ANNUAL REPORT

MASTER

SUPPORT RESEARCH FOR DEVELOPMENT OF IMPROVED GEOTHERMAL DRILL BITS

by

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FOREWORD

This represents the Phase II report for ERDA Contract E(10-1)-1546, aimed at providing background research to support development of improved geothermal roller drill bits.

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INTRODUCTION

Most present day drilling, whether for oil and gas recovery, large mining blastholes, or geothermal energy recovery is done with three-cutter roller bits which have evolved from the first roller-cone bit, patented by Howard Hughes in 1909. These bits are continuing to be improved by industry at the rate of 2-4% per year(1). Unfortunately for geothermal energy recovery, these bits have been optimized for lower temperatures and deliver only about one-fifth their life (designed footage) when used in some geothermal environments. Furthermore, the shortened bit life represents only one part of the increased drilling costs, which run at least fifty percent higher than for oil and gas wells. Penetration rates are generally lower since the bits are run at reduced load in order to prevent catastrophic failure. Additionally, bit replacement requires hours of non-productive "trip time" which ties up the rig and leads to other problems. Both require more time for well completion, and thereby increase well costs.

Drill bit manufacturers have not been able to devote very large efforts to develop the specialized high-temperature bits, since the geothermal bit market is small (relative to the oil, gas and mining bit markets, which are currently experiencing a "boom"). Hence the Energy Research and Development Administration, Division of Geothermal Energy initiated a program to provide the background research needed to develop drill bits for the geothermal environment. Terra Tek is the prime contractor, with Maurer Engineering as a subcontractor, and with support from the drill-bit manufacturers, particularly

Reed Tool Company. The program has, as a primary objective, the immediate "commercialization" of research findings, i.e. the design and manufacture of improved bits.

The results of Phase I of this contract are covered thoroughly in references 2 through 9. The major efforts for Phase I concentrated on obtaining backup information and performing baseline materials properties tests. A full-scale geothermal wellbore simulator and geothermal seal testing machine were designed in Phase I, with construction completed during Phase II.

This report covers Phase II. The major emphasis in the Phase II program, in addition to constructing the above research simulators, includes:

- Simulated tests on full-scale components: i.e. full-scale bits.
- Screening tests on elastomeric seals under geothermal conditions.
- Initial considerations of research needs for development of sealed high-temperature bits.

The "Semi-Annual Report"(10) presents much detailed information, and to make the present report complete, some information is briefly reviewed here. Six topical reports have been prepared and are shown in references 11 to 16, several of which have been published in the open literature.

The following pages provide a detailed discussion of the work done under each of the contract tasks of the Phase II effort.

TASK I. UNSEALED DRILL BITS

First-Generation Experimental Bits

One of the major tasks under Phase I of this contract was the investigation of the causes of geothermal drill bit failure. Examination of worn bits from the Geysers geothermal site revealed annealing of all constituent steels and resultant excessive bearing wear.(3) One such bit was subjected to a detailed metallurgical study by Prof. W. C. Leslie in order to determine peak temperatures.(2) His report noted that the nose bearing or friction pin experienced about 540°C , while the ball and roller raceways experienced about 425°C ; the temperature of the formation drilled by this bit was only 240°C . This large difference between formation temperature and bit temperatures occurs because drilling is done with air instead of mud or water at the Geysers. Although the Geysers is atypical of geothermal sites in this respect, it does represent worst-case drilling conditions, and in addition, accounts for over half of the present geothermal bit market. Hence, the first two experimental drill bits (referred to as MK I bits) were fabricated from steels which retain more hardness at temperature than the conventional steels. Table 1 below gives the measured hardness for these steels at various temperatures.

The steels selected for the experimental bits were of higher alloy content than the conventional steels. Table 2 below lists the nominal composition of all steels considered for the program thus far; the first three entries are the steels currently in wide use for conventional drill bits.

TABLE 1
Hardness Versus Temperature for Drill Bit Steels

BIT PART	BIT TYPE	MATERIAL	HARDNESS (ROCKWELL C)				
			ROOM TEMP.	260°C (500°F)	316°C (600°F)	371°C (700°F)	427°C (800°F)
CONES	CONVENTIONAL	AISI 4820 CARBURIZED	57	49	42		26
	RESEARCH	VASCO MA	59	57		55	53
LUGS	CONVENTIONAL	AISI 8620 CARBURIZED	60	53	47		28
	RESEARCH	H 13	54	49		46	44
BEARINGS	CONVENTIONAL	SOLAR STEEL	58	51	47		28
	RESEARCH	M 50	64	60		58	56

TABLE 2
Nominal Steel Composition

Steel	Composition (%)							
	C	Cr	Mn	Mo	Ni	Si	V	W
AISI #8620	0.20		0.80		0.55			
AISI #4820	0.20		0.60	0.25	3.50	0.27		
AISI S2 (Solar Steel)	0.50		0.40	0.5		1.00		
AISI M-50	0.80	4.00	0.25	4.50	0.10	0.25	1.00	
AISI H-13	0.37	5.25	0.35	1.30		1.00	1.00	
Vasco MA	0.51	4.50		2.75		0.22	1.00	2.00
Vasco X2	0.22	5.20	0.25	1.30	0.06	1.00	0.40	1.35
Vasco X2 Mod.	0.13	5.20	0.25	1.30	0.06	1.00	0.40	1.35

The cost of the more highly alloyed steels runs from two to three times that of the conventional steels due to the cost of the alloying metals, processing costs, and the lower volume of production.

The first geothermal drill bit tests were run in November, 1976 in the newly completed geothermal drilling test vessel, which is shown in Figure 1. The vessel is capable of simulating well temperatures of up to 427°C (800°F) and pressures to 35 MPa (5000 psi).

Figure 1. Geothermal wellbore simulator--the vessel is about 2 m tall.

The first two experimental, or "MK I" bits, were manufactured in three separate sections (Figure 2) so that they could be disassembled for wear measurements at various points in the test program. Two additional take-apart bits were fabricated from conventional materials in order to make direct comparisons of bearing wear between the MK I bits and conventional bits. Table 3 gives the constituent materials for the conventional bits (A and B), MK I experimental bits (C and D) and all other bits in the program.

Figure 2. Research drill bit and disassembled lug and cone from another research bit. Special non-welded design allows bits to be taken apart to measure bearing wear, then reassembled for more testing.

TABLE 3
Drill Bit Materials Summary

Bit #	Description	Size	Materials					Test Details/Test Date
			Lugs	Cones	Bearings	Bushing	Button	
A	Take-Apart M-83 (Conventional)	6-3/4"	8620	4820	S2	430 SS	M2	Not yet run
B	Take-Apart M-83 (Conventional)	6-3/4"	8620	4820	S2	430 SS	M2	316°C, 2 hr @ 20,000#, 2 hr @ 25,000#*/ Nov. 1976
C	MK I Experimental	6-3/4"	H-13 Tool Steel	Vasco MA	M50,R _c 64	none	none	316°C, run 5 min. at 5000#. Cones cracked/ Nov. 1976
D	MK I Experimental	6-3/4"	H-13 Tool Steel	Vasco MA (No inserts)	M50,R _c 64	none	none	316°C, 2 hr @ 20,000#, 2 hr @ 25,000#*/Nov. 1976
E	Conventional M-83	6-3/4"	8620	4820	S2	430 SS	M2	232°C, 2 hr @ 20,000#, 2 hr @ 25,000#/April 1977
F	Conventional M-83	6-3/4"	8620	4820	S2	430 SS	M2	149°C, 2 hr @ 20,000#, 40 min @ 25,000#/April 1977
G	MK II Experimental	6-3/4"	H-13 Tool Steel	Vasco X2CVM	M50,R _c 56	none	none	Delayed by manufacturing problems, probably August 1977
H	MK II Experimental	6-3/4"	H-13 Tool Steel	Vasco X2CVM	M50, R _c 56	none	none	Delayed by manufacturing problems, probably August 1977
I,J	MK III Experimental	8-3/4"	TBD	TBD	TBD	TBD	M2	Laboratory test, early 1978
K-P	MK III Experimental	8-3/4"	TBD	TBD	TBD	TBD	M2	Field tests, Geysers, early 1978
Q-V	Reed Y73 JA (Conventional)	8-3/4"	8620	4820	S2	430 SS	M2	Field tests, Geysers, early 1978

* Progressive test, also included 5 minutes at 5000#, 10 minutes at 10,000#, 20 minutes at 20,000# with take-apart inspection between each step.

The bits were evaluated in the geothermal wellbore simulator using a progressive sequence of five tests of increasing severity, detailed in Table 4.

TABLE 4
Progressive Test Sequence for Take-Apart Bits

Step	Load	Time	Temperature	Hydrostatic Pressure
1	22 KN(5000 lb)	5 minutes	316°C(600°F)	28 MPa(4000 psi)
2	44 KN(10,000 lb)	10 minutes	316°C(600°F)	28 MPa(4000 psi)
3	89 KN(20,000 lb)	20 minutes	316°C(600°F)	28 MPa(4000 psi)
4	89 KN(20,000 lb)	2 hours	316°C(600°F)	28 MPa(4000 psi)
5	111 KN(25,000 lb)	2 hours	316°C(600°F)	28 MPa(4000 psi)

After each test the bit was removed from the simulator, disassembled, cleaned, and measured for wear. The bit was then assembled and returned to the simulator for the running of the next step in the test sequence. This insured the availability of at least some wear data should any bit fail prematurely during the full-load tests (steps 4 and 5). Drill bit failure is detected by monitoring the torque required to turn the bit. This system is used for most tests run at the Drilling Research Laboratory, including those run in the geothermal simulator. The drive shaft of the drill tower is equipped with load cells for thrust and torque, which are monitored on strip-chart recorders. Prior to failure the torque generally rises erratically and the test is terminated. Subsequent examination of

the drill and test pad (or rock) usually reveals evidence of cone lock-up or more obvious failure such as lug breakage.

The first bit tested was the "C" MK I experimental bit. During step #1 the cutter cones developed cracks between almost every adjacent pair of tungsten-carbide inserts. This fracturing was undoubtedly due to the residual stress caused by pressing of the carbide inserts acting in concert with the stresses encountered in the drilling test. This cracking clearly indicated that the material in question (Vasco MA) had insufficient fracture toughness. Testing was halted on bit "C" at this point since it was feared that the cones might fail entirely and damage the wellbore simulator. The possibility of brittle failure of the cones had been foreseen and pressing of the tungsten-carbide inserts for the second experimental bit ("D") had been delayed. The "D" bit was, therefore, run without the inserts to minimize the possibility of cone failure. The entire test sequence was completed without cone-cracking or other damage and with very favorable wear results.

The third bit tested was conventional bit "B" which also completed the entire test sequence without cataclysmic failure and thus provided an ideal reference with which to compare the research bit. Although it could be argued that the conventional bit received a more severe test due to the presence of its tungsten-carbide inserts, the data indicate that this is unlikely. The thrust and torque records for both bits were nearly identical, indicating equal dynamic loading.

Test Results. The wear measurements for the two bits are presented in Table 5. Figure 3 indicates where the measurements for Table 5 were taken and illustrates the direction of taper. It is apparent that the MK I research

TABLE 5
Wear Data for Bits Tested in Geothermal Wellbore Simulator

		Bit B - Lug #1		Bit D - Lug #1		Bit E - Lug #3		Bit F - Lug #3	
		V ¹	H	V	H	V	H	V	H
Ball Bearing	Total Wear	.0094		.0019		.006	.014	-0.18 Max Wear +.017 Oversize .035 Out of Round	
Roller Bearing	Total Wear	.003		.0007		.002			
Lug Ball Race	Total Wear	.035	.012	.031	.024	.055	.024	.051 ³	.040
Lug Roller Race	In	.045	.001	.008	.000	.054	.007	.012	.000
	Out	.060	.003	.010	.000	.062	.000	.015	.000
	Taper ²	.015	.002	.002	.000	.008	-.007	.003	.000
Lug Friction Pin	In	+.0115	+.0025 ⁴	.0042	.0032	.013 ⁴	.000	.002	.003
	Out	-.0135	+.0035	.0052	.0032	.015	.006	.003	.000
	Taper	.0250	+.001	.0010	.0000	.002	.006	.001	+.003
Cone Ball Race	Total Wear	.0137		NA	NA	NA	NA	.005	.004
Cone Roller Race	In	.0345	.0325	.003	.003	.0275	.0285	.0005	.0005
	Out	.0195	.0180	.002	.002	.0135	.0135	.0007	.0005
	Taper	.015	.0145	.001	.001	.0140	.0150	.0002	.0000
Cone Friction Pin	In	.0034		.001	.001	.019 ⁵	.019	.002	.002
	Out	.0034		.001	.001	.002	.002	.001	.002
	Taper	.000		.000	.000	.017	.017	.001	.000

1. V = vertical dimension, H = horizontal dimension, cone dimensions taken at 0° and 90°.
2. Taper wear is most prominent in the roller raceways of the lug and cone. See Figure 3 for direction of positive taper.
3. The lip between the ball race and the roller race has pieces broken off, the broken pieces went into the raceways causing considerable damage to the ball bearings, rollerbearings and raceways.
4. The friction pin galled causing considerable metal migration on B and E lugs.
5. The bushing in cone 3E is not pressed in far enough (lacks .030 in.) causing excessive wear on the outer edge of the friction pin.

Figure 3. Measurement locations for wear data shown in Table 5. Notes correspond to notes in Table 5.

bit "D" experienced one-fifth to one-tenth as much wear as the conventional bit "B". The high wear observed on the conventional bit is similar to that found on failed bits taken from the Geysers(3) and would tend to support the contention that bearing wear has been the principal cause of drill bit failure in geothermal environments.

The cracking of the cutter cones on bit "C" was not totally unexpected since these cones were heat treated to a through-hardened condition. Conventional cones and lugs generally have a core hardness of about R_c 17-20 with the hardness rising gradually to R_c 25-27 at the surface. This is accomplished in the heat-treat and oil quenching of the parts. In addition, the bearing raceways are carburized to provide the necessary hardness. The soft non-carburized areas of conventional lugs and cones are, therefore, extremely tough and very resistant to cracking. Unfortunately, the Vasco MA used for the MK I cones is quenched in air and therefore has essentially

the same hardness ($R_c 59$) throughout. This high hardness was required since Vasco MA cannot be depth-carburized in the bearing areas (although stellite was applied to the friction-pin area.) Vasco MA does, however, exhibit excellent retention of hardness at elevated temperatures (see Table 1). It was hoped that the Vasco MA cones would be tough enough to complete the first round of tests. It should be noted that at the time that the steels were selected for the MK I bits very little information was available on the fracture toughness of any of the steels in question.

The major conclusions drawn from the MK I tests were that retention of bearing raceway hardness was indeed necessary for operation at elevated temperature and that sufficient fracture toughness is also essential. The steels for the MK II bits should, therefore, be carburizing types so that lower core hardness (i.e., higher core toughness) can be maintained while retaining adequate hardness in the bearing area.

Table 5 also gives wear data for two tests made on two conventional Reed M83 #JA 6-3/4" (171 mm) bits (designated E and F) at temperatures of 149°C (300°F) and 232°C (450°F). These tests were run in order to provide wear data at intermediate geothermal temperatures. Since bits "E" and "F" were of standard welded construction they could not be taken apart for periodic wear measurements.* The low-load test steps, 1-3 (see Table 4) were, therefore, deleted and only steps 4 and 5 were used. Bit "E" was run at 232°C (450°F) at 89 KN (20,000 pounds) load for two hours and 111 KN (25,000 pounds) load for two hours without incident. The data shown in Table 5 indicates wear which is roughly comparable to that of bit "B", the

* The cones will not slide clear of each other; disassembly is accomplished by cutting off the lugs with a torch.

conventional bit which was run at 316°C (600°F). In some cases the wear on bit "E" was worse than bit "B". Figure 4 shows the severe galling encountered on lug #3 of bit "E". Similar galling also occurred on bit "B".

Figure 4. Friction pin area of lug #3, bit "E".

Bit "F" was run at 149°C (300°F) for two hours at 89 KN but completed only forty minutes at 111 KN. The test was terminated at that time due to erratic torque. Upon removal from the simulator, two of the cones were found to be locked up. Tear-down of the bit revealed the cause of the locked cones to be breakage of the outside ball-race flange, shown in Figure 5. The pieces of broken flange embedded in the roller raceways and ball raceways of the two bits caused lock-up of the cones. Examination of the wear data indicates that much less wear was encountered on bit "F", even considering the reduced run time. This is an important observation; in many geothermal drilling operations the temperature can be held below 149°C

Figure 5. Broken flange on lug #3 of bit "F".

(300°F) using the cooling provided by the drilling mud or water. In these cases the extra cost of a geothermal drill bit may not be required.

The flange breakage is one of the more common failure modes found in roller-cone bits used for oil, gas, mining exploration and blast-hole service. However, the failure of bit "F" points up a problem frequently encountered when testing a drill bit against a steel pad. Excessive thrust is placed on the roller bearings because the cone receives little or no support from the outer diameter of the "hole". Thus, the roller bearing and friction pin must carry the entire vector force generated by the thrust load on the bit acting through the cone angle of 36°. Two things can be done to reduce this problem. The most immediate solution is to run the bit at slightly reduced loads. The second method is to machine tapered surfaces into the drill pad which roughly coincide with the profile of the bit. This provides even support for the bit and apparently prevents the load from concentrating in any one portion of the ball race. This can also be

achieved by running the bit against the pad at reduced loads, thereby generating a pattern which will support the cones. Since bit "F" was run long enough to have established a good pattern prior to its flange failures, it must be concluded that future tests should be run at reduced load, probably 89 KN (20,000 pounds) for the 6-3/4" (171 mm) bits. The pad break-in procedure is probably beneficial and should be continued.

Fracture Toughness of Drill Bit Steels

High fracture toughness is essential for materials used in drill bit steels due to the high impact loads which occur during drilling. Fracture toughness data is not generally available from the steel manufacturers and must be obtained through experimental measurements. The geothermal bit program required that fracture toughness measurements also be made at elevated temperatures for two reasons. The first is simply to insure that the steel retains sufficient toughness at temperature to withstand the dynamic loading, which was assumed to be roughly equal to the non-geothermal situation. The second reason is less obvious; fracture toughness and hardness are properties which are interrelated and must be "traded off" when selecting alloys and heat treatment. Since the geothermal bit requires a high degree of both hardness and fracture toughness, it becomes advantageous to be able to measure and control the toughness so that any additional capability of the alloy can be utilized to improve the hot hardness. For example, had this technique been available at an earlier date, some hardness could have been sacrificed on the cutter cones of the MK I bits, thereby increasing their toughness, and the cracking which occurred on bit "C" might have been avoided.

The measurement technique was perfected early in the Phase II year and was used to develop fracture toughness data on various steels under consideration or already in use. The method is based on the short rod technique(12-14) developed recently at Terra Tek. It is doubtful whether the standard ASTM tests could have measured the toughness of the bearing steels because of their brittleness. The short rod technique is applicable to both brittle and ductile materials, and the specimen configuration (Figure 6) was particularly well adapted to the tests of the two drill-bit-bearing steels inasmuch as the specimens could be made from actual drill-bit roller bearings.

Figure 6. Short rod specimen configuration.
The shaded area denotes the crack.

Figures 7a and 7b show the equipment used to make the fracture toughness measurements at elevated temperatures. Shown are the load frame, sample under test, failed sample, sample "oven", clip gauge extensometer, and the grips used to pull the sample apart. A detailed derivation of the short-rod technique is given in references 12 and 13.

Figure 7a. Apparatus for measuring fracture toughness at elevated temperatures.

Figure 7b. Detail of sample, grips, clip gauge and previously tested sample. Oven is pulled back for clarity.

The fracture toughness test results are shown in Figure 8, in which all of the data represent valid measurements except for the AISI 8620 data points. These are erroneous because the high temperature toughness and lower yield strength of the material dictate a larger sample size than that used in these tests for valid measurements. Room temperature measurements on some larger samples of AISI 8620 indicate a fracture toughness of at least $100 \text{ MPa } \sqrt{\text{m}}$, and the toughness may increase with temperature.

The fracture toughness values of the high temperature research drill bit steels are considerably smaller than the fracture toughnesses of the conventional drill bit steels (Figure 8). Since the toughness values of the research bit steels increase significantly with increasing temperature, higher temperature tests of the research bits are less severe than lower temperature tests as far as fracture is concerned. The fracture toughness measurements of the research bit steels and the Solar Steel were also reported in a paper presented at the Fourth International Conference on Fracture, held in Waterloo, Canada, June 1977.

Figure 8. Fracture toughness of drill bit steels to 427°C. All data points are valid except those of AISI 8620, which are significantly lower than the actual fracture toughness (see text).

Materials Selections for the MK II Drill Bits

Lugs - The H-13 tool-steel, used for the lugs on the MK I bits, exhibited very high fracture toughness at elevated temperatures, although it did fall off rapidly below 200°C. Since the H-13 has excellent hot-hardness characteristics, and since the bits would again be tested at 316°C (600°F), it was decided to use H-13 for the lugs of the MK II bits. Concurrently with the construction of the MK II bits, research was continued to locate a steel with adequate hot-hardness and better fracture-toughness characteristics, for fabrication of the lugs for the MK III bits.

Cones - The Vasco MA steel used for the MK I cones had two major problems, the first of which was its lack of fracture toughness (see Figure 8). The second problem was the steel's hardness: Vasco MA is not a carburizing grade of steel. Thus the entire part was heat-treated to full hardness ($R_c 59$) in order to obtain sufficiently hard bearing races. This required that the carbide insert holes be drilled prior to heat-treat, which is not common practice in the drilling industry due to the increased difficulty of obtaining a good fit for the pressed-in drilling inserts.

The material selected for the MK II lugs was Vasco X2 CVM, which is a carburizing steel that is otherwise somewhat similar to the MA (see Table 2). The X2 CVM should allow carburization of the bearing races to approximately $R_c 58-60$ (room temperature) while retaining a core hardness of about $R_c 40$. This should greatly improve the fracture-toughness of the core, and permit drilling of insert holes after heat treat.

Bearings - The M50 steel used in the rollers and balls of the MK I bits has hot-hardness characteristics that are much better than the S2 steel used in conventional bits (see Table 6, below). The fracture toughness data, however, revealed that the M50 steel in the fully heat-treated condition had fracture-toughness values less than half that of

TABLE 6

Properties of Various Steels Used in Conventional and Research Bits

Bit Part	Bit Type	Material	Yield Strength/Elongation, KSI/%, and Hardness, R _C				
			Room Temperature	260°C (500°F)	316°C (600°F)	371°C (700°F)	427°C (800°F)
Lugs	Conventional	8620*	84/16% core R _C 60 case	R _C 53 case	R _C 47 case		R _C 28 case
	MK I, MK II	H-13 [†]	238/9% R _C 54	R _C 49		R _C 46	R _C 44
Cones	Conventional	4820*	145/13% core R _C 57 case	R _C 49 case	R _C 42 case		R _C 26 case
	MK I	VASCO MA [†]	292/6% R _C 59	249/6% R _C 57		R _C 55	245/6% R _C 53
	MK II	VASCO X2CVM*	208/13% core R _C 50 core R _C 58 case			R _C 50 core	181/12% R _C 48 core
Bearings	Conventional	S2	312/2% R _C 58	R _C 51	292/7.5% R _C 47		220/10% R _C 28
	MK I	M50	338/1.6% R _C 64	308/4% R _C 60		R _C 58	262/6% R _C 56
	MK II	M50	R _C 56	R _C 52		R _C 50	R _C 48
Friction Pin Area of Lug	All	Stellite #1	R _C 59				R _C 55

* Case-carburized, typically 40 pt. at 1.8 mm (0.07 inch).

† Through-hardened

the S2 steel. The rollers and balls for the MK II bits will therefore be of M50 steel heat-treated to a hardness of R_C56 rather than R_C64 (20°C). Although fracture-toughness tests have not yet been made on this steel, a three-ball crush test has been run, and the crush strength of the M50 at R_C56 was roughly equivalent to results obtained with S2 at 20°C. It is therefore expected that the fracture-toughness tests will yield values similar to S2; the hot-hardness is expected to be about eight points lower on the Rockwell "C" scale at any temperature.

Pilot Bushing - The friction pin area of the cutter cones will be case-carburized; no pilot bushing will be required.

Hard-Facing - The Stellite used on conventional drill bits had excellent hot-hardness properties and will again be used for the friction-pin area on the MK II bits.

Summary

A complete summary of materials for all bits in the program is given in Table 3. A listing of the alloy compositions can be found in Table 2. Table 6, below gives some basic hardness and yield strength data for the steels.

TASK II. ELASTOMERIC SEAL DEVELOPMENT

The Geothermal Environment and Seal Failure Mechanisms

Prior to the design and construction of the geothermal seal test facility, a great deal of consideration was given to the factors affecting seal life. References 2 through 9, conducted under Phase I of this contract, provided an excellent starting point. This information was augmented through numerous conversations with the drill-bit, seal and elastomer manufacturers. It became apparent very early in the program that the seal test facility would have to simulate the geothermal environment as accurately as possible if meaningful results were to be expected. The following is a description of what are felt to be the key factors affecting seal life in a three-cutter drill bit.

1. Temperature. The formation temperature at Geysers, California reaches a maximum of 240°C ; Geysers currently accounts for about half of all geothermal well footage drilled in the U.S. The Roosevelt Hot Springs site in Utah has formation temperatures to 260°C , while the Imperial Valley, California site reaches 370°C . Several other sites in the 150°C - 200°C class dot the Western U.S.(5) It should be noted that geothermal prospecting technology is currently receiving a great deal of attention, and it may soon be possible to locate more sites in the 250°C to 350°C class.

Seals may experience additional heating due to friction in the drill bearings and at the seal interface, especially if air (not water) is used as the drilling fluid. Figure 9 shows a seal which has undergone melting at the seal interface due to excessive frictional heating. Elastomeric

Figure 9. Buna-N seal after 4 hours at 225°C and 90 RPM.

seals suffer from temperature-induced molecular degradation via broken molecular chains, or the formation of unwanted cross-links.(17) The melt condition depicted above is the result of the former. Unwanted cross-links tend to make the material too hard, and can actually result in a brittle failure, such as that shown in Figure 10.

Elevated temperatures can cause elastomers to plastically flow at decreased levels of compression, thus causing a permanent "compression set" in the material. This problem is aggravated by the fact that nearly all elastomers have a coefficient of expansion that is five to ten times greater than that of steel. This causes additional compression upon heating, with resultant increased plastic deformation. Upon cooling, the plastically deformed seal may actually contract away from one of the seal interfaces causing total failure.

Figure 10. Brittle failure of fluorosilicone seal. Tested 4 hours at 260°C, 200

2. Rotational Eccentricities. The cutter-cones do not run perfectly concentric to the lugs on which they turn, due to the drilling action. Each cone is studded with circular rows of teeth or drilling inserts. The rows are staggered such that inserts from each row do not all strike bottom at the same time. Thus, as the load transfers from one insert to another, a moment is generated which tends to make the cone "wobble" on the lug. This motion is on the order of ± 0.005 inch (± 0.13 mm) at the seal for a 7-7/8 inch (200 mm) bit at mid life, and increases markedly with bearing wear. (18)

Cutter wobble can cause seal failure in at least two distinct ways. As the cutter deviates from the axis of the lug, it traps grease between the two internal surfaces, causing pressure spikes which the seal must contain. The magnitude of these spikes depends on RPM, displacement, and grease viscosity. Figure 11 is an oscilloscope photo of some pulsations simulated in the seal test facility; these were obtained from a fluid-coupled transducer located just below the seal under test.

Figure 11. Oscilloscope photo showing pressure pulsation in grease at 22°C, 90 RPM. Radial wobble was ± 0.0075 inch (± 0.19 mm). 1 div = 100 m sec horizontal, 50 psi (0.35 MPa) vertical.

Note that pulsations have very short rise and fall times as compared to their period. This effectively prevents the compensator bellows from equalizing the pulsation before it reaches the seal. These pressure spikes will attempt to pump grease out past the seal and to draw well fluid and cuttings into the bearing area.

A second problem arises when a radial compression seal experiences the radial excursions of the cone. This motion must be successfully followed by the elastomer in order to maintain a good seal. This proves difficult in the instance where the gap to be sealed is widening, since the resiliency of the elastomer must provide the sealing force. A mathematical expression for the radial compression at a stationary point on the seal would be useful for determining how the wobble motion relates to rebound rate.

S = Instantaneous seal thickness;

then

$$S(t) = S_0 + W \sin(9\omega t) \quad (1)$$

S_o = Nominal installed thickness of seal

W = Amplitude of wobble motion, typ. 0.005 inch (0.13 mm)

ω = Cone angular velocity. Note: for most drills, the cutter cone RPM is about 150% of the drilling RPM. For 60 RPM drill speed, cone RPM = 90 (9.4 rad/sec).

For simplicity, it has been assumed that the radial motion due to wobble (or due to the rollers and pins in the seal tester) is sinusoidal. As can be seen from the oscilloscope photograph, Figure 11, the actual motion is more irregular. Nine pins were chosen as being roughly representative of the cutter wobble observed in a 7-7/8 inch (200 mm) bit designed for medium hardness formations.

The radial velocity then is:

$$\frac{ds}{dt} = 9\omega W \cos(9\omega t) \quad (2)$$

and $\frac{ds}{dt}_{\max} = 9\omega W \quad (3)$

This maximum radial velocity shown in Equation 3 cannot be allowed to exceed the seal's rebound rate, as installed, if a good seal is to be maintained. Equation 3 states that, for a given seal and temperature, the "rebound limit" is directly proportional to the product of the drilling RPM and amplitude of cutter wobble. (Cutter wobble can be expected to markedly increase with bearing wear.)

3. Abrasive Particles. Certain geothermal reservoirs, such as the Geysers (California), Raft River (Idaho), and Roosevelt Hot Springs (Utah), contain glassy volcanic materials such as rhyolite which generate highly abrasive cuttings when drilled. These particles can work their way into

the seal area, and become embedded in the elastomer. Penetration is probably assisted by cutter wobble, described above. The particles can then abrade the steel sealing surface, which in turn will wear the elastomer. The particles will also cause mechanical degradation of the elastomer; Figure 12 shows the results of the abrasive particles on a Buna-N seal, run in the laboratory, as compared to a seal tested without the abrasives. The abrasives used were cuttings from the Geysers geothermal site, to which some feldspar sand had been added.

Figure 12. Two Buna-N seals tested at 200°C for 4 hours with ± 0.005 inch (± 0.13 mm) radial motion and ± 0.30 inch (± 0.76 mm) axial motion. Seal on top tested with Geysers abrasives.

4. Starting Temperature. It was inadvertently discovered that preheating the seal prior to initiating a test would greatly mitigate the abrasive damage. In a typical seal test, rotation is started prior to heating the vessel. However, if the seal is warmed to 50°C prior to

initiating rotation, (then heating to the test temperature), damage to seal and shaft will be about 10% of what is seen on an identical test started from 20°C or less. This is most likely due to insufficient resiliency at the cooler temperatures, which allows gaps (caused by the radial "wobble") to open with subsequent penetration by the abrasives. Fortunately, preheating will occur automatically in geothermal drilling situations; this finding is probably more significant to non-geothermal drilling technology. All seal tests are now run with this preheating prior to test.

The chemical makeup of the geothermal brine and steam was investigated for its possible effect on elastomer properties. Although it was beyond the scope of the contract to determine the exact composition for all geothermal wells, Table 7 lists the composition of brine from one of the wells at the Niland, California Geothermal Loop Experimental Facility. Table 8 lists the percentages of various gases dissolved in the brine.

The data below were taken from a report entitled "Scaling Characteristics in the Geothermal Loop Experimental Facility at Niland, California" by Roland Quong of Lawrence Livermore Laboratories(19). The constituent in the brine or steam presenting the greatest threat to elastomers is the hydrogen sulfide gas. H_2S can cause embrittlement in elastomers by creating unwanted cross-links(17). Other data on geothermal hydrogen sulfide concentrations includes the figures 60 ppm-1600 ppm by Li and Alzheimer(20). and 100-225 ppm for wells currently being drilled at the Geysers(21). Although H_2S concentrations on the order of 100 to 1000 ppm are large enough to be a threat to some elastomers, there are two mitigating factors to be considered. While drilling, the bit area is constantly being purged by the drilling fluid; significant concentrations occur only

TABLE 7

Analysis of Brine at Niland Test Loop.
Output from Second-Stage Separator, Sample V7-B0-L28, after Quong.

Date taken:	6/16/76
Time:	1535
Method:	Condenser coil
pH:	5.9
Density:	1.18 g/cm ³ at 25°C
Suspended solids:	657 mg/kg
Dissolved solids:	252 g/kg
Noncondensable gas:	1-1.5% by mass

Element	Suspended Solids, wt%	Dissolved Solids, mg/liter	Reconstructed ^a Brine, mg/kg
Si	26.5	24	194
Fe	8.6	273	288
Cu	0.05	2	2.0
Ba	5.6	133	149
Cl		168,000	142,000
Mn	0.04	950	805
Pb	0.08	87	75

^a Suspended plus dissolved solids.

TABLE 8

Gas Composition, Niland Test Loop, after Quong.

Gas	Percent
CO ₂	98
CH ₄	1-1.5
H ₂	0.35%
NH ₃	100-300 ppm
H ₂ S	1500-2500 ppm

in the brief intervals when a steam or brine entry point is being penetrated, or when tripping. Since H_2S causes embrittlement problems in steel components such as the bit, drill pipe, drill collars and casing, efforts are usually made to control the H_2S with additives. In the case of air-drilling at the Geysers, a product known as "Unisteam" is injected into the drilling air. Unisteam contains coating amines which provide some shielding protection and possibly some neutralization of the H_2S , although the primary function of Unisteam is to coat the cuttings to reduce abrasion of the drill pipe. A corrosion (rust) inhibitor is also present.

Since the seal's actual exposure to H_2S in geothermal drilling appears to be almost negligible, and because H_2S presents problems of safety and equipment damage in the laboratory, H_2S is not used in the seal test environment. Data on chemical compatibility is generally available from the seal or elastomer manufacturers.

The Seal Test Facility

Figure 13 is a photograph of the seal testing facility developed to simulate the environment described previously. The facility consists of a pressure vessel, drive motor, pressurizing pump, control console, crane and exhaust stack. The tester utilizes several unique instrumentation subsystems which will be described in detail. Table 9 lists the controlled parameters of the tester, including the range and mode of adjustment for each.

The seal test is performed within a thick-walled pressure vessel. Figure 14 is a sectioned view of the vessel showing the heater box, insulation, test mechanism (in cutaway), and drive shaft. Above the drive

Figure 13. Seal test facility.

TABLE 9

Seal Tester Capabilities

Parameter	Range	Type of Adjustment
Temperature	20°C-425°C	External
RPM	50-2000	External
Pressure	0-3000 psi	External
Well Fluid	Water, Steam, Mud	Prior to test
Radial Motion	$\pm .001"$ to $\pm .017"$	Prior to test
Axial Motion	0" to $\pm 0.050"$	Change cam
Abrasives	any	Prior to test

Figure 14. Sectioned view of test vessel.

pully is a cam used to provide the axial stroke, and the solid rubber spring used to balance the hydraulic load placed on the shaft by the vessel pressure. Disassembly is accomplished by unscrewing the tapered top closure and hoisting the entire experiment out of the vessel. Figure 15 is a photo of the top plug assembly in the raised position; 16 shows the view into the vessel.

Figure 15. Seal tester top plug assembly.

Figure 16. Seal tester high pressure vessel.

Figure 17 is a sectioned view of the seal test mechanism; this hardware is also visible in the sectioned view of the test vessel, Figure 14. The seal under test is placed in a metal carrier and mounted over a stationary housing which also provides a grease reservoir; this reservoir is proportioned to simulate the inside of a cutter-cone.

The drive-shaft turns the bit-lug simulator; this part contains a stainless steel compensator bellows which equalizes the pressure in the grease reservoir. The surface contacting the seal under test is removable, thus allowing low-cost replacement should it become worn, or if a different size is required.

Figure 17. Detail of seal test hardware.

The "cutter wobble" is simulated as follows: The bit-lug simulator is hinged at the point where it attaches to the drive shaft. On the bottom of the bit-lug simulator are two interference rollers whose separation can be adjusted prior to a test. Located in the grease cup, or cutter-cone simulator, are nine hardened steel pins. The rotating rollers encounter these stationary pins and impart a rocking motion to the rotating bit-lug simulator. These parts are shown schematically in Figure 17 and in a photograph, Figure 18, below. The resultant motion seen at the seal is highly similar to the wobble in an actual drill bit caused by the inserts striking the formation. A fluid-coupled pressure transducer is situated directly below the seal to monitor the pressure pulsations generated.

Figure 18. Bit-lug simulator and pin-cup assembly.

An external cam imparts a sinusoidal stroking motion to the shaft to further simulate an actual cutter cone. The effects of this stroking (about ± 0.015 " or ± 0.4 mm) is felt to be much less harmful to the seal than the "wobble" action, since the stroke can be accommodated by sliding. The "well fluid" is water and/or steam depending on the temperature and pressure selected for the test. The well fluid is also present in the abrasives reservoir, directly over the seal. The instrumentation includes readouts for three thermocouples, RPM, and vessel pressure. The output of the fluid-coupled transducer can be observed on an oscilloscope and photographed.

Pressure Pulsation Monitoring Transducer System - As previously described, large transient pressure differentials occur across the seal due to the wobbling motion of the cutter cone. The amplitude of these pulses is a function of bearing wear, grease viscosity, grease temperature, and the rotational speed of the cone. The exact magnitude of the pressures involved has been the subject of a great deal of speculation, because instrumentation of a drill bit for such a measurement under actual drilling conditions is extremely difficult. Simulation of the wobble motion in the seal tester, however, was accomplished, and similar pressure pulsations should occur in the test set-up.

Monitoring of these pulsations requires that a pressure transducer be located as close to the sealing interface as possible. The transducer should also be small in size, since the pulsations vary in amplitude depending on the radial location relative to the instantaneous wobble motion taking place. The transducer must also be able to withstand geothermal

temperatures, up to 427°C (800°F), if possible. No such transducer is commercially available, hence one had to be built. The design selected is pictured in Figure 19.

Figure 19. Fluid-coupled transducer system.

The system utilizes a fluid coupling media to isolate the pressure transducer from the high temperatures. The ideal coupling fluid would be incompressible, have a very high boiling point, a low coefficient of thermal expansion, low density, and be safe to handle. The ideal transducer

would be very stiff, i.e. require a very small change in volume for large changes in pressure. Most of these properties are needed in order to insure a reasonably high frequency response, or "cutoff frequency". This limiting frequency is determined by the frequency at which the mass of the fluid in the column resonates against the spring-action of the pressure transducer diaphragm. A complete derivation of the design is given in Appendix A.

The system tended to oscillate at the resonant frequency, and attempts to dampen the oscillations with a series regulating valve proved unsatisfactory. A fourth-order Butterworth low-pass filter was used to electronically filter the ringing from the signal. The system had a usable frequency limit of 350 Hz, which is somewhat less than the goal of 500 Hz, but proved to be adequate. The fundamental frequency of the jogging motion should be as follows:

$$\frac{90 \text{ RPM} \times 9 \text{ pins} \times 2 \text{ rollers}}{60} = 27 \text{ Hz}$$

Hence the transducer system response is better than an order of magnitude above the fundamental frequency. The primary advantage of this system is that the calibration of the transducer is unaffected by vessel temperature.

An alternate design approach would be the construction of a metal diaphragm transducer located near the seal, utilizing welded metal strain gages. It would, however, be difficult to attain a small size, insulate the wire leads, feed the signal through the pressure vessel, and calibrate. Piezoelectric transducers are not feasible due to the high degree of electrical insulation required.

Seal Failure Detection - The most difficult engineering aspect of the seal tester proved to be the seal failure detection system. It was originally hoped that the fluid-coupled transducer system, previously described, would provide some positive indication of seal failure. This method proved unreliable: a slight seal leak caused no observable change in the pulses, and pulse amplitude varied greatly with lubricants and test temperatures. A second system, visible in Figure 17, utilized a fiber-optic scope to observe changes in the opaqueness of the lubricant. A great deal of difficulty was encountered in sealing the fiber bundle against the 3000 psi vessel pressure. Additional problems were encountered with breakage of the glass fibers, and sealing the bundle into the grease cup. We were able to locate only one manufacturer who was willing to attempt to make such a system; most fiber-optics products are now made with plastic fibers, whereas this application required glass. Since there was some question as to whether an observable change in opaqueness would indeed occur with all seal failures, further attempts to perfect the fiber-optics system were abandoned.

Some information on seal performance can be obtained from the seal thermocouple. Figure 20 gives a cross-sectional view of the placement of the thermocouple on the seal. The seal thermocouple generally reads from 2°C to 12°C hotter than the surrounding area due to friction at the sealing interface. Some sealing situations, such as the use of too much installed compression, will give an observable failure indication: the thermocouple temperature will record an abnormally high reading followed by an abrupt drop to ambient temperature. However, for most seal tests the thermocouple

Figure 20. Seal thermocouple placement.

signal did not exhibit a marked change at seal failure. The seal thermocouple may prove more useful if tests are to be run at higher RPM's; at the test conditions of 90 RPM in a water environment, frictional heating was probably not a significant factor in seal life. However, an ideal seal would allow the drill bit to operate at higher speeds, and higher RPM's may bear investigation if a promising candidate is located. In all seal tests run to date, the seal thermocouple is monitored for anomalous behavior, but is not considered a failure indicator.

Since a failure detection system was not yet at hand, and the seal tester was operating reliably, a series of fixed-duration tests were run. Approximately 115 seal tests were run using a fixed time interval of four hours.

During the months of fixed-interval testing, efforts continued on the seal failure detection problem. Two ideas given serious consideration involved detection of contamination in the lubricant or in the "wellbore" fluid. The first concept involved the placing of radioactive tracers in the lubricant, and drawing samples from the reservoir area directly above the seal for scintillation counting. The second method would have utilized the Karl Fisher titration method for detecting water in samples drawn from the grease. Both of these methods would have required extra care to avoid prior contamination, and both were indirect methods of failure detection.

The concept finally selected for development is pictures in Figure 21.

Figure 21. Differential pressure system for seal failure detection.

The system involves direct observation of the pressure across the seal via two electronic pressure transducers. The entire detection system, including the lubricant reservoir is filled with an S.A.E. 70-weight nondetergent motor oil. The stroking motion of the shaft and lug simulator causes pressure fluctuations in both the oil and in the vessel (water), via the crossover valve and bellows.* Due to the oil's viscosity, a transient differential pressure of about 40 psi is generated across the seal as the shaft strokes in or out. After the seal fails, these transients are no longer present; a strip-chart recorder provides a continuous record of the events. These fluctuations should not be confused with the pulsations generated by the wobble motion, which are of much higher frequency, and are not used for failure detection. A very accurate (± 2 psi) measurement of the pressure across the seal can be attained by making a bucking measurement across the outputs of the two transducers.

In its present form, the differential pressure system described above cannot use grease since unrealistically high pressure differentials would be applied to the seal. This is due to the positive-displacement caused by the stroke motion, the high viscosity of grease, and the long segments of tubing between the test chamber and the compensator bellows. Actual drill bits utilize grease, but the distance from the bearings to the compensator bellows is much shorter. Plans call for the addition of a grease/oil interface bellows directly below the bottom of the jogger pin cup. This will allow the seal to test in grease with reasonable differential pressure pulses (less than ± 50 psid), while the transducer system operates with low-viscosity oil.

* The original bellows, located in the lug simulator (see Figure 17), was removed and the passageway plugged off.

This system has just recently completed "debugging" and provides a distinct indication of seal failure. This development came too late in the year to include any "test-to-failure" results, although all future seal tests will probably be of this type.

Homogeneous Elastomeric Seal Development

Given the high reservoir temperatures of the more desirable geothermal formations (200°C - 370°C), and the low service temperatures of most elastomer seals (less than 200°C), one might reasonably question whether such seals have any role at all in geothermal drilling. There are, however, several factors which need to be considered. A considerable amount of cooling is provided by the drilling fluid when drilling with mud or water; this cooling effect in many cases will allow the use of a sealed bit in formation temperatures well over the service limit of the elastomeric seal and the lubricant. Unfortunately, this bottomhole temperature is very difficult and time-consuming to measure, and is also quite difficult to calculate due to the large number of variables involved. In order to learn more about the actual bottomhole temperatures that a geothermal bit experiences, Bill McDonald of Maurer Engineering conducted a study utilizing Exxon's computer program "CIRCUS"(4). Figure 22 below shows the calculated change in bottomhole temperature with time as mud circulation is started and stopped in a 5000-foot well.

Table 10 lists the test variables and resultant circulating temperatures for ten sets of parameters.

The simulations show that once circulation has been established, the bottomhole temperature usually drops to an acceptable value for the operation of elastomeric seals. Unfortunately, there is no established means

Figure 22. Calculated transient wellbore temperatures.

TABLE 10

Results of Ten Hypothetical Geothermal Well Simulations

GEOTHERMAL BOREHOLE TEMPERATURE CALCULATIONS

EXXON 'CIRCUS' PROGRAM

CASE	DEPTH (FT)	STATIC BOTTOM HOLE TEMPERATURE (°F)	CIRCULATION RATE (GPM)	GEOTHERMAL GRADIENT (°F/100 FT)	CIRCULATING BOTTOM HOLE TEMPERATURE (°F)	INLET TEMP. (°F)	OUTLET TEMP. (°F)
1	5,000	500	200	8.4	154	135	157
2	5,000	500	400	8.4	154	135	151
3	10,000	500	200	4.2	258	135	178
4	10,000	500	400	4.2	255	135	167
5	5,000	750	200	13.4	169	135	176
6	5,000	750	400	13.4	169	135	164
7	10,000	750	200	6.7	348	135	214
8	10,000	750	400	6.7	344	135	195
9	5,000	500	400	8.4	144	135	163
10	10,000	500	400	4.2	213	135	151

RUN CONDITIONS:

HOLE DIA. (IN.) : 8.625 (CASES 1-8)
 : 12.5 (CASES 9,10)
 DRILL PIPE DIA. (IN.) : 4.5
 MUD WT. (PPG) : 9.0
 MUD VISCOSITY (CP) : 30.
 SURFACE TEMP. (°F) : 80.

of keeping the bit cool while it is being lowered into the well. Hence, the seals are exposed to a very high temperature for thirty minutes or longer. Circulation can be established to cool the bit prior to starting the drilling. Some elastomeric seals may be able to survive the relatively short times at high temperatures during which the drill is tripping.

There are additional reasons for pursuing the development of high-temperature homogeneous elastomeric seals. Such a seal would be compatible with existing sealed bit technology, and could easily be incorporated without design changes. This eliminates costly tooling changes for geothermal bit manufacture, and also eliminates the need to coordinate seal development with bit design. When compared to heterogeneous designs (to be discussed in Task III), the homogeneous seals are found to require less space. This is a major advantage, since space in bit design usually translates into strength, and ultimately penetration rate and life.

A final advantage of the homogeneous "geothermal" seal lies in its ability to be easily retrofitted into existing models of sealed rock bits, used for oil, gas, blastholes, and exploration. A seal material having superior temperature characteristics would be of obvious benefit for deep oil or gas wells, where temperatures over 150°C (302°F) are common, and for the geopressured regions in the south, which reach 200°C (392°F). Sealed bits run with air, such as blasthole bits, are usually run at reduced rpms, as compared to unsealed bits, to reduce frictional heat. While this prolongs seal life, and gives longer overall bit life (footage), the penetration rate is lower than with the unsealed bit. An elastomer capable of higher temperature

(and having the required resistance to abrasives and chemicals) would allow the sealed bit to be run faster, with resultant higher penetration rate and better overall economy.

Elastomer Selection - The initial research on elastomeric seals was conducted by L. M. Barker of Terra Tek, Dr. K. L. DeVries, consultant to Terra Tek, and L. Matson of Maurer Engineering; this work is covered more thoroughly in the Phase I Annual Report(9), but will be summarized again briefly. Based on Maurer Engineering experiences, and on the experiences of several people contacted at the drill bit manufacturers, a list of critical elastomer properties for tricone bit seals was established:

1. Low compression set, less than 20%.
2. Good abrasion resistance and high tear strength.
3. Resistance to attack from petroleum-based lubricants, common mud additives, H_2S gas, and moderate shifts in ph.
4. Retention of above properties at elevated temperatures.

The search for an elastomer for geothermal operation was essentially a search for a material that could meet all of these goals, but at higher temperatures. Manufacturers of seals and elastomers were contacted and asked to recommend their best available product, without regard to cost. Table 11 lists the elastomers obtained during both Phase I and Phase II of the program.

GEM Program Interaction - Unfortunately, pure polymer chemistry research was too expensive and lengthy to be considered under the geothermal bit program. Manufacturers such as Dow Corning, DuPont, Olin, Firestone, 3M and Parker had, fortunately, already done a great deal of development work on problems similar to the geothermal seal.

TABLE 11

High-Temperature Elastomer Seal Materials.

Product Name	Producer	Material Class	Probable Temp Limit	Cost/Lb	Potential Compatability Problems	Potential Physical Properties Problems
Buna N (Nitrile)	At least 6 Co.'s	Butadiene & acrylonitrile	150°C (300°F)	Low	High temp water and steam	Low temp limit
Ethylene-propylene	At least 5 Co.'s	Ethylene-propylene	180°C (350°F)	Low	Petroleum base fluids	None
Viton	Du Pont	Fluorocarbon	260°C (500°F)	Low	High temp water and steam	Tear resistance
Fluorosilicone	Dow Corning	Fluorosilicone	260°C (500°F)	Low	Silicone lubes, water and steam	Tear & abrasion resistance; etc.
PNF-200	Firestone	Phosphonitrilic Fluoroelastomer	200°C (400°F)	\$100	None expected	Unknown
Kalrez	Du Pont	Perfluoroelastomer	260°C (500°F)	\$2000	None	Unknown
5AD-7004	Custom Rubber Products	Modified Buna N	230°C (450°F)	Low	None	Unknown
Viton, Peroxide-Cured	DuPont/Parker	Fluorocarbon	260°C (500°F)	Low	High temperature water and steam	Tear resistance
TR-70	Dow Corning	Fluorosilicone (clear)	260°C (500°F)	Low	Silicone lubes, water and steam	Tear resistance

In the latter half of the Phase II effort, contact was made with another program in the ERDA Division of Geothermal Energy engaged in developing new elastomeric compounds for specific geothermal applications. The contract is under the direction of Dr. Robert Reeber, of the Material Development and Corrosion Studies division of DGE, and is titled "Geothermal Elastomer Materials Program", contract WG-77-C-03-1308. Participating in the program are Hughes Aircraft, Inc., Jet Propulsion Laboratory, Inc., Sandia Laboratories, L'Garde, Inc., and Texas Research Institute. The main objectives for the GEM program are materials for geothermal wellbore packers, cable insulation, and static seals. The GEM goals for a static seal elastomer are particularly interesting, since they set an upper limit on what might be expected for the dynamic seals:

1. Service temperature 300°C (572°F) - at least 100 hours.
2. 65% maximum compression set, 25 hours at 300°C (572°F)
3. Resistance to steam, brine, NH_3 , and H_2S .

Since the GEM program is concurrent with the geothermal bit contract, only one elastomer, peroxide-cured Viton, has been recommended for testing thus far. However, a great deal has been learned through informal conversations with GEM researchers. Continued interaction with this program is expected throughout the remainder of the year, and should ultimately lead to the best possible elastomer for geothermal tricone bit seals.

Test Results for Homogeneous Elastomer Seals

Extensive testing was done on the following elastomers:

Buna-N, Parker #N4187 Compound

Ethylene-propylene, Parker E4183-80A Compound

Fluorocarbon, Parker V747-75A Compound

Fluorosilicone, Parker L4216-70A Compound

Fluoronated Polyphosphazene, Firestone PNF-200

Perfluoroelastomer, Kalrez, Compound #1050, 70A Durometer

Modified Buna-N, Custom Rubber Products

All of these were tested in the Reed oval configuration, with the exception of Kalrez, which was a #2-329 O-ring. The installed compression is about 8½% for the oval seals, and this rises with temperature due to thermal expansion of the elastomer; Buna-N has a calculated compression of about 10% at 200°C, for example. The Kalrez O-ring carrier was designed for an installed compression of 7%, rising to 14% at 260°C.

Buna-N - This elastomer is widely used for seals in conventional drill bits. It has excellent resistance to abrasion, good chemical resistance, and is low in cost. Thirty-eight Buna-N seals have been tested so far. Because of their low cost and widespread use in conventional bits, Buna-N was used to provide baseline data with which to compare the performance of other seals, and with which the performance of the seal test machine could be checked. Table 12 gives the data for the Buna-N seal tests.

Tests 1 through 37 showed that Buna-N could perform well in the region from 100°C to 200°C, with no abrasives present. Tests 45-69 were intended to determine the ultimate temperature limit for Buna-N in the 4-hour test with and without abrasives. However, it was this series of tests which led

TABLE 12

Test Data for Buna-N Elastomeric Seals

#	°C TEMP.	TIME (HRS.)	MOBILE MOTION (INCHES)	M-50°C C-25°C START TEMP.	ABRASIVES YES NO	GREASE LOSS	POST-TEST SEAL COND.	SHORE A DURIMETER BEFORE AFTER 0-30 SEC	COMMENTS
1	100	4	.004	C	N	1	Good	---	Good test - Lubriplate hi-temp grease.
2	150	4	.004	C	N	1	Good	77-70 74-69	Good test - Lubriplate hi-temp grease.
3	200	4	.004	C	N	1	Good	74-71 72-61	Good test - Lubriplate hi-temp grease.
7	125	1 hr 45 min	.0053	C	N	-	Good	76-71 74-69	This test used to photograph oscilloscope pattern.
8	200	4	.0053	C	Y	90	Poor	74-69	66-53 First test with cuttings - AT - 21°C for 20 minutes then 5" throughout rest of test.
10	200	4	.0053	C	N	5	Good	---	Run to check new 4:1 gear reducer.
11	Var- ious	-	.0072	C	N	-	Fair	---	New screw together diaphragm holder - Photos for baseline of future tests - New photos at 90 RPM.
15	200	4	.0072	C	N	25	Fair	76-70 68-55	Placed spring in bit lug bellows for this and all following tests.
18	299	4	.0072	C	N	10	Poor	75072 72-59	Seal failed during test 15 min. of test.
21	175	4	.0041	C	N	85	Poor	76-72 70-65	Lip on I.D. of seal carrier found to be bent up .050 in. causing cuts on I.D. of seal.
22	175	4	.0041	C	N	85	Good	---	71-67 Seal carrier straightened - bellows broke causing loss of grease.
23	175	4	.0041	M	N	-	Good	75-73 72-69	No seal damage - hole in bellows.
27	200	4	.004	C	N	4	Good	73-70 68-62	New carrier-not hardened (2.043 dia.) - 65 grease loss, no seal damage.
30	225	4	.004	C	N	20	Fair	75-72 63-54	Seal I.D. is .105 larger - seal took set.
37	200	4	.007	M	N	-	---	73-71 66-60	Bowtherm replaced by grease - run for system check only.
45	200	4	.007	M	N	8	Good	74-70 64-60	Run for comparison with other seals under same conditions.
47	225	4	.007	M	N	8	Fair	75-72 63-55	Seal turned in carrier - no damage. Seal took set I.D. = +.100.
64	175	4	.004	M	Y	15	Good	74-70 75-71	Minor seal damage - AT-10°C throughout test.
65	200	4	.004	C	Y	60	Poor	76-73	60 Extensive seal damage - AT-30°C during warm-up, then 25°C for 2 1/2 hr, 14°C at end of test.
66	200	4	.004	M	Y	9	Good	74-70 69-62	AT was 10°-12°C throughout the test.
67	200	4	.007	C	Y	20	Poor	75-71	61 Extensive seal damage AT-22° start and 11°C end.
68	200	4	.007	M	Y	2	Good	75-71 69-61	No seal damage - AT 10° throughout test.
69	200	4	.007	C	Y	90	Poor	76-72	62 Extensive seal damage - AT-22° start and 14°C end.
70	200	4	.007	M	Y	4	Good	75-72 69-65	Preheated to 50°C, 30 min. - AT was 10°C throughout test. No seal damage.
72	225	4	.007	M	Y	4	Fair	75-73 62-55	Top edge of seal I.D. is cut - major I.D. is worn out but is smooth, preheated to 50°C 30 min. - AT-17°C, 20 min. - 10°-14° during test.
76	225	4	None	M	Y	10	Poor	75-73 65-57	Preheated to 50° - AT-16° start, 13° end. Seal I.D. completely damaged, diameter=75 before and 65 after.
77	240 175	3 6	.007	240°C	Y	65	Poor	75-73 64-56	Preheated to 240°C for 3 hr. to simulate conditions during assembly of drill stem. Cooled to 175° and rotated for 6 hrs. Seal O.D. was .130 in. smaller after test. Burrometer was 78 before and 64 after test. No sand in grease.
78	260 175	3	.007	260°C	Y	55	Poor	76-74 60-51	Preheated to 260° for 3 hrs. cooled to 175 and rotated for 6 hrs. 55% loss of grease - seal O.D. was .125 smaller - seal burned in carrier - sand and cuttings were ground up and in the bottom of the grease cup.
87	20 50 100 200	10 min 25 min 1 hr 4 hr	.007	C	N	2	Good	75-72 74-66	This test was run to check new differential pressure system using two 3000 psi gauges. No seal failure - AP was 40 to 150 psi.
88	200	4	.007	C	N	2	Good	---	88 Duplicate of previous test - but could not maintain AP at end of test.
89	200	4	.007	C	N	2	Good	---	89 System test - too much travel in bellows - when AP is attained it ruins the bellows.
90	200 250	1 15 min	.007	M	N	94	Poor	---	90 Bellows removed from bit lug - AP as much as 900 psi - too much - tried to fail seal on purpose to check failure detection - seal did fail and was detected in 1 hr 15 min.
91	200	1	.007	C	N	-	Good	---	91 Bellows plug failed - AP too high.
92	200 250	1 3	.007	C	Y	20	Poor	---	92 Installed .005 shim stock on bellows plug - seal did not fail.
93	150 250	-	.007	M	N	-	---	---	93 AP went to 400 psi - blew out bellows plug - no seal failure.
94	250	3-3/4	.007	C	N	-	---	---	94 Ran out of time - no failure.
95	250 275 285	1 40 min	.007	M	Y	20	Poor	---	95 AP changed and test was stopped. Seal had failed after 3 hr. 10 min. Temp. was elevated to cause early failure.
96	250 275	1 15 min	.007	M	Y	-	---	---	96 Duplicate test - not enough control over AP. System modified for next test.
97	-	-	.007	C	N	-	---	---	97 AP system modified for better pressure control. Now able to control AP and pressure pulses.
98	250	1 1/2	.007	C	Y	-	---	---	98 Life test for failure detection - pulses had definite change when seal failed.
99	250	10 min	.007	C	Y	-	---	---	99 No AP - test terminated - installed new seal collar.
100	250	2	.007	C	Y	-	---	---	100 No AP - more problems.
101	250	2	.007	C	Y	-	---	---	101 AP check - no failure.
102	250 285	2 1 1/2	.007	C	Y	-	---	---	102 AP check - no failure.
103	250- 300	6	.007	C	N	-	---	---	103 Seal was very hard and brittle. Seal failed in 6 hours.
104	200	1-3/4	.007	C	Y	-	---	---	104 System check - worked good.

Need roller bit grease was used on all tests except 1, 2, 3, 18 and 23. Shaft RPM was 200 on tests 1 through 10 and 90 RPM on all subsequent tests. Axial shaft stroke was .040 inches on all tests. Tests 87 through 104 were used to "debug" and develop the differential pressure monitoring system.

to the discovery that the seal's temperature prior to rotational start-up was a determining factor in seal life when abrasives were present. Tests 65, 67 and 69, for example, were "cold starts" (20°C) which results in heavy abrasion damage, such as that shown in Figure 23 below.

Figure 23. Abrasive damage on Buna-N seal.

Tests 47, 64, 66, and 68 were afternoon tests, and received the unplanned benefit of a warm start-up, since the machine was still warm (around 50°C) due to the morning test. Test #70 was used to test this theory; the seal tester was initially cold (20°C), then the seal was installed and the tester was heated to 50°C for a half hour prior to starting rotation, with abrasives. This seal lasted for four hours at 200°C with very little abrasive damage. It is believed that warming of the seal increases the elastomer's flexibility and allows it to follow the "wobble"

motion more closely, thus preventing abrasives from dropping down between the shaft and seal, as is believed to happen on the "cold-start" tests. All subsequent seal tests were run with the pre-heat procedure.

Tests 72 and 75 were used to determine the seal's temperature limit for the four-hour test with abrasives present; it was concluded that this was approximately 225°C.

Tests 77 and 78 were designed to simulate what happens when a drill bit is "tripped" down into a hot well: once the bit reaches the bottom, circulation is initiated to provide cooling; drilling is begun after the bit has had time to cool to a lower temperature. In test 77, the seal was preheated at 240°C (464°F) for three hours, and 260°C (500°F) in test #78. In both tests, the seal was cooled to 175°C (347°F) prior to running a six-hour test. In both cases, the results were poor; grease was lost and the seal damage was extensive. The most likely explanation is that the higher temperature caused excessive thermal expansion and plastic deformation; upon cooling to the lower temperature the thermal contraction exceeded the remaining compression and an effective sealing force was no longer available.

The remaining Buna-N tests were primarily used to develop the differential seal failure detection system.

Ethylene-Propylene - A total of three tests have been run on this material, all of which were at 200°C. These tests were run using a silicone grease, Dow Corning #44, on the lubricant side of the seal since petroleum-base lubricants are known to attack ethylene-propylene. Table 13 gives the data for these tests.

TABLE 13

Test Data for Ethylene-Propylene Elastomeric Seals

#	°C TEMP.	TIME (HOURS)	MOBBLE MOTION (INCHES)	STROKE (INCHES)	RPM	LUBE	ABRASIVES YES/NO	W=50°C C=25°C START TEMP.	SHORE A DUROMETER BEFORE AFTER		COMMENTS
6	200	4	.004	.040	200	Dow 44 Silicone	N	C	78-73	70-66	Complete loss of grease - seal I.D. worn away.
16	200	4	.007	.040	90	Dow 44 Silicone	N	C	78-74	68-62	Seal indicated failure after 1 1/2 hours - 50% loss of grease after 4 hours.
17	200	4	.007	.040	90	Dow 44 Silicone	N	C	74-71	72-68	Seal indicated failure after 1 hour - 95% loss of grease after 4 hours.

The tests showed very poor retention of the grease for the four-hour tests, and subsequent examination revealed severe scoring of the sealing surface, shown in Figure 24. This was probably due to frictional heating caused by ethylene-propylene's high coefficient of friction. The bump visible on the top of the seal in Figure 24 is the result of plastic deformation of the elastomer into the thermocouple countersink hole (see

Figure 24. Wear and plastic deformation of ethylene-propylene seal tested four hours at 200°C.

Figure 20). Because of the ethylene-propylene's poor performance, and because of its incompatibility with petroleum-base lubricants, no tests were run with abrasives, and no further tests are planned.

Fluorocarbon Elastomer - This material is obtained from Parker Seal, who compounds it from DuPont "Viton" and/or 3M "Fluorel" stock. The Parker designation is V747-75A, meaning it measures 75 on the Shore-A durometer scale. This elastomer did very well in most of the 26 tests run on it so far. Table 14 gives the results of these tests.

With fluorocarbon, it was again found that pre-warming of the seal (see Buna-N results) was beneficial, although it was not as crucial as it was for the Buna-N. Test #60, for example, shows a cold-start with abrasives at 200°C, with good results. The ultimate usable temperature for this elastomer for the four-hour test appears to be 235°C.

Three tests were run which simulate tripping into a deep, hot well, circulating to cool the bit, then initiating drilling. The details of these tests are identical to those of tests #77 and #78 run on Buna-N (see Table 12). The fluorocarbon seals received an evaluation of "fair", while the Buna-N seals were rated "poor". Due to technical problems, test #48 should be discounted, although results did appear favorable. The fluorocarbon elastomer has a distinct advantage over Buna-N in this respect, whereas the 10°C advantage in operating temperature (235°C versus 225°C for the four-hour test) is slight. Attempts to push the fluorocarbon to temperatures above 235°C result in extensive plastic deformation, with the oval seal pushing out into the rectangular gland, and possibly extruding out the gap between the carrier and shaft. Figure 25 shows this effect on

TABLE 14

Test Data for Fluorocarbon Elastomeric Seals

TEST #	°C TEMP.	TIME (HRS.)	WOBBLE MOTION (INCHES)	W=50°C C=25°C START TEMP.	ABRASIVES		% GREASE LOSS	POST-TEST SEAL-COND.	SHORE A DUROMETER		COMMENTS
					NO	YES			BEFORE	AFTER	
4	200	4	.004	C	N		2	Good	80-71	77-70	Seal ID showed signs of wear
13	200	4	.007	C	N		8	Good	78-72	75-68	Seal ID showed signs of wear
19	175	4	.004	C	N		90	Fair	77-74	75-69	Seal ID damaged on top edge ΔT = 20°C for 10 min.
20	175	4	.004	C	N		80	Poor	76-70	75-71	Seal ID damaged on top and bottom edges caused by a bent seal carrier
24	175	4	.004	C	N		15	Good	77-73	75-69	New carrier-no seal damage-no wear on ID
28	225	4	.004	C	N		8	Good	76-70	74-70	No seal damage
29	200	4	.004	C	N		---	Good	76-73	75-70	Bellows broke-new batch of Viton- seems okay
32	225	4	.004	W	N		5	Good	76-70	74-70	New hardened seal carrier
33	250	4	.004	C	N		95	Poor	74-72	76-68	ID failed-seized on shaft. Water in grease cup
34	250	4	.004	W	N		95	Poor	75-73	76-70	ID failed-seized on shaft. Water in grease cup
35	225	4	.004	C	N		5	Fair	75-73	76-72	ID is worn
36	225	4	.004	C	N		5	Good	75-73	75-73	No wear
46	235	4	.007	C	N		15	Good	75-70	75-70	No wear
48	260 200	2 2	.007	C	N		---	Good	75-71	75-70	High temp. 2 hrs. to simulate conditions during assembly of drill stem. Fitting left out of bit lug causing loss of grease
49	235	4	.007	W	N		8	Good	75-71	74-69	No seal damage
58	100	4	.004	C		Y	1	Good	75-70	75-70	No change in grease texture
59	150	4	.004	C		Y	8	Good	76-72	76-70	Grease was liquid
60	200	4	.004	W		Y	4	Good	75-71	74-70	Oil separation in grease
61	200	4	.004	C		Y	4	Good	76-72	75-71	
62	225	4	.004	C		Y	4	Fair	76-72	75-71	ID worn
63	225	4	.004	C		Y	4	Fair	76-72	76-71	ID worn on top edge
73	235	4	.007	W		Y	4	Fair	75-73	75-71	ID worn on bottom edge
74	250	4	None	W		Y	5	Poor	74-71	79-73	ID has considerable damage
76	235	4	.010	W		Y	80	Poor	75-73	78-73	ID has considerable damage. Sand in joiner cup
79	240 175	3 6	.007	Hot		Y	2	Fair	76-73	76-72	Soaked 3 hrs. @ 240°C then run at 175° for 5 hrs. No major seal damage
80	260 175	3 6	.007	Hot		Y	2	Fair	76-72	78-74	Soaked 3 hrs. @ 260°C cooled to 175° * rotated for 6 hrs. - ID worn

seal #34, which was run at 250°C (480°F). This illustrates a problem in designing a seal gland for both low and high temperature operation: at high temperature, the gland may restrict the seal and cause excessive extrusion, and yet the gland must be small enough to prevent excessive movement at lower temperatures.

Figure 25. Fluorocarbon seal run at 250°C for four hours.

Fluorosilicone Rubber - The first seal to be tested at the facility was a fluorosilicone seal which was run at 260°C. This test is shown in Figure 10, and is the only example of brittle failure observed thus far. Seal tests 5, 14, and 105 were then run on this material with poor results. The test results are given in Table 15. In all tests, severe breakdown of the elastomer occurred. Figures 26 and 27 show the nature of this damage: deep cracks and pitting at the running surfaces. Note also the large drop in durometer, indicating molecular degradation throughout the material.

TABLE 15

Test Data for Fluorosilicone Elastomeric Seals

#	°C TEMP.	TIME (HOURS)	WOBBLE MOTION (INCHES)	STROKE (INCHES)	RPM	LUBE	ABRASIVES YES/NO	W=50°C C=25°C START TEMP.	SHORE A DUROMETER BEFORE AFTER		COMMENTS
A	260	4	.005	.030	200	LHT	N	C	67-73	---	Seal was separated into pieces about 1/8" x 1/8" square.
5	200	4	.004	.040	200	LHT	N	C	67-73	57-53	15% loss of grease.
14	200	4	.007	.040	90	RRB	N	C	68-63	47-41	85% loss of grease.
105	175	1	.007	.040	90	RRB	Y	W			Seal would no longer maintain any ΔP after 1 hour of running.

Figure 26. Fluorosilicone seal run four hours at 200°C, 200 RPM, Test #5.

Figure 27. Fluorosilicone seal run four hours at 200°C, 90 RPM, Test #14.

Fluorinated Polyphosphazene Rubber - Firestone PNF-200. This elastomer is an inorganic-backbone polymer based on a phosphorous-nitrogen group.

Table 16 below gives the test results for the one PNF-200 seal tested.

TABLE 16

Test Data for Fluorinated Polyphosphazene Elastomeric Seal

#	°C TEMP.	TIME (HOURS)	WOBBLE MOTION (INCHES)	STROKE (INCHES)	RPM	LUBE	ABRASIVES YES/NO	W=50°C C=25°C START TEMP.	SHORE A DUROMETER BEFORE AFTER	COMMENTS
12	200	4	.007	.010	90	RRB	N	C	63-59 50-39	95% loss of grease - seal I.D. ruined since this test was run with a cold start - it will be repeated later with a warm start.

Note the large loss of durometer, indicating molecular degradation.

Figure 28 shows the seal at the conclusion of test #12. In view of the degradation of the seal and the loss of durometer, further testing on

Figure 28. PNF-200 seal tested four hours at 200°C.

PNF-200 was temporarily set aside. Further tests will be run, however, since more data should be available prior to making a final decision on this material.

Perfluoroelastomer - Kalrez. The current selling price of Kalrez is about \$2,000 a pound, and the decision to purchase a large quantity for testing was postponed. Consequently, only one seal was tested, and this was a #2-329 O-ring, since the mold for this size was available. Table 17 below gives the particulars of the one test run on Kalrez. The seal is shown in Figure 29 (on top). The primary problem was that the seal took a compression set and failed to maintain a good seal. The surface was in good condition and the apparent durometer was about the same, both indicating that there was no molecular degradation. Plans call for the fabrication of some Kalrez seals in the oval configuration for future testing.

TABLE 17
Test Data for Perfluoroelastomer Seal

#	°C TEMP.	TIME (HOURS)	WOBBLE MOTION (INCHES)	STROKE (INCHES)	RPM	LUBE	ABRASIVES YES/NO	M=50°C C=25°C START TEMP.	SHORE A DUROMETER BEFORE AFTER	COMMENTS
9	200	4	.0053	.040	200	RRB	N	C	--- ---	98% loss of grease - seal was worn on I.D. and O.D. indicating that it was turning in the seal carrier. Will run again at lower RPM.

Modified Buna-N - Custom Rubber Products. The composition of this material was not made available to us by the manufacturer, as was the case with most of the elastomers, to one degree or another. Three tests were run on this material and the results are given in Table 18.

Figure 29. Kalrez seal run four hours at 200°C, 200 RPM.

TABLE 18

Test Data For Modified Buna-N

#	°C TEMP.	TIME (HOURS)	WOBBLE MOTION (INCHES)	STROKE (INCHES)	RPM	LUBE	ABRASIVES YES/NO	W=50°C C=25°C START TEMP.	SHORE A DUROMETER BEFORE AFTER		COMMENTS
31	200	4	.004	.040	90	RRB	N	C	98-97	98-98	95% loss of grease - seal material bonded to shaft.
56	225	4	.007	.040	90	RRB	N	C	98-98	98-98	50% loss of grease - ΔT was at 15°C for 2 hours then $\Delta T=10^\circ\text{C}$ rest of test.
83	225	6	.007	.040	90	RRB	N	W	99-98.5	96.5-95.5	50% loss of grease - $\Delta T=21^\circ\text{C}$ - 23°C during first 45 minutes of test - then 10°C rest of test -.108" wear on seal I.D. - no abrasion resistance.

None of the tests were encouraging: the material's exceptionally high durometer led to excessive frictional heating, with subsequent breakdown of the elastomer at the seal interface. In test #31, the elastomer

bonded to the seal collar (shaft), indicating that very high temperatures (above the curing temperature) had been attained. No future tests are contemplated for this material.

Conclusions

No elastomer tested appears to be better than Buna-N, which is already in wide, if not universal, use in the drilling industry. Fluorocarbon elastomer may have some advantage for applications which expose the seal to high soak temperatures such as when tripping into a geothermal well. In the tests which simulated tripping, with a cool-down period prior to seal testing, the V747 fluorocarbon seal showed a distinct advantage over Buna-N.

Future tests on Kalrez may show it to have a slight edge over Buna-N or fluorocarbon, but this remains to be proven. At the present time, no homogeneous elastomer has been tested which can provide a reliable seal for geothermal drill bits.

TASK III. HIGH-TEMPERATURE SEALED DRILL BIT

The construction of a high-temperature sealed geothermal drill bit requires improvements in steels, lubricants, carbides, and seals. The Task I effort of this contract covers the testing of steels for geothermal bits. The proposal for Phase III of this contract includes a major task on high-temperature lubricants for geothermal bits. This will include the construction of a high-temperature lubricant tester and the evaluation of numerous candidate lubricants. Much preliminary research has already been done on high-temperature lubricants, and testing thereof, during Phase I and Phase II of this contract. Proposals have been submitted to NSF-RANN and ERDA/DGE on the optimization of tungsten carbides for drilling at elevated temperatures.

The major effort for this year concentrated on the development of a high-temperature heterogeneous seal. It should be noted that the seal test work done under Task II is restricted to homogeneous elastomers, and is intended to establish the current state-of-the-art for such seals. Task III, however, is aimed at new seal designs which show promise of operating reliably at temperatures of at least 250°C. This has, of necessity, led to designs utilizing multiple components. The work done by Maurer Engineering, on subcontract to Terra Tek, was part of this task.

Heterogeneous Seal Development

The Maurer Engineering patent survey(6) located numerous heterogeneous sealing concepts, many of which utilize elastomeric components for sealing,

and metallic elements for spring members. Most of these sealing techniques are not in use, probably because the simpler compression seals work well, and require less space in the bit. Figure 30 illustrates one heterogeneous seal currently being used by the Smith Tool Company, while Figure 31 shows a Smith compression seal for comparison of size.

Seals for the roller-cone rock bits must do the following:

- a) Provide durable sealing surfaces which can exclude abrasive particles.
- b) Provide the sealing force.
- c) Accomodate rotational eccentricities.

Figure 30. Neilson inclined bearing seal. Smith Tool Co. U.S. Patent #3,096,835, 1963.

Figure 31. Neilson O-ring face seal. Smith Tool Co., U.S. Patent #3,127,942, 1964.

At geothermal temperatures (above 200°C) it becomes advantageous to utilize different mechanical elements to provide the sealing surface and the sealing force. The following are five such heterogeneous seal designs built and tested during Phase II.

The Schumacher Seal

The Schumacher seal, Figure 32, is covered by U.S. Patent #3,761,145. This seal was chosen for development because Reed Tool Company had tested it in rock bits with favorable results, and because the inventor, Bill Schumacher, was available for consultation. The patent states that the O-ring will provide the sealing force, but will also cause a slight bending of the sealing ring such that the psi loading is greater at the outer diameter of the seal interface. This is intended to provide increased resistance to the intrusion of abrasive particles. It should be noted that metal-to-metal sealing systems such as this require extremely flat

Figure 32. Schumacher seal.

and smooth sealing surfaces, such as a lapped finish. Any warping of the sealing members can cause leakage; the cross-section of the Schumacher ring is small enough that the sealing force supplied by the O-ring (about 120 pounds, or 530 Newtons) is probably sufficient to overcome a small amount of warp in the ring.

Test Results for Schumacher Seal - Fourteen tests have been run on this seal so far. A great deal of difficulty was encountered in keeping the O-ring in the intended groove; the predominant tendency was for the O-ring to escape out the back side of the ring.* Table 19 gives the data for all of the Schumacher seal tests. Several design modifications

TABLE 19
Test Data For Schumacher Seal

#	°C TEMP.	TIME (HRS)	WOBBLE MOTION (INS.)	O-RING MATERIAL	COMMENTS
25	175	4	± .004	Silicone	O-Ring severed. First design seal collar, no backup ring.
38	200	4	± .004	Silicone	First design collar with backup ring. Backup ring slipped and O-ring was severed.
39	200	4	± .004	Silicone	First design collar with backup ring. 4% grease loss. Good test. O-ring slightly cut
40	200	4	± .004	Kalrez	First design collar with backup ring. O-ring is too stiff, forced backup ring to shear bolts and slip. O-ring was severed.
42	200	4	± .004	Kalrez	Backup ring was pushed up with extreme force - O-ring appeared to have held about 1 hour. There were signs of high heat buildup during test due to friction at mating surface of seal and sealing surface. The O-ring was mutilated. Beryllium-copper sealing ring was blue.
43	200	4	± .004	Silicone	Still first design collar with backup ring. Spacer ring was installed to relieve compression of O-ring. 8% grease loss-good test
52	200	4	± .004	Kalrez	New backup ring, 6 screws, seal cut almost into two pieces lengthwise. 90% grease loss. Cross section severed 1 time.
55	200	4	± .004	Buna-N	O-ring came off - 90% loss of grease.
57	200	4	± .004	Viton	O-ring pulverized-98% loss of grease.
109	50	5 min.	± .007	Viton	New Schumacher collar with top lip built in. First Schumacher test using AP system. Stopped when no AP. O-ring severed, 90% loss of grease.
110	50	3 min.	± .007	Silicone	O-ring sliced around circumference in several places.
111	110	6 min.	± .007	Viton	O-ring severed.
112	80	12 min.	± .007	Silicone	O-ring sliced around entire circumference.
115	130	12 min.	± .007	Viton	5/8" ball for less O-ring compression. Spacer ring added to reduce back-side clearance. O-ring shredded into several long slices.

All Schumacher seal tests were run with .040 inches (1 mm) stroke, 90 RPM and with Reed roller bit grease, cold start up, and no abrasives.

*As pictured in Figure 32, the O-ring moves upward and out between the ring and the flange on the lug simulator.

were made to improve retention of the O-ring. Tests #25 through 43 were run with a movable retainer flange mounted on the lug simulator above the tapered portion. This allowed the clearance above the sealing ring to be reduced to about 0.050". Although the O-ring's cross-sectional diameter is 0.21" (5.3 mm), it still managed to work its way into the 0.050" (1.3 mm) gap, where it was usually cut by the stroke motion. Tests from #52 onward were done with a new tapered lug collar having a built-in flange which replaced the movable retainer flange, and provided smoother surfaces above the O-ring. In addition, the O-ring retention groove was deepened by 0.030" (0.76 mm) for tests #110-#115. None of these changes improved O-ring retention, however. Seal tests #39 and 43 were the only tests in which the O-ring was retained, and in these cases grease retention was good.

Conclusions - The Schumacher seal has several advantages which make it worthy of further testing. It is small in size and requires relatively minor alterations for incorporation into present drill bits. In addition, it should provide good resistance to abrasive particles, although this remains to be proven because none of the fourteen tests were run with abrasives. One possible reason for the difficulty encountered in retaining the O-ring may lie in the amount of stroke motion used for these tests. A 0.030" (0.76 mm) cam was used, since this represents a worst-case condition; the average stroke present in a drill bit is closer to 0.010" (0.25 mm). However, it was discovered that the actual stroke was 0.040" (1.0 mm), due to a tilt in the cam mount. Hence, the Schumacher seal may be capable of retaining the O-ring if less stroke motion is present.

The Schumacher seal, utilizing a Kalrez O-ring, should be capable of sustained operation at 200°C, and should also be capable of surviving high soak temperatures for several hours.* Additional tests should therefore be run with 0.010" stroke.

Spiral Helix Ring Seal

Figures 33 and 34 illustrate a unique seal concept invented by Larry Matson of Maurer Engineering, Inc. as part of the MEI subcontract effort. The sealing element consists of a helical ring of from four to six turns of a high-temperature plastic known as Ryton. The helical ring is held in an elastomeric surround which provides radial compliance and seals the outer edges of the Ryton. The ring is fabricated about 1.2% undersize and must be stretched over the shaft; thus the spring-like nature of the Ryton provides the sealing force. A primary advantage of this design is that the helical ring remains firmly in contact with the shaft and the wobble motion is accommodated by the stationary elastomeric mount. Ryton is a polyphenylene sulfide plastic manufactured by Phillips Petroleum, and has a maximum use temperature of about 315°C (600°F). The material used in these tests is a filled Ryton which was recommended and manufactured for MEI by International Polymers Corporation of Houston. The IPC designation is R-1832, which is basically a glass-filled Ryton, however the exact formulation is proprietary. Table 20, below, lists the parameters and results for the three tests run to date on this type of seal.

Note that the helix can be made in either the right-hand or left-hand direction. Since the seal tester's rotation is always clockwise (looking

* As previously described in tests simulating tripping of bit into deep, hot well, circulating to cool bit, then initiating rotation.

Figure 33. Spiral Helix Ring seal.

Figure 34. Spiral Helix Ring seal mounted in seal carrier for testing.

TABLE 20

Test Data for Spiral Helix Ring Seal

#	MATERIAL	°C TEMP.	TIME (HOURS)	WOBBLE MOTION (INCHES)	LUBE	W=50°C C=25°C START TEMP.	ABRASIVES YES/NO	GREASE LOSS	POST-TEST SEAL COND.	COMMENTS
26	Ryton(RH)	200	4	.004	LHT	C	N	50	Fair	RTV rubber crumbled - .018 wear on I.D.
41	Ryton(RH)	200	4	.004	RRB	C	N	--		RTV extruded - seal turned in carrier - bellows failed.
82	Ryton(LH) w/Viton	225	6	.007	RRB	C	Y	50	Poor	Seal collar O.D. worn .026. Sand in grease cup.

down on the shaft), a seal with a right-handed helix will tend to induct cuttings down into the grease cup, regardless of which way the seal is put on.

Seal Test #26 - The first helical seal to be tested was built with the right-hand helix in hopes that more lubricant would be retained; no cuttings were used. About half of the available grease escaped during the 4-hour test. However, the Ryton sealing surface remained in good condition; the helix lost about 0.8% of the 1.2% initial undersize fit. The elastomeric mount was made of G.E. high temperature silicone rubber (red), which was molded in place. This material disintegrated considerably during the test, and was thought to be the primary cause of the grease loss. Since an elastomeric mount still seemed feasible, and since the Ryton remained in good shape, test #26 was considered partially successful.

Seal Test #41 - The seal tested here was essentially identical to that tested in test #26. The primary difference being that test #41 used a rollerbit grease, supplied to us by Reed Tool Company. Unfortunately, the pressure compensation bellows ruptured during the test, allowing near total grease loss. Although this failure precluded the gathering of seal life

data, some observations were made. The seal again retained a good wear surface, and the ring retained some sealing force. The molded-in-place silicone rubber experienced even worse extrusion and disintegration than in test #26. Hence, the basic conclusions of test #26 were reinforced.

Seal Test #82 - Test #82 was more severe than the previous two: six hours at 225°C, with abrasives present. The seal was built with four turns of Ryton in the left-hand direction, to assist rejection of abrasives. The silicone rubber elastic mount was not used; instead, a Viton "washer" was installed on top of the Ryton. This seal pressed firmly between the helix and the top of the seal carrier, and thus provided the back seal. In addition, it was hoped that the Viton would keep the abrasives away from the helix.

The test caused severe wear of the seal collar (shaft), about 0.026" (0.66 mm), loss on the diameter, while the Ryton helix wore to the original diameter of the shaft, 2.075" (52.71 mm), an increase of 0.05" (1.27 mm) in diameter. The resulting loss of lubricant was about 50%. The principal problem with this design may be the lack of sufficient sealing force; the Ryton may be undergoing some stress relaxation at the elevated temperatures. The helical Ryton seal might work if driven by a metal spring; a relatively high sealing force could be used since the coefficient of friction is about 0.1 for unlubricated Ryton on steel. The helix would still be required so that the spring can transmit the sealing force through the helix without generating hoop stress in the Ryton. Ryton is a unique material which should be considered for future designs.

Wire-Reinforced Seal

Figure 35 below illustrates an elastomeric seal which contains a ring of reinforcing wire. The wire is woven and then pressed into a square cross section; it is then insert-molded into the elastomeric seal.

Figure 35. Wire-reinforced elastomeric seal.

The wire is restricted to the central portion of the seal so that it cannot interrupt the front or rear sealing surfaces. The wire is intended to reduce the compression-set of the elastomer, especially at elevated temperatures.

The original plans called for a 3 x 3 x 3 test matrix (total of 27 seals) with the following variables:

- a) Elastomers: Viton, Buna-N, Kalrez
- b) Wire mesh density: 10%, 20%, 30%
- c) Wire type: Beryllium-copper, 304SS, 316SS

Unfortunately, difficulties were encountered at two stages of production. The wire mesh manufacturer had difficulty producing rings of the required size. The elastomer molder experienced difficulty in keeping the

wire ring centered in the mold, and also with getting complete penetration of the elastomer into the wire mesh. In addition, the wire mesh altered the shrinkage of the elastomer, resulting in oversize seals. Table 21 gives the results of the two tests attempted thus far on the wire mesh reinforced seals.

TABLE 21
Test Data for Wire Mesh Reinforced Seals

#	°C TEMP.	TIME (HOURS)	WOBBLE MOTION (INCHES)	STROKE (INCHES)	RPM	LUBE	ABRASIVES YES/NO	W=50°C C=25°C START TEMP.	COMMENTS
116	200	2	.007	.040	90	RRB	N	W	A piece of rubber 1/8" x 1/8" was torn from the I.D. of the seal.
117	200	-	.007	.040	90	RRB	N	W	Seal dimensions too far out of tolerance. Not able to maintain differential pressure.

Test number 116 resulted in severe degradation of the sealing surface, with small chunks of elastomer breaking off; some plastic deformation was also noted. A second test was attempted, but the seal was so oversize that it would not seal the gland. Since both of these failures are a result of the seals being oversize, further testing should be done utilizing an oversize seal carrier and seal collar. It should be recognized that while the wire mesh may allow the elastomer to operate at a higher temperature than would otherwise be possible, this is still essentially an elastomer seal and will have a finite temperature limit which may not be high enough for the intended use.

Dacron-Reinforced Buna-N Seal

These seals were manufactured by Custom Rubber Products using their proprietary modified Buna-N compound, and a dacron fabric reinforcement material. As supplied, the seals were too thick for the seal carrier and were ground down to the standard thickness of 0.210" using a horizontal surface grinder. Table 22 gives the results of the two tests run on this material. The surfaces of these seals were too uneven to get accurate durometer data, but some embrittlement of the elastomer was noted in both tests, since the seals were harder to bend

TABLE 22
Test Data For Dacron-Reinforced Buna-N Seal

#	MATERIAL	°C TEMP.	TIME (HOURS)	WOBBLE MOTION (INCHES)	LUBE	W=50°C C=25°C START TEMP.	ABRASIVES YES/NO	GREASE LOSS	POST-TEST SEAL COND.	COMMENTS
44	Buna/ Dacron	200	4	.004	RRB	C	N	15	Fair	Not able to measure hardness, surface too uneven. Rubber was quite brittle at end of test. Seal height was milled down so seal would fit into carrier.
51	Buna/ Dacron	225	4	.007	RRB	C	N	60	Poor	Seal I.D. damaged brittle at end of test out of tolerance before test - too much compression.

The results of these two tests were not encouraging; the dacron-reinforcement provided little if any improvement over the homogeneous seals of modified Buna-N (see Table 18), and both of these types were inferior to the Parker Buna-N or Parker Fluorocarbon seals. No future testing is planned for fabric-reinforced seals.

Spider Seal

Figures 36 and 37 illustrate a new sealing system invented at Terra Tek as part of the Task III effort of this contract. The system consists of a metal seal ring which is driven towards the mating surface by twenty metal spring wires. In the seal tester, the seal ring rotates with the moving lug simulator, while the face-seal disc, shown in Figure 37, remains stationary. In a drill bit, the seal ring and wires would remain stationary on the lug, while the cutter cone rotated about the lug. The seal pictured in Figure 37 utilizes an elastomeric O-ring to provide the static seal between the ring and the lug simulator. Table 23 gives the test results for five tests run to date on this seal. Several of the tests were very encouraging, including one run at 250°C. This is the highest temperature that a seal has survived for the standard four-hour

Figure 36. Illustration of Spider Seal system.

Figure 37. Spider seal and face-seal disc.

TABLE 23

Test Results for Spider Seal

#	MATERIAL	°C TEMP.	TIME (HOURS)	WOBBLE MOTION (INCHES)	LUBE	W=50°C C=25°C START TEMP.	ABRASIVES YES/NO	% GREASE LOSS	POST-TEST SEAL COND.	COMMENTS
53	Viton O-Ring	200	4	.004	RRB	C	N	90	Poor	O-Ring was not properly seated in groove - not enough spring tension. Spring diameter too small and not enough down pressure was applied.
81	Viton O-Ring	200	4	.007	RRB	C	N	30	Good	.047 music wire - 3/4" ball- still not enough down pressure.
84	Viton O-Ring	200	4	.007	RRB	C	N	2	Good	.047 music wire - 13/16" ball good oil retention.
85	Viton O-Ring	225	4	.007	RRB	C	N	2	Good	Same as above.
86	New O-Ring	250	4	.007	RRB	C	N	5	Good	Same again - seems good to 250°C.

test. The same ring and test disc were used for all five tests, with no appreciable damage to either surface; both parts have been plasma-coated with Tribaloy T-800.*

A replacement is currently being sought for the elastomeric O-ring; metal C-rings and piston rings are under consideration. It would, however, be interesting to see how well Kalrez could perform in this application. The spring wire material is also being changed, since the high-carbon steel "music wire" currently used will begin to lose its temper around 250°C, and presents a corrosion problem at any temperature. Several grades of stainless steel are under consideration.

Future testing will include tests with abrasive particles and tests at higher temperatures. Pending favorable results, the spring system can then be redesigned to utilize a one-piece spring which will facilitate assembly. Although initial results with the Tribaloy T-800 seem favorable, there are other products available which can be substituted, such as Union Carbide's gun-deposited tungsten-carbide coatings. Union Carbide claims that the gun deposition provides a much stronger bonding to the substrate material.

Conclusions

Of the five design concepts tested this year, only one, the Spider seal, demonstrated any chance of success. Clearly, new designs must be brought into the program. Maurer Engineering and Terra Tek will each

* Tribaloy is a DuPont product, and the parts were coated by Plasma Technology, Inc. of Torrence, California. "Tribaloy intermetallic materials are two-phase compositions of hard hexagonal Laves phase crystals tightly bonded by a softer matrix." Dupont.

pursue at least one new seal design during the Phase III contract year.
Both will continue to consult with drill bit manufacturers concerning
the "manufacturability" of these designs.

TASK IV. DRILL-BIT INSERT RETENTION

Because tungsten carbide inserts have much smaller thermal expansions than the steels in which they are held, there has been concern that the inserts may fall out of their holes at elevated temperatures. The twelve conventional drill bits recovered from the Geysers and other sites were examined by Terra Tek and showed almost no loss of inserts. However, Blair, Tester and Mortensen of Los Alamos(21) reported typical losses of 15-25 inserts per bit on the LASL Hot Dry Rock Geothermal Project, which involved drilling of very hard granitic formations such as monzogranite gneiss and granodiorite gneiss. Typical bit lives of 40 hours for air drilling and 60 hours for water drilling would indicate a bottomhole temperature considerably below that experienced at the Geysers. Unfortunately, the LASL group did not attempt to calculate bottomhole temperatures, although they did report a formation temperature of 150°C. Since no bits had yet been run in the Terra Tek wellbore simulator at the onset of the Phase II contract year, there was no data available on insert retention at very high (300°C) operating temperatures.

An extensive patent survey(10) was conducted by Bill Maurer of Maurer Engineering on the subject of insert retention methods. Over 350 patents relating to roller bit design were reviewed, and of these, twenty-nine were found to be germane to the problem of high-temperature insert retention. Several of the patents could be put to use without modification, but would markedly increase the cost of the drill bit, at least for the present, since tooling modifications would be required. Drill bit tests run in the

wellbore simulator during Phase II, at temperatures to 316°C (600°F), experienced no insert loss (see Task I). However, these tests generally lasted from two to four hours, and longer tests may yet show insert loss in the laboratory.

Calculations involving actual insert pressing interferences now indicate that the inserts will not loosen at the temperatures anticipated in geothermal drilling, provided that the pressing interferences are not significantly decreased. Insert loss at high temperature is probably controlled more by the softening of the drill-bit roller cone steel, allowing plastic deformation, and the consequent working loose of the inserts, than by the difference in thermal expansion of the insert material and the retaining steel material. Since the proposed field test site for the Phase III effort is the Geysers, and since the conventional bits recovered from the Geysers experienced only minor insert losses, no special insert retention methods will be used for the next generation (MK III) experimental bits. Future geothermal bit applications involving very hard strata, such as those encountered by LASL in the Hot Dry Rock program, should consider the insert retention devices recommended by Maurer Engineering in the patent survey.

INTRODUCTION

Most present day drilling, whether for oil and gas recovery, large mining blastholes, or geothermal energy recovery is done with three-cutter roller bits which have evolved from the first roller-cone bit, patented by Howard Hughes in 1909. These bits are continuing to be improved by industry at the rate of 2-4% per year(1). Unfortunately for geothermal energy recovery, these bits have been optimized for lower temperatures and deliver only about one-fifth their life (designed footage) when used in some geothermal environments. Furthermore, the shortened bit life represents only one part of the increased drilling costs, which run at least fifty percent higher than for oil and gas wells. Penetration rates are generally lower since the bits are run at reduced load in order to prevent catastrophic failure. Additionally, bit replacement requires hours of non-productive "trip time" which ties up the rig and leads to other problems. Both require more time for well completion, and thereby increase well costs.

Drill bit manufacturers have not been able to devote very large efforts to develop the specialized high-temperature bits, since the geothermal bit market is small (relative to the oil, gas and mining bit markets, which are currently experiencing a "boom"). Hence the Energy Research and Development Administration, Division of Geothermal Energy initiated a program to provide the background research needed to develop drill bits for the geothermal environment. Terra Tek is the prime contractor, with Maurer Engineering as a subcontractor, and with support from the drill-bit manufacturers, particularly

Additional thanks goes to those who, through informal conversations, deepened our understanding of drill bit materials problems: Walter Littmann, Morgan Crow, W. S. Keller, Sam Penny, Jim Langford, Bob Williamson, Jim Young, Don Vandeford, Bob Clamon, Mike Conn, J. D. Morris, and Lee Goodman of Dresser Industries; Tom Oldham, Lee Stebbans, Bob Evans, Lloyd Garner, Bob Dixon and Stacy Oldham of Smith Tool Company.

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APPENDIX A

Derivation of cutoff frequency for the fluid-coupled pressure transducer.

Assumptions:

1. The cutoff frequency is determined by the resonance of the fluid column (mass) against the pressure transducer diaphragm (spring).
2. The walls of the tubing are much stiffer than the transducer.
3. The wavelength at the resonant frequency is much longer than the length of tubing.

The basic equation for simple harmonic motion is:

$$\omega_c = \frac{1}{\sqrt{MC}} \quad (1)$$

ω_c = cutoff frequency, radians/sec

M = mass of fluid in column

C = compliance of transducer/fluid column system

where

$$M = \rho L A$$

ρ = fluid density

L = column length

A = mean column cross-sectional area

C is the compliance of the transducer as seen by the fluid column.

However, the compliance figure for the transducer, if any is available,

is given in terms of volume change per unit pressure change. The transducer used on the seal test machine is a Tyco AB 3000, for which the compliance was available:

$$C_T = 4 \times 10^{-8} \text{ in}^3/\text{psi} \text{ or } 9.5 \times 10^{-17} \text{ M}^3/\text{Pa} \quad (2)$$

Making use of the fact that the volume change in the transducer equals the volume change in the fluid column:

$$\Delta V_T = \Delta V_C = C_T \Delta P = A \Delta X \quad (3)$$

ΔP = change in pressure applied to system

ΔX = movement of fluid column due to applied pressure

The force acting on the column due to the pressure change is:

$$\Delta F = A \Delta P \quad (4)$$

Equations (3) and (4) can be rearranged to yield an expression for C:

$$C \equiv \frac{\Delta X}{\Delta F} = \frac{C_T}{A^2} \quad (5)$$

Substituting (2) and (5) back into (1) yields:

$$W_c = \frac{1}{\sqrt{\frac{\rho L C_T}{A}}} \quad (6)$$

or

$$f_c = \frac{1}{2\pi \sqrt{\frac{\rho L C_T}{A}}} \quad (7)$$

The system built for the geothermal seal tester had the following approximate parameters:

$$C_T = 9.5 \times 10^{-17} \text{ M}^3/\text{Pa}$$

$$L = 0.6\text{M}$$

$$\rho = 1061 \text{ kg/m}^3, \text{ Dowtherm 'A'}$$

$$A = 4 \times 10^{-6} \text{ M}^2 \text{ (average for combination of .090" ID tubing and .19" ID tubing)}$$

This yields an f_c of 1.29 khz. The measured system resonance was at about 380 hz, indicating, most likely, that the second assumption was not valid. The compliance of the tubing may have been at fault, although the elastomeric O-ring, used to seal the transducer into the system, may also have been contributing to the problem. Future systems should use thick-walled tubing and all-metal seal rings to achieve higher frequency response.