUES FOR PRESSURE CALCULATIONS

<u>Air Flow Rate</u>	<u>Injection Pressure</u>	<u>Stall Torque</u>
	(Bypass Open / Closed)	(Bypass Open / Closed)
600	4.0% / -5.0%	4.3% / 10.5%
1200	9.8% / 11.0%	-8.9% / 0.0%
1800	- / 6.6%	-28.0% / -29.2%

Figure 25 shows the relationship between the calculated stall torques and those measured in the tests on the rig floor. The data are given for varying volumetric rates of air flow.

Figure 26 shows the relationship between the calculated turbine peak horsepowers for various volumetric rates of air flow.

Figure 27 shows the relationship between the calculated air injection (i.e., stand pipe) pressures and those measured on the rig floor. The data are given for varying volumetric rates of air flow.

Table 8 gives a numerical summary of the data in Figures 25 to 27.

From Table 8 it may be noticed that the measured values of the stall torques at 600 SCFM and at 730 SCFM are identical, indicating some error in measurements. Basically the same statement is applicable to the 1200 SCFM and the 1190 SCFM flow rates. Considering this, it may be concluded that the predicted (calculated) values for the stall torques are very reasonable. Again a very good correlation between the measured and the calculated values of the drill pipe air injection pressure is apparent.

Figure 25
Stall Torque vs Air Flow Rate

Figure 26
Peak Horsepower vs Air Flow Rate

Figure 27
Injection Pressure vs Air Flow Rate

TABLE 8
 NUMERICAL SUMMARY OF STALL TORQUES, HORSEPOWER,
 AND INJECTION PRESSURES FOR VARIOUS SCFM

<u>Air Flow Rate</u>	<u>Stall Torque</u>			<u>Injection Pressure</u>			<u>Peak HP</u>
(SCFM)	(ft-lbs)			(ft-lbs)			
	(Measured/Calculated)			(Measured/Calculated)			
335	300	/	287	10	/	9.6	2.3
600	800	/	716	20	/	21.0	8.0
730	800	/	807	32.5	/	29.3	9.7
1200	1550	/	1550	55	/	48.9	18.8
1190	1200	/	1537	-	/	53.7	18.6
1800	1800	/	2323	85	/	79.4	28.2

8.3 Drilling Test Results

Upon completion of the rig floor tests, the tool was run into the hole.

The bottomhole assembly was composed of a Smith 46A, 10 ⁵/₈ inch roller-insert type rock bit, two joints of 8 x 2 ⁷/₈ inch drill collar, nine joints of 7 x 2 ¹/₄ inch drill collar, and 4 ¹/₂ inch drill pipe. With the motor at the bottom of the hole, the hole was blown clean of residual cuttings (fill) and accumulated formation water. No weight on bit was applied during the cleaning operation. The air flow rate was 1200 SCFM, which resulted in a drill pipe injection pressure of approximately 65 psig.

After the hole was cleaned, approximately 8×10^3 lbs weight on bit was applied and the air volumetric flow rate was increased to 1800 SCFM. Wet cuttings and water returns were observed at the blowby line exit. The rotary table was locked to absorb the reaction torque. Strong vibrations were noted at the rotary table, indicating that the motor was drilling through a fracture zone. After drilling about seven feet, the air volumetric flow rate was increased to 2400 SCFM, and the weight on bit increased to approximately 16×10^3 lbs. The downhole motor continued to

drill ahead. A detailed log of the drilling rate, footage drilled, weight on bit, and other pertinent data are given in Appendix C. It may be seen from this log that between 440 ft and 490 ft, the drilling conditions were stable and consequently provided reliable information for analysis of the drilling process and of the tool performance. Also, drilling with this WOB (16×10^3 lbs) should produce a hole that is free of dog-legs, since the buckling weight is about 17.5×10^3 lbs for the bottomhole assembly that was used to perform the test.

A FORTRAN computer program, FIELD_PR, was developed to predict downhole motor performance and surface injection pressure. The program listing, with a sample data file and sample output, is given in Appendix E, Section E.2.

To calculate the frictional pressure losses through the turbine, around the transmission, through the main shaft of the thrust assembly, and through the bit orifices, the same equations were used as for modeling the rotary table test (i.e., Equations (9) and (10)). However, to calculate the bottomhole pressure and the surface drill pipe injection pressure, the effect of the drilled cuttings (on the flow in the annulus) and the gravitational effects must also be considered. The following equations were used to accomplish these calculations.

a) Flow in annular space:

$$P_{u(a)} = \left[(P_{d(a)}^2 + \frac{b}{a})e^{2aH} - \frac{b}{a} \right]^{0.5} \quad (11)$$

where

$$a = \frac{1 + \dot{m}_s / \dot{m}_a}{53.3(T_{av})}$$

$$b = \frac{0.01475(T_{av})(\dot{m}_a^2)}{(D - d)^{1.333}(D^2 - d^2)^2}$$

b) Flow in circular conduit:

$$P_{u(p)} = \left[\frac{b^*}{a^*} - \left(\frac{b^*}{a^*} - P_{d(p)}^2 \right) e^{-2a^*H} \right]^{0.5} \quad (12)$$

where

$$a^* = \frac{1}{53.3(T_{av})}$$

$$b^* = \frac{0.01475(\dot{m}_a^2)}{d^{5.333}}$$

and

$\dot{m}_s$ = Mass flow rate of drilled chips, (lb_m/sec)

H = Vertical hole depth, (ft).

All other notation and units are the same as for Equations (9) and (10).

The turbine efficiency and the utilization factors applied to predict the turbine performance are 0.35 and 0.46, respectively (the same values as used in Section 8.2). Performance curves (turbine horsepower versus bit shaft rotary speed) are presented

in Figure 28 for the 1800 SCFM flow rate, and in Figure 29 for the 2400 SCFM flow rate. The rotary speed has been scaled down by a factor of 168, to reflect the bit shaft speed, (i.e., the total gear reduction ratio in the transmission is 168 to 1).

In Figure 28 is presented the characteristic inverted parabolic curve of the turbine motor horsepower versus the bit shaft rotary speed for an air volumetric rate of flow of 1800 SCFM (HP_t). Also presented in this figure is the straight line relationship for the horsepower required to rotate the bit at a weight on bit of 8×10^3 lbs (HP_b). The intersection of the parabola and the straight line establishes the rotary speed (abscissa) at which the bit main shaft turns under this weight on bit condition, approximately 155 rpm. The ordinate of the intersection gives the actual horsepower developed by the downhole motor at this weight on bit (8×10^3 lbs), approximately 22.

Figure 29 represents the same relationships as those given in Figure 28, but for an air volumetric flow rate of 2400 SCFM, and a weight on bit of 16×10^3 lbs. The rotary bit speed in this case is approximately 120 rpm, and the actual downhole horsepower is about 32.

Figure 28
Horsepower vs RPM, 1800 SCFM

Figure 29
Horsepower vs RPM, 2400 SCFM

The rock bit horsepower is calculated as [8]

$$HP_b = (12.49)(10^{-5})(WOB)^{1.5}(D_b)^{2.5}(RPM) \quad (13)$$

where

WOB is the weight on bit, (10^3 lb)

D_b is the bit diameter, (in)

RPM is the bit rotary speed.

The empirical coefficient, 12.49×10^{-5} , was determined from knowledge of the stall torques and of the corresponding weight on bit. This coefficient also indicates a rather soft formation, which agrees with the drillability constant, 123.9 ft/hr.

To compare the drilling rates obtained with the downhole motor to the corresponding drilling rates obtained with conventional rotary drilling using water and then air as the drilling fluids, two assumptions are introduced:

- 1) The formation is microscopically homogeneous.
- 2) The drilling rate equation for the downhole motor is

$$ROP = K \left[\frac{W}{4D_b} \right]^{0.273} \left[\frac{N}{100} \right]^{0.75} \quad (14)$$

It is apparent that by using Equation (14) for the drilling performance of the downhole motor, the basic performance data exponents for conventional rotary drilling with air are being utilized. Further tests should be performed to verify the correctness of this assumption. Using the data in Appendix C, in particular $ROP = 96.66$ ft/hr for $WOB = 16 \times 10^3$ lbs with a

volumetric flow rate of 2400 SCFM, the drillability factor when using the downhole motor can be obtained from Equation (14). This factor is found to be 120.9 ft/hr, which is approximately the same as that obtained in conventional rotary drilling with air as the drilling fluid. This result is explained by the fact that the hole is relatively shallow (about 500 ft). At shallow depths, the dynamics of the drill string do not affect the rate of penetration as much as in a deeper hole. At this point a conclusion can be made that, before misting, the drilling rates, at 2400 SCFM, were basically comparable to conventional rotary drilling with air as the drilling fluid. (It should be noted that the recorded drill pipe injection pressures were very close to the predicted values).

At a hole depth of about 567 ft, 10 bbl/hr of water were added to the injected air (i.e., misting). This resulted in a higher drill pipe air injection pressure. The average drilling rate with misting was about 113 ft/hr, approximately 16 ft/hr higher than the rate before misting. This higher rate of penetration is probably associated with a higher turbine horsepower, due to the increased mass flow rate through the turbine nozzles (i.e., the additional mass flow of the water). A computer program developed for predicting the turbine horsepower produced values that were significantly lower than the values measured while misting. A

significantly lower than the values measured while misting. A separate mathematical model will be developed to simulate mist drilling.

Finally, the air flow rate was increased to 3000 SCFM, and the weight on bit to 24×10^3 lbs. Again, a relatively high drill pipe pressure (450 - 470 psig) was recorded, and the drilling rate increased to 182 - 186 ft/hr. Stalling the bit, even by placing as much as 40×10^3 lbs on bit, became extremely difficult. There is no doubt that at this time the turbine produced very high horsepower, indicating the high drilling potential of this downhole motor.

Drilling at high penetration rates, with comparatively low air flow rates, in very difficult downhole conditions (significant amounts of water and solids in the annulus) resulted in the formation of plugs in the annular space of the borehole. To re-establish air circulation, the boosters were used. The drill pipe pressures then increased from about 470 psig to approximately 900 psig. A large mud slug was produced from the hole, and the test was halted. The tool was pulled for inspection.

8.4 Comparison of Bench Test Results and Drilling Test Results

The horsepowers predicted from the drilling test data (see Section 8.3) are about 50% higher than those recorded during the bench tests of the second prototype (see Section 7.2). Two factors probably contributed to this difference. First, the pressure model used during the drilling test did not account for the high pressure losses observed around the transmission when the tool was run on the test stand. Inclusion of this effect would reduce the values of the pressure based field prediction. The field horsepowers were also predicted based on the weight on bit and on the rate of penetration, i.e., the equations used were developed for standard rotary drilling. With the downhole motor, the tremendous reduction in the dynamic action of the drill string is likely to reduce the required power for drilling.

Whether the field predictions or the bench test predictions are most accurate is important only in predicting future downhole motor performance. The actual bit power required for drilling with this downhole motor is not of critical interest. What is important is that the downhole motor drilled exceedingly well in the field test, even though it was run at only two-thirds of the design flow rate, i.e., at approximately two-thirds of the power available.

9. Proposed Design Changes
for
All Three Prototype Downhole Motors

The components of the first prototype motor survived the bench tests and the drilling test carried out in the Farmington, New Mexico area. With the design changes recommended below, the first prototype, as well as the second and third prototypes, can be available for future field testing in the Geysers area near Santa Rosa, California.

There are four design changes that should be made prior to testing the prototype motors in the deep, hot wells at the Geysers area. These recommended design changes are:

1. The torque carrying connections between the gearbox housing and the main housing, and between the turbine housing and the main housing, must be strengthened. This lack of strength was displayed during the first attempt to drill with the motor on March 16, 1987 (see Section 4.2).
2. A ratcheted system or a clutch system should be designed into the thrust assembly to prevent or restrict backward rotation of the main shaft and, subsequently, of the gears in the gearbox, and the turbine rotor. Such an effective

backward rotation will occur when the entire drill string is rotated and the motor is not operating. The rotation of the drill string is a common drilling practice when a downhole motor is used to drill a straight borehole. This problem became apparent during the first attempt to drill with the motor on March 16, 1987 (see Section 4.2).

3. The bypass valve should be redesigned and/or repositioned to eliminate the entry of water and cuttings into the main housing above the turbine motor, during well clean-out operations when entering the borehole and while making connections to drill ahead. This problem was noticed during the last day of drilling, at the end of the drilling operation on March 23, 1987 (see Section 4.4).

4. The air flow passage around the gearbox should be modified for more aerodynamic flow to reduce the power loss produced in this section, thereby improving the horsepower output of the turbine.

Although these design changes are important, especially in terms of the overall mechanical integrity of the motor, they are rather straightforward improvements of the current design. In addition, these solutions affect primarily the external housing features of the prototypes rather than the principal moving components (i.e., the turbine motor or the gearbox).

Basically, the fundamental operation of the downhole pneumatic motor has been proven. The design changes recommended above are those which will improve the reliability of the motor and protect the turbine and gearbox components, while allowing the motor to be used in operational modes that are consistent with the use of other downhole (mud) motors in the field.

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Appendix A

Bench Test Data for First Prototype

A.1 Details on the Compressor Calculations

In general, the compressors used for the bench tests and for the drilling test were those from Four Corners Air Drilling Service. These were Quincy Rotary Screw compressors, Model 850-200. Four Corners Air Drilling Service operates these compressors so that their output is approximately 715 actual cfm. The atmospheric pressure during the test was about 12 psia, and the temperature about 40°F. Therefore, the specific weight of air used by the compressors was [1]

$$\gamma = \frac{12(144)}{53.3(505)} = 0.0642 \text{ lb/ft}^3$$

The weight rate of flow for each compressor was

$$G = \frac{0.0642(715)}{60} = 0.76505 \text{ lb/sec}$$

and the equivalent SCFM from each compressor was

$$Q = \frac{0.76505}{0.0765}(60) = 600 \text{ SCFM}$$

A.2 Gearbox Run-In Data

Date	01/25/87			01/27/87		
Time (24 hr)	15:07	15:50	16:05	20:16	20:26	20:36
Pressure - manifold (psi)	0.0	-	-	12.0	12.0	12.0
Pressure - brake (psi)	0.0	-	-	265.0	265.0	260.0
Torque (ft-lb)	0.0	-	-	345.0	377.5	377.5
RPM - turbine	0.0	4032.0	4200.0	5040.0	5040.0	5040.0
Horsepower - bit	0.0	-	-	2.0	2.2	2.2
Temperature - input (F)	61/61	-/103	97/99	63/61	74/72	77/77
Temperature - 1st stage (F)	69/72	116/110	101/100	54/53	-/67	77/76
Temperature - 2nd stage (F)	68/67	104/102	99/98	-/65	76/78	82/86
Temperature - 3rd stage (F)	65/61	95/94	97/97	68/70	81/81	85/86
Temperature - 4th stage (F)	61/61	90/-	96/-	64/69	77/81	83/86

Date

Time (24 hr)	20:46	20:56	21:06	21:16	21:26	21:36	21:46
Pressure - manifold (psi)	12.0	12.0	12.0	12.0	12.0	12.0	12.0
Pressure - brake (psi)	270.0	275.0	290.0	300.0	290.0	310.0	310.0
Torque (ft-lb)	375.0	375.0	400.0	390.0	390.0	400.0	390.0
RPM - turbine	5376.0	6048.0	5880.0	5880.0	5880.0	6048.0	6384.0
Horsepower - bit	2.3	2.6	2.7	2.6	2.6	2.7	2.8
Temperature - input (F)	-/86	102/102	-/112	112/114	114/116	127/128	133/134
Temperature - 1st stage (F)	91/94	108/110	117/119	119/122	122/125	-/137	144/144
Temperature - 2nd stage (F)	93/94	110/112	119/119	123/124	126/127	141/141	149/147
Temperature - 3rd stage (F)	92/97	109/111	116/120	120/123	124/128	136/139	144/146
Temperature - 4th stage (F)	88/86	106/104	110/109	117/117	121/120	134/131	141/140

Date		01/28/87				
Time (24 hr)	21:56	13:47	13:57	14:07	14:17	14:27
Pressure - manifold (psi)	12.0	15.0	15.0	15.0	6.0	7.0
Pressure - brake (psi)	310.0	320.0	300.0	250.0	310.0	350.0
Torque (ft-lb)	360.0	340.0	250.0	200.0	225.0	225.0
RPM - turbine	8064.0	9240.0	11424.0	11760.0	13776.0	14784.0
Horsepower - bit	3.3	3.6	3.2	2.7	3.5	3.8
Temperature - input (F)	142/144	-/73	121/123	-/165	-/204	-/229
Temperature - 1st stage (F)	154/156	79/89	128/123	168/170	205/208	-/235
Temperature - 2nd stage (F)	159/159	86/87	122/126	165/164	205/200	231/227
Temperature - 3rd stage (F)	156/159	84/85	119/117	160/159	198/198	225/225
Temperature - 4th stage (F)	154/150	81/80	115/114	156/155	194/194	221/220

Date	01/28/87					
Time (24 hr)	14:37	14:47	14:57	19:24	19:34	19:44
Pressure - manifold (psi)	14.0	15.0	15.0	10.5	10.5	10.5
Pressure - brake (psi)	275.0	250.0	250.0	300.0	300.0	315.0
Torque (ft-lb)	135.0	130.0	130.0	175.0	165.0	175.0
RPM - turbine	13776.0	14280.0	14280.0	10080.0	10080.0	10248.0
Horsepower - bit	2.1	2.1	2.1	2.0	1.9	2.0
Temperature - input (F)	261/262	298/297	329/330	131/125	141/136	-/145
Temperature - 1st stage (F)	-/269	-/302	-/333	128/133	138/141	145/151
Temperature - 2nd stage (F)	265/260	302/294	325/315	134/132	144/142	154/152
Temperature - 3rd stage (F)	257/256	293/292	315/311	133/132	142/142	151/152
Temperature - 4th stage (F)	250/249	286/282	308/307	130/129	138/137	147/146

Date

Time (24 hr)	19:57	20:07	20:17	20:27	20:37	20:47	20:57
Pressure - manifold (psi)	13.0	11.0	11.0	11.0	11.0	11.0	11.0
Pressure - brake (psi)	500.0	420.0	430.0	420.0	420.0	420.0	420.0
Torque (ft-lb)	300.0	250.0	210.0	230.0	225.0	230.0	230.0
RPM - turbine	11760.0	10920.0	13272.0	11424.0	11760.0	11928.0	11928.0
Horsepower - bit	4.0	3.1	3.2	3.0	3.0	3.1	3.1
Temperature - input (F)	-/164	183/181	188/188	204/198	205/209	218/214	-/218
Temperature - 1st stage (F)	-/168	172/180	192/188	201/197	204/207	201/211	203/213
Temperature - 2nd stage (F)	169/167	184/179	193/188	203/197	213/206	218/211	221/214
Temperature - 3rd stage (F)	166/164	178/176	189/188	198/197	207/205	212/210	216/214
Temperature - 4th stage (F)	161/160	172/169	183/179	191/187	200/196	205/201	210/206

Date

Time (24 hr)	21:07	21:17	21:27	21:37	21:47	21:57	22:07
Pressure - manifold (psi)	11.0	11.0	12.0	12.0	12.0	12.0	12.0
Pressure - brake (psi)	420.0	420.0	395.0	410.0	425.0	420.0	410.0
Torque (ft-lb)	240.0	180.0	175.0	175.0	165.0	150.0	195.0
RPM - turbine	12264.0	12432.0	13440.0	12600.0	12768.0	13104.0	13272.0
Horsepower - bit	3.3	2.5	2.7	2.5	2.4	2.2	2.9
Temperature - input (F)	227/229	239/236	247/244	256/253	260/256	265/264	-/265
Temperature - 1st stage (F)	212/222	229/231	-/237	-/244	-/245	248/251	253/238
Temperature - 2nd stage (F)	229/222	238/229	243/235	247/239	254/245	265/255	270/255
Temperature - 3rd stage (F)	224/222	230/229	237/234	243/236	246/246	254/253	258/252
Temperature - 4th stage (F)	217/213	223/219	228/223	233/230	238/234	246/242	254/248

A.3 Bench Test Data

HORSEPOWER VS BIT RPM : 1200 SCFM, 50 psig

TORQUE VS BIT RPM : 2400 SCFM, 50 psig

HORSEPOWER VS BIT RPM : 2400 SCFM, 50 psig

TORQUE VS BIT RPM : 2400 SCFM, 100 psig

HORSEPOWER VS BIT RPM : 2400 SCFM, 100 psig

Appendix B

Bench Test Data for Second Prototype

B.1 Temperature Test Data

TRANSMISSION TEMPERATURE VS TIME : 6116 RPM

TRANSMISSION TEMPERATURE VS TIME : 7978 RPM

TRANSMISSION TEMPERATURE VS TIME : 8883 RPM

TRANSMISSION TEMPERATURE VS TIME : 10000 RPM

B.2 Lock-Rotor Torque Test Data

LOCK-ROTOR TORQUE TEST

Date 04/14/87

Time 1:00 pm

Air Temperature 58°F

Atmospheric Pressure 12 psia

No. of Compressors 3

Pressures (psig)

<u>Casing</u>	<u>Trans</u> <u>Outlet</u>	<u>Trbn</u> <u>Outlet</u>	<u>Trbn</u> <u>Inlet</u>	<u>Upstream</u> <u>of Turbine</u>	<u>Manifold</u>	<u>Torque</u> <u>(ft-lbs)</u>
98	-	105	110	130	-	532
76	-	82	99	113	-	580
50	-	60	79	100	-	807
25	-	41	62	91	-	1112

LOCK-ROTOR TORQUE TEST

Date 04/15/87
 Time 10:00 pm
 Air Temperature 65°F
 Atmospheric Pressure 12 psia
 No. of Compressors 4

Pressures. (psig)

<u>Casing</u>	<u>Trans</u> <u>Outlet</u>	<u>Trbn</u> <u>Outlet</u>	<u>Trbn</u> <u>Inlet</u>	<u>Upstream</u> <u>of Turbine</u>	<u>Manifold</u>	<u>Torque</u> <u>(ft-lbs)</u>
127	135	150	165	180	-	-
99	108	125	152	160	-	-
75	86	100	131	145	-	-
50	67	80	120	132	-	-
24	50	67	110	124	-	-

LOCK-ROTOR TORQUE TEST

Date 04/15/87
 Time 5:42 pm
 Air Temperature 76°F
 Atmospheric Pressure 12 psia
 No. of Compressors 4

Pressures (psig)

<u>Casing</u>	<u>Trans</u> <u>Outlet</u>	<u>Trbn</u> <u>Outlet</u>	<u>Trbn</u> <u>Inlet</u>	<u>Upstream</u> <u>of Turbine</u>	<u>Manifold</u>	<u>Torque</u> <u>(ft-lbs)</u>
125	130	146	-	177	207	1184
97	109	125	-	158	195	1546
70	85	100	-	144	180	1827
50	70	85	-	135	173	2054
25	53	70	-	130	180	2344

LOCK-ROTOR TORQUE TEST

Date 04/15/87
 Time 5:30 pm
 Air Temperature 78°F
 Atmospheric Pressure 12 psia
 No. of Compressors 5

Pressures (psig)

<u>Casing</u>	<u>Trans</u> <u>Outlet</u>	<u>Trbn</u> <u>Outlet</u>	<u>Trbn</u> <u>Inlet</u>	<u>Upstream</u> <u>of Turbine</u>	<u>Manifold</u>	<u>Torque</u> <u>(ft-lbs)</u>
137	145	163	-	205	245	1667
101	120	139	-	185	228	2054
75	96	113	-	172	212	2436
48	77	93	-	163	206	2706
26	68	85	-	159	201	2948

LOCK-ROTOR TORQUE TEST

Date 04/15/87
 Time 5:00 pm
 Air Temperature 79°F
 Atmospheric Pressure 12 psia
 No. of Compressors 6

Pressures (psig)

<u>Casing</u>	<u>Trans</u> <u>Outlet</u>	<u>Trbn</u> <u>Outlet</u>	<u>Trbn</u> <u>Inlet</u>	<u>Upstream</u> <u>of Turbine</u>	<u>Manifold</u>	<u>Torque</u> <u>(ft-lbs)</u>
137	150	170	-	219	260	2030
100	121	143	-	196	241	2368
75	100	120	-	187	230	2706
50	81	100	-	180	227	3045
30	73	90	-	173	220	3190

B.3 Bench Test Data

PTP HORSEPOWER TEST

DATE : 4/ 14/ 87
TIME : 13.15
TEMPERATURE (Deg F): 60.0
OF COMPRESSORS : 3
ATMOSPHERIC PRSSR : 12.0

SYSTEM PRESSURES (psig)

MANIFOLD : 0.
INLET LINE : 97.
TURBINE INLET : 92.
TURBINE OUTLET : 73.
TRANSMISSION OUTLET: 58.
CASING : 49.

TORQUE AND RPM DATA

Ntrbne/1000	T(read)	T(ft-lbs)	Horsepower
0.00	195.00	942.44	0.0
6.50	170.00	821.61	6.1
7.40	165.00	797.45	6.7
10.25	150.00	724.95	8.4
11.50	125.00	604.12	7.9
12.50	125.00	604.12	8.6
14.80	118.00	570.29	9.6
17.00	103.00	497.80	9.6

BIT TORQUE VS TURBINE RPM - 1800 SCFM, 50 psig

THE REGRESSION POLYNOMIAL OF LINE 1 -

$$(9.717E+02) + (-5.530E+01)*X$$

THE VARIANCE - 8.773E+02

THE REGRESSION POLYNOMIAL OF LINE 1 -

$$(4.483E-02) + (2.217E+01)*X + (-2.610E+01)*X^2$$

THE VARIANCE - 2.429E-02

PTP HORSEPOWER TEST

DATE : 4/ 14/ 87
 TIME : 13.15
 TEMPERATURE (Deg F): 60.0
 # OF COMPRESSORS : 3
 ATMOSPHERIC PRSSR : 12.0

SYSTEM PRESSURES (psig)

MANIFOLD : 0.
 INLET LINE : 112.
 TURBINE INLET : 110.
 TURBINE OUTLET : 95.
 TRANSMISSION OUTLET: 80.
 CASING : 75.

TORQUE AND RPM DATA

Ntrbne/1000	T(read)	T(ft-lbs)	Horsepower
0.00	138.00	666.95	0.0
4.75	105.00	507.47	2.7
8.00	95.00	459.14	4.2
10.50	75.00	362.48	4.3

THE REGRESSION POLYNOMIAL OF LINE 1 -

$$(6.604E+02) + (-5.553E+01)*X$$

THE VARIANCE - 2.405E+02

THE REGRESSION POLYNOMIAL OF LINE 1 -

$$(-1.420E-02) + (1.526E+01)*X + (-2.607E+01)*X^2$$

THE VARIANCE - 3.322E-03

PTP HORSEPOWER TEST

DATE : 4/ 15/ 87
 TIME : 7.05
 TEMPERATURE (Deg F): 73.0
 # OF COMPRESSORS : 4
 ATMOSPHERIC PRSSR : 12.0

SYSTEM PRESSURES (psig)

MANIFOLD : 170.
 INLET LINE : 137.
 TURBINE INLET : 0.
 TURBINE OUTLET : 70.
 TRANSMISSION OUTLET: 53.
 CASING : 24.

TORQUE AND RPM DATA

Ntrbne/1000	T(read)	T(ft-lbs)	Horsepower
0.00	485.00	2344.01	0.0
6.00	450.00	2174.85	14.8
7.00	445.00	2150.69	17.1
9.00	435.00	2102.35	21.5
11.00	420.00	2029.86	25.3
13.00	400.00	1933.20	28.5
15.00	375.00	1812.38	30.8
16.00	365.00	1764.05	32.0
17.00	345.00	1667.39	32.1
18.00	330.00	1594.89	32.5
19.00	320.00	1546.56	33.3
20.00	305.00	1474.07	33.4
21.00	295.00	1425.74	33.9

BIT TORQUE VS TURBINE RPM - 2400 SCFM, 25 psig

THE REGRESSION POLYNOMIAL OF LINE 1 -

$$(2.460E+03) + (-9.258E+01)*X$$

THE VARIANCE - $3.303E+03$

HORSEPOWER VS TURBINE RPM - 2400 SCFM, 25 psig

THE REGRESSION POLYNOMIAL OF LINE 1 -

$$(-2.617E-01) + (9.242E+01)*X + (-1.228E+02)*X^2$$

THE VARIANCE - 4.768E-02

PTP HORSEPOWER TEST

DATE : 4/ 15/ 87
 TIME : 7.14
 TEMPERATURE (Deg F): 72.0
 # OF COMPRESSORS : 4
 ATMOSPHERIC PRSSR : 12.0

SYSTEM PRESSURES (psig)

MANIFOLD : 180.
 INLET LINE : 144.
 TURBINE INLET : 0.
 TURBINE OUTLET : 99.
 TRANSMISSION OUTLET: 84.
 CASING : 73.

TORQUE AND RPM DATA

Ntrbne/1000	T(read)	T(ft-lbs)	Horsepower
0.00	378.00	1826.87	0.0
6.00	330.00	1594.89	10.8
8.00	320.00	1546.56	14.0
10.00	305.00	1474.07	16.7
12.00	280.00	1353.24	18.4
14.00	255.00	1232.42	19.6
15.00	240.00	1159.92	19.7
16.00	230.00	1111.59	20.2
17.00	220.00	1063.26	20.5
18.00	210.00	1014.93	20.7
19.00	200.00	966.60	20.8
20.00	193.00	932.77	21.2
21.00	180.00	869.94	20.7

THE REGRESSION POLYNOMIAL OF LINE 1 -

$$(1.888E+03) + (-9.537E+01)*X$$

THE VARIANCE - $9.362E+02$

THE REGRESSION POLYNOMIAL OF LINE 1 -

$$(-3.168E-03) + (6.645E+01)*X + (-1.056E+02)*X^2$$

THE VARIANCE - 1.363E-02

PTP HORSEPOWER TEST

DATE : 4/ 15/ 87
 TIME : 7.26
 TEMPERATURE (Deg F): 71.0
 # OF COMPRESSORS : 5
 ATMOSPHERIC PRSSR : 12.0

SYSTEM PRESSURES (psig)

MANIFOLD : 202.
 INLET LINE : 158.
 TURBINE INLET : 0.
 TURBINE OUTLET : 85.
 TRANSMISSION OUTLET: 67.
 CASING : 26.

TORQUE AND RPM DATA

Ntrbne/1000	T(read)	T(ft-lbs)	Horsepower
0.00	610.00	2948.13	0.0
6.00	550.00	2658.15	18.1
8.00	535.00	2585.66	23.5
10.00	520.00	2513.16	28.5
12.00	505.00	2440.67	33.2
14.00	480.00	2319.84	36.8
15.00	465.00	2247.34	38.2
16.00	455.00	2199.02	39.9
17.00	440.00	2126.52	41.0
18.00	425.00	2054.03	41.9
19.00	415.00	2005.70	43.2
20.00	400.00	1933.20	43.8
21.00	380.00	1836.54	43.7

BIT TORQUE VS TURBINE RPM - 3000 SCFM, 25 psig

THE REGRESSION POLYNOMIAL OF LINE 1 -

$$(2.999E+03) + (-1.036E+02) * X$$

THE VARIANCE - 1.453E+03

HORSEPOWER VS TURBINE RPM - 3000 SCFH, 25 psig

THE REGRESSION POLYNOMIAL OF LINE 1 -

$$(-2.384E-01) + (1.089E+02)*X + (-1.273E+02)*X^2$$

THE VARIANCE - 4.372E-02

PTP HORSEPOWER TEST

DATE : 4/ 15/ 87
 TIME : 7.34
 TEMPERATURE (Deg F): 70.0
 # OF COMPRESSORS : 5
 ATMOSPHERIC PRSSR : 12.0

SYSTEM PRESSURES (psig)

MANIFOLD : 207.
 INLET LINE : 164.
 TURBINE INLET : 0.
 TURBINE OUTLET : 93.
 TRANSMISSION OUTLET: 80.
 CASING : 52.

TORQUE AND RPM DATA

Ntrbne/1000	T(read)	T(ft-lbs)	Horsepower
0.00	560.00	2706.48	0.0
6.00	490.00	2368.17	16.1
8.00	480.00	2319.84	21.0
10.00	465.00	2247.34	25.5
12.00	440.00	2126.52	28.9
14.00	422.00	2039.53	32.4
15.00	402.00	1942.87	33.0
16.00	390.00	1884.87	34.2
17.00	375.00	1812.38	34.9
18.00	360.00	1739.88	35.5
19.00	350.00	1691.55	36.4
20.00	340.00	1643.22	37.3
21.00	332.00	1604.56	38.2

THE REGRESSION POLYNOMIAL OF LINE 1 -

$$(2.735E+03) + (-1.071E+02)*X$$

THE VARIANCE - $9.147E+02$

THE REGRESSION POLYNOMIAL OF LINE 1 -

$$(-1.013E-01) + (9.644E+01)*X + (-1.203E+02)*X^2$$

THE VARIANCE - 2.837E-02

FTP HORSEPOWER TEST

DATE : 4/ 15/ 87
 TIME : 7.41
 TEMPERATURE (Deg F): 69.0
 # OF COMPRESSORS : 5
 ATMOSPHERIC PRSSR : 12.0

SYSTEM PRESSURES (psig)

MANIFOLD : 225.
 INLET LINE : 183.
 TURBINE INLET : 0.
 TURBINE OUTLET : 133.
 TRANSMISSION OUTLET: 120.
 CASING : 100.

TORQUE AND RPM DATA

Ntrbne/1000	T(read)	T(ft-lbs)	Horsepower
0.00	425.00	2054.03	0.0
6.00	375.00	1812.38	12.3
8.00	350.00	1691.55	15.3
14.00	280.00	1353.24	21.5
15.00	270.00	1304.91	22.2
16.00	260.00	1256.58	22.8
17.00	250.00	1208.25	23.3
18.00	240.00	1159.92	23.7
19.00	225.00	1087.43	23.4
20.00	220.00	1063.26	24.1
21.00	210.00	1014.93	24.2

THE REGRESSION POLYNOMIAL OF LINE 1 -

$$(2.084E+03) + (-1.028E+02) * X$$

THE VARIANCE - $3.073E+02$

THE REGRESSION POLYNOMIAL OF LINE 1 -

$$(7.382E-02) + (7.014E+01)*X + (-1.034E+02)*X^2$$

THE VARIANCE - 9.005E-03

PTP HORSEPOWER TEST

DATE : 4/ 15/ 87
 TIME : 6.55
 TEMPERATURE (Deg F): 74.0
 # OF COMPRESSORS : 6
 ATMOSPHERIC PRSSR : 12.0

SYSTEM PRESSURES (psig)

MANIFOLD : 220.
 INLET LINE : 176.
 TURBINE INLET : 0.
 TURBINE OUTLET : 92.
 TRANSMISSION OUTLET: 75.
 CASING : 30.

TORQUE AND RPM DATA

Ntrbne/1000	T(read)	T(ft-lbs)	Horsepower
0.00	695.00	3358.94	0.0
8.00	615.00	2972.30	27.0
10.00	590.00	2851.47	32.3
12.00	573.00	2769.31	37.7
14.00	540.00	2609.82	41.4
15.00	510.00	2464.83	41.9
16.00	512.00	2474.50	44.9
18.00	480.00	2319.84	47.3
20.00	440.00	2126.52	48.2
21.00	425.00	2054.03	48.9

BIT TORQUE VS TURBINE RPM - 3600 SCFM, 30 psig

THE REGRESSION POLYNOMIAL OF LINE 1 -

$$(3.446E+03) + (-1.263E+02)*X$$

THE VARIANCE - $2.911E+03$

THE REGRESSION POLYNOMIAL OF LINE 1 -

$$(-9.146E-02) + (1.241E+02)*X + (-1.530E+02)*X^2$$

THE VARIANCE - 6.041E-02

PTP HORSEPOWER TEST

DATE : 4/ 15/ 87
 TIME : 6.13
 TEMPERATURE (Deg F): 77.0
 # OF COMPRESSORS : 6
 ATMOSPHERIC PRSSR : 12.0

SYSTEM PRESSURES (psig)

MANIFOLD : 220.
 INLET LINE : 168.
 TURBINE INLET : 0.
 TURBINE OUTLET : 100.
 TRANSMISSION OUTLET: 85.
 CASING : 51.

TORQUE AND RPM DATA

Ntrbne/1000	T(read)	T(ft-lbs)	Horsepower
0.00	630.00	3044.79	0.0
5.00	550.00	2658.15	15.1
7.00	535.00	2585.66	20.5
9.00	525.00	2537.33	25.9
10.00	510.00	2464.83	27.9
11.00	500.00	2416.50	30.1
13.00	480.00	2319.84	34.2
14.00	465.00	2247.34	35.7
15.00	455.00	2199.02	37.4
15.00	450.00	2174.85	37.0
16.00	440.00	2126.52	38.6
17.00	425.00	2054.03	39.6
18.00	410.00	1981.53	40.4
19.00	400.00	1933.20	41.6
20.00	385.00	1860.71	42.2
21.00	378.00	1826.87	43.5

THE REGRESSION POLYNOMIAL OF LINE 1 -

$$(3.018E+03) + (-1.129E+02) * X$$

THE VARIANCE - 8.360E+02

THE REGRESSION POLYNOMIAL OF LINE 1 -

$$(-1.714E-01) + (1.057E+02)*X + (-1.241E+02)*X^2$$

THE VARIANCE - 1.899E-02

PTP HORSEPOWER TEST

DATE : 4/ 15/ 87
 TIME : 6.00
 TEMPERATURE (Deg F): 76.0
 # OF COMPRESSORS : 6
 ATMOSPHERIC PRSSR : 12.0

SYSTEM PRESSURES (psig)

MANIFOLD : 240.
 INLET LINE : 190.
 TURBINE INLET : 0.
 TURBINE OUTLET : 141.
 TRANSMISSION OUTLET: 125.
 CASING : 100.

TORQUE AND RPM DATA

Ntrbne/1000	T(read)	T(ft-lbs)	Horsepower
0.00	510.00	2464.83	0.0
4.00	430.00	2078.19	9.4
5.00	415.00	2005.70	11.4
6.00	420.00	2029.86	13.8
7.00	400.00	1933.20	15.3
9.00	395.00	1909.04	19.5
10.00	365.00	1764.05	20.0
10.50	375.00	1812.38	21.6
14.00	320.00	1546.56	24.5
15.00	295.00	1425.74	24.2
15.50	300.00	1449.90	25.5
17.00	285.00	1377.41	26.5
18.00	280.00	1353.24	27.6
19.00	270.00	1304.91	28.1
20.00	265.00	1280.74	29.0
21.00	255.00	1232.42	29.3

BIT TORQUE VS TURBINE RPM - 3600 SCFM, 100 psig

THE REGRESSION POLYNOMIAL OF LINE 1 -

$$(2.356E+03) + (-1.123E+02)*X$$

THE VARIANCE - 2.674E+03

THE REGRESSION POLYNOMIAL OF LINE 1 -

$$(1.858E-01) + (7.547E+01)*X + (-9.984E+01)*X^2$$

THE VARIANCE - 8.242E-02

Appendix C

Drilling Test Data for First Prototype

Bit Type : Smith 46A

Bit Size : 10 5/8 in

Initial Hole Depth : 389 ft

Hole Depth (ft)	Footage Drilled (ft)	Time (min)	Height on Bit (lbs x 1000)	Flow Rate of Air (SCFM)	Drill Pipe Pressure (psig)	Rate of Penetration (ft/hr)	Remarks
389.00	0.00	5.00	5.00	1200	85.00	0.00	03/22/87 Hole clearing (water)
394.00	5.00	3.75	8.00	1800	120.00	80.00	Strong vibrations
394.00	0.00	10.00	18.00	1800	120.00	0.00	Hole clearing (water)
396.00	2.00	2.00	10.00	1900	120.00	60.00	
396.00	0.00	3.00	27.00	2400	165.00	0.00	Hole clearing (dust)
405.00	9.00	5.40	16.00	2400	170.00	100.00	Heavy vibrations
410.00	5.00	3.00	15.00	2400	170.00	100.00	"
410.00	0.00	2.00	27.00	2400	170.00	0.00	Stall to add joint
410.00	0.00	8.00	0.00	0	0.00	0.00	Connecting joint
441.82	31.82	23.00	15.00	2400	170.00	83.00	Vibrations
441.82	0.00	2.00	26.00	2400	170.00	0.00	Stall & blow
441.82	0.00	7.50	0.00	0	0.00	0.00	Connecting joint
473.00	31.18	20.00	15.00	2400	170.00	93.50	Smooth, good returns
473.00	0.00	2.00	26.00	2400	170.00	0.00	Stall & blow
473.00	0.00	6.00	0.00	0	0.00	0.00	Connecting joint
504.29	31.29	18.00	15.00	2400	170.00	104.30	Heavy water, inc dpp
504.29	0.00	8.00	23.00	2400	173.00	0.00	Stall & trip out
=====							
504.29	0.00	10.00	0.00	1200	80->195	0.00	03/23/87 Water dump @ 145 psi
504.29	0.00	15.00	20.00	1800	180->200	0.00	Water slugs
504.29	0.00	15.00	25.00	2400	260->285	0.00	Water returns
520.89	16.60	10.00	17.00	2400	250.00	100.00	Hot cuttings
536.11	15.22	15.00	18.00	2400	260.00	85 - 45	Water, unsteady conditions
536.11	0.00	2.00	22.00	2400	260.00	0.00	Water/mud returns
536.11	0.00	95.00	0.00	0	0.00	0.00	Add joint, g-meter, mister
567.96	31.85	19.00	17.00	2400	240->300	100.50	Misting 10 bbl/hr, erratic
567.96	0.00	2.00	36.00	2400	300.00	0.00	Stall to add joint
567.96	0.00	5.00	0.00	0	0.00	0.00	Connecting joint
599.91	31.95	20.00	36.00	2400	375.00	95.85	Unsteady returns, dpp
599.91	0.00	2.00	0.00	2400	340.00	0.00	Stall to add joint
599.91	0.00	9.00	18.00	0	0.00	0.00	Connecting joint
631.54	31.63	18.00	35.00	2400	345.00	105.40	Variable dpp, returns
631.54	0.00	2.00	0.00	2400	320.00	0.00	Stall to add joint
631.54	0.00	13.00	16.00	0	0.00	0.00	Connecting joint
663.17	31.63	15.00	36.00	2400	340.00	126.52	
663.17	0.00	2.00	0.00	2400	340.00	0.00	Stall to add joint
663.17	0.00	8.00	0.00	0	0.00	0.00	Connecting joint
694.64	31.47	20.00	8 - 30	2400	360.00	94.41	Sandia G test
694.64	0.00	2.00	34.00	2400	360.00	0.00	Stall to add joint
694.64	0.00	8.00	0.00	0	0.00	0.00	Connecting joint
726.22	31.58	14.00	19.00	2400	340.00	135.34	Hot dirt returns
726.22	0.00	2.00	38.00	2400	340.00	0.00	Stall to add joint
726.22	0.00	4.00	0.00	0	0.00	0.00	Connecting joint
757.30	31.08	10.00	24.00	3000	470.00	186.48	5 compressors
757.30	0.00	2.00	40.00	3000	470.00	0.00	Stall attempt-drop cmpr
757.30	0.00	0.50	40.00	2400	not rcrd	0.00	Stall to add joint
757.30	0.00	6.00	0.00	0	0.00	0.00	Connecting joint
787.72	30.42	10.00	30.00	3000	450.00	182.52	Hot dirt returns
787.72	0.00	10.00	40.00	3000	450.00	0.00	Stall attempt-drop cmpr
787.72	0.00	0.50	40.00	2400	not rcrd	0.00	Stall to add joint
787.72	0.00	6.00	0.00	0	0.00	0.00	Connecting joint
788.72	1.00	2.00	40.00	3600	500->900	none	no return
788.72	0.00	1.00	40.00	3600	900->580	none	Mud plug - trip out

Total Footage Drilled :
Net Drilling Time (hrs) :399.72
3.77Average Rate of Penetration (ft/hr) : 106.05
Total Time (hrs) : 8.39

Appendix D
Bench Test Programs

D.1 PROGRAM TTST

```
*****  
** Calculate the maximum theoretical performance parameters for **  
** the turbine, under the given test conditions, and print out **  
** test data in dimensionless form. P.W. Johnson 2/28/87 **  
*****
```

```
INCLUDE 'CELE'
```

```
* DO 200 J=1,3  
CALL DATA  
DO 100 i=1,Nseg  
  IF (rL(i).eq.0) THEN  
    A = PI/4.0*D2(i)**2  
    write(6,*)'a=',a,'p(i-1)=',p(i-1),'tmp=',tmp  
    P(i) = Pchku(A,P(i-1),Tmp,Ginj,rk)  
  ELSE  
    Dftr = (D2(i)-D1(i))**1.33*(D2(i)**2-D1(i)**2)**2  
    P(i) = SQRT(P(i-1)**2+.0375*Ginj**2*Tmp*rL(i)/Dftr)  
  ENDIF  
100 CONTINUE  
CALL TURBNE(P(Nseg),P(Nseg-1))  
CALL OUTPUT  
*200 CONTINUE  
  
STOP  
END
```

SUBROUTINE DATA

```
*****
**  Read in the flow data.                                **
*****
```

```
CHARACTER DAT*15
INCLUDE 'CELE'
```

```
WRITE (6,*) 'ENTER THE INPUT FILE NAME:'
```

```
READ (5, '(A)') DAT
```

```
OPEN (UNIT=10, FILE=DAT)
```

```
READ (10,*) Tmp, Ncomp, Pb, Psurf, RtrD, NT, Nseg
```

```
PI = 3.14159
```

```
Tmp = Tmp+460.0
```

```
Q = FLOAT(Ncomp)*750.*520./tmp*Psurf/14.7
```

```
Ginj = Q*29.0/(379.0*60.0)
```

```
Pb = (Pb+Psurf)*144.0
```

```
P(0) = Pb
```

```
RtrCro = RtrD*3.14159/12.0
```

```
rk = 1.4
```

```
rkftr = (rk-1.0)/rk
```

```
DO 100 i=1, Nseg
```

```
  READ (10,*) D2(i), D1(i), rL(i)
```

```
  D1(i) = D1(i)/12.0
```

```
  D2(i) = D2(i)/12.0
```

```
100 CONTINUE
```

```
DO 200 i=1, NT
```

```
  READ (10,*) rN(i), T(i)
```

```
  T(i) = T(i)*4.833
```

```
  rN(i) = rN(i)*1000.0/2.0
```

```
200 CONTINUE
```

```
RETURN
```

```
END
```

```

FUNCTION Pchku(A,P2,T1,Ginj,rk)
*****
** Calculate the upstream choke pressure for critical or sub- **
** critical bit chokes or for a supersonic turbine choke. **
*****

    rKftr=(rk-1.0)/rk
    CrPrto=(2.0/(rk+1.0))*(rk/(rk-1.0))

***** Find the root *****
    P =1.01*P2
    TEST=1.0
    del=P/1000.0
60  IF (TEST.GT.(.0001)) THEN
        F1=F(A,P,P2,T1,Ginj,rk,rKftr)
        Px=P+del
        F2=F(A,Px,P2,T1,Ginj,rk,rKftr)
        dP=-(del*F1)/(F2-F1)
        TEST=ABS(dP/P)
        P=P+dP
        GOTO 60
    ENDIF
    Pchku=P

***** Check for critical flow at the bit *****
    IF ((P2/Pchku).LT.CrPrto) THEN
        Pchku=Ginj/(A*SQR(32.2*rk/(53.3*T1)
1  *((2.0/(rk+1.0))*(rk+1.0)/(rk-1.0))))
    ENDIF

    RETURN
    END

FUNCTION F(A,P,P2,T1,Ginj,rk,rkftr)
*****
** Choke flow equation, so posed as to be zero if the upstream **
** pressure has been correctly found. P is the upstream pressure.**
*****

    X=64.4/(rkftr*53.3*T1)*((P2/P)**(2.0/rk)-(P2/P)**((rk+1.0)/rk))
    IF (X.LE.0.0) THEN
        WRITE(Ndvce,*)' X=',X
        WRITE(Ndvce,*)' P2,P:',P2,P
    ENDIF
    X=(A*SQR(X))
    F=Ginj/X-P

    RETURN
    END

```

SUBROUTINE TURBNE(Pin,Pout)

** Calculates turbine operating parameters. **

INCLUDE 'CBLK'

*** Tm, HPm, RPMm ***

Vcfn = $2.0 \times 32.2 \times 53.3 / \text{rkftr}$
Pfctr = $1.0 - (\text{Pout} / \text{Pin}) \times \text{rkftr}$
Vinlet = $\text{SQRT}(\text{Vcfn} \times \text{Tm} \times \text{Pfctr})$
HPm = $.5 \times \text{Ginj} / (550.0 \times 32.2) \times \text{Vinlet}^2$
rNm = $60.0 \times \text{Vinlet} / \text{RtrCrc}$
Tm = $4. \times \text{HPm} \times 168. \times 5250. / \text{rNm}$

RETURN
END

SUBROUTINE OUTPUT

```
*****
**      Print out the results.                      **
*****
```

```
CHARACTER OUTFILE*15
INCLUDE 'CBLK'
```

```
WRITE (6,*) 'ENTER THE OUTPUT FILE NAME:'
READ (5,'(A)')OUTFILE
OPEN(UNIT=11,FILE=OUTFILE)
```

```
Abt = 3.14159/4.0*D2(Nseg-3)**2*144.0
Ant = 3.14159/4.0*D2(Nseg)**2*144.0
Pb = Pb/144.0-Psurf
Prto= P(Nseg-1)/P(Nseg)
Tmp = Tmp-460.0
WRITE (11,'(A,F12.3)')'TURBINE NOZZLE AREA      (sq in):',Ant
WRITE (11,'(A,F12.3)')'BIT      CHOKE AREA      (sq in):',Abt
WRITE (11,'(A,F12.3)')'AIR FLOW RATE      (scfm ): ',Q
WRITE (11,'(A,F12.3)')'AIR TEMPERATURE      (deg F): ',Tmp
WRITE (11,'(A,F12.3)')'SYSTEM BACK PRESSURE  (psig ): ',Pb
WRITE (11,'(A,F12.3)')'TURBINE PRESSURE RATIO      : ',Prto
WRITE (11,'(A,F12.3)')'Nm, MAX THRTCL TRBN SPD(rpm ): ',rNm
WRITE (11,'(A,F12.3)')'HPm, MAX THRTCL HORSEPOWER      : ',HPm
WRITE (11,'(A,F12.3)')'Tm, MAX THRTCL TORQUE (ft-#s): ',Tm
```

```
IF (NT.GT.0) THEN
```

```
  WRITE (11,*)' '
```

```
  WRITE (11,'(T3,A,T12,A,T24,A,T33,A,T41,A,T50,A)')
1'N(rpm)', 'T(ft-#s)', 'HP', 'N/Nm', 'T/Tm', 'HP/HPm'
  WRITE (11,'(A)')
```

```
1' _____ ,
```

```
DO 200 i=1,NT
```

```
  Z = T(i)/Tm
```

```
  X = rN(i)/rNm
```

```
  H = T(i)*rN(i)/(168.*5250.)
```

```
  E = H/HPm
```

```
  WRITE (11,'(2F9.2,4F9.3)')rN(i),T(i),H,X,Z,E
```

```
*      WRITE (12,'(F6.3,A,F6.3)')X,', ',E
```

```
*      WRITE (13,'(F6.3,A,F6.3)')X,', ',Z
```

```
200 CONTINUE
```

```
ENDIF
```

```
RETURN
```

```
END
```

** SAMPLE DATA FILE T5-25 - 3000 SCFM, 25 psig **

71,5,26,12,5.5,13,7
15.0,5.0,2.67
15.0,8.5,15.5
2.1,0,0
1.948,0.0,0.0
2.0,0.0,5.0
7.0,6.2,47.5
1.2037,0.0,0.0
0,610
6,550
8,535
10,520
12,505
14,480
15,465
16,455
17,440
18,425
19,415
20,400
21,380

 ** SAMPLE OUTPUT FROM TTST USING T5-25 **

TURBINE NOZZLE AREA (sq in): 1.138
 BIT CHOKE AREA (sq in): 2.980
 AIR FLOW RATE (scfm): 2997.809
 AIR TEMPERATURE (deg F): 71.000
 SYSTEM BACK PRESSURE (psig): 26.000
 TURBINE PRESSURE RATIO : 0.659
 Nm, MAX THRCTL TRBN SPD(rpm): 35253.289
 HPm, MAX THRCTL HORSEPOWER : 77.254
 Tm, MAX THRCTL TORQUE (ft-#s): 7731.267

N(rpm)	T(ft-#s)	HP	N/Nm	T/Tm	HP/HPm
0.00	2948.13	0.000	0.000	0.381	0.000
3000.00	2658.15	9.041	0.085	0.344	0.117
4000.00	2585.66	11.726	0.113	0.334	0.152
5000.00	2513.16	14.247	0.142	0.325	0.184
6000.00	2440.67	16.603	0.170	0.316	0.215
7000.00	2319.84	18.411	0.199	0.300	0.238
7500.00	2247.34	19.110	0.213	0.291	0.247
8000.00	2199.02	19.946	0.227	0.284	0.258
8500.00	2126.52	20.494	0.241	0.275	0.265
9000.00	2054.03	20.959	0.255	0.266	0.271
9500.00	2005.70	21.603	0.269	0.259	0.280
10000.00	1933.20	21.918	0.284	0.250	0.284
10500.00	1836.54	21.864	0.298	0.238	0.283

D.2 PROGRAM MAIN

```
*****  
** Select system parameters that give the best fit to the **  
** pressure data. P.W. Johnson 4/18/87 **  
*****
```

```
INCLUDE 'CELK'
```

```
CALL DATA
```

```
CALL GOLDEN(DL,DH,X,Y)
```

```
CALL OUTPUT(X,Y)
```

```
STOP
```

```
END
```

SUBROUTINE DATA

```
*****
**  Read in the flow data.  **
*****
```

```
CHARACTER DAT*15
INCLUDE 'CBLK'
```

```
WRITE (6,*) 'ENTER THE INPUT FILE NAME:'
READ (5, '(A)') DAT
OPEN (UNIT=10, FILE=DAT)
```

```
PI      = 3.14159
rk      = 1.4
rKftr   = (rk-1.0)/rk
```

```
READ (10,*) Nseg, DH, DL
```

```
DO 100 i=1, Nseg
  READ (10,*) D2(i), D1(i), rL(i)
  D1(i) = D1(i)/12.0
  D2(i) = D2(i)/12.0
```

```
100 CONTINUE
```

```
READ (10,*) Ntst
```

```
Ndata = 0
DO 200 i=1, Ntst
  READ (10,*) Temp, Psurf, rNcomp(i)
  Tmp(i)=Temp+460.0
  G(i) = rNcomp(i)*750.*Psurf/14.7*520./Tmp(i)*29./(379.*60.)
  DO 200 j=0, 4
    READ (10,*) Pg
    IF (Pg.NE.0) THEN
      Pt(i, j)=(Pg+Psurf)*144.0
      Ndata = Ndata+1
    ENDIF
```

```
200 CONTINUE
```

```
RETURN
END
```

SUBROUTINE GOLDEN(X1,X2,X,Y)

** Select the optimum system parameter. **

INCLUDE 'CBLK'

IF (X1.EQ.X2) THEN

X=0

Y=FN(X)

GOTO 300

ELSEIF (X1.GT.X2) THEN

XA=X2

XD=X1

ELSE

XA=X1

XD=X2

ENDIF

YA = FN(XA)

YD = FN(XD)

XB = XD-.618034*(XD-XA)

YB = FN(XB)

XC = XA+.618034*(XD-XA)

YC = FN(XC)

100 IF (YC.LT.YB) THEN

XA = XB

YA = YB

XB = XC

YB = YC

XC = XA+.618034*(XD-XA)

YC = FN(XC)

ELSE

XD = XC

YD = YC

XC = XB

YC = YB

XB = XD-.618034*(XD-XA)

YB = FN(XB)

ENDIF

TEST = 2.*(XD-XA)/(X1+X2)

IF (TEST.LT.(.0001)) THEN

X=(XD+XA)/2.0

Y=FN(X)

ELSE

GOTO 100

ENDIF

300 RETURN

END

FUNCTION FN(X)

```

*****
** Find the average pressure error as a function of the system **
** parameter, X. **
*****

```

INCLUDE 'CBLK'

```

FN = 0.0
Nj = Nseg-3
DO 200 i=1,Ntst
  Pc(0)=Pt(i,0)
  DO 100 j=1,Nseg
    IF (rL(j).EQ.0) THEN
      IF (D2(j).EQ.0.0) THEN
        A = PI/4.0*X**2
      ELSE
        A = PI/4.0*D2(j)**2
      ENDIF
      Pc(j) = Pchku(A,Pc(j-1),Tmp(i),G(i),rk)
    ELSE
      Dftr = (D2(j)-D1(j))**1.33*(D2(j)**2-D1(j)**2)**2
      IF (rL(j).LT.0) THEN
        Temp=Pc(j-1)**2+.0375*G(i)**2*Tmp(i)*X/Dftr
      ELSE
        Temp=Pc(j-1)**2+.0375*G(i)**2*Tmp(i)*rL(j)/Dftr
      ENDIF
      Pc(j)=SQRT(Temp)
    ENDIF
    Njt = j-(Nj-1)
    IF ((j.GE.Nj).AND.(Pt(i,Njt).NE.0))
      FN=FN+ABS(Pt(i,Njt)-Pc(j))
1
100 CONTINUE
200 CONTINUE

FN = FN/(144.*FLOAT(Ndata))

RETURN
END

```

```

FUNCTION Pchku(A,P2,T1,Ginj,rk)
*****
** Calculate the upstream choke pressure for critical or sub- **
** critical bit chokes or for a supersonic turbine choke. **
*****

    rKftr=(rk-1.0)/rk
    CrPrto=(2.0/(rk+1.0))*(rk/(rk-1.0))

***** Find the root *****
    P =1.01*P2
    TEST=1.0
    del=P/1000.0
60  IF (TEST.GT.(.0001)) THEN
        F1=F(A,P,P2,T1,Ginj,rk,rKftr)
        Px=P+del
        F2=F(A,Px,P2,T1,Ginj,rk,rKftr)
        dP=-(del*F1)/(F2-F1)
        TEST=ABS(dP/P)
        P=P+dP
        GOTO 60
    ENDIF
    Pchku=P

***** Check for critical flow at the bit *****
    IF ((P2/Pchku).LT.CrPrto) THEN
        Pchku=Ginj/(A*SQRT(32.2*rk/(53.3*T1)
        1 *((2.0/(rk+1.0))*(rk+1.0)/(rk-1.0))))
    ENDIF

    RETURN
END
FUNCTION F(A,P,P2,T1,Ginj,rk,rkftr)
*****
** Choke flow equation, so posed as to be zero if the upstream **
** pressure (P) has been correctly found. **
*****

    X=64.4/(rkftr*53.3*T1)*((P2/P)**(2.0/rk)-(P2/P)**((rk+1.0)/rk))
    IF (X.LE.0.0) THEN
        WRITE(Ndvcce,*)' X=',X
        WRITE(Ndvcce,*)' P2,P:',P2,P
    ENDIF
    X=(A*SQRT(X))
    F=Ginj/X-P

    RETURN
END

```

SUBROUTINE OUTPUT(X,Y)

```
*****
** Print out the results.
*****
```

```
CHARACTER OUTFILE*15
INCLUDE 'CELR'
```

```
WRITE (6,*) 'ENTER THE OUTPUT FILE NAME:'
READ (5, '(A)') OUTFILE
OPEN(UNIT=11, FILE=OUTFILE)
```

```
IF (X.NE.0) THEN
  DO 20 j=1,Nseg
    IF (rL(j).EQ.0) THEN
      IF (D2(j).EQ.0) THEN
        Xt = X*12.0
        WRITE (11, '(T3,A,T35,F7.2,/)' )
1      'DUMMY CHOKE DIA (in):',Xt
      ENDIF
    ELSE
      IF (rL(j).EQ.-1) THEN
        WRITE (11, '(T3,A,T35,F7.2,/)' )
1      'DUMMY LENGTH (ft):',X
      ENDIF
    ENDIF
  20 CONTINUE
ENDIF
```

```
WRITE (11, '(T3,A,T35,F6.2)' )
1 'AVERAGE PRESSURE ERROR (psi):',Y
```

```
Perr=0.0
Nj = Nseg-3
DO 200 i=1,Ntst
  N=INT(rNcomp(i))
  WRITE (11, '(/,T3,A,T28,I2)' ) 'NUMBER OF COMPRESSORS:',N
  WRITE (11, '(T3,A,T22,A,T37,A)' )
1 'TEST PRSSR', 'CALC PRSSR', 'DIFF'
  Pc(0)=Pt(i,0)
  DO 100 j=1,Nseg
    IF (rL(j).EQ.0) THEN
      IF (D2(j).EQ.0.0) THEN
        A = PI/4.0*X**2
      ELSE
        A = PI/4.0*D2(j)**2
```

```

        ENDIF
        Pc(j) = Pchku(A,Pc(j-1),Tmp(i),G(i),rk)
    ELSE
        Dftr = (D2(j)-D1(j))**1.33*(D2(j)**2-D1(j)**2)**2
        IF (rL(j).LT.0) THEN
            Temp=Pc(j-1)**2+.0375*G(i)**2*Tmp(i)*X/Dftr
        ELSE
            Temp=Pc(j-1)**2+.0375*G(i)**2*Tmp(i)*rL(j)/Dftr
        ENDIF
        Pc(j)=SQRT(Temp)
    ENDIF
    del = 0.0
    Ptt = 0.0
    IF (j.GE.Nj) THEN
        Pcc=Pc(j)/144.0-Psurf
        Njt=j-(Nj-1)
        IF (Pt(i,Njt).NE.0) THEN
            Ptt=Pt(i,Njt)/144.0-Psurf
            del=pcc-ptt
        ENDIF
        Perr=Perr+ABS(del)
        WRITE (11,'(T5,F8.1,T20,F8.1,T35,F7.1)')Ptt,Pcc,del
    ENDIF
100    CONTINUE
200    CONTINUE

    Perr=Perr/FLOAT(Ndata)
    WRITE (11,'(T3,A,T35,F6.2)')
1    'AVERAGE PRESSURE ERROR (psi):',Perr

    RETURN
    END

```

** SAMPLE DATA FILE T5 - ALL LOCK ROTOR TORQUE TEST PRESSURE DATA **

8,100,4
15.0,5.0,2.67
15.0,8.5,15.5
2.1,0,0
1.948,0.0,0.0
2.0,0.0,5.0
7.0,6.2,-1
1.2037,0.0,0.0
2.0,0.0,20.0

24
58,12,3
25
41
62
79
91
58,12,3

50
60
79
0
100
58,12,3

76
82
99
0
113
58,12,3

98
105
110
0
130
65,12,4

24
50
67
110
124
65,12,4

50
67
80
120

132
65, 12, 4
75
86
100
131
145
65, 12, 4
99
108
125
152
160
65, 12, 4
127
135
150
165
180
76, 12, 4
25
53
70
0
130
76, 12, 4
50
70
85
0
135
76, 12, 4
70
85
100
0
144
76, 12, 4
97
109
125
0
158
76, 12, 4
125
130
146
0
177

78,12,5

26

68

85

0

159

78,12,5

48

77

93

0

163

78,12,5

75

96

113

0

172

78,12,5

101

120

139

0

185

78,12,5

137

145

163

0

205

79,12,6

30

73

90

0

173

79,12,6

50

81

100

0

180

79,12,6

75

100

120

0

187

79,12,6

100
121
143
0
196
79,12,6
137
150
170
0
219

 ** SAMPLE OUTPUT DATA FILE FROM MAIN USING T5 **

DUMMY LENGTH (ft): 47.52

AVERAGE PRESSURE ERROR (psi): 3.10

NUMBER OF COMPRESSORS: 3

TEST PRSSR	CALC PRSSR	DIFF
41.0	40.4	-0.6
62.0	54.5	-7.5
79.0	82.9	3.9
91.0	90.2	-0.8

NUMBER OF COMPRESSORS: 3

TEST PRSSR	CALC PRSSR	DIFF
60.0	59.9	-0.1
79.0	70.7	-8.3
0.0	93.3	0.0
100.0	100.0	0.0

NUMBER OF COMPRESSORS: 3

TEST PRSSR	CALC PRSSR	DIFF
82.0	83.1	1.1
99.0	91.6	-7.4
0.0	109.5	0.0
113.0	115.3	2.3

NUMBER OF COMPRESSORS: 3

TEST PRSSR	CALC PRSSR	DIFF
105.0	103.8	-1.2
110.0	110.8	0.8
0.0	125.8	0.0
130.0	131.0	1.0

NUMBER OF COMPRESSORS: 4

TEST PRSSR	CALC PRSSR	DIFF
50.0	49.6	-0.4
67.0	70.1	3.1
110.0	110.6	0.6
124.0	120.6	-3.4

NUMBER OF COMPRESSORS: 4

TEST PRSSR	CALC PRSSR	DIFF
67.0	66.8	-0.2
80.0	83.6	3.6
120.0	118.1	-1.9
132.0	127.5	-4.5

NUMBER OF COMPRESSORS: 4		
TEST PRSSR	CALC PRSSR	DIFF
86.0	87.4	1.4
100.0	101.3	1.3
131.0	130.1	-0.9
145.0	138.8	-6.2

NUMBER OF COMPRESSORS: 4		
TEST PRSSR	CALC PRSSR	DIFF
108.0	108.9	0.9
125.0	120.5	-4.5
152.0	145.1	-6.9
160.0	153.0	-7.0

NUMBER OF COMPRESSORS: 4		
TEST PRSSR	CALC PRSSR	DIFF
135.0	135.0	0.0
150.0	144.7	-5.3
165.0	165.4	0.4
180.0	172.4	-7.6

NUMBER OF COMPRESSORS: 4		
TEST PRSSR	CALC PRSSR	DIFF
53.0	49.7	-3.3
70.0	69.8	-0.2
0.0	109.6	0.0
130.0	119.5	-10.5

NUMBER OF COMPRESSORS: 4		
TEST PRSSR	CALC PRSSR	DIFF
70.0	66.4	-3.6
85.0	83.1	-1.9
0.0	117.0	0.0
135.0	126.3	-8.7

NUMBER OF COMPRESSORS: 4		
TEST PRSSR	CALC PRSSR	DIFF
85.0	82.9	-2.1
100.0	97.0	-3.0
0.0	126.4	0.0
144.0	135.2	-8.8

NUMBER OF COMPRESSORS: 4		
TEST PRSSR	CALC PRSSR	DIFF
109.0	106.9	-2.1
125.0	118.4	-6.6
0.0	142.9	0.0
158.0	150.7	-7.3

NUMBER OF COMPRESSORS: 4		
TEST PRSSR	CALC PRSSR	DIFF
130.0	132.9	2.9
146.0	142.6	-3.4
0.0	163.1	0.0
177.0	170.1	-6.9

NUMBER OF COMPRESSORS: 5		
TEST PRSSR	CALC PRSSR	DIFF
68.0	61.2	-6.8
85.0	87.2	2.2
0.0	138.4	0.0
159.0	150.8	-8.2

NUMBER OF COMPRESSORS: 5		
TEST PRSSR	CALC PRSSR	DIFF
77.0	73.3	-3.7
93.0	96.4	3.4
0.0	143.0	0.0
163.0	155.1	-7.9

NUMBER OF COMPRESSORS: 5		
TEST PRSSR	CALC PRSSR	DIFF
96.0	93.5	-2.5
113.0	113.0	0.0
0.0	153.1	0.0
172.0	164.5	-7.5

NUMBER OF COMPRESSORS: 5		
TEST PRSSR	CALC PRSSR	DIFF
120.0	115.6	-4.4
139.0	132.2	-6.8
0.0	166.7	0.0
185.0	177.3	-7.7

NUMBER OF COMPRESSORS: 5		
TEST PRSSR	CALC PRSSR	DIFF
145.0	148.3	3.3
163.0	161.7	-1.3
0.0	190.3	0.0
205.0	199.7	-5.3

NUMBER OF COMPRESSORS: 6		
TEST PRSSR	CALC PRSSR	DIFF
73.0	74.4	1.4
90.0	106.0	16.0
0.0	167.9	0.0
173.0	182.8	9.8

NUMBER OF COMPRESSORS: 6		
TEST PRSSR	CALC PRSSR	DIFF
81.0	84.0	3.0
100.0	113.2	13.2
0.0	171.3	0.0
180.0	185.9	5.9

NUMBER OF COMPRESSORS: 6		
TEST PRSSR	CALC PRSSR	DIFF
100.0	101.0	1.0
120.0	126.6	6.6
0.0	178.7	0.0
187.0	192.9	5.9

NUMBER OF COMPRESSORS: 6		
TEST PRSSR	CALC PRSSR	DIFF
121.0	120.9	-0.1
143.0	143.3	0.3
0.0	189.6	0.0
196.0	203.0	7.0

NUMBER OF COMPRESSORS: 6		
TEST PRSSR	CALC PRSSR	DIFF
150.0	153.1	3.1
170.0	171.6	1.6
0.0	210.5	0.0
219.0	222.7	3.7
AVERAGE PRESSURE ERROR (psi):		3.10

Appendix E
Drilling Test Programs

E.1 PROGRAM SURFVLV

```

*****
** COMPUTES PRESSURES THROUGH SYSTEM ON RIG FLOOR WITH BYPASS **
** OPEN OR CLOSED. **
*****

```

```

IMPLICIT REAL(A-Z)
INTEGER IN,OUT,I,J,MAX,TRBN,BYPASS
CHARACTER COUT,FNAME*10
DIMENSION PRSSR(0:20),TMP(0:20),LNGTH(20),Anz(20),ID(20),OD(20)

```

```

*****
** INPUT DATA - FROM SV--DAT **
*****
** Q          FLOW RATE                      SCFM          **
** E          TURBINE EFFICIENCY              **
** Uf         UTILIZATION FACTOR              **
** MAX        NUMBER OF NODES IN STRING       **
** TRBN, BYPASS PLACEMENT OF THESE NODES IN STRING **
** PRSSR(0)    AMBIENT PRESSURE                PSI           **
** T(0)        AMBIENT TEMPERATURE             DEG F         **
** TMP(I)      TEMPERATURES AT NODES           "              **
** Anz(I)      AREA OF NOZZLES AT NODES (0 = NONE) SQ FT      **
** ID(I)       ID AT NODES                     IN             **
** OD(I)       OD AT NODES (0 = PIPE)          "              **
** LNGTH(I)    LENGTH OF NODES (0 = CHOKE)     FT             **
*****

```

```

PI=3.1415927
FT=12.
FT2=FT**2
FtoR=460.
R=53.3
DENa=0.0764
Dtw=0.4583
GRAV=32.2

```

```

WRITE(*,*)' ENTER DATA FILE NAME : '
READ(*,'(A)')FNAME
IN=3
OPEN(IN,FILE=FNAME)

```

```

WRITE(*,*)' OUTPUT TO CONSOLE OR PRINTER (C/P) : '
READ(*,'(A)')COUT
IF(COUT.EQ.'P'.OR.COUT.EQ.'p') THEN
  OUT=2
  FNAME='PRN'

```

```

ELSE
  OUT=6
  FNAME='CON'
ENDIF
OPEN(OUT,FILE=FNAME)

READ(IN,*)Q,E,Uf
READ(IN,*)PRSSR(0),TMP(0)
PRSSR(0)=PRSSR(0)*FT2
TMP(0)=TMP(0)+FtoR
READ(IN,*)MAX, TRBN, BYPASS
DO 10 I=1,MAX
  READ(IN,*)TMP(I),LNGTH(I),Anz(I),ID(I),OD(I)
  TMP(I)=TMP(I)+FtoR
  ID(I)=ID(I)/FT
  OD(I)=OD(I)/FT
10 CONTINUE

J=0
MFR=Q*DENA/60.
F=0.35
IF(Anz(BYPASS).EQ.0) F=0
1 MV=F*MFR
MT=MFR-MV

DO 20 I=1,MAX
  IF(LNGTH(I).EQ.0) THEN
    PRSSR(I)=PRSSR(I-1)*(1.+(MT/(2.0564*Anz(I)*PRSSR(I-1))))**2
    *TMP(I))**3.5
  C IF(PRSSR(I-1)/PRSSR(I).LT.0.52) THEN
    PRSSR(I)=MT*SQRT(TMP(I))/(0.532*Anz(I))
  ENDIF
  ELSE
    PRSSR(I)=SQRT(PRSSR(I-1)**2+LNGTH(I)*0.0375*MT**2*TMP(I)/
  C ((ID(I)-OD(I))**1.333*(ID(I)**2-OD(I)**2)**2))
  ENDIF

  IF(I.EQ.BYPASS) THEN
    IF(F.GT.0) THEN
      PCHK=PRSSR(0)*(1.+(MV/(2.0564*Anz(I)*PRSSR(0))))**2*TMP(0))
    C **3.5
      IF(PCHK/PRSSR(0).LT.0.52) THEN
        PCHK=MV*SQRT(TMP(0))/(0.532*Anz(I))
      ENDIF
      DIFF=PRSSR(I)-PCHK
      IF(ABS(DIFF).GT.FT2*2) THEN
        J=J+1
        IF(MOD(J,20).EQ.0) PAUSE
        WRITE(*,'(IX,'Pby =',F12.4,' PCHK =',F12.4)')

```

```

C      PRSSR(1),PCHK
      IF(DIFF.LT.0) THEN
        F=F-.001
      ELSE
        F=F+.001
      ENDIF
      GOTO 1
    ENDIF
  ENDIF
ENDIF
20  CONTINUE

Vn=SQRT(11980.*TMP(TRBN)*(1.-(PRSSR(TRBN-1)/PRSSR(TRBN))
C    **0.2857))
HPt=E*0.5*MT*Vn**2/(GRAV*550)
RPMt=60*Uf*Vn/(PI*Dtw)
RPMb=RPMt/168.
TRQb=2.*5250.*HPt/RPMb

WRITE(OUT,'(1X,'AT',15,' SCFM '))INT(Q)
IF(Anz(BYPASS).EQ.0) THEN
  WRITE(OUT,'(10X,'WITH BYPASS CLOSED'))
ELSE
  WRITE(OUT,'(10X,'WITH BYPASS OPEN'))
ENDIF
WRITE(OUT,'(1X)')
DO 30 I=1,MAX
  WRITE(OUT,100)'PRESSURE AT ',I,' :',(PRSSR(I)-PRSSR(0))
C    /FT2,' psig'
30  CONTINUE
WRITE(OUT,'(1X)')
WRITE(OUT,200)'MFR THRU THE VALVE      :',MV,' lb-mass/sec'
WRITE(OUT,200)'MFR THRU THE TURBINE    :',MT,' lb-mass/sec'
WRITE(OUT,'(1X)')
WRITE(OUT,200)'VELOCITY AT NOZZLES      :',Vn,' ft-sec'
WRITE(OUT,200)'PEAK TURBINE HP          :',HPt,' '
WRITE(OUT,200)'PEAK TURBINE RPM         :',RPMt,' '
WRITE(OUT,200)'PEAK BIT RPM             :',RPMb,' '
WRITE(OUT,200)'STALL TORQUE AT BIT      :',TRQb,' ft-lbs'

100  FORMAT(1X,A,14,A,F12.4,A)
200  FORMAT(1X,A,F12.4,A)
STOP
END

```

 ** SAMPLE DATA FILE SV18DAT - 1800 SCFM WITH BYPASS OPEN **

1800, 0.35, 0.46	;Q,e,Uf
12, 40	;P,T AT SURFACE
8, 4, 5	;MAX # OF NODES, TURB, BYPASS NODE
35, 0, 2.070860E-2, 0, 0	;T,L,Anz,ID,OD AT BIT
40, 5, 0, 2, 0	;T,L,Anz,ID,OD AT THRUST
40, 4, 0, 7, 6.25	;T,L,Anz,ID,OD AT GEARBOX
40, 0, 7.902777E-3, 0, 0	;T,L,Anz,ID,OD AT TURBINE
65, 4, 6.3916E-3, 2, 0	;T,L,Anz,ID,OD AT BYPASS
65, 30, 0, 2.75, 0	;T,L,Anz,ID,OD AT KELLY
65, 55, 0, 3.00, 0	;T,L,Anz,ID,OD AT HOSE
65, 30, 0, 3.80, 0	;T,L,Anz,ID,OD AT STAND PIPE

 ** SAMPLE OUTPUT AT 1800 SCFM WITH BYPASS OPEN **

PRESSURE AT	1 :	9.3432	psig
PRESSURE AT	2 :	12.5644	psig
PRESSURE AT	3 :	13.9665	psig
PRESSURE AT	4 :	44.2100	psig
PRESSURE AT	5 :	45.3043	psig
PRESSURE AT	6 :	46.7729	psig
PRESSURE AT	7 :	48.4216	psig
PRESSURE AT	8 :	48.6725	psig

MFR THRU THE VALVE	:	0.7701	lb-mass/sec
MFR THRU THE TURBINE	:	1.5219	lb-mass/sec

VELOCITY AT NOZZLES	:	1089.0353	ft-sec
PEAK TURBINE HP	:	17.8355	
PEAK TURBINE RPM	:	20876.1934	
PEAK BIT RPM	:	124.2631	
STALL TORQUE AT BIT	:	1507.0690	ft-lbs

E.2 PROGRAM FIELDPR

```
*****
** COMPUTES TURBINE CHARACTERISTICS, DRILLING PARAMETERS, AND **
** PRESSURES THROUGH ANNULUS, MOTOR AND DRILL STRING WHILE **
** DRILLING. **
*****
```

```
IMPLICIT REAL(A-Z)
INTEGER IN,OUT,I,J,N
CHARACTER COUT,FNAME*10
```

```
*****
** INPUT DATA - FROM FP--DAT **
*****
** Q          FLOW RATE          SCFM          **
** E          TURBINE EFFICIENCY **
** Uf         UTILIZATION FACTOR **
** DENch      CHIP DENSITY       LB/FT        **
** ROP        RATE OF PENETRATION FT/HR        **
** WOB        WEIGHT ON BIT      1000 LB       **
** Ps         AMBIENT PRESSURE   PSI          **
** Ts         AMBIENT TEMPERATURE DEG F       **
** Tbh        TEMPERATURE OF BOREHOLE          **
** Tbn        TEMPERATURE AT BIT NOZZLES       **
** Tub        TEMPERATURE UPSTREAM OF BIT      **
** Tutr       TEMPERATURE UPSTREAM OF TURBINE  **
** Abn        AREA OF BIT NOZZLES              SQ FT    **
** Atn        AREA OF TURBINE NOZZLES          **
** Lbh        LENGTH OF BOREHOLE              FT      **
** Lth        LENGTH OF THRUST                 **
** Lgb        LENGTH OF GEARBOX                 **
** Lby        LENGTH OF BYPASS                 **
** Ldc2       LENGTH OF LOWER DRILL COLLARS     **
** Ldc1       LENGTH OF UPPER DRILL COLLARS     **
** Ldp        LENGTH OF DRILL PIPE              **
** Lk         LENGTH OF KELLY                  **
** Lh         LENGTH OF ROTARY HOSE             **
** Lsp        LENGTH OF STAND PIPE              **
** IDbh       ID OF BOREHOLE                   IN      **
** IDth       ID OF THRUST                     **
** IDc        ID OF HOUSING                    **
** IDby       ID OF BYPASS                     **
** IDdc2      ID OF LOWER DRILL COLLARS         **
** IDdc1      ID OF UPPER DILL COLLARS          **
** IDdp       ID OF DRILL PIPE                 **
** IDk        ID OF KELLY                      **
** IDh        ID OF ROTARY HOSE                **
** IDsp       ID OF STAND PIPE                 **
```

```

** ODdc2      OD OF LOWER DRILL COLLARS      "      **
** ODdc1      OD OF UPPER DRILL COLLARS      "      **
** ODdp       OD OF DRILL PIPE                "      **
** ODgb       OD OF GEARBOX                   "      **

```

```

*****
** PROGRAM VARIABLES
*****

```

```

** MFR        MASS FLOW RATE, AIR              LBm/SEC      **
** MFRch      MASS FLOW RATE, CHIPS            "              **
** Pdpa  ANNLR PRSSR DOWNSTREAM OF PIPE      (Ps)   PSF      **
** Pupa  ANNLR PRSSR UPSTREAM OF PIPE        "              **
** Pdca1 ANNLR PRSSR DOWNSTREAM OF COLLARS    (Pupa)  "        **
** Puca1 ANNLR PRSSR UPSTREAM OF LOWER COLLARS "              **
** Pdca2 ANNLR PRSSR DOWNSTREAM OF UPPER      "              **
**              COLLARS (Puca1)              "              **
** Puca2      PRSSR UPSTREAM OF COLLARS        "              **
** Pdb        PRSSR DOWNSTREAM FROM BIT      (Puca2)  "        **
** Pub        PRSSR UPSTREAM FROM BIT         "              **
** Pdth       PRSSR DOWNSTREAM OF THRUST      (Pub)   "        **
** Puth       PRSSR UPSTREAM OF THRUST         "              **
** Pdgb       PRSSR DOWNSTREAM OF GEARBOX     (Puth)  "        **
** Pugb       PRSSR UPSTREAM OF GEARBOX        "              **
** Pdtr       PRSSR DOWNSTREAM OF TURBINE     (Pugb)  "        **
** Putr       PRSSR UPSTREAM OF TURBINE        "              **
** Pdby       PRSSR DOWNSTREAM OF BYPASS      (Putr)  "        **
** Puby       PRSSR UPSTREAM OF BYPASS         "              **
** Pdcp2  PIPE PRSSR DOWNSTREAM OF COLLARS    (Pby)   "        **
** Pucp2  PIPE PRSSR UPSTREAM OF LOWER COLLARS "              **
** Pdcp1  PIPE PRSSR DOWNSTREAM OF UPPER      "              **
**              COLLARS (Pucp2)              "              **
** Pucp1  PIPE PRSSR UPSTREAM OF COLLARS        "              **
** Pdpp  PIPE PRSSR DOWNSTREAM OF PIPE      (Pucp1)  "        **
** Pupp  PIPE PRSSR UPSTREAM OF PIPE         "              **
** Pdk    PRSSR DOWNSTREAM OF KELLY          (Puby)  "        **
** Puk    PRSSR UPSTREAM OF KELLY             "              **
** Pdh    PRSSR DOWNSTREAM OF HOSE            (Puk)   "        **
** Puh    PRSSR UPSTREAM OF HOSE              "              **
** Pdsp   PRSSR DOWNSTREAM OF ST PIPE        (Puh)   "        **
** Pusp   PRSSR UPSTREAM OF STAND PIPE        "              **

```

```

*****

```

```

PI=3.1415927
FT=12.
FT2=FT**2
FtoR=460.
R=53.3
Dtw=0.4583
DENA=.0764
GRAV=32.2

```



```

WRITE(*,*)' ENTER DATA FILE NAME : '
READ(*, '(A)')FNAME
IN=3
OPEN(IN,FILE=FNAME)

WRITE(*,*)' OUTPUT TO CONSOLE, FILE OR PRINTER (C/F/P) : '
READ(*, '(A)')COUT
IF(COUT.EQ.'P'.OR.COUT.EQ.'p') THEN
    OUT=2
    FNAME='PRN'
ELSEIF(COUT.EQ.'F'.OR.COUT.EQ.'f') THEN
    OUT=11
    WRITE(*,*)' ENTER OUTPUT FILENAME : '
    READ(*, '(A)')FNAME
ELSE
    OUT=6
    FNAME='CON'
ENDIF
OPEN(OUT,FILE=FNAME)

READ(IN,*)Q
READ(IN,*)E,Uf
READ(IN,*)Ps
READ(IN,*)Ts,Tbn,Tub,Tutr
READ(IN,*)Abn,Atn
READ(IN,*)IDth,IDc,ODgb,IDby,IDk,IDh,IDsp
READ(IN,*)Lth,Lgb,Lby,Lk,Lh,Lsp

READ(IN,*)IDbh,DENch,ROP,WOB
READ(IN,*)Tbh
READ(IN,*)ODdp,ODdc1,ODdc2,IDdp,IDdc1,IDdc2
READ(IN,*)Lbh,Ldc1,Ldc2

Tbh=Tbh+FtoR
IDbh=IDbh/FT
ODdp=ODdp/FT
ODdc1=ODdc1/FT
ODdc2=ODdc2/FT
IDdp=IDdp/FT
IDdc1=IDdc1/FT
IDdc2=IDdc2/FT
Lm=16.
Ldc2=Ldc2+Lm
Ldp=Lbh-(Ldc1+Ldc2)
Tdc1=Ldp+Ldc1
Tdc2=Tdc1+Ldc2
Db=IDbh*FT
Ps=Ps*FT2

```

$T_s = T_s + F_{toR}$
 $T_{bn} = T_{bn} + F_{toR}$
 $T_{ub} = T_{ub} + F_{toR}$
 $T_{utr} = T_{utr} + F_{toR}$
 $ID_{th} = ID_{th} / FT$
 $ID_c = ID_c / FT$
 $OD_{gb} = OD_{gb} / FT$
 $ID_{by} = ID_{by} / FT$
 $ID_k = ID_k / FT$
 $ID_h = ID_h / FT$
 $ID_{sp} = ID_{sp} / FT$

$MFR = Q * DENA / 60.$
 $MFR_{ch} = PI / (4. * 3600.) * ID_{bh} * DEN_{ch} * ROP$

$A = (1. + MFR_{ch} / MFR) / (T_{bh} * R)$

$Pd_{pa} = P_s$
 $B = (0.014753 * T_{bh} * MFR^{**2}) / ((ID_{bh} - OD_{dp})^{**1.333} * (ID_{bh}^{**2} -$
C $OD_{dp}^{**2})^{**2})$
 $Pupa = \sqrt{(Pd_{pa}^{**2} + B/A) * \exp(2. * A * L_{dp}) - B/A}$
CALL CHECK(OUT, R, MFR, ID_{bh}, OD_{dp}, T_{bh}, Pupa)

$Pd_{ca1} = Pupa$
 $B = (0.014753 * T_{bh} * MFR^{**2}) / ((ID_{bh} - OD_{dc1})^{**1.333} * (ID_{bh}^{**2} -$
C $OD_{dc1}^{**2})^{**2})$
 $Pu_{ca1} = \sqrt{(Pd_{ca1}^{**2} + B/A) * \exp(2. * A * T_{dc1}) - B/A}$
CALL CHECK(OUT, R, MFR, ID_{bh}, OD_{ca1}, T_{bh}, Pu_{ca1})

$Pd_{ca2} = Pu_{ca1}$
 $B = (0.014753 * T_{bh} * MFR^{**2}) / ((ID_{bh} - OD_{dc2})^{**1.333} * (ID_{bh}^{**2} -$
C $OD_{dc2}^{**2})^{**2})$
 $Pu_{ca2} = \sqrt{(Pd_{ca2}^{**2} + B/A) * \exp(2. * A * T_{dc2}) - B/A}$
CALL CHECK(OUT, R, MFR, ID_{bh}, OD_{ca2}, T_{bh}, Pu_{ca2})

$Pd_b = Pu_{ca2}$
 $Pub = Pd_b * (1. + (MFR / (2.0564 * A_{bn} * Pd_b))^{**2} * T_{bn})^{**3.5}$
IF(Pd_b/Pub.LT.0.52) Pub=MFR*SQRT(T_{bn})/(0.532*A_{bn})

$Pd_{th} = Pub$
 $Pu_{th} = \sqrt{(Pd_{th}^{**2} + L_{th} * 0.0375 * MFR^{**2} * T_{ub} / ID_{th}^{**5.333})}$

$Pd_{gb} = Pu_{th}$
 $A = (ID_c - OD_{gb})^{**1.333} * (ID_c^{**2} - OD_{gb}^{**2})^{**2}$
 $Pu_{gb} = \sqrt{(Pd_{gb}^{**2} + L_{gb} * 0.0375 * MFR^{**2} * T_{ub} / A)}$

$Pd_{tr} = Pu_{gb}$
 $Putr = Pd_{tr} * (1. + (MFR / (2.0564 * A_{tn} * Pd_{tr}))^{**2} * T_{bn})^{**3.5}$
IF(Pd_{tr}/Putr.LT.0.52) Putr=Pd_{tr}/0.52

```

Vn=SQRT(11980.*Tutr*(1.-(Pdtr/Putr)**0.2857))

Pdby=Putr
Puby=SQRT(Pdby**2+Lby*0.0375*MFR**2*Tutr/IDby**5.333)

A=1./(R*Tutr)
Tdc1=Ldc2+Ldc1
Tdp=Tdc1+Ldp
Tk=Tdp+Lk

Pdcp2=Puby
B=0.014753*Tutr*MFR**2/IDdc2**5.333
Pucp2=SQRT(B/A-(B/A-Pdcp2**2)*EXP(-2.*A*Ldc2))

Pdcp1=Pucp2
B=0.014753*Tutr*MFR**2/IDdc1**5.333
Pucp1=SQRT(B/A-(B/A-Pdcp1**2)*EXP(-2.*A*Tdc1))

Pdpp=Pucp1
B=0.014753*Tutr*MFR**2/IDdp**5.333
Pupp=SQRT(B/A-(B/A-Pdpp**2)*EXP(-2.*A*Tdp))

Pdk=Pupp
Puk=SQRT(Pdk**2+Lk*0.0375*MFR**2*Tutr/IDk**5.333)

Pdh=Puk
Puh=SQRT(Pdh**2+Lh*0.0375*MFR**2*Tutr/IDh**5.333)

Pdsp=Puh
Pusp=SQRT(Pdsp**2+Lsp*0.0375*MFR**2*Tutr/IDsp**5.333)

HPT=E*0.5*MFR*Vn**2/(GRAV*550)
RPMt=60*Uf*Vn/(PI*Dtw)
RPMb=RPMt/168.
TRQb=2.*5250.*HPT/RPMb

WRITE(OUT, '(1X, ''AT '', 15, '' SCFM, AND AT A DEPTH OF '', 16,
C  '' FT : '' ) ) INT(Q), INT(Lbh)
WRITE(OUT, '(1X)')
WRITE(OUT, 100) 'PRESSURE BELOW PIPE      : ', (Pupa-Ps)/FT2, ' psig'
WRITE(OUT, 100) 'PRESSURE BELOW COLLAR1   : ', (Pucal-Ps)/FT2, ' psig'
WRITE(OUT, 100) 'PRESSURE BELOW COLLAR2   : ', (Puca2-Ps)/FT2, ' psig'
WRITE(OUT, '(1X)')
WRITE(OUT, 100) 'PRESSURE ABOVE BIT        : ', (Pub-Ps)/FT2, ' psig'
WRITE(OUT, 100) 'PRESSURE ABOVE THRUST     : ', (Puth-Ps)/FT2, ' psig'
WRITE(OUT, 100) 'PRESSURE ABOVE GEARBOX    : ', (Pugh-Ps)/FT2, ' psig'
WRITE(OUT, 100) 'PRESSURE ABOVE TURBINE    : ', (Putr-Ps)/FT2, ' psig'
WRITE(OUT, 100) 'PRESSURE ABOVE BYPASS     : ', (Puby-Ps)/FT2, ' psig'

```

```

WRITE(OUT,'(1X)')
IF(OUT.EQ.6) PAUSE
WRITE(OUT,100)'PRESSURE ABOVE COLLAR2 :',(Pucp2-Ps)/FT2,' psig'
WRITE(OUT,100)'PRESSURE ABOVE COLLAR1 :',(Pucp1-Ps)/FT2,' psig'
WRITE(OUT,100)'PRESSURE ABOVE PIPE :',(Pupp-Ps)/FT2,' psig'
WRITE(OUT,'(1X)')
WRITE(OUT,100)'PRESSURE ABOVE KELLY :',(Puk-Ps)/FT2,' psig'
WRITE(OUT,100)'PRESSURE ABOVE HOSE :',(Puh-Ps)/FT2,' psig'
WRITE(OUT,100)'PRESSURE ABOVE ST-PIPE :',(Pusp-Ps)/FT2,' psig'
WRITE(OUT,'(1X)')
WRITE(OUT,100)'VELOCITY AT NOZZLES :',Vn,' ft-sec'
WRITE(OUT,100)'PEAK TURBINE HP :',HPt,' '
WRITE(OUT,100)'PEAK TURBINE RPM :',RPMt,' '
WRITE(OUT,100)'PEAK BIT RPM :',RPMb,' '
WRITE(OUT,100)'STALL TORQUE AT BIT :',TRQb,' ft-lbs'

```

```

CALL HPDATA(OUT,HPt,RPMb,WOB,Db)

```

```

100 FORMAT(1X,A,F12.4,A)

```

```

STOP
END

```

```

SUBROUTINE CHECK(OUT,R,MFR,IDbh,OD,Tbh,Pu)

IMPLICIT REAL(A-Z)
INTEGER OUT

Va=R*MFR/(0.7854*(IDbh**2-OD**2))
DEN=Pu/(R*Tbh)
A=0.5*DEN*Va**2

IF(A.LT.130.) THEN
  WRITE(OUT, '(1X, ''INADEQUATE HOLE CLEANING, A ='',
C  F12.4)')A
  PAUSE
ENDIF

RETURN
END

```

SUBROUTINE HPDATA(OUT,HPd,RPMd,WOB,Db)

IMPLICIT REAL(A-Z)

INTEGER OUT,I,J,N,IMAX

DIMENSION RPM(100),HPt(100),HPb(100)

RPMmax=2.*RPMd

C=12.49E-06

DIV=20

STEP=RPMmax/DIV

RPM(1)=0.

DO 10 I=1,100

HPt(I)=HPd*(2.*RPM(I)/RPMd-(RPM(I)/RPMd)**2)

HPb(I)=C*WOB**1.5*Db**2.5*RPM(I)

IF(RPM(I).GT.(RPMmax-0.1)) GOTO 99

RPM(I+1)=RPM(I)+STEP

10 CONTINUE

99 IMAX=I

IF(OUT.EQ.6) PAUSE

WRITE(OUT,'(1X)')

WRITE(OUT,'(9X,''RPM'',11X,''HPt'',11X,''HPb''))')

DO 20 I=1,IMAX

WRITE(OUT,'(1X,3(2X,F12.4))')RPM(I),HPt(I),HPb(I)

IF(OUT.EQ.6) THEN

IF(MOD(I,20).EQ.0) PAUSE

ENDIF

20 CONTINUE

RETURN

END

 ** SAMPLE DATA FILE FP24DAT - 2400 SCFM DOWNHOLE **

2400	;Q	(scfm)
0.35,0.46	;E,UF	
14.0	;Ps	(psia)
40,35,40,65	;Ts,Tbn,Tub,Tutr	(deg F)
2.07086E-2,7.902777E-3	;Abn,At	(sq ft)
2,7.0,6.25,2,2.75,3,3.8	;IDth,IDc,ODgb,IDby,IDk,IDh,IDsp	(in)
5,4,4,30,55,30	;Lth,Lgb,Lby,Lk,Lh,Lsp	(ft)
10.625,160.0,100,16	;IDbh(in),DENch(lb/cu ft),ROP(ft/hr),WOB	
50	;Tbh	(deg F)
4.5,7.0,8.0,3.8,2.25,2.875	;ODdp,ODdc1,ODdc2,IDdp,IDdc1,IDdc2	(in)
405,270,60	;Lbh,Ldc1,Ldc2	(ft)

 ** SAMPLE OUTPUT AT 2400 SCFM AND AT A DEPTH OF 405 FT **

PRESSURE BELOW PIPE : 0.1456 psig
 PRESSURE BELOW COLLAR1 : 2.3500 psig
 PRESSURE BELOW COLLAR2 : 7.6777 psig

 PRESSURE ABOVE BIT : 28.8579 psig
 PRESSURE ABOVE THRUST : 35.3260 psig
 PRESSURE ABOVE GEARBOX : 38.1417 psig
 PRESSURE ABOVE TURBINE : 86.2725 psig
 PRESSURE ABOVE BYPASS : 88.7396 psig

 PRESSURE ABOVE COLLAR2 : 93.5932 psig
 PRESSURE ABOVE COLLAR1 : 157.0545 psig
 PRESSURE ABOVE PIPE : 158.3586 psig

 PRESSURE ABOVE KELLY : 160.3413 psig
 PRESSURE ABOVE HOSE : 162.5991 psig
 PRESSURE ABOVE ST-PIPE : 162.9456 psig

 VELOCITY AT NOZZLES : 1035.2825 ft-sec
 PEAK TURBINE HP : 32.3661
 PEAK TURBINE RPM : 19845.7812
 PEAK BIT RPM : 118.1297
 STALL TORQUE AT BIT : 2876.8743 ft-lbs

RPM	HPt	HPb
0.0000	0.0000	0.0000
11.8130	6.1496	3.4748
23.6259	11.6518	6.9495
35.4389	16.5067	10.4243
47.2519	20.7143	13.8990
59.0648	24.2746	17.3738
70.8778	27.1875	20.8485
82.6908	29.4532	24.3233
94.5037	31.0715	27.7980
106.3167	32.0424	31.2728
118.1297	32.3661	34.7475
129.9426	32.0424	38.2223
141.7556	31.0715	41.6970
153.5685	29.4532	45.1718
165.3815	27.1875	48.6465
177.1945	24.2746	52.1213
189.0074	20.7143	55.5960
200.8204	16.5067	59.0708
212.6333	11.6518	62.5455

224.4463
236.2592

6.1496
0.0000

66.0203
69.4950