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**PROGRAM TO DEVELOP IMPROVED
DOWNHOLE DRILLING MOTORS**

Semi-Annual Report

**By
William C. Maurer**

November 1, 1976

Work Performed Under Contract No. EG-76-C-07-1581

**Terra Tek, Inc.
Salt Lake City, Utah**

and

**Maurer Engineering, Inc.
Houston, Texas**

**U. S. DEPARTMENT OF ENERGY
Geothermal Energy**

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by

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November 1, 1976

FOR

ENERGY RESEARCH AND DEVELOPMENT AGENCY
GEOTHERMAL ENERGY DIVISION
Contract No. E(10-1)-1581
Attention: Mr. Cliff Carwile

Prime Contractor: Terra Tek, Inc.
Subcontractor: Maurer Engineering, Inc.

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FOREWORD

This report is divided into three sections:

- (1) Downhole Drilling Motor Design by Maurer Engineering, subcontractor;
- (2) Design of Downhole Motor Seal, Bearing and Lubrication Test Systems by Terra Tek, prime contractor; and
- (3) Program Schedule.

The report includes progress made on the contract to November, 1976.

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INTRODUCTION

Downhole drilling motors were first extensively tested in the 1920's. They did not find widespread use until the 1950's when turbodrills began being used in the Soviet Union. By the early 1960's, 85 percent of the wells in the Soviet Union were being drilled with turbodrills. This has decreased to 70 percent because conventional drills are now being used in most Soviet wells deeper than 4000 meters because of the increased trip time required to replace worn turbodrills. Dyna-Drill type motors are now being used in the Soviet Union which further reduces the amount of holes drilled with turbodrills.

Drilling motors have found widespread use in the United States for drilling directional holes, but they are not widely used for straight hole drilling because of bearing and seal problems. Current seal designs limit bit pressure drops to only 100 to 300 psi which is not adequate for a good bottomhole cleaning bit.

Commercial downhole motors operate at speeds of 300 to 1000 rpm whereas roller bits operate most effectively at speeds of 50 to 150 rpm. At high motor speeds, the bit bearings fail after only 5 to 15 hours whereas at lower speeds the bit bearings last up to 200 hours. The high rotary speeds also cause failure of the motor bearings in 20 to 40 hours which is not adequate for straight hole drilling. Several bit companies are working on high speed roller bits (200 to 300 rpm) and their development should increase the economic use of downhole motors.

Improved motors which will: 1) operate at speeds of 100 to 200 RPM. 2) allow bit pressure drops of 1000 to 2000 psi, and 3) have bearing life in excess of 200 hours would find widespread use in deep wells and in the straight hole

portion of offshore wells. The commercialization of borehole data telemetry systems will increase the use of drilling motors since these systems will perform real time monitoring of the downhole drilling variables and will guide the motors in directional wells.

The first phase of this ERDA funded project is aimed at developing an improved bearing package that can be used on new and existing drilling motors. The goal of this project is to develop a sealed bearing assembly which allows the bearings to operate in a lubricant filled reservoir. This should increase the bearing life to over 200 hours. An additional goal is to develop improved rotary seals that will allow the bits to operate at pressure drops in excess of 1000 psi to overcome current bit cleaning problems with existing motor.

Maurer Engineering is responsible for the design of the drilling motor bearings assemblies and components to be tested by Terra Tek during this project. This design program is well underway and components are being prepared for testing at Terra Tek.

I. TURBODRILL DEVELOPMENT

Downhole drilling motors were patented soon after the development of the rotary drilling rigs in the 1860's. Cross patented the single-stage turbodrill shown in Fig. 1 in 1873. This turbodrill developed very low torques; therefore, produced low drilling rates.

Fig. 1 Cross Single-Stage Turbodrill (1873)

Baker patented the single-stage axial flow turbodrill shown in Fig. 2 in 1884. This type of blading is similar to that used in modern turbodrills. Because of the single stage, this motor was also limited to low torques and low drilling rates.

Fig. 2 Baker Single-Stage Axial Flow Turbodrill (1884)

C. Scharpenburg (Standard of California) patented the multistage turbodrill shown in Fig. 3 in 1924. This multistage turbodrill was the forerunner of the turbodrills currently in use.

Fig. 3 Scharpenburg Axial-Flow, Multi-Stage Turbodrill (1924)

Fig. 4 Capeliuschnicoff Single-Staged Turbodrill (1928)

The Scharpenberg turbodrill contained a lubrication system which allowed the thrust bearing to operate in oil or grease. Drilling fluid acting on a floating piston pressurized the lubricant in the system. The bearings in modern turbodrills operate directly in the abrasive drilling mud, resulting in rapid failures, which limit application of these tools. New turbodrills currently being developed utilize lubrication systems similar to that tested by Scharpenberg in the 1920's. Development of these sealed lubrication systems could greatly improve the performance and life of these tools.

Capeliuschnicoff patented the single-staged, geared turbodrill shown in Fig. 4 in 1928. The Soviets conducted extensive tests of these geared turbodrills from 1924 to 1934. The gear reduction system was used to reduce the speed of the single stage turbine.

The Russians had severe problems with the speed reducers in the Capeliuschnicoff turbodrill; consequently, in 1934 they switched to the turbodrill being developed by Scharpenberg. Sumilov, Tagiev, Iaonesyan and Gusman perfected the multi-stage turbodrill during the 1940's and 1950's and by the early 1960's, the Russians were drilling 80 to 90 percent of their wells with axial flow turbodrills. This has decreased to about 65 to 75 percent at present because rotary drills are being used in most Russian wells that are deeper than 4000 meters.

In 1956, Russia licensed its turbodrill technology to the following companies:

Dresser - U.S.A.
Neyrpic - France
Salzgitter- Germany
Trauzl - Austria

They sold a large number of Russian turbodrills to these companies, including 40 to Dresser. Some of the turbodrills sold to Salzgitter and Trauzl in 1956 are still in used, testifying to the durability of turbodrills as drilling tools.

The bearings in these Russian turbodrills were located above the turbine blades, as shown in Fig. 5. This design created problems because the turbodrills had to be completely disassembled to replace the bearings.

Fig. 5 Turbodrill With Thrust Bearings Located Above Turbine Blades (1950's)

A radial bearing containing a rubber wear sleeve d, located at the bottom of the turbodrill, provided a rotary mud seal, as shown in Fig. 6. Approximately 20 to 40 percent of the mud leaked through this seal and cooled the bearings while the remainder of the mud flowed through the bit. These turbodrills could be operated at bit pressure drops of only 100 to 200 psi, which is insufficient to effectively clean the cuttings from beneath the bits.

Fig. 6 Typical Turbodrill Radial Bearing

During the late 1950's and early 1960's, extensive R&D was conducted on turbodrills in Russia, France and the U.S.A. One of the major improvements was to move the bearings to the bottom of the turbodrill (i.e., below the turbine blades) as shown in Fig. 7. This allowed the worn bearings to be replaced more readily, greatly facilitating repairs and maintenance of these turbodrills.

Fig. 7 Modern Turbodrill With Bearings Below Turbine Blades

Salzgitter went out of the turbodrill business in 1969 and sold their Russian-built turbodrills and spare parts to three companies:

Deutag (Bentheim, Germany)	(20-30 Turbodrills)
Preussag (Berkopen Uber Peine, Germany)	(26 Turbodrills)
Winterfall	(5-10 Turbodrills)

These companies are still using these Russian turbodrills, even though they are over 20 years old. The Salzgitter turbodrill is shown in Fig. 8.

Aufbau der E- und A-Turbine

Abb. 4

Fig. 8 Salzgitter Turbodrill

The Trauzl turbodrill is currently manufactured by Gebr, Bohler and Co. in Vienna, Austria. These turbodrills are similar to those purchased from the Russians in 1956 with only minor improvements having been made (Fig.9). The thrust bearing in this turbodrill is still located at the top of the turbodrill as originally designed by the Russians.

Fig. 9 Trauzl Turbodrill

During the late 1950's and early 1960's, four new turbodrills were developed:

Voith:	Germany
Turboservice:	France
RediDrill:	U.S.A.
Eastman:	U.S.A.

Voith in Germany started manufacturing the 6-5/8 inch O. D. turbodrill shown in Fig. 10 in 1958. This short turbodrill, which is designed for directional drilling and for mining applications, develops low torque because it contains only 40 stages. Several of these motors can be connected together to provide higher torque turbodrills, containing 80, 120 or 160 stages.

Fig. 10 Voith Turbodrill

In 1968, Voith went out of business and Deutag bought all of the remaining turbodrills and spare parts. These turbodrills are still being used in Deutag's mining and oilfield operations.

The Turboservice turbodrill, designed by Tiraspol'sky, also contains the bearings below the turbine blades as shown in Fig. 11.

Fig. 11 Turboservice Turbodrill

In 1960, Dresser sold their turbodrills to Eastman who did extensive development on these tools during the 1960's. Eastman moved the thrust bearings to the bottom of the turbodrill and put them in a bearing package which could be easily replaced on the rig floor (Fig.12). This design greatly facilitates maintenance and eliminates the need for sending the turbodrills into land-based repair shops from offshore rigs. The Neyrpic turbodrills also utilize similar bearing packages.

TURBINE
SECTION

REPLACEABLE
BEARING PACK
SECTION

DIRECTIONAL
INTEGRAL
TURBODRILL

Fig. 12 Eastman Turbodrill

In 1953, Sir Frank Whittle began work on the sophisticated turbodrill shown in Fig. 13. This turbodrill contained numerous innovations including contra-rotating rotors and stators, both turning faster than the bit through a gear reduction system, which was sealed in oil. The turbodrill also contained a servo-controlled hydraulic loading ram which automatically controlled the speed of the turbine by varying the weight on the bit.

Fig. 13 Whittle Turbodrill

The Whittle turbodrill was manufactured by Bristol-Siddeley in England and was extensively tested in the U.S.A. from 1963 to 1970. Over 200 runs were made on land and offshore in the Gulf Coast during this testing program. Numerous problems were encountered with this tool, especially with the seals. These problems arose primarily due to the complexity of the tool. Work was abandoned on this tool in about 1970 after approximately \$7.5 million was spent on the program. Some efforts have been made during the past year to revive interest in this tool.

Neyrpac in Grenoble, France has continued development of the turbodrill and currently markets the motor shown in Fig. 14. This turbodrill contains a bearing package which can easily be replaced in the field. Neyrpac turbodrills are widely used outside of the U.S.A. and account for about 50 percent of the directional drilling in the North Sea. These turbodrills contain rubber friction bearings, but Neyrpac is reportedly testing turbodrills containing ball bearings.

Fig. 14 Neyrpac Turbodrill

Whipstock purchased Eastman and in the 1970's, Eastman Whipstock acquired Turboservice. Eastman Whipstock currently markets both the Eastman and the Turboservice turbodrills.

As a result of the acquisitions over the years, the only companies actively marketing turbodrills at present are:

1. Neyrpic -France
2. Eastman-Whipstock -U.S.A.
3. RediDrill -U.S.A.
4. Bohler-Schoeller-Bleckmann (Austria) (Trauzl Turbodrill)

In addition, the East Block countries utilize turbodrills to drill 65 to 75 percent of their wells.

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II. POSITIVE DISPLACEMENT MOTOR DEVELOPMENT

Field testing of positive displacement downhole drilling motors began in the early 1930's and has continued to date. Dyna-Drill introduced the first successful positive drilling motor in 1965. This Dyna-Drill is a Moineau motor as shown in Fig. 1. This motor utilizes an eccentric steel rotor in a rubber stator to produce torque to turn the bit. Wallace Clark patented this Moineau drilling motor in 1959.

Fig. 1. Dyna-Drill Moineau Motor

This motor has found widespread use and is currently used in about 90 percent of the directional holes in the U.S.A. and in about 40 to 50 percent of the directional holes in the North Sea. The Dyna-Drill is a good tool and has played an important part in the development of domestic oil and gas reserves. Despite its widespread use in directional holes, the Dyna-Drill has the following limitations, which precludes its widespread use in straight hole drilling:

1. The rubber stator limits the tool to a maximum operating temperature of about 350°F.
2. The tungsten carbide ball bearings operate in the abrasive drilling mud and wear out in 20 to 40 hours.
3. The motor operates at excessive speeds of 300 to 500 rpm and delivers insufficient torque to operate roller bits effectively.

Dyna-Drill is reportedly working on sealed bearing systems, which allow the bearings to operate in oil or grease, so it is likely that the performance of these motors will be significantly improved in the next few years.

The major advantage of the Dyna-Drill over the turbo-drills is that it is a positive displacement motor and the flow rate is a direct measure of the rotary speed. The speed of turbodrills is usually not known because the

pressure drop across the tool is independent of the rotary speed. It is possible to stall a turbodrill and not know it at the surface until the driller notices that drilling has ceased.

This turbodrill limitation will be overcome with the development of borehole data telemetry systems which will allow data to be transmitted up the borehole while drilling. Over 25 companies are working on these systems and it is likely that one or more systems will become commercially available in the next few years.

Numerous attempts have been made to develop other types of downhole positive-displacement motors, but to date, none of these projects has been successful. In 1965, L. E. Gilreath tested the unbalanced sliding vane motor shown in Fig. 2 in Houston. This motor was unsuccessful because the high pressure drilling mud acted only on one side of the unbalanced rotor, pushing the rotor against the stator and causing high friction between these parts. This friction prevented the motor from turning when tested with muds on a drilling rig.

Fig. 3 Gilreath Sliding Vane Motor (1965)

In 1965, Göttker tested the unbalanced flexing vane motor shown in Fig. 4 in Celle, Germany. This motor also failed to turn when tested with drilling mud; again, due to the high unbalanced force on the rotor. These tests demonstrate that it is necessary to utilize balanced pressure systems when designing positive displacement motors.

Fig. 4 Göttker Unbalanced Flexing Vane Motor (1965)

In 1966, Garrison began testing the balanced flexing vane motor shown in Fig. 5. The forces on the rotor in this motor are balanced and the motor was used to drill short distances in field wells. Several companies have unsuccessfully attempted to develop the Garrison motor. Carmet, a division of Allegheny-Ludlum, is currently testing prototypes of this motor. Carmet is using a speed reduction system with this motor.

Fig. 5 Garrison Balanced Flexing Vane Motor (1966)

Several companies are working on new Moineau and vane motors and it is likely that one or more new motors will become commercially available in the next few years.

The Russians are developing the multiple lobe Moineau motors. The motor shown in Fig. 6 utilizes a two-lobe rotor and a three cavity stator.

Fig. 6 Russian Moineau Motor (1974)

The latest Russian motors utilize a nine lobe rotor in a ten cavity stator as shown in Fig. 7. Both of these Russian motors develop higher torques and lower speeds than Dyna-Drills. The Russians have made over 1000 field runs with these Moineau motors and it appears that they will find widespread application and that they will replace turbodrills in some areas.

Fig. 7 Russian Moineau Motor (1974)

Baker Oil Tools in Houston is developing the Moineau motor shown in Fig. 8. This motor utilizes a three lobe rotor and a four cavity stator. This motor also develops higher torque and lower speed than Dyna-Drills.

Fig. 8 Baker Moineau Motor (1974)

As a result of the extensive efforts underway, it appears that several new positive-displacement motors will find commercial application soon. If long life motors (100 to 200 hours) can be developed which will operate at speeds of 100 to 200 rpm and deliver high torques, it is likely that these motors could find increased use for straight-hole drilling, especially in deep wells or offshore wells where costs are high.

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III. THEORY OF TURBODRILLS

The torque developed by a turbodrill is maximum when the turbodrill stalls. This stall torque, T_0 , equals:

$$T_0 = \eta_m \frac{nw (1-\epsilon) Q^2}{g\lambda\pi b} (\tan \beta) \text{ (ft-lb)} \quad (1)$$

where

- η_m = Mechanical efficiency
- n = Number of turbine stages
- w = Fluid density (lb/ft.³)
- ϵ = Fluid loss coefficient
- Q = Fluid flow rate (ft³/sec)
- g = Gravitational constant (32.2 ft/sec²)
- λ = Fluid contraction coefficient
- b = Rotor vane width (ft)
- β = Exit angle of rotor and stator blades

Equation 1 shows that the turbine torque can be increased by increasing the number of turbine stages (n), by increasing the fluid flow rate (Q) or by increasing the exit angle (β) of the turbine blades.

The turbine torque decreases linearly to zero as the rotary speed increases to the runaway speed (N_R) as shown in Figure 1.

Figure 1 - Turbodrill output Torque.

The power output of the turbodrill is proportional to the product of the speed (N) and torque (T):

$$P = 2\pi NT \quad (2)$$

Consequently, the power out is maximum at a speed equal to half the runaway speed as shown in Figure 2. The torque at this optimum condition equals half the stall torque as shown in Figure 1.

Figure 2 - Turbodrill Power Output

The runaway speed of a turbodrill equals:

$$N_R = \frac{120 (1-\epsilon) Q}{\lambda (\pi D)^2 b} (\tan \beta) \text{ (rpm)} \quad (3)$$

where D is the nominal rotor vane diameter (ft). Equation 3 shows that the larger the turbodrill (i.e., the larger b and D), the slower the rotary speed. The runaway speed of commercial turbodrills ranges from 1200 to 2000 rpm which corresponds to optimum rotary speeds of 600 to 1000 rpm.

The torque and power curves for actual turbodrills deviate slightly from the theoretical curves shown in Figure 1 and 2. Figure 3 shows the performance curves for the Neyrpic turbodrill.

Figure 3 - Neyrpic Turbodrill Characteristic Curves.

Turbodrills are torque limited and therefore their operation in the field is often complicated. The torque available to turn the bit, T_{Bit} , equals the output torque, T_{Output} , minus the friction in the bearings, $T_{Bearings}$:

$$T_{Bit} = T_{Output} - T_{Bearings}$$

Existing turbodrills use a stack of 20 to 30 rubber friction bearings which create high friction and often prevent the turbodrills from rotating. It is often necessary to apply weight to the bit to counterbalance the hydraulic downthrust on the rotor before the turbine will rotate. Tiraspolsky has discussed this in detail in the reference cited.

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IV. THEORY OF POSITIVE DISPLACEMENT MOTORS

The power output, P_{output} , of a positive displacement motor equals:

$$P_{\text{output}} = \eta P_{\text{input}} \quad (1)$$

where η is the overall efficiency of the motor.

As the fluid powers the motor, power is lost due to leakage of fluid through the motor and due to friction loss between the sliding elements. The overall efficiency of the motor therefore equals:

$$\eta = \eta_v \times \eta_m \quad (2)$$

where

η_v = Volumetric efficiency

η_m = Mechanical efficiency

The actual flow rate, Q , is greater than the theoretical flow rate Q_t due to the leakage:

$$Q = Q_t / \eta_v \quad (3)$$

Similarly, the actual torque, T , is less than the theoretical torque, T_t , due to the friction losses:

$$T = \eta_m T_t \quad (4)$$

The power input to the motor equals:

$$P_{\text{input}} = \frac{pQ}{1714} \quad (\text{hp}) \quad (5)$$

where

p = Pressure drop across motor (psi)

Q = Flow rate (gpm)

The power output equals (from Equation 1 & 5)

$$P_{\text{output}} = \eta \frac{pQ}{1714} \quad (\text{hp}) \quad (6)$$

The power output also equals:

$$P_{\text{output}} = \frac{2\pi NT}{33,000} \quad (\text{hp}) \quad (7)$$

where

N = Rotary Speed (rpm)

T = Torque (ft-lb)

Equating equations 6 and 7:

$$T = 3.07\eta \frac{pQ}{N} \quad (\text{ft-lb}) \quad (8)$$

The volumetric displacement through the motor per revolution, q, equals:

$$q = \frac{Q_t}{N} = \eta_v \frac{Q}{N} \quad (\text{gallons/revolution}) \quad (9)$$

For a given motor design $q = \text{constant}$, and speed therefore varies as

$$N = \eta_v \frac{Q}{q} \quad (\text{rpm}) \quad (10)$$

This shows that the speed of a positive displacement motor is proportional to the flow rate. This is a major advantage of positive displacement motors since the speed is always known whereas with turbodrills the speed is independent of the flow rate and it is unknown. Speed varies inversely with the volumetric displacement, q , so it is desirable to make the volumetric displacement as large as possible in order to reduce the speed of the motors.

The motor torque varies as: (from equation 8 & 9)

$$T = 3.07 \eta_m p q \quad (\text{ft} - \text{lb}) \quad (11)$$

This shows that the torque is directly proportional to the pressure drop across the positive displacement motor and to the volumetric displacement, q , of the motor. It is desirable to make both of these as large as possible in order to maximize the torque output of the motor. There is a limit to the pressure drop that can be used across the motor, because at high pressure drops, leakage becomes a problem and the parts erode quickly.

The new multi-lobe Moineau Motors being developed by the Russians and by Baker deliver approximately twice the torque of Dyna-Drills at half the speed. This improved performance is the result of larger volumetric displacements due to the multilobe designs. These lower speed motors should find widespread use because they are ideally suitable for use with roller bits.

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V. BASIC MOTOR COMPONENTS

All downhole drilling motors have four basic components:

1. Motor Section
2. Thrust Bearings
3. Radial Bearings
4. Rotary Seal

as shown in Figure 1. In addition, two optional systems are used in many motors currently being developed:

5. Sealed Lubrication System (optional)
6. Speed Reducer (optional)

Figure 1 - Downhole drilling motor components.

The bearings and seals can be placed in a separate package at the bottom of the motor which can be used on any type of motor (i.e., Dyna-Drills, Turbodrills, Vane Motors, etc.).

Motor Section

There are two basic types of downhole drilling motors:

1. Turbodrills
2. Positive-Displacement

Turbodrills utilize the momentum change of drilling fluid passing through the curved turbine blades to provide torque to turn the bit. Diamond bits are used on most turbodrills because these motors turn at speeds of 600 to 1000 rpm whereas rock bits operate effectively only at speeds up to about 150 rpm. At high speeds (500 to 1000 rpm), the bit bearings wear out in 5 to 10 hours whereas at lower speeds, sealed bearing bits last in excess of 100 hours. Diamond bits do not contain bearings and therefore work effectively at speeds of 500 to 1000 rpm.

Positive-Displacement motors have a fixed volumetric displacement and their speed is directly proportional to the flow rate. There are three basic types of new positive displacement motors currently under development:

1. Moineau
2. Flexing Vane
3. Sliding Vane

These new motors have large volumetric displacement and therefore deliver higher torques at lower speeds than existing Dyna-Drills.

Thrust Bearings

Thrust bearing failures in downhole motors is a problem because of high dynamic loads produced by the action of the bits and by drillstring vibrations. Exxon placed a recorder at the hole bottom and found that the dynamic loads were often 50 percent higher than the applied bit weight. They found that on occasion the bit bounced off bottom and produced loads in excess of 120,000 pounds when drilling at an applied bit weight of 40,000 pounds. These high loads can cause rapid failure of the thrust bearings; consequently these bearings must be greatly oversized to operate in the hostile downhole environment.

Two basic types of bearings are used in downhole drilling motors:

1. Rubber friction bearings
2. Ball or roller bearings

In existing motors, these bearings operate directly in the abrasive drilling mud and therefore wear out in 20 to 200 hours. In addition the rubber friction bearings have high friction and therefore absorb 30 to 40 percent of the output torque of the turbodrills. The lift of the thrust bearings can be increased by operating at bit weights which nearly balance the hydraulic downthrust thereby removing most of the load from the thrust bearings.

Radial Bearings

Radial Bearings are required on each side of the motor and on each side of the thrust bearings. These radial bearings are usually subjected to lower loads than the

thrust bearings and therefore have much longer life.

Two basic types of radial bearings are used in downhole motors:

1. Marine bearings
2. Roller or ball bearings

Most motors contain marine bearings made out of brass, rubber or similar bearing materials. The marine bearings are cooled by circulating mud through them.

Rotary Seals

Rotary seals are currently the weakest link in downhole motors. Improved seals would allow the bearings to be sealed in lubricant, thereby greatly increasing their life. Improved seals would allow the bits to be operated at higher pressures (1000 to 2000 psi) thereby greatly increasing drilling rate.

There are six basic types of seals that have been tested in downhole motors.

1. Packing
2. Face
3. Labyrinth
4. Radial Lip
5. Constrictions (Friction bearings and Marine Bearings)
6. Flow metering

Existing drilling motors allow drilling mud to continuously leak through the rotary seals by constricting the flow with labyrinth seals, rubber friction thrust bearings, marine radial bearings, flow metering devices or a combination of these.

Sand and other abrasive particles are filtered out of the mud causing rapid failure of the seals. To produce long-life seals, it will be necessary to develop positive seals (e.g., packing, face or lip) which allow no leakage. This will overcome the problem of filtering abrasive particles in the rotating seal.

Lubrication Systems (Optional)

The bearings in existing Dyna-Drills and turbodrills operate directly in the abrasive drilling mud and therefore wear out rapidly. Attempts to use a sealed lubrication system which allows the bearings to operate in oil or grease have been unsuccessful, primarily due to rotary seal failures. Emphasis should be placed on developing improved rotary seals, since the development of reliable sealed lubrication systems will greatly increase the life and application of downhole drilling motors.

Speed Reducers (Optional)

Attempts to reduce the speed of downhole drilling motors using speed reducers have been underway since the early 1920's, but to date these attempts have been unsuccessful. Two major problems involved are 1) the space constraints which results in high stresses on the speed reducer components and 2) the failure of the lubrication system seals and subsequent flooding of the speed reducer with abrasive drilling mud. The development of reliable rotary seals and lubrication systems should eventually lead to the development of reliable speed reducer systems.

VI. FORCES ON THRUST BEARINGS

When a downhole motor is operating, the force on the thrust bearing, F_B , equals:

$$F_B = F_H + W_R - W \quad (1)$$

where

F_H = Hydraulic downthrust on rotor (lbs)

W_R = Weight of rotor (lbs)

W = Weight applied to bit (lbs)

These forces are shown in Figure 1.

Figure 1 - Forces on downhole motor thrust bearing

When the bit weight equals the sum of the downthrust forces ($W = F_H + W_R$) these loads cancel out and there is no

load on the thrust bearing. This "balanced" condition often causes "chattering" and premature failure of the thrust bearing due to the dynamic loads on the bit. In practice, the bit weight is adjusted so that the bearing load is as low as possible without causing chattering (e.g., 5000 pounds).

Hydraulic Downthrust

As fluid is pumped through the motor, pressure drops occur across the motor and across the bit as shown in Figure 2.

Figure 2 - Hydraulic Downthrust on rotor

These pressure drops produce a hydraulic downthrust, F_H , on the rotor equal to:

$$F_H = p_m \times A_R + p_b \times A_s \quad (\text{lbs}) \quad (2)$$

where

p_m = Pressure drop across motor (psi)

p_b = Pressure drop across bit (psi)

A_R = Effective area of rotor (psi)

A_s = Effective area of rotating seal (psi)

Typical values for a 7 1/4 inch turbodrill are:

p_m = 1000 psi

p_b = 300 psi

A_R = 25 in²

A_s = 20 in²

In this case, the hydraulic downthrust equals:

$$F_H = 1000 \times 25 + 300 \times 20 = 31,000 \text{ pounds}$$

Seal problems limit bit pressure drops on existing motors to 200 to 400 psi compared to the 1000 to 2000 psi required to effectively remove the cuttings from the hole bottom. As a result, drilling rates are greatly reduced due to regrinding of cuttings on the hole bottom.

Drilling motors with improved seals which would allow higher bit pressures would greatly increase drilling rate. In addition, the higher pressure drops would reduce the bearing thrust loads at higher bit weights. For example, at a bit pressure drop of 1000 psi, the hydraulic downthrust is increased to :

$$F_H = 1000 \times 25 + 1000 \times 20 = 45,000 \text{ pounds}$$

which would allow low bearing loads at high bit weights.

Rotor Weight

The rotor in a 7 1/4 inch turbodrill weighs approximately:

$$W_R = 3000 \text{ pounds}$$

The rotors in positive displacement motors are shorter than those in turbodrills and consequently they weigh less.

Bearing Load

For the above example the thrust bearing load will equal:

$$F_B = 31,000 + 3000 = 34,000 \text{ pounds}$$

When the motor is operated off bottom $W=0$. This often corresponds to the worst operating condition for a turbodrill since the turbodrill operates near runaway speed (up to 2000 rpm) when it is off bottom and the combination of high bearing thrust and high speed quickly wears out the rubber friction bearings. The bearings in turbodrills are often ruined by operating the turbine off bottom for a few minutes.

As weight, W , is applied to the bit, the load on the thrust bearing decreases as the weight increases (EQ.1). For example, with a bit weight of 20,000 pounds, the bearing load is reduced from 34,000 pounds to:

$$F_B = 34,000 - 20,000 = 14,000 \text{ pounds}$$

At a bit weight of 29,000 pounds there is a downthrust on the bearing equal to:

$$F_B = 34,000 - 29,000 = 5000 \text{ pounds}$$

a bit weight of 39,000 pounds would produce an upthrust of :

$$F_B = 34,000 - 39,000 = -5000 \text{ pounds}$$

on the thrust bearings. Either of these loads would produce long bearing life without producing chattering of bearing. This example shows that thrust bearings in motors must be designed to handle both upward and downward thrusts.

One problem with turbodrill is that the rubber friction bearings cannot operate at high thrusts; consequently the bits must be operated at thrusts near the balanced condition. These thrusts are often not optimum for the drilling conditions which results in non-optimum bit operation. In some areas, these near balanced bit weights produce bit torques which are higher than the output torque of the drilling motor. In this case the bit weight must be reduced to the point where the motor will operate. The reduced bit weight produces higher bearing thrust and short bearing life. This shows that thrust bearings limitations complicates the operation of existing motors and in many cases precludes their economic application.

VII. THRUST BEARING DESIGN

Two types of thrust bearings are currently used in down-hole drilling motors - rubber friction bearings and rolling bearings.

Friction Bearings

American, German, Austrian and French turbodrills utilize rubber friction bearings (30) as shown in Figure 1.

Figure 1 - Turbodrill rubber friction bearings.
(Tiraspolsky and Rouvieve, 1962).

The thrust capability of these bearings is limited because of the sliding friction; consequently, a stack of 20 to 30 friction bearings is required even when the turbodrill

is operated at near balanced conditions (i.e., where the bit weight approximately equals the hydraulic downthrust).

Drilling mud is continually leaked through the friction bearings to coat and lubricate them. The abrasive particles in the drilling mud filter out in the bearings and cause rapid failure of the rubbing elements.

Ball Bearings

The Dyna-Drill utilizes tungston carbide bearings as shown in Figure 2. These bearings operate in abrasive drilling mud and wear out in 20 to 50 hours. This short life seriously limits application of Dyna-Drills for straight hole drilling.

Figure 2 - Dyna-Drill ball bearings.

One problem with ball bearings is that they wear in an elliptical manner. The Russians have reduced this problem by using split bearing races as shown in Figure 3. The split races reportedly allow the balls to maintain their spherical

shape as they wear. These split races are utilized in nearly all Russian downhole motors.

Figure 3 - Russian ball bearings with split races (laonesian, 1976).

Figure 4 & 5 show Russian motors utilizing split bearing races.

Figure 4 - Russian turbodrill bearing assembly (Laonesian, 1968)

Figure 5 - Russian Moineau motor bearing assembly (Ogly, 1974).
(Ogly, 1974).

A key element in manufacturing ball or roller bearings is to forge the parts so that the fiber patterns are paralleled to the wear surfaces as shown in Figure 6. The dynamic capacities of the controlled fiber races are twice those of conventional races. The Russians have carried this technology one step further and manufacture the balls so that the fibers are paralleled to the surface over the entire surface of the ball. This increases the life of the balls and allows them to wear in a spherical manner.

Figure 6 - Fiber flow in bearing races.

The need for fiber orientation greatly increases the cost of manufacturing experimental bearings since they must be manufactured from forged parts. Testing unforged bearings

will give much shorter life and will not give a true indication of the potential of the bearing.

Various ball thrust bearings will be tested during this program to compare their performance to roller bearings.

Angular Contact Bearings

Angular contact bearings absorb both axial and radial thrust as shown in Figure 7. These bearings are used in helicopters and other critical applications.

Figure 7 - Angular contact bearings

The angular contact bearings can be placed back-to-back to accommodate rotor loads in either direction or in series to reduce the unit bearing load as shown in Figure 8.

Back-to-Back

Series

Figure 8 - Angular contact bearings.

Angular contact bearings were used in the Whittle turbodrill as shown in Figure 9. This turbodrill was unsuccessful primarily due to problems with the rotary seals. Engineering Enterprises tested angular contact bearings in its turbodrill and found that the bearings failed due to the high radial loads.

Figure 9 - Angular contact bearings used in Whittle turbodrill.

Roller Bearings

Roller bearings can handle higher thrust loads than ball bearings because they have line contacts whereas balls have point contacts. There are two basic types of roller bearings - cylindrical roller and tapered roller. Cylindrical rollers are not true rolling. They are often split into 2 or 3 parts as shown in Figure 10 in order to reduce skidding as the rollers turn. The self-aligning bearings absorb misalignment between the rotor and housing.

Conventional

Self-Aligning

Figure 10 - Cylindrical roller thrust bearings.

Figure 11 shows a drilling motor assembly which utilizes cylindrical roller thrust bearings (105) in conjunction with radial ball bearings (110).

Figure 11 - Drilling motor utilizing cylindrical roller thrust bearing (Kern & Striegler, 1974).

Tapered roller bearings (Figure 12) are true rolling and therefore can operate at high thrusts. The axial force produces an outward radial force on the tapered rollers which, in turn, produces friction between the rollers and the races. With the narrow bearings used in drilling motors, this friction can be more detrimental than the skidding which occurs with cylindrical rollers. Both cylindrical and tapered rollers will be tested during this program.

Figure 12 - Drillings motor utilizing tapered roller bearings (Peterson, 1974).

The inclined spherical roller bearings shown in Figure 13 are able to accommodate combine thrust-radial loads. These spherical rollers can operate at high speeds and are therefore used in pumps in deep oilwells.

Figure 13 - Drilling motor utilizing inclined spherical roller bearings (Warder, 1975).

Cylindrical and tapered roller bearings will be extensively tested during this program. Based on available information it appears that roller bearings are the leading candidate for use in an improved drilling motor.

Thrust Bearing Load Distribution

Because of space limitations, it has been necessary to distribute the thrust load across several sets of bearings in order to reduce the unit load on the bearings. Several systems have been used to distribute the load including:

1. Rubber donuts or pads
2. Rubber friction bearings
3. Belleville springs
4. Metal wire pads
5. Metal field rings
6. Hydraulic pistons

Tiraspol'sky and Rouviere utilized wire mesh pads (16) bearings in a turbodrill as shown in Figure 14.

Figure 14 - Wire mesh pad thrust load equalizes
(Tiraspolsky and Rouviere, 1969).

Garrison utilized rubber compression pads (68) to distribute the load across two bearings as shown in Figure 15.

Figure 15 - Rubber pad thrust load equalizers (Garrison, 1971).

Kern utilized hydraulic pistons (27' and 28') to equalize the load across ball thrust bearings as shown in Figure 16.

Figure 16 - Hydraulically balanced thrust bearings (Kern, 1973).

Both Gruber and Striegler proposed using Belleville springs (180 and 180') to distribute the load across the thrust bearings as shown in Figure 17.

Figure 17 - Belleville spring thrust load equalizers (Striegler, 1973).

Berryman (1974) utilized a yieldable elastic ring (85) to distribute the thrust load as shown in Figure 18.

Figure 18 - Yieldable elastic ring thrust load equalizer (Berryman, 1974).

These thrust load balancing systems create problems because the elements usually do not distribute the loads evenly and the components tend to fatigue and break up, putting foreign matter into the bearings and causing catastrophic failure. A system containing a single bearing that would handle the entire thrust could be highly desirable and will be a goal of this program. If this is not possible, various load distribution systems similar to those shown in Figures 14 to 18 will be tested.

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VIII. RADIAL BEARING DESIGN

Radial bearings are not a major problem in downhole motors because the radial loads are much lower than the thrust loads and because the space constraints are not so great.

Three types of radial bearings are used in downhole motors:

1. Sleeve bearings
2. Ball bearings
3. Roller bearings

The radial bearings are subjected to high loads when drilling through broken and dipping formations and when drilling in highly deviated wells. The radial bearings in turbodrills must prevent excessive deflection of the rotor or stator so that the blades do not come into contact and cause catastrophic failure.

Sleeve Bearings

Drilling motor marine bearings utilize an elastomer wear surface as shown in Figure 1. Drilling mud passes through the grooves in the sleeve and cools and lubricates the bearing.

Figure 1 - Radial Marine Bearing

Figure 2 shows a turbodrill which utilizes a marine bearing (26) at the bottom of the tool. The turbine blades (2 and 9) are located at the top of the tool with the rubber friction bearings (30) located directly below.

Figure 2 - Turbodrill components (Tiraspolsky and Rouviere, 1962)

Radial Ball Bearings

Radial ball bearings similar to the one shown in Figure 3 are being tested in downhole drilling motors. These bearings have the advantage that they are compact, they take up little space and they have long life when operated in lubricant.

Figure 3 - Radial Ball Bearings

Self Aligning radial ball bearings can also be used as shown in Figure 4.

Figure 4 - Self-Aligning Radial Ball Bearings

Yost (1942) patented the radial ball bearing shown in Figure 5 where the balls are placed in inclined passageways. This distributes the wear over a much larger area, thereby increasing the life of the radial bearing.

Figure 5 - Ball Radial Bearing (Yost, 1942)

Radial Roller Bearings

Needle radial roller bearings can be used in downhole drilling motors as shown in Figure 6. These roller bearings can operate at much higher loads than ball bearings, but they require more axial space.

Figure 6 - Needle Radial Roller Bearing

In larger diameter motors where the space constraints are not so great, self-aligning spherical roller bearings can be used as shown in Figure 7. These bearings will allow some misalignment between the rotor and the housing.

Figure 7 - Self-Aligning Spherical Roller Bearing.

Sleeve, ball and roller radial bearings will give adequate life when operated in a sealed lubrication system. Ball or roller radial bearings will be used during this program because of their compactness.

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IX. ROTARY SEAL DESIGN

Five basic types of rotary seals have been used in drilling motors:

1. Constricted Flow (Marine Bearings)
2. Labyrinth
3. Face
4. Packing
5. Radial Lip

Constricted Flow Seal

Marine bearings and rubber friction bearings are used to constrict the flow and serve as rotary seals in turbodrills shown in Figure 1.

Approximately 20 to 40 percent of the mud leaks through the marine bearing while the remainder flows through the bit.

Figure 1 - Turbodrill Rotary Mud Seal (Alliquander, 1968)

Labyrinth Seal

Labyrinth seals, which were just used in turbodrills in the 1950's, provide a long, tortuous path for the fluid to leak through, as shown in Figure 2. Because of the long, narrow flow path, the flow rate through the seal is small. A disadvantage of this seal is that the abrasive solids in the mud filter out in the seal and cause rapid wear.

Figure 2 - Labyrinth Seal (Dicky, 1972).

Face Seal

Face seals have been tested extensively in downhole drilling motors including the Dresser and Whittle turbodrills. Figure 3 shows the spring loaded face seal used in the Whittle turbodrill.

Figure 3 - Whittle Turbodrill Face Seal (Ratcliff, 1969).

To date face seals have been unsuccessful because abrasive particles from the drilling mud get between the lapped surfaces, scoring them and causing excessive leakage. This is possibly caused by pressure fluctuations during drilling or during drillpipe connections.

After a connection is made, the turbodrill is lowered to the bottom producing a pressure surge in the annulus. If the annular pressure exceeds the pressure in the drillpipe it will tend to pump the face seals open and allow abrasive mud to come between the faces. The problem may also arise

while drilling when the bit bounces on the hole bottom and produces high annular pressure surges which may momentarily exceed the pressure inside the drillpipe (i.e., inside the seal). Because of the dynamic pressures involved, face seals do not appear to be a good candidate for motor used with abrasive drilling muds. They could possibly be used in geothermal drilling applications where clear water is used.

Packing Seals

Packing seals provide positive seals that allow essentially no leakage. They are a good candidate for use in improved drilling motors since they overcome the problem of filtering abrasive solids out into leaking seals (e.g., marine bearings and labyrinth seals). These seals are being successfully tested in Russian turbodrills and in the Cullen electrodrill. Figure 4 shows a turbodrill bearing assemble utilizing packing seals.

Figure 4 - Turbodrill Bearing Utilizing Packing Seals
(Alliquander, 1968).

Figure 5 shows a high pressure swivel seal (15,000 psi) that operates effectively in abrasive drilling mud for 60 hours at 150 rpm. This seal should last 200 to 300 hours in a downhole drilling motor due to the lower operating pressure (1000 to 2000 psi).

Figure 5 - High Pressure Swivel Seal (Heilhecker, et.al, 1973).

The key to the long life of this seal are the beryllium copper (39 and 43) and Teflon (38 and 42) backup rings.

Radial Lip

Radial lip seals utilize an interference between the flexible sealing element (54) and the rotating member (20) to provide the seal as shown in Figure 6. The radial seal

acts against an annular sleeve (55) which is laterally moveable in order to absorb any non concentric motion of the rotor.

Figure 6 - Radial Lip Seal (Kern, & Striegler, 1973).

No data are available on drilling motor tests using these radial lip seals.

Redundant Seal

When the seals in a downhole motor fail, it is necessary to make a "round trip" in order to replace the seals in the motor. This is expensive since it may take up to 20 hours to make a round trip in a deep well where rig costs are on the order of \$5000 per day. It would therefore be advantage to have spare or redundant rotary seals which are not subjected to pressure until the primary seals fail. The redundant seals would activate automatically or by a command from the surface, (e.g., go-devil or pressure pulse). This would reduce the time to replace the seal from 10 to 20 hours to a few minutes.

The high pressure swivel seal shown in Figure 7 contains redundant seals (41 and 54) which are not activated until the primary seals leak (40 and 53). This redundant seal assembly was successfully used on Exxon's high pressure drilling project.

Figure 7 - Rotary Seal Containing Spare Seal (Maurer, et al., 1973).

Once the primary seals fail and fluid leaks through open ports 43 and 59, the ports are closed by plugs 66 and 67. This activates the spare seals and drilling is resumed.

A similar spare packing concept that could be used in down-hole drilling motors. In this case, it would be desirable to activate the spare seals automatically when the primary seals fail.

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X. SEALED LUBRICATION SYSTEMS

The bearings in existing Dyna-Drills and turbodrills have short lives because they operate directly in the abrasive drilling mud. It appears that with improved sealing materials and sealing techniques, the technology exists to build drilling motors with sealed lubrication systems where the bearings operate in oil or grease. It is a major goal of this program to develop a reliable sealed bearing assembly that can be utilized on any existing or new drilling motor.

Unbalanced Bearing Lubrication Systems

In most sealed lubrication systems, the pressure drop across the bit acts across the lower seal which seals the lubricant in the bearing reservoir as shown in Figure 1.

Figure 1 - Unbalanced Bearing Lubrication System (Striegler, 1973).

In this system, the mud inside the drillpipe (pressure p_i) acts above floating piston 80 pressurizing the lubricant around the bearings. The lower rotary seal (33) is subjected to annular pressure p_a on the outside and drillpipe pressure p_i on the inside; consequently this seal is subjected to a differential pressure equal to the pressure drop across the bit ($p_i - p_a$).

The Baker Moineau Motor (Figure 2) utilizes a similar arrangement except that a differential area piston (115) is used at the top which produces a slightly higher pressure in the lubricant reservoir than in the drillpipe. The bearings are flooded with abrasive drilling mud when the bottom rotary seal (114) fails.

Figure 2 - Unbalanced Bearing Lubrication System (Berryman, 1974).

A similar system is shown in Figure 3 where the thrust bearings (24) operate in lubricant at annulus pressure since mud from the annulus acts on the bottom of the floating piston (27).

Figure 3 - Unbalanced Pressure Bearing Lubricating System
(Dicky, 1972).

The pressure drop across the bit acts across the upper rotating seal (21). When this seal leaks the thrust bearings are flooded with abrasive mud. The lower radial bearing (22) operates in mud at all times.

These unbalanced systems have the advantage of simplicity but the disadvantage that the bearings are flooded with abrasive mud when the primary rotary seal fails.

Most drilling motors currently being developed utilize unbalanced lubrication systems similar to those shown above. In some cases, bellows or diaphragms are used in place of the floating pistons.

Balanced Bearing Lubrication Systems

If a separate rotary mud seal is used, it is possible to use a balanced pressure bearing lubrication system where the rotary seals are not subjected to differential pressure as shown in Figure 4.

Figure 4 - Balanced Bearing Lubrication System (Striegler, 1973).

The labyrinth seal (43) produces annular pressure above the upper rotary seal (21) while mud acting beneath floating piston (27) produces annular pressure within the bearing lubrication system; consequently there is no differential pressure across the rotary seals containing the lubricant in the bearing chamber. The lower radial bearing (23) operates in the drilling mud at all times. If the labyrinth seal fails, mud will flow to the annulus without flooding the bearings.

A similar balanced pressure lubrication system is shown in Figure 5 where mud acting below floating piston 255 produces annular pressure in the lubricant reservoir. The bit pressure drop is taken across rotary face seals 244A. Vent holes (246) allow annulus mud to come into contact with seal 244, so there is no pressure drop across rotary seal 244 or rotary seal 245. If the face seals leak, mud will flow to the annulus through vent 246 and not flood the bearings.

Figure 5 - Balanced Bearing Lubrication System (Fox, 1976).

Diaphragms and bellows can be used in place of the floating piston to pressurize the bearing lubrication system. Warder, 1975, used a bellows (24) to produce annulus pressure in the bearing reservoir as shown in Figure 6. The upper rotary seal (32) was subjected to the differential pressure while the lower rotary seal (32) was subjected to annulus pressure on both sides. If the upper seal fails, the bellows will be broken and the mud will flow directly to the annulus without flooding the bearings.

Figure 6 - Bearing Lubrication System (Warder, 1975).

Dyna-Drill patented the balanced pressure lubrication system shown in Figure 7.

Figure 7 - Balanced Bearing Lubrication System (Tschirky, 1975).

The drilling mud pressure drops from drillpipe pressure to annulus pressure as the mud flows through flow constrictions (24) and out vent (29) to the annulus. Mud at annulus pressure acts above rotary seals (30) while lubricant in the bearing is at annulus pressure due to annulus mud acting on bellows ring (70). Consequently, the lubrication system is balanced

and the bearings are not flooded with mud if the flow constrictor fails.

Both balanced and unbalanced bearing lubrication systems should be considered for use in new drilling motors. The balanced systems have the advantage that the bearings are not flooded when the primary rotary seals fail, but at the expense of increased complexity. If rotary seals are developed that are reliable for 200 to 300 hours, the unbalanced systems may be preferred since the bearings will have to be replaced anyway if the seals fail after this length of time. If the seals fail after 50 to 100 hours, the balanced systems would be preferred since the bearing life is much greater than this except in extremely rough drilling.

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XI. LUBRICANTS

The primary function of lubricants is to reduce the frictional resistance occurring at the surfaces of two solids when one is moving with respect to the other. In addition, the lubricant acts as a heat-transfer medium, protects against rust and corrosion, acts as a sealing media and may be a scavenger for contaminants.

Physical tests used to characterize lubricants include viscosity, density, pour, gravity, flash and fire, demulsibility, odor and color. Chemical tests include carbon residue, oxidation, corrosion, acidity, oiliness, extreme pressure, sulfur, ash and precipitation number.

Petroleum Oils

Petroleum oils are widely used because of their effectiveness in most existing equipment. These lubricating oils are refined from naturally occurring hydrocarbons. The three main crude oil sources in the United States are characterized as pure paraffin (Pennsylvania), semi-paraffin (Gulf Coast) and naphthene base (California). These petroleum oils are not suitable for use in geothermal drilling because of the loss of viscosity with increasing temperature as shown in Figure 1.

Figure 1 - Effect of Temperature on the Viscosity of Petroleum Oils (Baumeister & Marks, 1967).

Animal, Vegetable and Fish Oil

Animal, vegetable and fish oils differ from petroleum oils by being saponifiable with caustic alkalies. These fatty oils are added to petroleum oils to increase the absorbed film on the working surface. This increases the "oiliness" and the load bearing capacity of the oilfilm, an important factor in heavily loaded bearings such as in rock bits or drilling motors. Table 1 lists the properties of typical fatty oils.

Table 1. Typical Properties of Some Animal, Vegetable, and Fish Oils
(Commonly used in compounded petroleum products)

	Free fatty acid or oleic, percent	Specific gravity at 60 F	Saponification number	ASTM pour point, deg F	Flash, open cup, deg F	Viscosity Saybolt sec (at 100 F)	Price, cents per lb
Blown rapeseed.....	4-9	0.968	198-215	40	460	800-900	25
Castor, hydrogenated.....	0.15-2	0.963	178-181	15	505	1,485	21
Corn.....	1-10	0.922	189-192	14	480	180-190	17
Cottonseed.....	1-3	0.922	187-197	40	580	180-195	16
Degras, common.....	15-25	0.951	95-120	65	520	175-190*	11
Lard oil, prime.....	1.5-2	0.915	192-198	35	460	200-210	14
Lard No. 1.....	15-18	0.915	192-198	50	440	200-210	10
Neat's foot.....	10-12	0.915	193-204	40	440	200-210	16
Porpoise jaw.....		0.925	269-273	0	415	80-85	\$15
Rosin, kidney.....		0.990	70-90	25	270	220-230	9
Sperm, 45 NW.....	1-2	0.880	120-140	45	550	95-105	15
Tallow, acidless.....	0.2-0.5	0.927	193-198	80	610	205-215	10
200 Vis. neutral, 95 VI.....	0	0.890	0-30	15	410	200-210	3

* At 210 F; it is semisolid at 100 F.

These organic fatty oils also have severe temperature limitations and are unsuitable in most geothermal drilling applications. At high temperatures they tend to decompose to corrosive acids.

Petroleum Grease

A grease is defined as "a solid to semisolid product of dispersion of a thickening agent in a liquid lubricant". Other ingredients imparting special properties may be included.

Most greases are thickened with soaps which are formed by the chemical reaction of fats or fatty acids with alkali. Alkalies used in grease manufacture are usually calcium, sodium, lithium, borium, aluminum, and lead while the fat may be animal, vegetable or fish. Characteristics of some common greases are given in Table 2. Dropping point is the temperature when the grease changes from a semisolid to a liquid as tested by a standard ASTM test (ASTM D566-64).

Table 2. General Characteristics of Greases

Soap base	Texture	Dropping point (approx)	Maximum continuous usable temperature	Effect of water	Primary uses
Calcium.....	Buttery to smooth	200-225 F	175 F	Resistant	General purpose industrial lubricant for plain bearing and line shafting.
Sodium.....	Fibrous or smooth	350-450 F	260 F	Susceptible	Ball and roller bearings over wide speed and temperature range, automobile wheel bearings, universal joints, spring shackles.
Mixed base, sodium-calcium	Fairly buttery to fibrous	345 F	250 F	Susceptible	All types of ball and roller bearings, and special applications at both high and low temperatures, depending upon composition.
Lithium.....	Buttery to stringy	375 F	300 F	Resistant	Aircraft lubrication at temperatures from -100 to +300 F, and many applications in automotive and industrial use.
Aluminum.....	Some buttery, some stringy, but never fibrous	200 F	150 F	Resistant	Special applications where resistance to centrifugal force or adhesiveness is desired.

Most organic greases have serious temperature limitations and are not suitable for use in high temperature geothermal applications. The lithium base greases have higher operating temperatures than the calcium and sodium base greases.

Synthetic Fluids

Synthetic oils or greases are manmade as opposed to naturally occurring petroleum fluids. Most of these synthetic lubricants are organic chemicals such as:

1. polyolefins
2. polyalkylene glycols
3. esters
4. polyphenol ethers
5. halogenated hydrocarbons

while others are inorganic:

1. silicones

These synthetic materials have been found to have some outstanding characteristics which make them especially good lubricants (e.g., high viscosity index or thermal stability) in special applications. These synthetic fluids are expensive and are used only where their special properties are essential such as in jet aircraft engines.

Because of the high temperatures and extreme loads, synthetic lubricants will have to be used in geothermal drilling motors.

Lubrication

Most new drilling motors utilize ball or roller bearings. The function of the lubricant is to:

1. Lubricate the sliding contacts
2. Provide cooling
3. Prevent rusting
4. Prevent scuffing and wear
5. Seal against drilling mud

Experience has shown that the lowest-viscosity lubricant which can be retained in the bearing should be used. Leakage through the seals increases as the viscosity decreases; consequently, very thin lubricants cannot be used in drilling motors.

The choice of lubricant is influenced by:

1. Temperature
2. Speed
3. Load
4. Sealing
5. Bearings Size

The critical factor in drilling motors is to prevent the abrasive drilling mud from leaking into the lubricant and causing rapid wear of the bearings. If the seal problem can be solved, there are several commercially available high temperature lubricants that could be used in drilling motors.

REFERENCES

1. Baumeister, T. and Marks, L. S., 1967, Standard Handbook For Mechanical Engineers, Lubricants and Lubrication, p.6-243 to 6-258, McGraw-Hill.
2. Bearings, Machine Design, Section 5 - Lubricants, June 20, 1974.
3. Faires, V. M. 1965, Design of Machine Elements, Macmillan Company, New York.
4. Fuller, D. D., 1966, Theory And Practice of Lubrication For Engineers, John Wiley & Sons.

XII. PROJECT STATUS

Maurer Engineering is responsible for the design of the motor components tested on this project including the following:

1. Bearing Pack Assembly
2. Thrust Bearings
3. Radial Bearings
4. Rotary Seals
5. Lubricants

Bearing Pack Assembly

Because of the space constraints involved, the design of downhole motors is a compromise. As the bearings are made larger, the housing becomes thinner and less stiff resulting in higher tool deflections while the rotor becomes smaller thereby increasing the possibility of twisting it in two. The life of the bearings increases rapidly with increased bearing size; consequently it is important to make the bearings as large as possible while still maintaining adequate housing stiffness and rotor strength.

Because of these constraints, it was essential to design the bearing pack assembly to accommodate the maximum sized bearings that can be used. A detailed design effort aimed at maximizing the size of the thrust bearings resulted in the bearing assembly shown in Figure 1.

Figure 1 - ERDA Drilling Motor Bearing Assembly.

This ERDA bearing pack can be screwed onto existing motors or can be used on motors to be designed in later phases of this program. This bearing pack is designed for use on 7-3/4 inch diameter motors. It allows the use of larger bearings than contained in any commercial drilling motor of this size.

In this assembly, the upward thrust produced while drilling is carried by upper roller bearing 18' while the downward thrust produced when the motor is rotated off bottom is carried by lower roller bearing 18. The bearings operate in lubricant pressurized by mud acting on top of floating piston 29. Rotary seals 4 are subjected to the differential pressure drop between the lubricant in the reservoir at internal drillpipe pressure (p_i) and the mud at annulus pressure (p_a). This differential pressure ($p_i - p_a$) equals the pressure drop across the bit nozzles. If the rotary seals leak, the floating piston moves downward and maintains pressure on the lubricant reservoir. Radial roller bearings 12, 14 and 21 absorb the radial loads on the rotor.

Thrust Bearings

Bearing life is a direct function of the size of the bearings; consequently it is desirable to use as large thrust bearings as possible in the bearing assembly. The bearing assembly shown in Figure 1 allows the use of larger thrust bearings than are used in any commercial 7 3/4 inch downhole drilling motors (Table 1).

TABLE 1

COMPARISON OF MOTOR THRUST BEARINGS

<u>Motor</u>	<u>Bearing Type</u>	<u>Bearing Number</u>	<u>Inner Diameter (in)</u>	<u>Outer Diameter (in)</u>	<u>Basic Load Rating (pounds)</u>
Turbodrill	Ball	920	3.94	5.91	27,600
ERDA*	Ball	725	4.92	6.89	33,100
ERDA	Roller	RTH-734	4.00	7.00	122,000

*Similar to 7-3/4 inch Dyna-Drill Bearings

Existing drilling motors cannot use commercial roller bearings because these bearings come in one inch incremental sizes and they are too large to fit into the motors. In the new ERDA bearing pack, the bearing size has been increased to 4.00 inch I.D. X 7.00 inch O.D. which allows the use of the RTH-734 roller bearing. The specifications for roller and ball thrust bearings are given in Tables 2 and 3, respectively.

TABLE - 2

ROLLER THRUST BEARINGS

RTM

Single Acting
MEDIUM
SECTION
Inch series
(continued)

BEARING NUMBER	EXTERNAL DIMENSIONS				INTERNAL DIMENSIONS				LOAD RATING (lbs)		Shaft & Housing Fillet
	B	D	H		b	d	f	g	Dynamic	Static	
RTM-616	1.937	3.500	1.000	-.006	2.13	3.41	.25	.500	35600	40800	.062
RTM-617	2.000	3.625	1.000	-.008	2.19	3.50	.25	.500	35400	40800	.062
RTM-618	2.125	3.750	1.000	-.008	2.38	3.65	.25	.500	37400	44500	.062
RTM-619	2.250	3.875	1.000	-.008	2.44	3.75	.25	.500	37100	44500	.062
RTM-620	2.375	4.000	1.000	-.008	2.63	3.90	.25	.500	39000	48250	.062
RTM-621	2.500	4.125	1.000	-.008	2.67	4.00	.25	.500	38800	48250	.062
RTM-622	2.625	4.343	1.000	-.008	2.88	4.22	.25	.500	40600	51900	.062
RTM-623	2.750	4.468	1.000	-.008	3.06	4.34	.25	.500	42400	55700	.062
RTM-624	3.000	4.718	1.000	-.010	3.25	4.59	.25	.500	44000	59400	.062
RTM-625	3.250	4.968	1.000	-.010	3.50	4.84	.25	.500	43400	59400	.062
RTM-626	3.500	5.218	1.000	-.010	3.75	5.09	.25	.500	46900	66800	.062

RTH

Single Acting
HEAVY
SECTION
Inch series *ERDA* →

RTH-727	2.000	6.000	1.375	-.008	2.25	5.88	.375	.625	103000	154000	.062
RTH-728	2.000	7.000	1.375	-.008	2.25	6.88	.375	.625	113000	177000	.062
RTH-729	2.000	8.000	1.375	-.008	2.25	7.88	.375	.625	144000	248000	.062
RTH-730	3.000	6.000	1.375	-.010	3.25	5.88	.375	.625	95000	143000	.062
RTH-731	3.000	7.000	1.375	-.010	3.25	6.88	.375	.625	117000	191000	.062
RTH-732	3.000	8.000	1.375	-.010	3.25	7.88	.375	.625	145000	259000	.062
RTH-733	3.000	9.000	1.375	-.010	3.25	8.88	.375	.625	173000	334000	.062
RTH-734	4.000	7.000	1.750	-.010	4.25	6.88	.50	.750	122000	186000	.062
RTH-735	4.000	8.000	1.750	-.010	4.25	7.88	.50	.750	165000	282000	.062
RTH-736	4.000	9.000	1.750	-.010	4.25	8.88	.50	.750	194000	352000	.062
RTH-737	4.000	10.000	1.750	-.010	4.25	9.88	.50	.750	205000	384000	.062
RTH-738	5.000	8.000	1.750	-.010	5.25	7.88	.50	.750	136000	224000	.062
RTH-739	5.000	9.000	1.750	-.010	5.25	8.88	.50	.750	184000	337000	.062
RTH-740	5.000	10.000	2.000	-.010	5.25	9.88	.562	.875	226000	402000	.125
RTH-741	5.000	11.000	2.000	-.010	5.25	10.88	.562	.875	264000	500000	.125
RTH-742	5.000	12.000	2.000	-.010	5.25	11.88	.562	.875	304000	611000	.125
RTH-743	6.000	9.000	2.000	-.015	6.38	8.75	.562	.875	157000	253000	.125
RTH-744	6.000	10.000	2.000	-.015	6.38	9.75	.562	.875	208000	371000	.125
RTH-745	6.000	11.000	2.000	-.015	6.38	10.75	.562	.875	189000	332000	.125
RTH-746	6.000	12.000	2.000	-.015	6.38	11.75	.562	.875	214000	396000	.125
RTH-747	7.000	10.000	2.000	-.015	7.38	9.75	.562	.875	160000	271000	.125
RTH-748	7.000	11.000	2.000	-.015	7.38	10.75	.562	.875	228000	433000	.125
RTH-749	7.000	12.000	2.000	-.015	7.38	11.75	.562	.875	276000	562000	.125
RTH-750	7.000	14.000	3.000	-.015	7.38	13.75	.812	1.375	483000	873000	.250
RTH-751	8.000	12.000	3.000	-.015	8.38	11.75	.812	1.375	334000	539000	.250
RTH-752	8.000	14.000	3.000	-.015	8.38	13.75	.812	1.375	444000	797000	.250
RTH-753	8.000	16.000	3.000	-.015	8.38	15.75	.812	1.375	573000	1130000	.250
RTH-754	10.000	16.000	3.000	-.020	10.38	15.75	.812	1.375	488000	944000	.250
RTH-755	10.000	18.000	3.750	-.020	10.38	17.75	1.031	1.687	724000	1394000	.250
RTH-756	10.000	20.000	3.750	-.020	10.38	19.75	1.031	1.687	864000	1782000	.250
RTH-757	12.000	18.000	3.750	-.020	12.50	17.75	1.031	1.687	649000	1242000	.250
RTH-758	12.000	20.000	4.500	-.020	12.50	19.75	1.25	2.000	929000	1787000	.250
RTH-759	12.000	24.000	4.500	-.020	12.50	23.75	1.25	2.000	1296000	2825000	.250
RTH-760	14.000	20.000	3.750	-.020	14.50	19.75	1.031	1.687	700000	1420000	.250
RTH-761	14.000	22.000	3.750	-.020	14.50	21.75	1.031	1.687	890000	1968000	.250
RTH-762	14.000	24.000	3.750	-.020	14.50	23.75	1.031	1.687	1054000	2516000	.250
RTH-763	16.000	22.000	4.500	-.020	16.50	21.50	1.25	2.000	797000	1548000	.250
RTH-764	16.000	24.000	4.500	-.020	16.50	23.75	1.25	2.000	1067000	2284000	.250

• Thrust plates and roller assemblies may be purchased separately.

TABLE - 3

SKF ball thrust bearings

SKF BEARING NUMBER	Aligning Washer Number ¹	Nominal Bearing Dimensions								Aligning Washer Dimensions						Max. Fillet Radius ²	Basic Dynamic Load Rating C	Brg. Wt. lb ³
		Bore d		d ₂		Outside Diam. D		Height H		d ₃	D ₁		H ₁		h ₁			
		mm	in.	mm	in.	mm	in.	mm	in.		in.	mm	in.	mm		in.	in.	
802		15	.5906	16	.630	35	1.3780	14	.551						.024	3610	.1	
804		20	.7874	21	.827	40	1.5748	14	.551						.024	3870	.1	
805		25	.9843	26	1.024	48	1.8898	15.5	.610						.024	4990	.2	
806		30	1.1811	31	1.220	53	2.0866	15.5	.610						.024	5030	.3	
807		35	1.3780	36	1.417	62	2.4409	18	.709						.024	7080	.4	
808		40	1.5748	42	1.654	64	2.5197	18	.709						.024	6280	.4	
809		45	1.7717	47	1.850	73	2.8740	22	.866						.024	8490	.7	
810		50	1.9685	52	2.047	78	3.0709	22	.866						.024	9790	.8	
811		55	2.1654	57	2.244	88	3.4646	24.5	.965						.024	12000	1.1	
812		60	2.3622	62	2.441	90	3.5433	24.5	.965						.024	10400	1.1	
813		65	2.5591	67	2.638	100	3.9370	27	1.063						.039	12900	1.6	
814		70	2.7559	72	2.835	103	4.0551	27	1.063						.039	13200	1.6	
less-Bright Series 9																		
medium type																		
815		75	2.9528	77	3.031	110	4.3307	27	1.063						.039	13300	1.6	
816		80	3.1496	82	3.228	115	4.5276	29	1.142						.039	15400	2.1	
817		85	3.3465	88	3.465	125	4.9213	30.5	1.201						.039	19200	2.1	
818		90	3.5433	93	3.661	135	5.3150	30.5	1.201						.039	23000	3.1	
818		95	3.7402	98	3.858	140	5.5118	32.5	1.280						.039	24900	3.1	
820		100	3.9370	103	4.055	150	5.9055	32.5	1.280						.039	27600	4.1	
821		105	4.1339	108	4.252	155	6.1024	40	1.575						.039	31300	5.1	
823		115	4.5276	118	4.646	165	6.4961	43	1.693						.039	31700	6.1	
825		125	4.9213	128	5.039	175	6.8898	46.5	1.831						.039	33100	7.1	
826		130	5.1181	133	5.236	180	7.0866	46.5	1.831						.039	31800	9.1	
828		140	5.5118	143	5.630	200	7.8740	52	2.047						.039	41800	11.1	
702	702 U	15	.5906	17	.669	35	1.3780	15	.591	.866	38	1.496	17	.669	.197	.024	3610	
704	704 U	20	.7874	22	.866	40	1.5748	16	.630	.984	45	1.772	18	.709	.217	.024	3870	
705	705 U	25	.9843	27	1.063	48	1.8898	17	.669	1.417	50	1.969	19	.748	.236	.024	4990	
706	706 U	30	1.1811	32	1.260	53	2.0866	18	.709	1.673	59	2.323	20	.787	.236	.024	5030	
707	707 U	35	1.3780	37	1.457	62	2.4409	21	.827	1.811	67	2.638	23	.906	.276	.024	7080	
708	708 U	40	1.5748	42	1.654	64	2.5197	21	.827	1.890	69	2.717	23	.906	.276	.024	6280	
709	709 U	45	1.7717	47	1.850	73	2.8740	25	.984	2.087	78	3.071	27	1.063	.315	.024	8490	
710	710 U	50	1.9685	52	2.047	78	3.0709	25	.984	2.283	83	3.268	27	1.063	.315	.024	9790	
less-Bright Series 7																		
medium type																		
711	711 U	55	2.1654	57	2.244	88	3.4646	28	1.102	2.795	94	3.701	30	1.181	.315	.024	12000	1.1
712	712 U	60	2.3622	62	2.441	90	3.5433	28	1.102	2.717	96	3.780	30	1.181	.315	.024	10400	1.1
713	713 U	65	2.5591	67	2.638	100	3.9370	32	1.260	2.913	105	4.134	34	1.339	.394	.039	12900	1.6
714	714 U	70	2.7559	72	2.835	103	4.0551	32	1.260	3.189	109	4.291	34	1.339	.354	.039	13200	1.6
715	715 U	75	2.9528	77	3.031	110	4.3307	32	1.260	3.504	114	4.488	34	1.339	.354	.039	13300	1.6
716	716 U	80	3.1496	82	3.228	115	4.5276	35	1.378	3.504	124	4.882	37	1.457	.394	.039	15400	2.1
717	717 U	85	3.3465	88	3.465	125	4.9213	38	1.496	3.858	138	5.433	40	1.575	.413	.039	19200	2.1
718	718 U	90	3.5433	93	3.661	135	5.3150	38	1.496	4.331	141	5.551	40	1.575	.413	.039	23000	3.1
719	719 U	95	3.7402	98	3.858	140	5.5118	41	1.614	4.213	151	5.945	43.5	1.713	.492	.039	24900	3.1
720	720 U	100	3.9370	103	4.055	150	5.9055	41	1.614	4.646	156	6.142	44	1.732	.512	.039	27600	4.1
721	721 U	105	4.1339	108	4.252	155	6.1024	46	1.811	4.646	163	6.417	49	1.929	.551	.039	31300	5.1
723	723 U	115	4.5276	118	4.646	165	6.4961	49	1.929	5.039	173	6.811	52	2.047	.551	.039	31700	6.1
725	725 U	125	4.9213	128	5.039	175	6.8898	53	2.086	5.472	186	7.323	56	2.205	.551	.039	33100	7.1
728	728 U	140	5.5118	143	5.630	200	7.8740	60	2.362	6.142	212	8.346	62	2.441	.610	.039	41800	11.1

- ¹To obtain a complete set of bearing and aligning washer order for example: 702 and 702 U.
²The maximum fillet on the shaft or in the housing, which will be cleared by the bearing corner.
³Does not include weight of the washer.

Note: Bearings of Series 9 are also available in extra precision tolerances. See page 51.
 *This series is in the process of being discontinued. Consult for availability.

The basic load rating (C) of a bearing is defined as the bearing load at which 10 percent of the bearings will fail after 1 million revolutions. Table 1 shows that the load factor of the roller bearing is 122,000 pounds compared to 33,100 pounds for the largest ball bearing that can be used in the ERDA assembly and 27,600 pounds for the bearing used in the commercial turbodrill.

The life of a bearing can be calculated as follows:

$$L_{10} = \frac{16667}{N} \left(\frac{C}{P} \right)^3 \quad (\text{Ball})$$

$$L_{10} = \frac{16667}{N} \left(\frac{C}{P} \right)^{3.33} \quad (\text{Roller})$$

where

L_{10} = Bearing Life (hours)

C = Basic Load Rating (lbs)

P = Bearing Load (lbs)

N = Rotary Speed (rpm)

For a motor rotating at 500 rpm with 30,000 pounds bearing load, the life of the bearings in Table 1 will equal:

$$\begin{aligned} \text{Turbodrill (Ball)} &= \frac{16667}{500} \left(\frac{27,600}{30,000} \right)^3 = 26.0 \text{ hours} \\ \text{ERDA (Ball)} &= \frac{16667}{500} \left(\frac{33,100}{30,000} \right)^3 = 44.8 \text{ hours} \\ \text{ERDA (Roller)} &= \frac{16667}{500} \left(\frac{122,000}{30,000} \right)^{3.33} = 3580 \text{ hours} \end{aligned}$$

This calculation shows that the predicted life of the roller bearing is 3580 hours compared to 44.8 and 26.0 hours for the ball bearings. This order of magnitude increase in

predicted life shows that there is potential for major improvement with the ERDA assembly provided the rotary seals operate effectively and prevent the bearings from being flooded with drilling mud. The actual life of these bearings may be somewhat less than predicted by the calculations because of the high dynamic loads produced by drillstring vibrations and by the cutting action of the bits.

The thrust bearings will be made of M50 tool steel which has a draw temperature of 1100°F compared to a draw temperature of 450°F for the 5210 steel used in most bearings. In addition to the higher draw temperature, the M50 steel bearings are capable of higher shock and dynamic loads.

Radial Bearings

Radial bearings are not a major problem in most drilling motors because 1) the radial loads are much smaller than the thrust loads and 2) the space constraints are not so great on the radial bearings. The ERDA bearing assembly will utilize MR-64 and MR-88 roller radial bearings (Table 4) which have basic load factors of 37,210 and 82,710 pounds, respectively. These radial bearings should not fail unless the rotary seals fail and allow the bearings to be flooded with drilling mud.

TABLE - 4
ROLLER RADIAL BEARINGS

MR® series *CAGEROL*™ BEARINGS

Outer Ring and Roller Assembly with Separable Inner Ring

McGILL NUMBER outer ring & roller assy.	McGILL NUMBER inner ring only	B		D		W		WI	H	R	SHAFT DIA.			HSG. BORE DIA.			AIRCRAFT STATIC CAPACITY	BASIC DYNAMIC CAPACITY
		+ .0005	TOL.	+ .0005	TOL.	+ .0005 - .0005	+ .0005 - .0005	HOLE DIA.	MAX. FILLET for shaft & hsg.	ROTAT. SHAFT	TOL. + .0005	STAT. SHAFT	TOL. + .0005	ROTAT. HSG.	TOL. + .0005	STAT. HSG.		
MR-36-N	MI-28-N	1.7500		3.0000		1.500	1.510	3/16	.060	1.7505		1.7496		2.9996		3.0007	67700	17640
MR-36	MI-28	1.7500		3.0000		1.750	1.760	3/16	.060	1.7505		1.7496		2.9996		3.0007	82100	20360
MR-36	MI-30	1.8750		3.0000		1.750	1.760	3/16	.060	1.8755		1.8746		2.9996		3.0007	82100	20360
MR-40-N	MI-32-N	2.0000		3.2500		1.500	1.510	3/16	.080	2.0005		1.9996		3.2496		3.2507	73900	18420
MR-40	MI-31	1.9375		3.2500		1.750	1.760	3/16	.080	1.9380		1.9371		3.2496		3.2507	89600	21270
MR-40	MI-32	2.0000		3.2500		1.750	1.760	3/16	.080	2.0005		1.9996		3.2496		3.2507	89600	21270
MR-40	MI-34	2.1250		3.2500		1.750	1.760	3/16	.080	2.1258		2.1247		3.2496		3.2507	89600	21270
MR-44-N	MI-36-N	2.2500		3.5000		1.500	1.510	3/16	.080	2.2508		2.2497		3.4995		3.5008	80100	18990
MR-44	MI-35	2.1875		3.5000		1.750	1.760	3/16	.080	2.1883		2.1872		3.4995		3.5008	97100	21900
MR-44	MI-36	2.2500		3.5000		1.750	1.760	3/16	.080	2.2508		2.2497		3.4995		3.5008	97100	21900
MR-48-N	MI-40-N	2.5000		3.7500		1.500	1.510	3/16	.080	2.5008		2.4997		3.7495		3.7508	83100	19220
MR-48	MI-38	2.3750		3.7500		1.750	1.760	3/16	.080	2.3758		2.3747		3.7495		3.7508	100800	22180
MR-48	MI-39	2.4375		3.7500		1.750	1.760	3/16	.080	2.4383		2.4372		3.7495		3.7508	100800	22180
MR-48	MI-40	2.5000		3.7500		1.750	1.760	3/16	.080	2.5008		2.4997		3.7495		3.7508	100800	22180
MR-52	MI-42	2.6250		4.2500		1.750	1.760	3/16	.080	2.6258		2.6247		4.2495		4.2508	116600	29370
MR-52	MI-44	2.7500		4.2500		1.750	1.760	3/16	.080	2.7508		2.7497		4.2495		4.2508	116600	29370
MR-56-N	MI-48-N	3.0000		4.5000		1.750	1.760	1/4	.080	3.0008		2.9997		4.4995		4.5008	127200	30780
MR-56	MI-46	2.8750		4.5000		2.000	2.010	1/4	.080	2.8758		2.8747		4.4995		4.5008	151630	35170
MR-56	MI-47	2.9375		4.5000		2.000	2.010	1/4	.080	2.9383		2.9372		4.4995		4.5008	151630	35170
MR-56	MI-48	3.0000		4.5000		2.000	2.010	1/4	.080	3.0008		2.9997		4.4995		4.5008	151630	35170
MR-60	MI-50	3.1250		4.7500		2.000	2.010	1/4	.100	3.1260		3.1246		4.7495		4.7508	157950	35910
MR-60	MI-52	3.2500		4.7500		2.000	2.010	1/4	.100	3.2510		3.2496		4.7495		4.7508	157950	35910
MR-64	MI-54	3.3750		5.0000		2.000	2.010	1/4	.100	3.3758		3.3746		4.9999		5.0011	170590	37210
MR-64	MI-56	3.5000		5.0000		2.000	2.010	1/4	.100	3.5008		3.4996		4.9999		5.0011	170590	37210
MR-68	MI-58	3.6250		5.2500		2.000	2.010	1/4	.100	3.6258		3.6246		5.2499		5.2511	183220	38560
MR-68	MI-60	3.7500		5.2500		2.000	2.010	1/4	.100	3.7508		3.7496		5.2499		5.2511	183220	38560
MR-72	MI-62	3.8750		6.0000		2.250	2.260	1/4	.100	3.8758		3.8746		5.9999		6.0011	224400	57900
MR-80	MI-64	4.0000		6.5000		2.250	2.260	1/4	.100	4.0008		3.9996		6.4999		6.5011	243900	62320
MR-80	MI-68	4.2500		6.5000		2.250	2.260	1/4	.100	4.2508		4.2496		6.4999		6.5011	243900	62320
MR-88-N	MI-72-N	4.5000		7.0000		2.500	2.515	1/4	.100	4.5008		4.4996		6.9999		7.0011	298900	64310
MR-88	MI-72	4.5000		7.0000		3.000	3.015	1/4	.100	4.5008		4.4996		6.9999		7.0011	369800	82710
MR-96-N	MI-80-N	5.0000		7.5000		2.500	2.515	3/16	.120	5.0010		4.9995		7.4998		7.5011	321000	66190
MR-96	MI-80	5.0000		7.5000		3.000	3.015	3/16	.120	5.0010		4.9995		7.4998		7.5011	397200	77760
MR-104-N	MI-88-N	5.5000		8.0000		2.500	2.515	3/16	.120	5.5010		5.4995		7.9998		8.0011	343100	68240
MR-104	MI-88	5.5000		8.0000		3.000	3.015	3/16	.120	5.5010		5.4995		7.9998		8.0011	424600	80170
MR-116	MI-96	6.0000		9.1250		3.000	3.015	3/16	.120	6.0012		5.9995		9.1248		9.1261	476600	101560
MR-124	MI-104	6.5000		9.6250		3.000	3.015	3/16	.120	6.5012		6.4995		9.6250		9.6265	494200	103190
MR-132	MI-112	7.0000		10.1250		3.000	3.015	3/16	.120	7.0012		6.9995		10.1250		10.1265	529500	106910
MR-140	MI-120	7.5000		10.6250		3.000	3.015	3/16	.160	7.5010		7.4995		10.6250		10.6265	564800	110010
MR-148	MI-128	8.0000		11.1250		3.000	3.015	3/16	.160	8.0010		7.9995		11.1250		11.1265	600100	113390

PRECISION BEARINGS **McGILL**

Rotary Seals

Marine, Labyrinth and Flow Restrictor rotary seals have been ruled out for use in the ERDA bearing assembly because they leak continuously and are not positive seals.

Face seals have been ruled out because of bad experiences with these seals in numerous drilling motors. These include the problem of "pumping" the faces open and allowing abrasive solids to come between the faces and scoring the lapped surfaces. Face seals will be reconsidered if packing seals do not operate effectively.

Packing and Radial Lip seals will be tested initially on this program. Meetings have been held with representatives of over 20 different seal companies and the following seals have been selected as candidates for use in the ERDA drilling motor bearing assembly:

1. Chevron Packing (V) - Bayou City Rubber, Inc.
2. Chevron Packing (V) - Custom Rubber Products
3. Chevron Packing (V) - Engineering Enterprises
4. Uneepac Packing (W) - Johns-Manville
5. Radial Lip - Utex Industries, Inc.

The Chevron and Uneepac Packing seals operate in a similar manner and the primary difference between them is in the compounds used in making them. The Radial Lip seals differ from packing seals in that only one seal is loaded at a time whereas the Chevron and Uneepac seals are all loaded simultaneously.

The Bayou City Rubber Company Chevron packing seals shown in Figure 2 are composed of a Dupont EPDM base material.

Figure 2 Bayou City Rubber Chevron Packing Seal

Bayou City Rubber representatives state that by careful compounding of the EPDM base material, these seals should operate effectively at temperatures of 500 to 550°F.

The proposed compound has the following physical properties:

Tensile Strength	-	2100 psi
Elongation	-	700 percent
Hardness	-	59 Shore A
Comp. Str. (22 hrs. @ 2.57°F)	-	22 percent

These Chevron "V" rings will be molded in a curved plane to eliminate line contact with the shaft. Plans are being made to make a single cavity mold to produce this packing.

The Custom Rubber Company Chevron seals are similar to those shown in Figure 2 except that they will be manufactured from a different compound.

The Engineering Enterprises Packing seal assembly consist of alternating layers of thin GRAFOIL seal rings and beryllium copper backup rings as shown in Figure 3.

Figure 3 - Engineering Enterprises Chevron Packing Seal.

The GRAFOIL rings are made of pure graphite and therefore have good lubricity and good thermal conductivity. The high thermal conductivity is important in dissipating the heat generated at the sliding seal surface since heat buildup is the major cause of rotary seal failures. These seals are currently being tested in the Engineering Enterprises turbo-drill and are a leading candidate for use in the ERDA bearing assembly.

The Johns-Manville "Uneepac" seal assembly shown in Figure 4 is also a candidate for use in the ERDA bearing assembly. These seal rings were successfully used in the Exxon 15,000 psi swivel.

Figure 4 - Johns-Manville Uneepac Seal.

The Uneepac rings come in conventional and deep base designs as shown in Figure 5.

Figure 5 - Johns-Manville "Uneepac" Packing Seals.

These Uneepac seal rings come in several different compounds that operate at temperatures up to 500°F as shown in Table 5. The Style 470 VITON rings will be used in the initial tests.

TABLE 5
MATERIALS USED IN
JOHN-MANSVILLE UNEEPAC SEALS

J-M MATERIAL STYLE NO.	DESCRIPTION	SERVICE GUIDE			J-M MATERIAL COLOR CODE†
		MEDIA	PSI	TEMPERATURE	
HOMOGENOUS STYLES					
410	Neoprene, 90 Durometer	CF	1000	0 +250°F	Yellow
413	Buna-N, 90 Durometer	CF	2000	0 +250°F	
470	Viton*, 90 Durometer	SF	1500	- 40 +500°F	
FABRIC REINFORCED STYLES					
405	Neoprene with Heavy Cotton Fabric	CF	3000	20 +250°F	White
406	Neoprene with Asbestos Cloth	CF	3000	+ 250 +400°F	Orange
446	Butyl with Heavy Cotton Fabric	PEF	3000	-20 +200°F	Yellow & Pink
475	Viton* with Heavy Cotton Fabric	SF	3000	-50 +350°F	Yellow
486	Neoprene with Light Weight Cotton Fabric	CF	1000	- 20 +250°F	Red

*DuPont Trademark

CF - Common Hydraulic Fluids

SF - Severe Fluids

PEF - For Phosphate Ester type fluids. Not for petroleum base fluids.

†For ease of identification, all parts produced from the above materials, except #410 and #413, are identified by a color code of a single stripe (one color) or a double stripe (two colors).

Utex Industries has two seal designs that are candidates for use in downhole drilling motors. The conventional Radial Lip seal arrangement shown in Figure 6 utilizes Style RD-136 seal rings composed of a ceramic fiber core sheathed with a flurocarbon treated aramid fabric impregnated with graphite. The flurocarbon treated fabric is resistant to 450° F while the graphite impregnant is used to reduce friction and heat.

Figure 6 - Utex Conventional Radial Lip Seal.

Aluminum bronze backup rings support each of the seal rings so that only one of the seal rings is loaded at a time. This arrangement limits wear to the seal ring providing the seal. This is in contrast to Chevron and Uneepac seals (Figures 2 to 4) where each seal ring loads the seal ring beneath it and they all wear simultaneously. This is a major advantage of the Utex seals over the packing seals.

One problem with rotary seals is that sealing occurs along a line around the rotating shaft resulting in high heat generation and high temperatures at this interface. The high temperature often degrades the lubricant and causes high friction at the line contact which leads to rapid failure of the seal. Utex has proposed the unconventional "canted" seal shown in Figure 7 which overcomes this problem. As the shaft rotates, the shaft surface is continually lubricated since the seal comes into contact with each point on the shaft surface only twice every revolution. This, should result in cooler operation and longer life. These canted seals will be tested during this program.

Figure 7 - Utex Canted Radial Lip Seal.

Dupont has developed a new perfluoroelastomer trademarked KALREZ which has better high temperature properties than TEFLON. This material is a hybrid of TEFLON (PTFE) and VITON (Fluoroelastomer) and as such combines the high temperature properties of TEFLON with the resistance of VITON. This perfluoroelastomer has much better high temperature properties than TEFLON coated VITON as shown in Figure 8. KALREZ is a candidate material for use in downhole motor seals and will be tested if the more readily available seals do not perform satisfactorily.

Figure 8 - Comparison Of Seal Retention Force at 500°F.

Lubricants

An extensive review of high temperature lubricants has been completed and the lubricants shown in Table 6 have been selected as candidates for use in drilling motors. Data sheets on these lubricants are contained in Appendix 1.

TABLE 6
CANDIDATE HIGH TEMPERATURE LUBRICANTS

<u>Company</u>	<u>Lubricant Number</u>	<u>Thickener</u>	<u>Oil</u>	<u>Temperature Limitations (°F)</u>
Amoco	31052	Arylurea	Silicone	+450
Amoco	M-125	Teflon	Silicone	+450
Amoco	LG-1191	?	Dimethyl Silicone	+500
Amoco	Amdex #2	Calcium	?	+500
Dow-Corning	560	?	Chlorophenyl Silicone	650
Dow-Corning	710	Graphite	Silicone	500
Dow-Corning	M-77	MoS ₂	Mineral	450
Pacer*	Eon-E11	?	Mineral	?
Superior	White	?	?	+500
Superior	Moly	?	?	+500
Tenneco	Anderol 456	?	Diester	Over 350

* Tested in Engineering Enterprises turbodrill.

REFERENCES

Asti, F. J., 1976, "Perfluoroelastomer For Seals In Natural Energy Production Applications", 51st Annual SPE Fall Meeting, New Orleans, Oct 3-6.

Precision Bearings, 1975 Handbook H-80A, McGill Manufacturing Company, Inc., Valparaiso, Indiana, 46383, March.

Roller Thrust Bearings, 1969, Andrews Bearing Corporation, Spartanburg, South Carolina, 29304.

SKF Product Catalogue, 1971, Catalogue 450A, SKF Industries, Incorporated, Philadelphia, Pa.

APPENDIX I

HIGH TEMPERATURE LUBRICANTS

AMERICAN OIL COMPANY

ENGINEERING SERVICE

SUPERMIL GREASES - FOR AVIATION AND AEROSPACE TYPICAL TEST DATA

	SUPERMIL Grease No. 1177	SUPERMIL Grease No. A72832	SUPERMIL Grease No. 73552	SUPERMIL ASU Grease No. 31052	SUPERMIL Grease M-125	SUPERMIL Grease No. 94532
Military Specification	MIL-G- 7711-A	MIL-G- 23827-A	MIL-G- 25537-A	MIL-G- 25013-D	None	MIL-G- 23549
Temperature Range, °F. (Higher for short periods)	-40 to +250	-100 to +250	-65 to 165°F.	-100 to +450	-30 to +450	0 to +400
Type of Thickener	Lithium Soap	Lithium Soap	Calcium (OH)Stear- ate	Arylurea	* Teflon	Arylurea
Type of Oil	Mineral	Diester	Mineral	Silicone	Special Silicone	Mineral
Oil Viscosity at 100°F. SUS	165	63	69	350	2360	4200
Worked Penetration - 77°F.	280	285	270	300	290	285
Dropping Point, °F.	374	372	303	450+	400+	500+
Water Resistance, % Loss	1.4	1.3	1.8	1.4	4.0	None
Oil Separation, 30 hrs. % Loss	2.8	1.9	2.1	6.8	6.8	1.0
at Temperature, °F.	210	212	210	450	400	350
Evaporation in 22 hrs. at Temperature °F.		250	210	400	400	
Loss, % by Wt.		0.8	3.1	4	5	
Oxidation Bomb Sta- bility lbs. drop, 100 hrs. at 210°F.	4	2	1	1	1	
Bearing Performance, 10,000 rpm at Temperature, °F.	250	250		450	400	
Time, hours	1000+	1000+		500+	1000+	
Corrosion Resistance	Good	Good	Good	Good	Good	Good
Load Carrying Capacity	Moderate	Moderately High	Moderate	Low	Very High	High
Color	Amber	Amber	Amber	Lavender	White	Black
Recommended Use	Aircraft General Purpose	Aircraft and in- struments, Actuators and Gears	Helicop- ter Oscil- lating Bearing	For Air- craft ball and roller bearings extreme high & low temp.	High temp. & loads *DuPont Trade- mark	Catapult foot pad High temp. and low sliding speed

	<u>LG-1157</u>	<u>LG-1191</u>	<u>LG-1195</u>
Type Fluid	Special Silicone	DiMethyl Silicone	Mineral Oil
Thickener	Teflon Type	ASU Type	Lithium Complex
Vis at 100°F., SSU	330	475	900
Pen. Unworked	283	272	240
Pen. Worked 77°F.	272	279	267
Drop Point, °F.	450+	500+	473
Leakage (Cone & Beaker), %			
24 Hrs. at 212°F.	2.2	None	4.9
24 Hrs. at 350°F.	5.1	2.8	
24 Hrs. at 450°F.	7.9	3.2	
Rust Test			Pass
4-Ball Weld	250	126	200
Load Wear Index	21.3	20.4	22.4
L-35 Bearing Test			
Hours at 350°F.			750-1,000
Hours at 450°F.	1150+	650	

American Oil Company Regional Offices

CENTRAL REGION

20 North Wacker Drive
Chicago, Illinois 60606
312-856-6111

EASTERN REGION

Blaustein Building
1 North Charles Street
Baltimore, Maryland 21201
301-727-6700

GREAT LAKES REGION

16025 Northland Drive
Southfield, Michigan 48076
313-444-4000

WESTERN REGION

114 East Armour Boulevard
Kansas City, Missouri 64111
816-561-4140

SOUTHERN REGION

6 Executive Park Drive, N.E.
Atlanta, Georgia 30329
404-634-6611

TYPICAL TEST DATA

	AMDEX Grease No. 0 EP	AMDEX Grease No. 1 EP	AMDEX Grease No. 2 EP
Type of thickener	Calcium Complex		
Worked penetration, 60 strokes	365	325	287
Worked penetration, 100,000 strokes	382	351	321
Viscosity of Oil @ 100°F.	900	900	900
Viscosity Index of Oil	90	90	90
Timken E.P., lbs. pass	55	55	55
Dropping point °F.	500+	500+	500+
Bearing Rust Protection Test	No. 1	No. 1	No. 1
Bomb Oxidation, psi drop/100 hrs.	1.5	1.5	1.5
Copper Corrosion	Pass	Pass	Pass
Roll Stability, 20 hrs. 1% Change	+4.6	+5.1	+6.4
Oil Separation, 24 hrs. @ 210°F. %	2.9	2.0	0.5
Color	Blue	Blue	Blue
Texture	Smooth	Smooth	Smooth

STORAGE & HANDLING

All greases are made unfit for use when contaminated. Keep containers covered and exercise care when transferring grease. The less times grease is handled, from original container to final use, the less chance there is of damaging contamination.

RECOMMENDATIONS

AMDEX is available in 3 grades to meet different consistency requirements. AMDEX Grease No. 1EP is a general purpose grade. AMDEX Grease No. 0EP is a soft grade for use where better flow is needed and AMDEX Grease No. 2EP is a heavy grade to provide better sealing and resistance to flow.

CONTAINER SIZE

AMDEX EP Greases are available in 14½ oz. cartridges, 35 lb. pails, 120 lb. drums, and 400 lb. barrels. Bulk deliveries are also available.

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DISPERSIONS AND OILS (cont.)

OILS (SILICONE)

	DESCRIPTION	TEMPERATURE RANGE (F)	COLOR	EVAPORATION PER CENT MAXIMUM	LOADING IN FAULT PER CENT	ASTM GRADE	ISO GRADE	POUR POINT (F)	FLASH POINT (F)	FIRE POINT (F)
DOW CORNING 4X SPRAY	Dimethyl silicone that provides non-melting, non-freezing, non-gumming, protection of plastic and rubber, metal parts	-40 to 400	Clear			.940		-85	325	
DOW CORNING 580 OIL	Chlorophenyl silicone oil for use at extremely low or high temperatures. Gives good ferrous metal lubricity.	-65 to 650	Light Straw	4.5 (120 hrs., 392F)	1000 lbs. at failure	1.04	75	-85	550	660
DOW CORNING 710 OIL	General purpose lubricant offering low volatility and high resistance to oxidation and gumming. 710-G contains 5% colloidal graphite. 710-R contains a heat stable rust inhibitor.	0 to 500	Colorless to Light Straw	24 (120 hrs., 482F)	100 lbs. at failure	1.11	500	-8	575	645

GREASES

Use of Dow Corning FS-3452 valve lubricant extended plug valve relubrication from every two weeks to more than eight months.

Industrial Applications

Lubricating solids blended in medium to low percentages to form high quality lubricating greases for extreme-pressure fortification. Greases are preferred where it is desired to add EP characteristics or to extend service interval to an ordinary grease application.

They are used for:

- antifriction ball and roller bearings, splines, keyed shafts, and sleeves
- high-temperature bearings
- threaded connections . . . to reduce torque loss due to friction, galling, or seizure

Silicone and Fluorosilicone lubricants are used for: anti-friction, ball, roller and journal bearings, valves, shafts and packings where solvents, acids, chemicals are present.

Continued on next page

GREASE

The information and data contained herein are based on information we believe reliable. You should thoroughly test any application, and independently conclude satisfactory performance before commercialization. Suggestions of uses should not be taken as inducements to infringe any particular patent.

DOW CORNING CORPORATION, MIDLAND, MICHIGAN 48640

Atlanta Boston Brussels Chicago Cleveland Dallas Detroit Greensboro
Los Angeles New York San Francisco Sydney Tokyo Toronto

DOW CORNING

Specific Applications:

Molykote G paste

- Wear-in for bearings, splines, cams and ways.
- Threads and bolt heads.
- Press fitting.
- Prevents fretting corrosion and cold welding of ball and roller bearings, bushings and splines.
- Ovens and kilns.
- Prevents galling, welding and metal pickup on tools and dies in machinery, stamping and drawing operations, bending, thread rolling, cold heading, swaging and forging.
- Extreme pressure lubricant for sliding bridge pads, turbine pedestals and rock crushers.

Molykote U paste

- Prevents sticking and tearing of O-rings.
- Packings and glands on compressors and pumps.
- High temperature threads, couplings and static joints.
- Elastomers, rubber seals and gaskets (static or sliding).

Molykote M-77 paste

- O-rings, packings, seals (except silicone rubber)
- Leather washers and plastic or fiber bearings
- Threads, couplings, splines, static joints
- Gears and cams

APPLICATION METHODS:

- Burnishing—rub on with lint-free cloth
- Brushing—preferably using brush with short stiff bristles
- Aerosol spray (Molykote G paste only)
- Grease gun—Due to the high solids content of these products and their high specific gravity there is a tendency for oil to separate; thus the products are not recommended for use in pressure lines, nor should they remain in guns for long periods. The products can be re-mixed.

REMOVAL:

After a thorough degreasing, the following two strippers will generally remove MoS₂ even after metal working operations: Oakite Stripper R-6, Oakite Products, Inc., New York, N. Y., and Wyandotte WLG Cleaner, Wyandotte Chemical Corp., Wyandotte, Michigan. These strippers are primarily for use with ferrous metals. If there is any question about their effect on a specific metal, the suppliers should be consulted and/or test panels should be processed before stripping finished parts.

PACKAGING:

Molykote G, U and M-77 pastes are available in 2 oz., 1 lb., 4 lb., 15 lb., and 50 lb. containers.

Molykote G paste is also available in ¼ oz., 16-oz. aerosol and 250 lb. containers.

TYPICAL PROPERTIES

(not to be used as specifications)

	<u>G</u>	<u>U</u>	<u>M-77</u>
Color.....	Black.....		
Penetration (worked 60), ASTM D 217.....	345	320	325
Temperature Range, F.....	0 to 250	-40 to 350	-50 to 450
Density, lbs./gal.	15.5	16.0	16.0
Bleed (16 hrs., 302 F), percent...	3.6	4.3	6.0
Evaporation (16 hrs., 302 F) percent.....	1.5	0.4	0.3
Coefficient of friction, maximum*.....	0.12	0.12	0.2
Effect on copper strip (2 hrs. at 212 F) ASTM D 130-65.....	#2 slight tarnish	#1 no dis- coloration	#1 no dis- coloration
4 Ball wear scar, mm. (40 kg, 1 hr., 1740 rpm, steel/steel).....	0.91	0.87	2.94
Falex seizure load, lbs.	3300	3875	1400

* Test conditions—Alpha LPW-4—Press Fit Test; Bearing conditions: steel vs. steel; velocity, inches per minute—8.6; pressure, psi—10,000; environment ambient.

The information and data contained herein are based on information we believe reliable. You should thoroughly test any application, and independently conclude satisfactory performance before commercialization. Suggestions of uses should not be taken as inducements to infringe any particular patent.

DOW CORNING CORPORATION, MIDLAND, MICHIGAN 48640

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DOW CORNING

PACER PETROLEUM COMPANY • 7600 NORTH LOOP EAST • HOUSTON, TEXAS 77028

EON E-11 SYNTHETIC MOTOR FLUID

I. "SE" PERFORMANCE TEST

SPECIFICATION

EON E-11

Sequence IIC	Average Engine Rust	8.6 Min.	8.7
Sequence IIIC	Viscosity Increase, 100°F		
	40 Hours, %	400 Max.	2.0
	64 Hours, %	---	1.0
	Average Engine Ratings at 64 Hours		
	Piston Skirt Varnish	9.5 Min.	9.5
	Oil Ring Land	6.0 Min.	6.3
	Sludge Rating	9.0 Min.	9.4
	Ring Sticking	None	None
	Lifter Sticking	None	None
	Scuffing and Wear, 64 Hours		
	Cam or Lifter Scuffing	None	None
	Cam and Lifter Wear, Inches - Average	0.0010	0.0009
	Maximum	0.0020	0.0015
Sequence VC	Average Engine Sludge	8.5 Min.	9.5
	Average Piston Skirt Varnish	8.2 Min.	8.8
	Average Engine Varnish	8.0 Min.	8.8
	Oil Screen Clogging, %	5 Max.	0
	Oil Ring Clogging, %	5 Max.	0
	Compression Ring Sticking	None	None
L-38, Method 3405	Bearing Weight Loss, mg.	40 Max.	31.6
Physical Test	Viscosity - SSU @ 100°F	80-83	
	Pour Point	-50°F Max.	
	Cloud Point	-50°F Max.	
	Flash Point	460°F	
	Fire Point	500°F	
	Appearance	Bright and Clear	

II. PHYSICAL SPECIFICATIONS

Viscosity, SSU @ 100°F	445-455	Cloud Point, °F	-50°F Max.
210°F	80-83	Flash Point, °F	460°F
Viscosity Index	200	Fire Point, °F	500°F
Pour Point, °F	-50°F Max.	Appearance	Bright & Clear

AUTOMOTIVE AND INDUSTRIAL LUBRICANTS • DIESEL FUEL • GASOLINE • TRANSPORT • DRUMS • CONTAINERS

SUPERIOR PRODUCT DATA

-16-

SUPERIOR WHITE LUBRICANT

DESCRIPTION

- * SUPERIOR WHITE LUBRICANT IS A HIGH TEMPERATURE, GENERAL PURPOSE LUBRICANT
- * NON-MELTING-EFFECTIVE TEMPERATURE RANGE OF -40°F TO +500°F- WILL RESIST HEAT AND COLD WITHOUT CHANGING ITS CONSISTENCY
- * WATER RESISTANT-CONTAINS AN OXIDATION AND RUST INHIBITOR AND AN EXTREME PRESSURE ADDITIVE
- * VERY ADHESIVE, HAS A GOOD METAL WETTING ABILITY ENABLING IT TO LUBRICATE WHERE OTHER GREASES FAIL
- * ECONOMICAL-SAVES TIME AND MONEY BY LASTING LONGER AND REDUCING GREASE CONSUMPTION
- * MEETS SPECIFICATION NLGI GRADE #2

APPLICATIONS

FOOD PROCESSORS	MACHINERY	
MINES	AUTOS & TRUCKS	
FACTORIES	FARM EQUIPMENT	SHIPS

PACKAGED

SUPERIOR WHITE LUBRICANT -- CASE OF 10-14½ oz. CARTRIDGES

SUPERIOR WHITE LUBRICANT -- CASE OF 12-2 Lb. CANS

SUPERIOR WHITE LUBRICANT -- INDIVIDUAL 14½ oz. CARTRIDGE

SUPERIOR WHITE LUBRICANT -- INDIVIDUAL 2 Lb. CAN

SUPERIOR WHITE LUBRICANT -- INDIVIDUAL 35 Lb. PAIL

SUPERIOR WHITE LUBRICANT -- INDIVIDUAL 400 Lb. DRUM

SUPERIOR LUBRICANT CO. INC. — HOUSTON, TEXAS

SUPERIOR PRODUCT DATA

SUPERIOR MOLY LUBRICANT

-17-

DESCRIPTION

- * SUPERIOR MOLY LUBRICANT IS A LOW FRICTION, HIGH TEMPERATURE, GENERAL PURPOSE LUBRICANT CONTAINING MOLYBDENUM DISULFIDE
- * NON-MELTING-EFFECTIVE TEMPERATURE RANGE OF -40°F TO +500°F- WILL RESIST HEAT AND COLD WITHOUT CHANGING ITS CONSISTENCY
- * WATER RESISTANT-CONTAINS AN OXIDATION AND RUST INHIBITOR AND AN EXTREME PRESSURE ADDITIVE
- * VERY ADHESIVE, HAS A GOOD METAL WETTING ABILITY ENABLING IT TO LUBRICATE WHERE OTHER GREASES FAIL
- * ECONOMICAL-SAVES TIME AND MONEY BY LASTING LONGER AND REDUCING GREASE CONSUMPTION
- * MEETS SPECIFICATION NIGI GRADE #2

APPLICATIONS

FACTORIES

MACHINERY

MINES

AUTOS & TRUCKS

SHIPS

FARM EQUIPMENT

PACKAGED

SUPERIOR MOLY LUBRICANT -- CASE OF 10-14½ oz. CARTRIDGES

SUPERIOR MOLY LUBRICANT -- CASE OF 12-2 Lb. CANS

SUPERIOR MOLY LUBRICANT -- INDIVIDUAL 14½ oz. CARTRIDGE

SUPERIOR MOLY LUBRICANT -- INDIVIDUAL 2 Lb. CAN

SUPERIOR MOLY LUBRICANT -- INDIVIDUAL 35 Lb. PAIL

SUPERIOR MOLY LUBRICANT -- INDIVIDUAL 400 Lb. DRUM

SUPERIOR LUBRICANT CO. INC. — HOUSTON, TEXAS

Turner Place, P.O. Box 2
Piscataway, New Jersey 08854
(201) 981-5000

PRODUCT DATA

ANDEROL (R) 456

SYNTHETIC MEDIUM VISCOSITY LUBRICATING OIL

ANDEROL 456 is a synthetic diester-based lubricant which is formulated to provide a wide operating temperature range, low wear rates, moderate load carrying, minimum rpm fluctuations, and low noise levels.

TYPICAL PROPERTIES

Optimum Operating Range, °F	-	-60 to 350
Viscosity @ 100°F, cSt	ASTM D445	36.2 (169.5 SUS)
Viscosity @ 210°F, cSt	ASTM D445	8.2 (53.1 SUS)
Viscosity @ -40°F, cSt	ASTM D445	6600
Viscosity Index	ASTM D2270	226
SAE Viscosity Number	SAE J309a	5W-20
Pour Point, °F	ASTM D97	-70
Flash Point, °F	ASTM D92	445
Specific Gravity, 77°F	ASTM D1298	0.93
Evaporation, 22 hours @ 210°F, %	ASTM D972	0.9
Four-Ball Wear, 1200 rpm, 167°F, 40 kg, hr, mm	ASTM D2266	0.3
Four-Ball EP, Weld Point, kg	ASTM D2783	190

APPLICATIONS

ANDEROL 456 is recommended for the following applications:

Impregnating fluid for powdered metal bearings, parts and wicks
Office, computing and accounting machinery
Measuring and dispensing pumps and meters
Gear lubrication
Mechanical spring and actuator mechanisms
Antifriction bearings
General purpose medium viscosity lubricating oil
Mold and die release agent
SAE 5W-20 crankcase oil for spark ignition engines
Low temperature oil
High VI hydraulic fluid
Speed reducers

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SEMI-ANNUAL REPORT
DESIGN OF DOWNHOLE MOTOR
SEAL, BEARING AND LUBRICATION TEST SYSTEMS

By
Alan D. Black
Sidney J. Green

Prime Contract: ERDA E(10-1)-1581
TERRA TEK, INC.
SALT LAKE CITY, UTAH

November 1, 1976

TR76-59

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INTRODUCTION

Terra Tek's portion of the contract is the establishment of the necessary equipment and testing to systematically evaluate existing and new seal, bearings, and lubrication components and packages designed and specified by sub-contractor Maurer Engineering for the purpose of improving the bearing section of downhole motors. The goal is to improve and extend the reliability, life and temperature capabilities of downhole motor bearing sections.

The test equipment provided is unique and will allow direct comparative testing of candidate seals, bearings and lubricants, which are subjected to the near downhole conditions of pressure, temperature and dynamics, but carefully controlled in the laboratory environment. Since actual downhole conditions are not well understood and complete laboratory simulation is not possible, the program is aimed at testing under the most realistic, controlled and understood conditions presently known and testing and evaluating the performance of given components by direct one on one comparison.

SEAL, BEARING AND LUBRICATION TEST DESIGN CRITERIA:

A great deal of effort has been spent to develop a realistic design criteria to provide laboratory facilities for testing seals, bearings and lubricants for downhole motors. The following general guidelines were used:

- (1) Downhole conditions should be closely simulated whenever possible; therefore, the range of conditions specified for the test facility, such as pressure, temperature, axial displacement, radial eccentricity, dynamics, RPM, and pressure fluctuations must be based on existing experience and knowledge of actual downhole conditions. Actual data measuring downhole conditions directly is limited. Exxon¹ made measurements with a downhole recorder which showed the dynamic conditions and pressure fluctuations at the drill bit. Conditions can also be inferred by observing wear and damage after running the tool downhole.
- (2) Provisions should be made to easily vary the simulated conditions over the pre-determined range of values.
- (3) The proper scale (full-scale) of the components tested should be maintained wherever possible. The size of seals and bearings should be the same as is used and needed in a downhole motor. Tests on smaller seals, however, could be used to screen candidate materials since these seals are less expensive than full-scale seals.

---

¹ Deily, F.H., et al: "Downhole Measurements of Drill String Forces and Motions", Translation, ASME Journal of Engineering for Industry, May, 1968.

(4) The test apparatus should be versatile, reliable and should not adversely influence the parameters you would like to control and/or measure. Installation and removal of the test seals and bearings must be done quickly, easily and without imparting any damage to these components. Reliability of the test is very important since tests will normally be of long duration and any premature failure of the test equipment could be detrimental to the test results.

The nature of the test apparatus will nearly always impart either some adverse condition or some ideal condition that may not be found downhole or under normal operating conditions. This influence needs to be minimized and where possible understood and measured to give added meaning to the interpretation of the test results.

(5) It is imperative to define a criteria for failure. What constitutes failure? Has a seal failed when an increase in torque first occurs, when leakage first begins, after a certain amount of wear is measured or at catastrophic failure and destruction? The criteria selected must reflect the condition that would cause adverse and damaging results in the field and must be used consistently to judge failure on all tests.

(6) Since there is nearly an infinite variety of combinations of test conditions available, it is important to choose the reasonable number and type of conditions necessary to obtain meaningful results. The extent of the testing program of course will depend greatly upon time and money constraints.

(7) Since it is impossible to completely simulate downhole conditions in the laboratory, the tests should show the relative performance of different candidate seals, bearings and lubricants. All test data should be comparative in nature.

Downhole Drill Motors
Semi-Annual Report

BASIC TEST CONDITIONS:

The basic range of test conditions was chosen and forms the basis for design of the test apparatus.

<u>Parameters</u>	<u>Seals</u>	<u>Seal & Bearing</u>
Pressure	5,000 psi	5,000 psi
Differential Pressure Across Seals	0-5,000 psi	0-5,000 psi
Pressure Pulsation	0-500 psi	0-500 psi
Temperature	32-600°F	32-600°F
RPM	100-1700 RPM	100-1700 RPM
Radial Eccentricity	0-.010 inch	0-.010 inch
Axial Displacement	0-.020 inch	0-.020 inch
Axial Load	0	100,000 lbs
Radial Load	0	0
Dynamics Load	From axial and radial eccentricity only	Axial load - 0- 100,000 lbs
Pressure Media	Lubricant, water, mud, abrasives	Lubricant, water, mud, abrasives

INDEPENDENT AND DEPENDENT VARIABLE PARAMETERS:

A large number of independent and dependent variables were identified for both testing seals and seal-bearing packages and summarized below:

Independent Variables:

- (1) Seals
  - (a) Type - face seals, packing seals, lip seals
  - (b) Seal configuration
  - (c) Number of seal rings and overall seal lengths
  - (d) Seal cross-section
  - (e) Seal back-up rings
  - (f) Pre-compression of seals
  - (g) Seal compounds and materials
- (2) Radial Eccentricity
- (3) Axial Displacement
- (4) Pressure
- (5) Differential pressure across the seal
- (6) Dynamic pressure fluctuation or pulsation
- (7) RPM
- (8) Temperature
  - (a) Maintain seal at constant temperature
  - (b) Allow seal temperature to rise at a controlled rate
  - (c) Allow the seal temperature to rise uncontrolled from seal friction
- (9) Shaft and housing material
- (10) Shaft surface finish
- (11) Geometry of surface finish of shaft
- (12) Lubricants
  - (a) Type - grease, oil, dry
  - (b) Viscosity
  - (c) Lubricant additives

Downhole Drill Motors
Semi-Annual Report

- (13) Mud and abrasives
- (14) Metal particles in lubricant
- (15) Bearing type
  - (a) Ball
  - (b) Roller
  - (c) Tapered roller
  - (d) Friction bearing
- (16) Bearing material, hardness and surface finish
- (17) Bearing size and configuration
- (18) Single or multiple bearings
- (19) Bearing fits and tolerances
- (20) Bearing load
- (21) Dynamic loading of bearings
  - (a) Frequency
  - (b) Magnitude of loading
  - (c) Uniform cyclic loading
  - (d) Random cyclic loading

DEPENDENT VARIABLES:

- (1) Torque
- (2) Seal Temperature
- (3) Bearing Temperature
- (4) Cooling water inlet and outlet temperatures
- (5) Lubricant or mud leakage
- (6) Seal wear and degradation
- (7) Bearing wear and damage
- (8) Lubricant degradation
- (9) Time or life

ALTERNATIVE CONCEPTS FOR SEAL TEST APPARATUS:

Various alternative designs were considered for testing seals. A brief description of each concept considered and logic for either choosing or eliminating the alternative are given in the pages to follow.

A. The concept illustrated in Figure 1 would consist of a large diameter cantilevered shaft inside a pressure vessel and sealed at two locations (test seal and outer seal). The large pressure reaction on the shaft would be counteracted by a large thrust bearing outside the vessel and the shaft would also be supported by two external radial bearings. Radial eccentricity could be machined into a sleeve on the shaft, while axial displacement would be obtained by a high force dynamic actuator (with difficulty and great expense).

Advantages:

- (1) The test fixture would be simple, particularly if axial displacement was not built in;
- (2) Only two seals are involved;
- (3) Bearings are outside vessel and could be made large to react the large pressure reaction force;
- (4) Relatively easy to assemble and disassemble.

Disadvantages:

- (1) Since the outer seal and test seal are the same size and have the same pressure times velocity relation and the outer seal will normally see a greater differential pressure, the outer seal would be expected to fail sooner than the test seal and disrupt the test;
- (2) Axial displacement would be very difficult to obtain;
- (3) The cantilever design could allow some inherent instability in the rotating shaft;
- (4) The outer seal life would also be reduced since it experiences axial displacement along with the test seal.

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FIGURE 1

TOLERANCES ON DIMENSIONS UNLESS OTHERWISE SPECIFIED				CONTRACT NO.		TERRA TEK INC. SALT LAKE CITY, UTAH
				DRAWN <i>Tom Sutter</i>		
BASIC DIM.	DECIMAL	2 PLC	3 PLC	FRACTION	CHECKED	
UNDER 6"	±.02	±.010	± 1/32	MECH.		
6" TO 24"	±.03	±.020	± 1/16	ELECT.		
OVER 24"	±.06	±.030	± 1/8	PROJ. ENG.		
ANGLES ±1/2°						
MATERIAL				APPROVED BY		DATE
FINISH				APPROVED BY (OTHERS)		DATE
				11		
				DWG A		REV
				SCALE NTS		SHEET OF

B. Figure 2 illustrates basically the same concept described in (A), but with a smaller diameter outer shaft.

Advantages:

- (1) Decrease the pressure times velocity relation from Concept A, and thus increase the outer seal life;
- (2) Reduce the magnitude of the pressure reaction.

Disadvantages:

- (1) Increase the physical difficulty of reacting the pressure reaction force by a thrust bearing outside the vessel;
- (2) The cantilever problem is increased since the large diameter shaft is cantilevered by a small diameter shaft;
- (3) The difficulty to assemble and disassemble is increased.

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FIGURE 2

TOLERANCES ON DIMENSIONS
UNLESS OTHERWISE SPECIFIED

BASIC DIM.	DECIMAL 2 PLC	3 PLC	FRAC- TION
UNDER 6"	±.02	±.010	± 1/32
6" TO 24"	±.03	±.020	± 1/16
OVER 24"	±.06	±.030	± 1/8
ANGLES ±1/2°			

MATERIAL

FINISH

CONTRACT NO.

DRAWN

CHECKED

MECH.

ELECT.

PROJ. ENG.

APPROVED BY

DATE

APPROVED BY (OTHERS) DATE

13

TERRA TEK INC.

SALT LAKE CITY, UTAH

SEAL TESTER
CONCEPT

DWG
A
SIZE

SCALE NTS

SHEET

OF 1

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C. The concept illustrated in Figure 3 is similar to (B) except for the addition of a radial bearing inside the vessel.

Advantage:

Eliminate the instability resulting from the cantilever design.

Disadvantage:

This will place a bearing inside the high pressure zone where drilling mud and abrasives are introduced when testing a mud seal. When lubricant is used, there will be no problem.

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FIGURE 3

TOLERANCES ON DIMENSIONS UNLESS OTHERWISE SPECIFIED				CONTRACT NO. <i>10m Suth</i>		TERRA TEK INC. SALT LAKE CITY, UTAH	
BASIC DIM.	DECIMAL	2 PLC	3 PLC	FRAC- TION	CHECKED		
UNDER 6"	±.02	±.010	± 1/32	MECH.			
6" TO 24"	±.03	±.020	± 1/16	ELECT.			
OVER 24"	±.06	±.030	± 1/8	PROJ. ENG.	<i>93</i>		
ANGLES ±1/2°							
MATERIAL				APPROVED BY		DATE	DWG A SIZE SCALE MTS
FINISH				APPROVED BY (OTHERS)		DATE	
						15	REV SHEET / OF 1

D. This concept (Figure 4) is a pressure balanced test fixture and includes a symmetrical stepped shaft with two test seals on the large diameter and two outer seals on the small diameter. High pressure is introduced between the two test seals and equal lower pressure between the outer and test seals. The shaft is stabilized by radial bearings outside the vessel, radial eccentricity can be built into the shaft sleeve while axial displacement could be introduced with a small dynamic actuator or an eccentric cam and spring arrangement outside the vessel.

Advantages:

- (1) With a pressure balanced system, there is no large pressure reaction unless a seal fails;
- (2) High pressure mud and abrasives can be introduced between the test seals with no contact with bearings;
- (3) Either two identical seals or two different seals could be tested under the same or different conditions;
- (4) Two shafts extending out of the vessel gives greater access to the instrumentation lead thru's and cooling ports.

Disadvantages:

- (1) The radial bearings outside the vessel also make it difficult to provide stability at the enlarged shaft inside the vessel making dynamic balancing a necessity;
- (2) When a seal fails, the shaft would be displaced into the vessel closures due to the pressure reaction force and could potentially result in damage to the test fixture;
- (3) With two outer seals, the probability of experiencing a premature failure is increased;
- (4) The outer seals will see axial displacement and thus reduce the seal life.

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FIGURE 4

TOLERANCES ON DIMENSIONS UNLESS OTHERWISE SPECIFIED				CONTRACT NO.	
BASIC DIM.	DECIMAL 2 PLC	DECIMAL 3 PLC	FRACTION	CHECKED	DRAWN
UNDER 6"	±.02	±.010	± 1/32	MECH.	Tom Sutton
6" TO 24"	±.03	±.020	± 1/16	ELECT.	
OVER 24"	±.06	±.030	± 1/8	PROJ. ENG.	AS
ANGLES ±1/2°					
MATERIAL				APPROVED BY	DATE
FINISH				APPROVED BY (OTHERS)	DATE 17

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SALT LAKE CITY, UTAH

SEAL TESTER
CONCEPT

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E. The concept illustrated in Figure 5 is basically the same as (D) except tapered radial/thrust bearings have been moved inside the vessel. Radial eccentricity can be machined into the shaft sleeve and axial displacement is created by displacing in the shaft sleeve with a cam and cam follower mechanism.

Advantages:

- (1) Radial bearings inside the vessel will add dynamic stability to the shaft;
- (2) When leakage occurs, and the shaft becomes pressure imbalanced, the tapered bearings will temporarily take up the thrust until the rotation is stopped instead of causing damage to the shaft or vessel closure;
- (3) The shaft can be assembled and disassembled more easily;
- (4) The outer seals will not see the axial eccentricity and their life will be enhanced.

Disadvantages:

- (1) The concept of imparting axial displacement to the shaft sleeve is complicated and poses several problems;
  - (a) Since the sleeve must be keyed to the shaft, the points of sliding contact between the sleeve and key will create a frictional force that must be overcome by the cam mechanism;
  - (b) High frictional forces between the test seals and sleeve will exist;
  - (c) Frictional forces will also exist at the inside diameter of the sleeve and seals on the shaft.

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LTR	DESCRIPTION	DATE	APPROVED

FIGURE 5

TOLERANCES ON DIMENSIONS UNLESS OTHERWISE SPECIFIED				CONTRACT NO.		TERRA TEK INC. SALT LAKE CITY, UTAH
BASIC DIM.	DECIMAL	2 PLC	3 PLC	FRACTION	CHECKED	
UNDER 6"	±.02	±.010	± 1/32	MECH.		
6" TO 24"	±.03	±.020	± 1/16	ELECT.		
OVER 24"	±.06	±.030	± 1/8	PROJ. ENG.		
ANGLES ±1/2°						
MATERIAL				APPROVED BY		DATE
FINISH				APPROVED BY (OTHERS)		DATE 19
DWG A SIZE						REV
SCALE NTS						SHEET 1 OF 1

DESIGN OF THE SEAL TEST SYSTEM:

The test stand (Figure 6) will have the following main components:

- (1) Pressure/temperature vessel
- (2) Vessel closures
- (3) Rotating shaft and bearings
- (4) Outer seals
- (5) Test seals
- (6) Heating and cooling system
- (7) Pressurization system
- (8) Drive system
- (9) Instrumentation and control

(1) Pressure/temperature vessel:

- (a) Approximate vessel dimensions = 7 - 7 1/2" ID, 10 - 11" OD, 24" long;
- (b) Straight bore vessel with threaded ends and pressure, cooling and thermocouple ports;
- (c) Working pressure 5,000 psi maximum, temperature 600°F maximum;
- (d) Vertically mounted.

(2) Vessel Closures:

- (a) Identical upper and lower threaded closures;
- (b) Contain the static seal on vessel ID and the housing for the outer seals and outer seal cooling ports;
- (c) House the cone of the tapered bearing;
- (d) House the eccentric cam plates and cam return spring;
- (e) Keys to react the torque between the test seal housings and the vessel closure;
- (f) Wrench slots for easy removal - the top closure is removed and the shaft removed from the top;

(g) Water swivel with pressure and return ports located in both closures;

(3) Rotating Shaft and Bearings:

(a) 100-800 RPM in four or five steps - could go faster if needed, about 1500-1700 maximum;

(b) Tapered bearing to support shaft radially and axially if test seal fails and unbalanced condition exists;

(c) Shaft OD approximately 4", with cooling ports, thermocouple lead thru's, keyways and static seals. Approximately 5" OD x 4" ID x 12" long sleeves with different eccentricities in their OD can be slipped over the shaft OD;

(d) OD of outer seal shaft approximately 2 1/2";

(e) Cooling water ports are located at each end of the shaft, allowing separate cooling for each test seal. The shaft bottom is threaded for a torque cell and drive spline.

(3) Outer Seals:

(a) Maximum pressure 5,000 psi; however, this seal will normally see lower pressure and cooling on both sides to keep the seal cool. The outer seals will always seal lubricants;

(b) Shaft OD is 2 1/2" - this will give lower pressure velocity factor than test seals - also this seal will see very little eccentricity in both the radial and axial direction.

(c) Outside seals can be similar to high pressure seals used by Maurer on the Exxon high pressure swivel;

(d) Outer seals will be in cartridge that is easily replaced should it fail.

(5) Test Seals:

(a) Seal housings will be 5" ID and approximately 7 - 7 1/2" OD x 6" long;

(b) Different housings could be made up to accommodate seal cross sections from 1/4" - 3/4" - also the fit in the seal housing could be varied to pre-compress the seals;

(c) Both test seals will see the same conditions - high pressure 5,000 psi maximum on inside of vessel (center port) and 0 - 5,000 psi outside (outer ports);

(d) As mentioned, the shaft sleeve can be varied for different degrees of eccentricity, surface finish, different grinding techniques and geometry of finish, different materials and hardness. The surface finish eventually could be varied for each test seal;

(e) A radial lip seal on a floating sleeve could be tested;

(f) Through an eccentric cam and spring, the shaft sleeve can be forced to move back and forth axially with different displacements and frequency equal to a multiple of the RPM;

(g) The sleeve is keyed to the shaft.

(6) Heating and Cooling:

(a) The test vessel will have heating capacity to 600°F;

(b) Cooling will be provided on both sides of each seal to control the temperature of the seals - the outer seals will be run as cool as possible while the test seals can be maintained at desired temperature or varied.

(7) Pressurization System:

(a) The higher pressure will be maintained through the center port - fluid can be either oil, grease, drilling mud, abrasive additives, etc. With drilling mud, the fluid must be circulated - particularly at the high temperatures;

(b) The lower pressure side of the test seals is lubricant since the bearings and outer seals would be affected by abrasives;

(c) The lubricant under pressure will be supplied to the seals from a cylinder/piston chamber - as the lubricant flows in or out of the chamber (representing seal leakage) the piston will be displaced - the displacement is measured and related to the volume leakage;

(d) Using the cylinder/piston arrangement - pressure pulses can be introduced into the system;

(e) In and out ports are provided for filling the chambers with lubricant and circulating if necessary.

(8) Drive System:

(a) Drive will consist of either an AC motor and pulley system to vary the speed or variable speed drive;

(b) RPM - 100-800 in four or five fixed steps with provision to go as high as 1500-1750 RPM;

(c) Rough estimates of horsepower needed is 10 - 20 HP;

(d) Drive shaft will be attached to the seal test shaft by easily removable socket, spline or coupling.

(9) Instrumentation and Control

(a) Need controls for heating and cooling, safety shutdown, drive;

(b) Instrumentation and transducers to measure RPM; torque, horsepower; temperature on either side of test seals T_1 , T_2 , T_3 , T_4 ; inlet and outlet temperature of cooling water, pressures, displacement of lubricant pistons, flow of mud;

(c) Two shaft temperatures, T_1 and T_2 , and torque transducer signal will be brought out through slip rings.

BEARING TEST SYSTEM DESCRIPTION:

Several alternative methods were investigated to test single thrust bearings. After reviewing a number of these preliminary designs, it was determined that the concept of testing a single bearing was not acceptable since the proper downhole conditions could not readily be simulated. Therefore, it was decided to design a test fixture for testing bearing, seal and lubrication package. Preliminary design of the test apparatus is shown in Figure 7. The existing geothermal pressure vessel will be modified to accommodate either full-size seal-lubricant or seal-bearing-lubricant packages. The shaft of the package is splined inside the vessel to the rotary drive shaft at one end and is connected through thrust bearings to the dynamic actuator at the opposite end of the vessel. Since the drive shaft and package shaft is splined, the high pressure will act hydrostatically on the end of the test package and will apply a net force equal to the pressure times the area of the shaft exiting the opposite end of the vessel. Therefore, the bearing package is pre-loaded to about one half the maximum load as the high pressure zone is pressurized and then the dynamic actuator will apply additional load or reduce the load on the preloaded bearing. With this scheme, a smaller, less massive, less expensive and more responsive actuator can be used to load the bearing package dynamically.

Due to the size of the existing geothermal vessel, the bearing and seal package is constrained to the overall dimensions of 8" OD by 48" long. With these dimensions, full size bearings and seals and various schemes of lubricating the bearing can be tested. Also, multiple, stacked bearings can be run. The effects of run-out and cumulative tolerance deviation in the bearing housings can also be accounted for as well as actual deformation of the various components due to the differential pressures and loading of the seal and bearing housings.

APPENDIX

John Jeter, consultant with Texas Dynamatics, Inc., and Jeddy Nixon, subcontractor with Maurer Engineering, reviewed the preliminary design of the Seal Tester and Seal Bearing Tester and made many helpful comments and suggestions. Our correspondence with John Jeter and Jeddy Nixon and the written response of John Jeter follow.

September 16, 1976

John Jeter
201 West Clara
Iowa Park, Texas 76367

Dear Mr. Jeter:

Enclosed are two copies of our preliminary concept and design for a seal tester and some general comments on the proposed features and specifications. Basically, the proposed concept is to test two seals in a test fixture that is pressure balanced, vary the differential pressure across the seals, the radial and axial displacement seen at the seal-shaft face, control temperature and RPM, and various other parameters, such as lubrication, surface finishes, etc. I would appreciate your review and comments on the concept and design. I realize that you will have many questions and perhaps later we can get together in discussing them in detail.

There are, however, several key questions that concern us and we feel need to be answered soon. I would appreciate you addressing these questions as well as any other comments and recommendations you may have.

(1) Is the idea of testing two seals at once acceptable, or will the results be too difficult to interpret? In other words, will the two seals interfere with each other and give results that cannot be interpreted? I have looked at a system for testing one seal - it's a massive system with a large thrust bearing, one test seal and probably two outer seals. When the system is unbalanced, it appears to be extremely difficult to create the axial displacement of the shaft and it creates many other problems, but it is something that we could pursue further, if a two-seal concept does not appear feasible.

(2) Is it necessary to measure torque created by the individual seals, or can we measure the total torque and obtain acceptable results?

(3) Is axial displacement necessary?

(4) Are the range of parameters, such as pressures, temperatures, RPM, axial and radial eccentricity, etc., acceptable?

I have also enclosed an initial design concept for a seal-bearing-lubrication package test fixture. The basic pressure vessel with heating and threaded closures is in existence at this time. Basically, we would design and make the end plugs to adapt the seal bearing package to the vessel and provide the drive system and dynamic actuator. In the system, fluid pressure acts on the bearing seal package and applies approximately 50% of the desired load to the thrust bearing, then the dynamic actuator provides the additional 50% of the load. There is no provision for loading the bearings in the radial direction. Our first priority is to provide the seal tester, but any comments you might have on this seal-bearing-lubrication package test fixture would be helpful.

I am going to proceed with the basic design of the seal test fixture and would appreciate receiving your comments as soon as possible. I look forward to hearing from you.

Sincerely,

Alan D. Black
Director
Drilling Research Laboratory

Enclosures

cc: Sidney J. Green
William C. Maurer
Jeddy Nixon

ADB/mn

Septemer 16,1976

Jeddy Nixon
Maurer Engineering
10301 Northwest Freeway, Suite 202
Houston, Texas 77018

Dear Jeddy:

It was good to meet with you on the 26th of August. I appreciate the input you had on our preliminary design concept for the seal tester for the Downhole Motor Project. I have reviewed this initial design and incorporated many of your suggestions and other ideas I feel will simplify and make a more reliable system. Enclosed are two copies of the assembly and also some general comments I made on the features and specification proposed for the seal tester.

Will you review this concept in detail and make comments? In particular, there are several questions I think still need some thought:

(1) Is the idea of testing two seals at once acceptable, or will the results be too difficult to interpret? In other words, will the two seals interfere with each other and give results that cannot be interpreted? I have looked at a system for testing one seal - it's a massive system with a large thrust bearing, one test seal and probably two outer seals. When the system is unbalanced, it appears to be extremely difficult to creat the axial displacement of the shaft and it creates many other problems, but it is something that we could pursue further, if a two-seal concept does not appear feasible.

(2) Is it necessary to measure torque created by the individual seals, or can we measure the total torque and obtain acceptable results?

(3) Is axial displacement necessary?

(4) Are the range of parameters, such as pressures, temperatures, RPM, axial and radial eccentricity, etc., acceptable?

I will be roughing out the basic design of this new concept and I would appreciate receiving your comments as soon as possible. John Jeter will also be looking at the design concept and specifications and make comments. Again, I appreciate your input and look forward to your response.

Sincerely,

Alan D. Black
Director
Drilling Research Laboratory

Enclosures

cc: Sid Green
William C. Maurer

ADB/mn

PRELIMINARY DESIGN
OF
BALANCED SEAL TEST STAND

MAIN FEATURES OF SEAL TEST STAND

The test stand will have the following main components: (1) pressure/temperature vessel; (2) vessel closures; (3) rotating shaft and bearings; (4) outer seals; (5) test seals; (6) heating and cooling system; (7) pressurization system; (8) drive system; and (9) instrumentation and control.

- (1) Pressure/temperature vessel
  - (a) Approximate vessel dimensions - 7 - 7 1/2" ID, 10 - 11" OD, 24" long
  - (b) Straight bore with threaded ends and pressure, cooling and thermocouple ports
  - (c) Working pressure 5,000 psi maximum, temperature 600°F maximum
  - (d) Vertical mount
- (2) Vessel Closures
  - (a) Identical upper and lower threaded closure
  - (b) Static seal on vessel ID and housing for outer seals with cooling ports
  - (c) House cone of tapered bearing
  - (d) House eccentric cam plates and cam return spring
  - (e) Key to react torque on test seal housings
  - (f) Wrench slots for easy removal - normally the top closure is removed and the shaft removed from top
- (3) Rotating Shaft and Bearings
  - (a) 100 - 800 RPM in steps - could go faster if needed, ~1500 - 1700 maximum

(b) Bearing support shaft radially and axially if test seal fails and unbalances

(c) Shaft OD approximately 4" with cooling ports, thermocouple lead throughs, keyways and static seals; approximately 5" OD x 4" ID x 12" long sleeves with different eccentricity in their OD can be slipped over the shaft OD.

(d) OD of outer seal shaft approximately 2 1/2"

(e) Top of shaft threaded for water swivel and bottom shaft threaded for torque cell and drive spline

(4) Outer Seals

(a) Maximum pressure 5,000 psi, however, this seal will normally see lower pressure - also with cooling on both sides, the seal is kept cool, also the outer seals will always seal lubricants

(b) Shaft OD is 2 1/2" - this will give lower pressure velocity factor than test seals - also this seal will see very little eccentricity in both the radial and axial direction

(c) Outside seals can be similar to high pressure seals used by Maurer on Exxon high pressure swivel

(d) Outer seal will be in cartridge that is easily replaced should it fail

(5) Test Seals

(a) Seal housings will be 5" ID and approximately 7 - 7 1/2" OD x 6 - 8" long

(b) Different housing could be made up to accommodate seal cross sections from 1/4"-3/4" - also the fit in the seal housing could be varied

(c) Both test seals will see the same conditions - high pressure 5,000 psi maximum on inside of vessel (center port) and 0 - 5,000 psi outside (outer ports)

(d) As mentioned, the shaft sleeve can be varied for different degrees of eccentricity, surface finish, different grinding techniques and geometry of finish, different materials and hardness, etc. The surface finish eventually could be varied for each test seal

- (e) A floating sleeve idea could easily be tested
- (f) Through an eccentric cam and spring, the sleeve can be forced to move back and forth axially with different displacements and frequency equal to a multiple of the RPM.
- (g) The sleeve is keyed to the shaft

(6) Heating and Cooling

- (a) The test vessel will have heating capacity to 600°F
- (b) Cooling will be provided on both sides of each seal to control the temperature of the seals - the outer seals will be run as cool as possible while the test seals can be maintained at desired temperature or varied

(7) Pressurization System

- (a) The higher pressure will be maintained through the center port - fluid can be either oil, grease, drilling mud, abrasive additives, etc. With drilling mud, the fluid must be circulated - particularly at the high temperatures
- (b) The lower pressure side of the test seals is lubricant since the bearings and outer seals would be affected by abrasives
- (c) The lubricant under pressure will be supplied to the seals from a cylinder/piston chamber - as the lubricant flows in or out of the chamber (representing seal leakage) the piston will be displaced - the displacement is measured and related to the volume leakage
- (d) Using the cylinder/piston arrangement - pressure pulses can be introduced into the system
- (e) Provide in and out ports for filling with lubricant and circulation if necessary

(8) Drive System

- (a) Drive will consist of either AC motor and pulley system to vary the speed or variable speed drive
- (b) RPM - 100-800 in fixed steps - could go as high as 4500-1750 RPM

- (c) Rough estimates of horsepower needed is 10 - 20 HP
  - (d) Drive shaft will be attached to seal test shaft by easily removable socket, spline or coupling.
- (9) Instrumentation and Control
- (a) Need controls for heating and cooling, safety shutdown, drive
  - (b) Instrumentation and transducers to measure RPM; torque; horsepower; temperature on either side of test seals T_1 , T_2 , T_3 , T_4 ; inlet and outlet temperatures of cooling water, pressures, displacement of lubricant pistons, flow of mud
  - (c) Two shaft temperatures, T_1 and T_2 , and torque transducer signals will be brought out through slip rings

SUMMARY

Proposed range of Variables

- (1) Pressure and ΔP - 0-5,000 psi
  - (2) Temperature - 0-600°F
  - (3) RPM - 0-800 RPM, 4-5 fixed steps or variable - could be greater
  - (4) Radial eccentricity - 0-.010
  - (5) Axial eccentricity - 0-.020
- } could be greater with no problem

Independent variables for seal tester

- (a) Seals - type, cross section, length, backup rings, pre-compression
- (b) Radial eccentricity
- (c) Axial eccentricity
- (d) Pressures and differential pressure across seals
- (e) RPM
- (f) Temperature
- (g) Surface finish, geometry of surface finish, material of shaft sealing surface
- (h) Lubricants

- (i) Mud and abrasives
- (j) Metal particles in lubricant
- (k) Lubricant or mud pulsation

Dependent Variables for Seal Tester

- (a) Torque
- (b) Seal Temperatures T_1 , T_2 , T_3 , T_4
- (c) Cooling water inlet and outlet temperatures
- (d) Lubricants or mud flow
- (e) Seal wear and deformation

Possible test results to look for:

TEXAS DYNAMATICS INC.

5952 ROYAL LANE
SUITE 254
DALLAS, TEXAS 75230
PHONE 363-5493

September 25, 1976

Terra Tek, Inc.
University Research Park
420 Wakara Way
Salt Lake City, Utah 84108

Attention: Mr. Alan D. Black, Director, Drilling Research Lab.

Dear Mr. Black:

Your letter of September 16 has been received and the prints 317240 and 317250 have been reviewed. You suggested haste in the review and I have not gone through the reconstruction of apparent reasons for each design feature. I have concentrated upon your specific questions.

Comments offered in general were invited in your letter, and some are included. These comments are offered now because you have indicated that you are about to cut steel and they might be useless later on. Keep in mind that after a more thorough background study I might cancel out these comments each or all.

It is possible, in an evolving project, that I do not have the latest guidelines on ambients to be expected in the use of the developed product in application. I have, therefore, quantified only in terms of the application ambients. My answers then will survive changes in certain features of the project.

I have tried to envision the activities that I think will surround this project in the long term, whether planned or not, and have evaluated the impact the test machine design will have on such activity. This affects general comments only.

Addressing now your specific questions by number and content from your letter of September 16:

1. The two seals will not interfere and the balanced design should pose no problems. Your temperature probes should distinguish between seals as heat sources. Torque anomalies will likely be less reliable than temperature anyway. Torque background should be higher with the unbalanced system so the masking effect of the second seal, when balanced, should not be a comparative detriment.

2. Total torque measurement should be adequate. Measuring torque created by an individual seal is extremely difficult. Remember, you will likely be testing multi-element seals anyway and they usually fail element-by-element. Seals having frail contact at the sliding surface probably fail due to heat generated by friction plus ambient or by microscopic particulate action (not considering possible pressure related strain destruction). It is quite possible for a failing seal to cause less torque -- not more. It would be useful to have seal torque displayed in the order of .01% of total for real time alertness to anomalies if the source could be limited to the test seal. This I believe would be a difficult project in itself and probably should not even be attempted.

3. Axial displacement is necessary. We may even have to employ high frequency axial displacement before this project bears fruit. The Maurer design accommodates axial displacement in the supports and it is always present to some degree anyway. Tests without axial displacement may well serve a purpose but any favorable outcome must also survive axial displacement in system tests. In application, conventional seal systems see axial displacement in the presence of radial displacement.

4. The pressure limits are all right. The temperature limits must be as high as the expected application ambient. The rpm. capability must be as high as the seal will ever see in service and should be 25% higher. Axial and radial eccentricities, as defined, are probably all right. As you point out, the surface point excursions can be increased and probably should be before a favorable product outcome gets its final certification.

To elaborate, seals of interest seem to fail at all pressures of consequence both differential and hydrostatic.

Seals having small heat flow section area from contact surfaces are very sensitive to temperature. They may fail sometime after fatal

peak speeds. The temperature generated in a few revolutions over certain speeds may be fatal to seals. The seal sees the sum of ambient temperature and friction temperature so both speed and temperature should be capable of quantities somewhat above expected product ambient so that an approved seal will not be operated, say at 99% of a destructive threshold. Since we may expect to operate a successful seal on a turbine expected to drive a diamond drill bit at, say, 800 rpm, we can expect a start-up speed as high as 1600 rpm. (Such turbines often see 2400 rpm.) You should have latest figures on planned application. Positive displacement motors in common use do not have the speed problem. Guidelines on speed may change and I prefer not to quote quantities for design objectives.

The following are general comments related to the test machine design.

The design of the seal tester appears to be almost integrated into the material being tested. It is good practice normally to avoid the waste of the cantilever designs and it is tempting to minimize stress as is evident in the planned machine. In laboratory test gear, however, every effort should be made to provide an independent chamber for the material to be tested. As I go through the mental exercise of putting in test material and removing it I find myself with bits and pieces of machinery all over the place. This often destroys evidence of cause and effect relationships in test material, maximizes down time of test machines and greatly elevates the level of operational expertise required to do a specified amount and quality of laboratory work.

If the experience of others in attempting to develop a workable seal is a guideline, we will spend a lot of time putting in and removing test material. If we give our predecessors credit, we have to assume that, if we miss any cause and effect evidence we won't be successful either. We can assume that the costs of operating this machine is going to vastly overshadow the initial machine cost. If the machine causes operational inefficiency, then we will pay long and hard for that mistake in money and manpower, and because there is just so much of either we may fail. Lab. test machines are highly perfected for a very good reason.

It should be possible and convenient to insert and remove test seals and the mating sliding surface as well as the seal housing as a unit without axial sliding of the mated parts until the unit is on the lab bench

under observation of your arch seal specialist. Your test machine should be back in operation immediately testing a seal cartridge assembled on the bench by a seal specialist.

The ideal test machine should have a standing spindle and a standing housing. An annular space should accommodate test cartridges. The spindle should be removable in the manner of a milling machine arbor. A massive drive assembly should have a hollow shaft like a milling machine and be driven independently of the removable spindle.

It should be possible to clamp the seal sliding cylindrical member to the seal carrier to make up a rigid cartridge. Upon insertion of the cartridge into the annular chamber, the inner member should be locked to the spindle, the seal carrier should be locked to the housing, then the seal inner and outer members should be unlocked for relative rotation. For removal of tested material, the process should be reversible.

Cooling and instrument leads could come from below much as you show it. It should be practical to instrument the seal cartridge and lead to slip rings above.

Ideally, the housing should be adaptable to radial translation during some tests to apply radial loads.

The massive bearing and drive assembly should not be very costly. Large bearings and gears in stocked numbers are reasonable. Over the long period, manpower to assemble and disassemble the planned machine will cost much more.

I believe that the residual value of a true laboratory type test machine at project end is worth consideration.

With an annular test chamber, it will be relatively easy to run a fuse bar from a high frequency vibration test machine into the test region. We may have to go to something like that to succeed where others have failed in this form of undertaking.

You would, by the way, still need the upper and lower seals but not the bearings as shown. All seals could be the same diameter.

In practice, the various "O" rings under pressure might provide all security needed for both rotating and stationary tested members.

These recommendations have certainly not had exhaustive study and any part may not be usable. Certainly give them objective scrutiny, however, because of the great volume of standard laboratory gear that embraces the principles prescribed. I will give this matter some more effort and if gross errors appear present in this paper, you will be promptly notified.

The seal-bearing tester has problems outlined above and the same comments apply.

This report constitutes my input and appears to satisfy the work prescribed in your September 16 letter. If there is any part of the continuing effort that I can productively contribute, please let me know.

Sincerely,

A handwritten signature in cursive script, appearing to read "John Jeter", written in dark ink.

John Jeter

SCHEDULE FOR REMAINING TASKS

ESTIMATED SCHEDULE