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Factors Affecting Seal Life in Downhole Motors

D. W. Dareing

Sandia National Laboratories

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FACTORS AFFECTING SEAL LIFE IN DOWNHOLE MOTORS*

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ABSTRACT

Life expectancy of rotary seals in downhole motors depends both on the temperature generated by sliding friction and on the ambient temperature. A parameter study which led to an improved, temperature-reducing seal configuration is described in this report.

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SUMMARY

The life expectancy of rotary seals in downhole motors depends on temperature generated by sliding friction as well as ambient temperature. Heat transfer calculations show that sliding friction can produce a significant rise in temperature across seal assemblies, great enough to deteriorate the seal material and cause premature failure.

Thermal conductivities of seal materials and thicknesses of shaft, sleeve, and housing are major design factors influencing steady state temperature profiles across seal assemblies. In general, smaller dimensions and higher thermal conductivities allow the friction generated heat to dissipate at a lower temperature.

A parameter study led to an improved rotary seal configuration which will significantly lower peak seal temperatures in downhole motors. The design will channel drilling mud near the sliding friction surface for better dissipation of the friction generated heat. Plans are being made to incorporate this improvement into the bearing seal test assembly developed earlier under this contract for Terra Tek by Maurer Engineering.

It is doubtful that seals made of Buna-N will perform successfully on downhole motors, even when used in the improved design. On the other hand, calculated maximum temperatures are within material limiatations of Grafoil.

BACKGROUND

Two basic components of turbodrills are the turbine section and the bearing pack. The turbine section converts hydraulic horsepower, carried by the drilling mud, to mechanical power at the drill bit in the form of rotary speed and bit torque. Thrust bearings in the bearing pack transfer axial thrust, generated by a combination of pressure drop across the turbine blades and bit weight, to the turbine housing and drill collars. Radial bearings in the bearing pack centralize the inner rotating shaft. Figure 1 shows how the turbine section and bearing pack are combined to form a turbodrill. Drilling mud enters at the top of the turbine section, is diverted through the

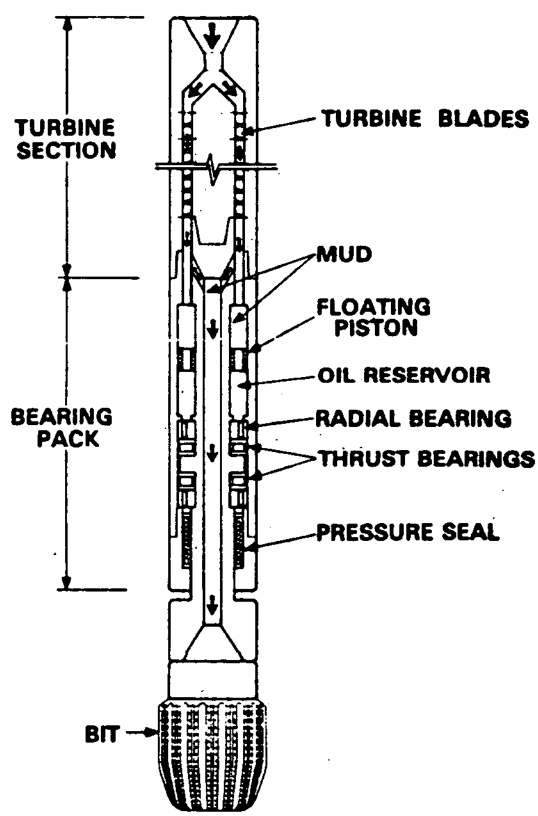

Figure 1 Turbodrill Assembly

turbine blades, and exits at the bottom end of the bearing pack. There is a pressure drop across the turbodrill and an additional pressure drop across the drill bit. The drilling mud cleans and cools the bit and flows back up the wellbore in the annular space between the turbodrill and wellbore wall.

If the bearings are allowed to operate in the drilling mud environment, their operating life is greatly reduced. One design scheme for protecting these bearings from drilling mud (see Figure 2) is to operate the bearings in lubricant. The lubricant is pressurized by a floating piston which is displaced by pressure differential across the bit. A pressure seal controls leakage from the oil reservoir and prevents drilling mud in the annulus from seeping into the bearings. Normally, the bearings fail shortly after the pressure seal fails.

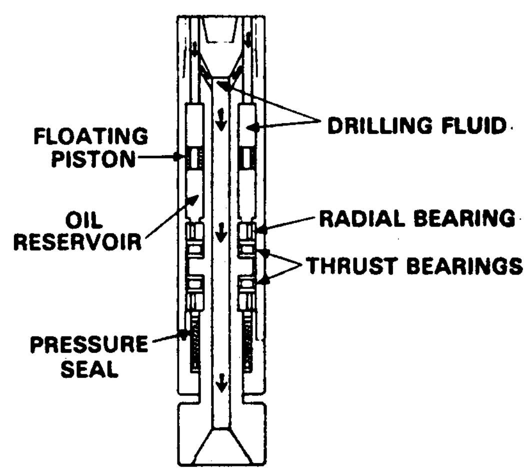

Figure 2 Bearing Pack Assembly

Designing effective rotary seals has been a problem for several years, and in many pieces of equipment rotary seals are the weakest component. Recognizing this, the U.S. Department of Energy/Division of Geothermal Energy funded a laboratory study of rotary seals in downhole drilling motors. The test program is set up at Terra Tek (Salt Lake City) and is designed to establish the life expectancy of different types of rotary seals. Hopefully, the study will lead to a long life seal for downhole motors.

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TERRA TEK SEAL TESTER

The Terra Tek seal tester utilizes a rotating shaft and a stationary housing as shown in Figure 3. The seals operate against a polished wear sleeve plated with a hard material such as chrome. Thermocouples placed in the sleeve and in the housing are used to monitor seal temperatures during the tests. The coolant fluid inside the tester flows adjacent to the inside of the sleeve, consequently, no heat transfer takes place across the shaft. Heat is dissipated more effectively in the seal tester than in the bearing pack where the heat must flow through the drive shaft before it can be convected by the mud inside the motor. For a detailed description of the tester and a summary of test results, refer to an earlier report (Black et al., 1978)[1]

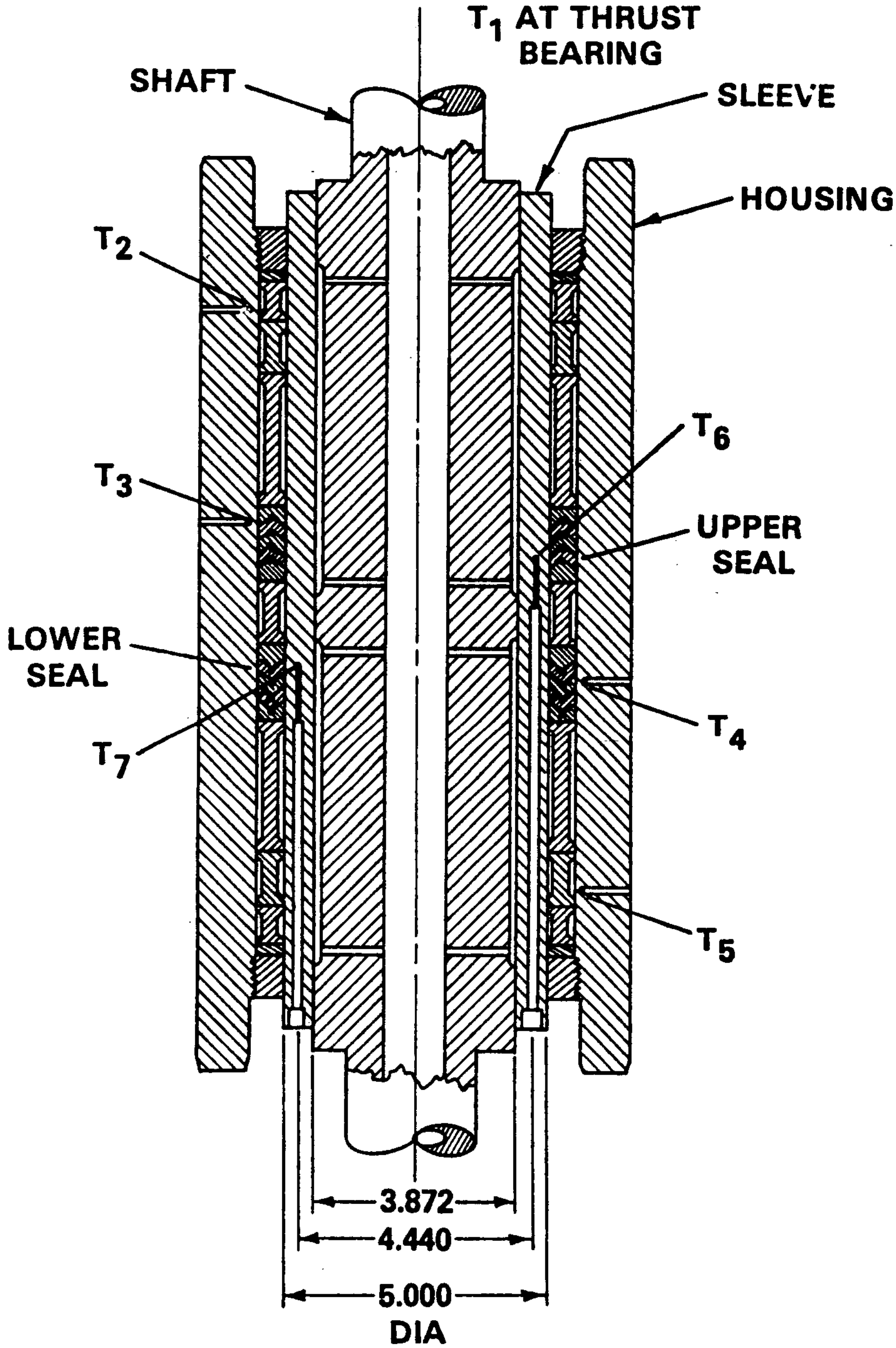

Figure 3 Cross Section of Seal Tester

BEARING SEAL PACK

As subcontractor to this project, Maurer Engineering designed and built the bearing seal test assembly shown in Figure 2. Cross-sectional dimensions of the seal assembly are given in Figure 4. It is the heat transfer through this section that was analyzed and is covered in this report.

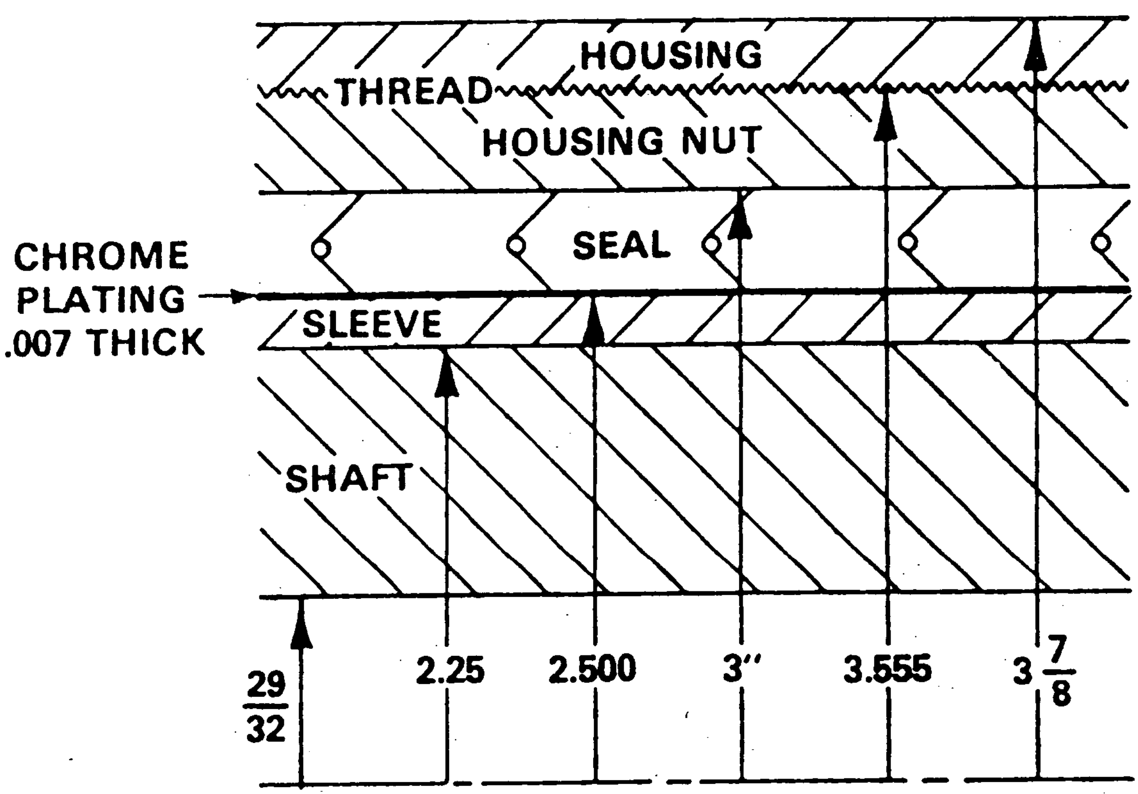

Figure 4 Cross-Sectional Dimensions of Bearing Seal Assembly

Since this assembly was designed and submitted to Terra Tek, much has been learned during the project about heat generation and temperature distribution across rotary seal assemblies. Excessive local temperature can reduce the strength of the seal material and is one possible cause of failure initiation in rotary seals. We now know that it is important to design seal assemblies so that heat, caused by sliding friction between seal and sleeve, can be removed by sufficiently low temperature gradients while keeping steady state temperatures within acceptable limits.

It was shown earlier (Dareing, 1978)[2] that calculations based on a one dimensional heat transfer model gave reasonable predictions of temperature profiles across rotor seal assemblies. Temperature profiles predicted by the one dimensional theory compared favorably with temperatures measured by thermocouples in the Terra Tek seal tester.

The computer program presented in reference 2 has since been used as a design aid to determine the affects of seal profiles and the intensity of maximum local temperature at the sliding seal/sleeve interface. Our study has lead to a seal assembly which is an improvement over the one shown in Figure 4.

TEMPERATURE CALCULATIONS

Heat Transfer Formulation

The temperature profile across a seal assembly depends on ambient temperature, both inside and outside the assembly, as well as the rate at which heat is being generated at the sliding seal/sleeve interface. The rate of heat generation at the sliding surface is determined by

$$Q = \frac{2\pi TN}{788} \text{ Btu/min} \tag{1}$$

where

T - Friction Torque, ft-lbs
N - Rotary Speed, rpm

Note that friction torque is

$$T = \mu p_c \frac{\ell}{12} (2\pi c)c \tag{2}$$

where

μ - coefficient of friction
p_c - seal/sleeve contact pressure, psi
ℓ - length of seal contact, inches
c - radial distance to sliding surface, inches

In general, this heat is dissipated by conduction inwardly through the shaft and outwardly through the housing; during drilling the heat is carried away by drilling mud once it reaches the fluid contact boundaries of the turbine.

Following the one dimensional heat transfer model, heat travels only in a radial direction and the temperature profile across each cylinder (such as shaft, seal, housing) is defined by

$$T(r) = \frac{1}{\ln \frac{r_o}{r_i}} \left[T_i \ln \frac{r_o}{r} + T_o \ln \frac{r}{r_i} \right] \tag{3}$$

The boundary condition at the sliding interface for the sleeve is

$$Q_i = -2\pi r \ell \ K_2 \left. \frac{dT_2}{dr} \right|_{r=c} \tag{4}$$

while the boundary condition at the sliding interface for the seal is

$$Q_o = -2\pi r \ell \ K_3 \left. \frac{dT_3}{dr} \right|_{r=c} \tag{5}$$

In equations 4 and 5, the subscript 2 refers to the sleeve while the subscript 3, refers to the seal. Boundary conditions at the fluid contact boundaries are ambient temperatures. Assumptions made on the one dimension heat transfer model are

1. Thermal conductivities are independent of temperature
2. Thermal conductivity of fluid boundary layer is ignored.

While the mathematical model is not an exact representation of reality, it does give temperature predictions which compare favorably with measured data. Since Reference 2 was written, the computer program was expanded to include four cylindrical layers to account for

1. Shaft
2. Sleeve
3. Seal
4. Housing

A schematic of a typical temperature profile is shown in Figure 5.

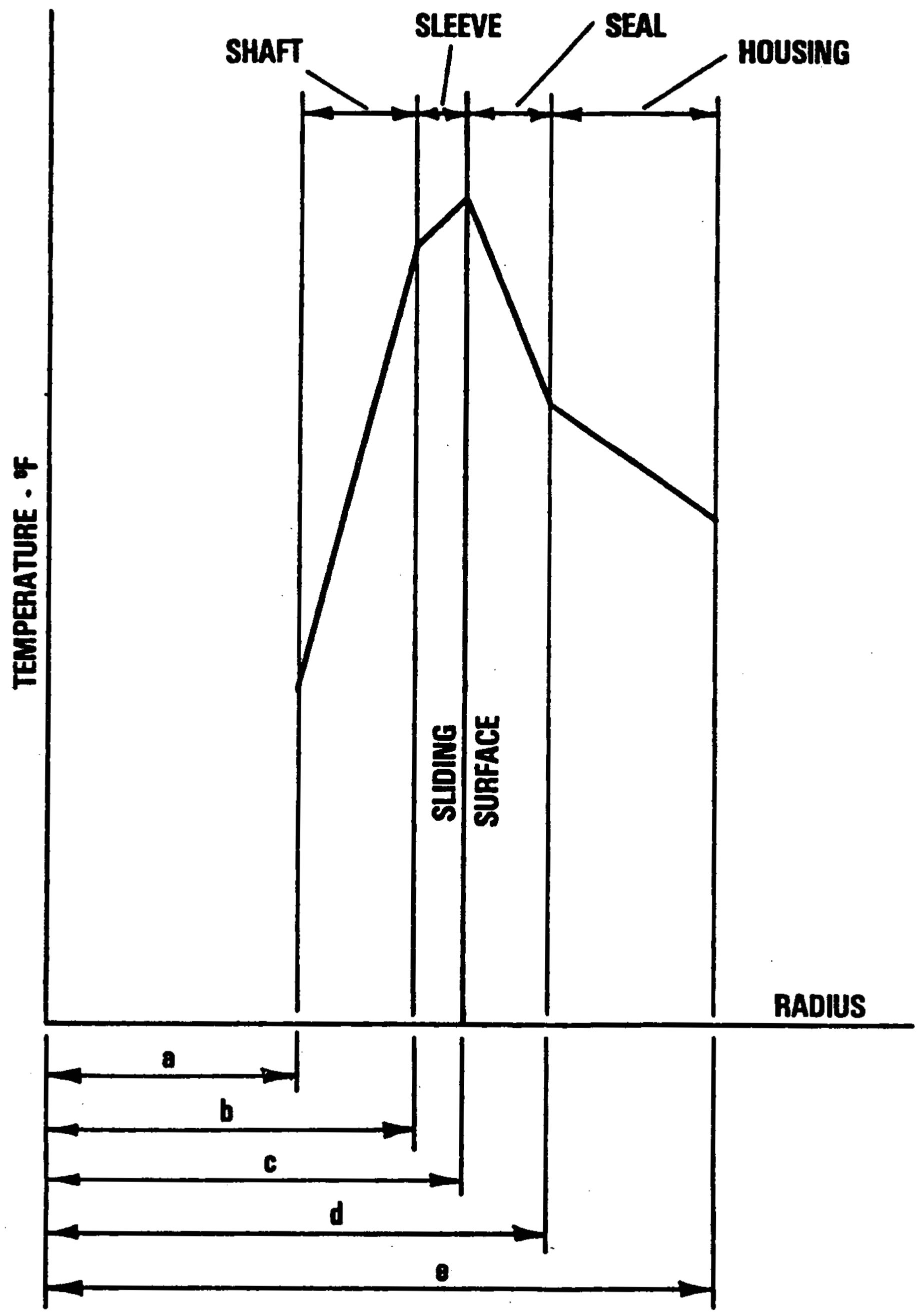

Figure 5 Schematic of Temperature Profile

Reference Parameters

The seal assembly (Figure 4) in the bearing pack has different dimensions from the Terra Tek seal tester (Figure 3). Dimensions of the two seal assemblies are given in Table 1.

Table 1

	Terra Tek Seal Tester	Bearing Seal Assembly
Shaft Wall Thickness, inches	0 *	1.34
Sleeve Thickness, inches	0.564	0.25
Housing Wall Thickness, inches	1.125	.875
Radius to Sliding Surface, inches	2.5	2.5
Seal Thickness, inches	.5	.5

* Effectively zero since coolant contacts sleeve.

To date, seal tests have been conducted with an average coolant fluid temperature of 80°F. The outside temperature has been adjusted so that thermocouple temperature, T_3, (see Figure 3) is maintained at 250°F. During tests of Buna-N seals, the outside temperature is approximately equal to T_3 because Buna-N insulates the seal housing from friction heat. On the other hand, during tests of Grafoil seals, the outside temperature is much less than 250°F because Grafoil conducts friction heat to T_3.

Drilling mud temperature inside and outside of a downhole motor will typically be 180°F in hot geothermal wells (McDonald, 1976)[3]. The mud reaches higher temperatures as it flows up the hot wellbore annulus, but this does not affect the seal temperature in the motor.

Several parameters can be selected as input for computer calculation of temperature. For each set of calculations only one parameter was varied at a time while the others were fixed. The reference design parameters were the same as those selected for the bearing seal assembly (Figure 4) while the reference operating parameters were the same as the ones measured from the seal tester; the reference ambient temperature for the bearing seal assembly was 180°F. Tables 1 and 2 summarize the reference parameters.

TABLE 2

REFERENCE PARAMETERS FOR COMPUTER CALCULATIONS

	SEAL TESTER	BEARING PACK
Rotary Speed (rpm)	212	212
Seal Pressure Drop (psi)	1,500	1,500
Friction Torque (ft-lbs)	50*	50*
Friction Power (hp)	2	2
Shaft** Thermal Conductivity (Btu/hr/ft/°F)	25	25
Sleeve** Thermal Conductivity (Btu/hr/ft/°F)	25	25
Grafoil Seal Thermal Conductivity (Btu/hr/ft/°F)	80	80
Buna-N Seal Thermal Conductivity (Btu/hr/ft/°F)	0.1	0.1
Inner Fluid Mud Temperature T_a (°F)	80	180
Outer Fluid Temperature T_e (°F)	***	180

* 50 ft-lbs per seal
** Steel
*** Varied to maintain T_3 at 250°F

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PREDICTED TEMPERATURES IN SEAL TESTER

Temperature profiles given in this section are reproduced from reference 2 for comparison with temperature profiles predicted for the bearing pack seal assembly. As shown in reference 2, these calculated temperatures compare favorably with temperatures measured in the seal tester.

Figure 6 shows that a maximum seal temperature of 535°F is generated at the sliding interface between the seal and the wear sleeve. This temperature is much higher than the 350°F temperature at which Buna-N deteriorates and is consistent with the severe deterioration of the Buna-N seals observed during the tests. In some test cases, the seals were vulcanized together to the extent that they could not be pulled apart. Figure 6 shows that reducing the seal coefficient of friction by 50%, (i.e., reducing the friction losses from 2 hp to 1 hp) would reduce the seal temperature from 535°F to 308°F.

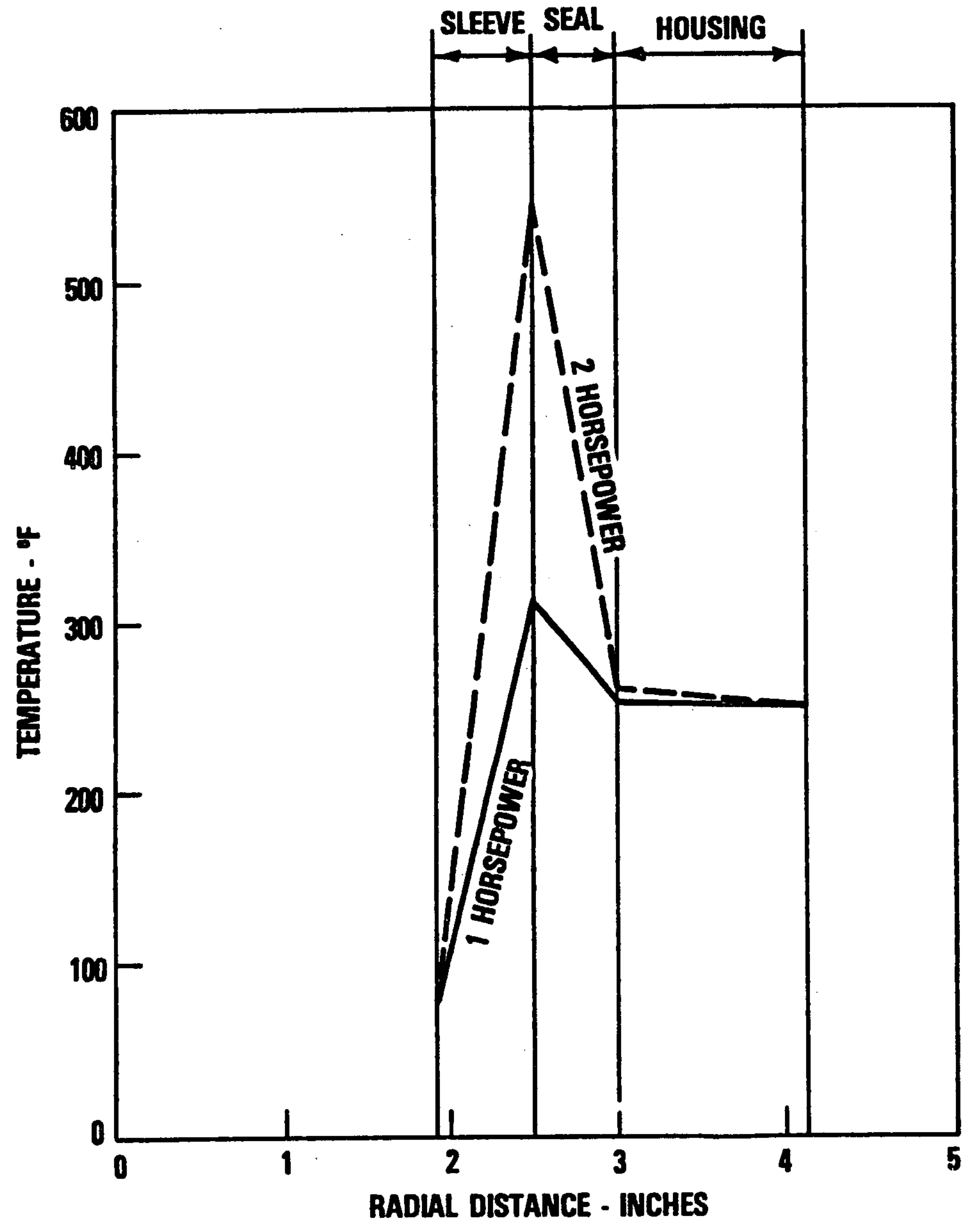

Figure 6 Buna-N Seal Temperatures in Seal Tester

PREDICTED TEMPERATURES IN BEARING SEAL PACK

Calculations show that the temperatures in the bearing seal pack will be much higher than the temperatures in the seal tester (Figure 7). The computer model predicts a peak temperature of 1,950°F in the bearing seal assembly compared to a peak temperature of 535°F in the seal tester. This high seal temperature is produced because heat must flow through the shaft in the bearing pack whereas it does not in the seal tester because fluid flows adjacent to the wear sleeve in the tester. The high seal temperature indicates that the seal will fail much faster in the bearing pack than in the seal tester. <u>It is therefore critical that the bearing pack be modified so that the seal temperatures will be reduced to acceptable limits</u>. The objective of this study is to analyze the various parameters that affect seal temperature in the bearing pack and to make recommendations on how to redesign the existing bearing pack to reduce seal temperatures.

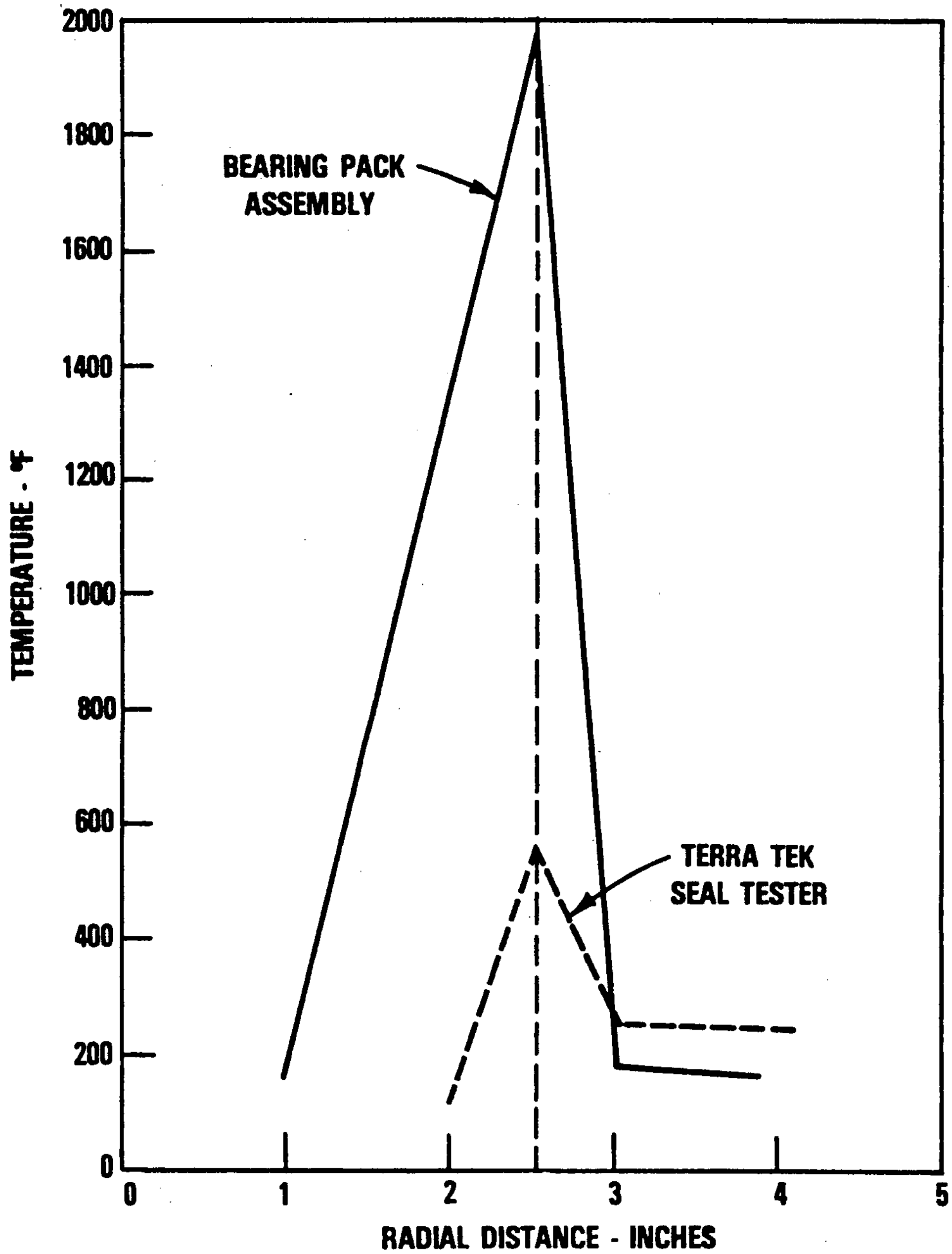

Figure 7 Buna-N Seal Temperature
(50 ft-lbs, 212 rpm)

Effect of Seal Material

Seal temperature is highly dependent upon the thermal conductivity of the seal material because any friction heat that flows to the outside of the tool must flow through the seal. Figure 8 shows the effect of the two different thermal conductivities on temperature profiles for the bearing pack assembly. Buna-N has a thermal conductivity of .1 Btu/hr/ft/°F compared to 80 Btu/hr/ft/°F for Grafoil. An increase in the thermal conductivity by a factor of 800 reduces maximum seal temperature in the bearing seal pack from 1,950°F to 607°F. The improved heat flow is one of the important advantages of using Grafoil as a seal material. Another advantage is that Grafoil is 99+ percent carbon and can operate at temperatures in excess of 1,000°F compared to the 350°F limitation of Buna-N rubber.

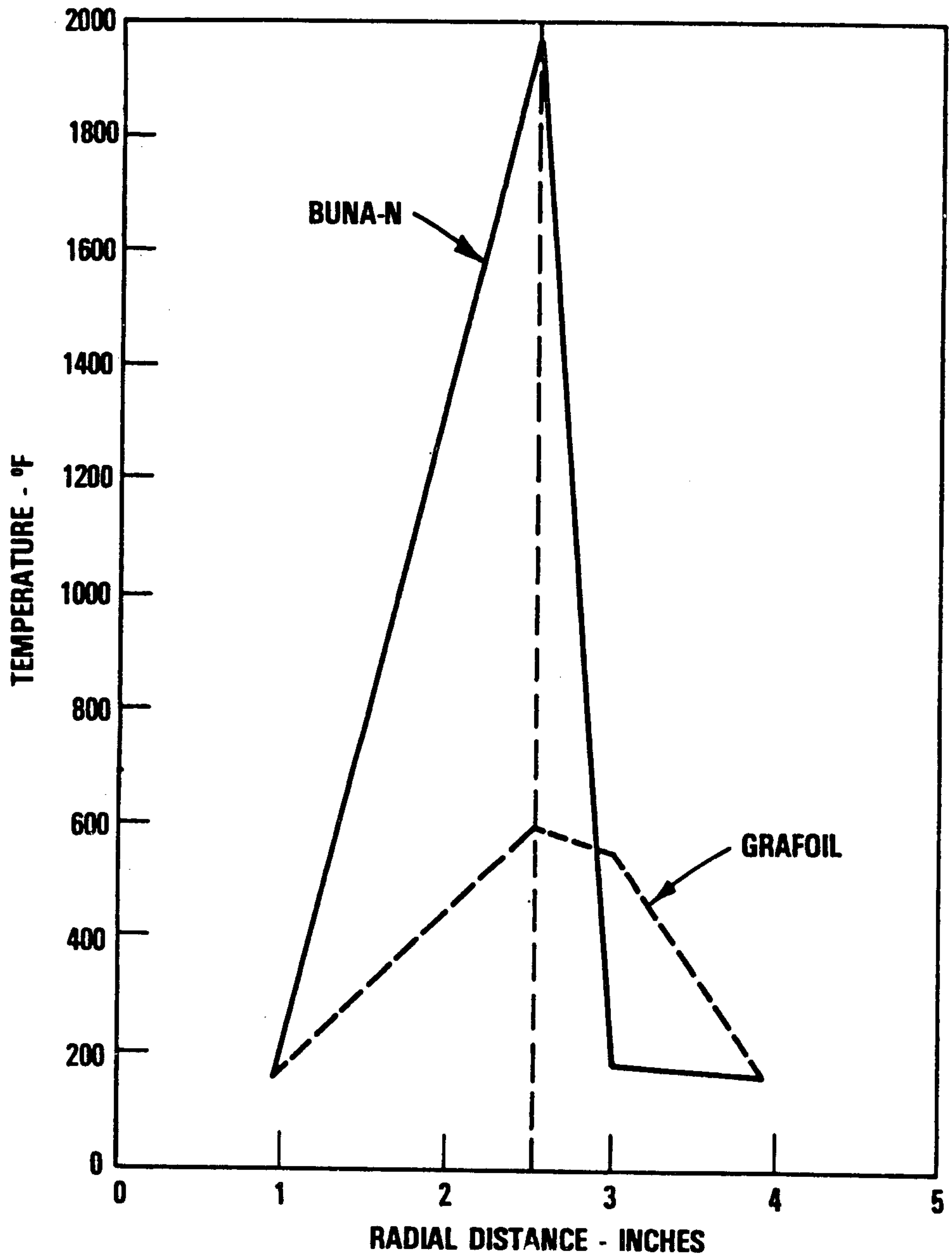

Figure 8 Bearing Pack Assembly Seal Temperatures
(50 ft-lbs, 212 rpm)

Effect of Shaft Wall Thickness

As shown earlier, excessive seal temperatures will be produced in the bearing pack because of low heat flow through the motor shaft. Figure 9 shows that the Buna-N seal temperature can be reduced from 1,950°F to 370°F by reducing the shaft thickness from 1.34 inches to zero. A design shown later can effectively reduce the shaft thickness to zero by flowing drilling mud next to the wear sleeve, thereby reducing the seal temperature to 370°F.

One conclusion from these data is that even with only a .25 inch steel sleeve, (i.e., zero shaft thickness) maximum temperature exceeds the 350°F allowable operating temperature of Buna-N. Seals made of Buna-N have therefore little chance of working as rotary seals in downhole motors. This is consistent with past experience as researchers have been unsuccessfully attempting to use various types of rotary seals containing Buna-N rubber since the mid 1920's.

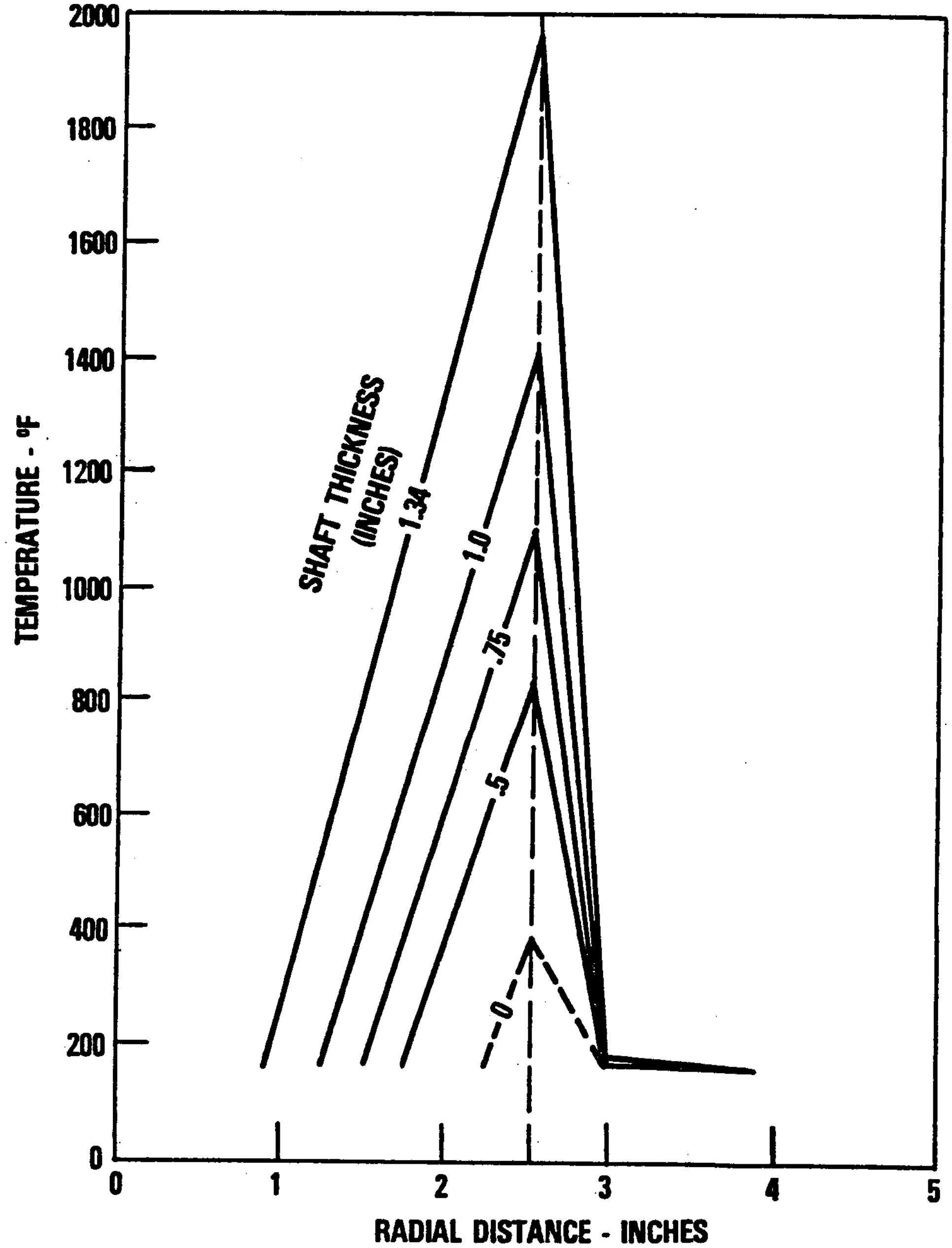

Figure 9 Effect of Shaft Thickness on Buna-N Bearing Pack Seal Temperatures

Figure 10 shows that the bearing pack Grafoil seal temperature can be reduced from 607°F to 321°F by reducing the shaft thickness from 1.34 to zero inches. Changes in shaft thickness have less affect on the Grafoil seal temperature than on Buna-N seal temperature because more heat flows through Grafoil and across the housing to the drilling mud in the annulus.

Peak temperature data from Figure 9 and 10 are summarized in Figure 11. The intersection of the dashed line with the two curves identifies peak temperature corresponding to each seal material when used in the bearing seal assembly (Figure 4).

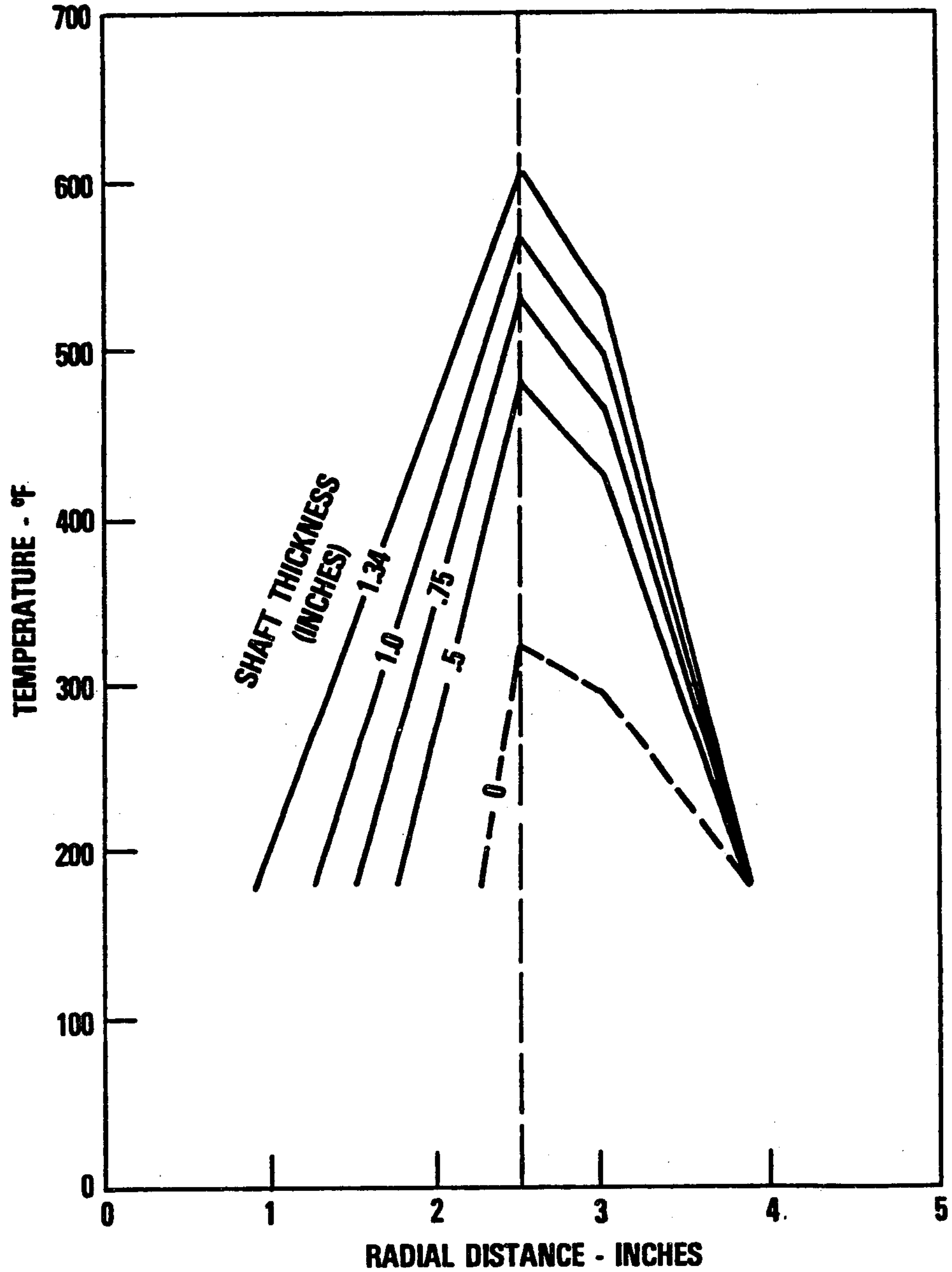

Figure 10 Effect of Shaft Thickness on Grafoil Bearing Pack Seal Temperatures

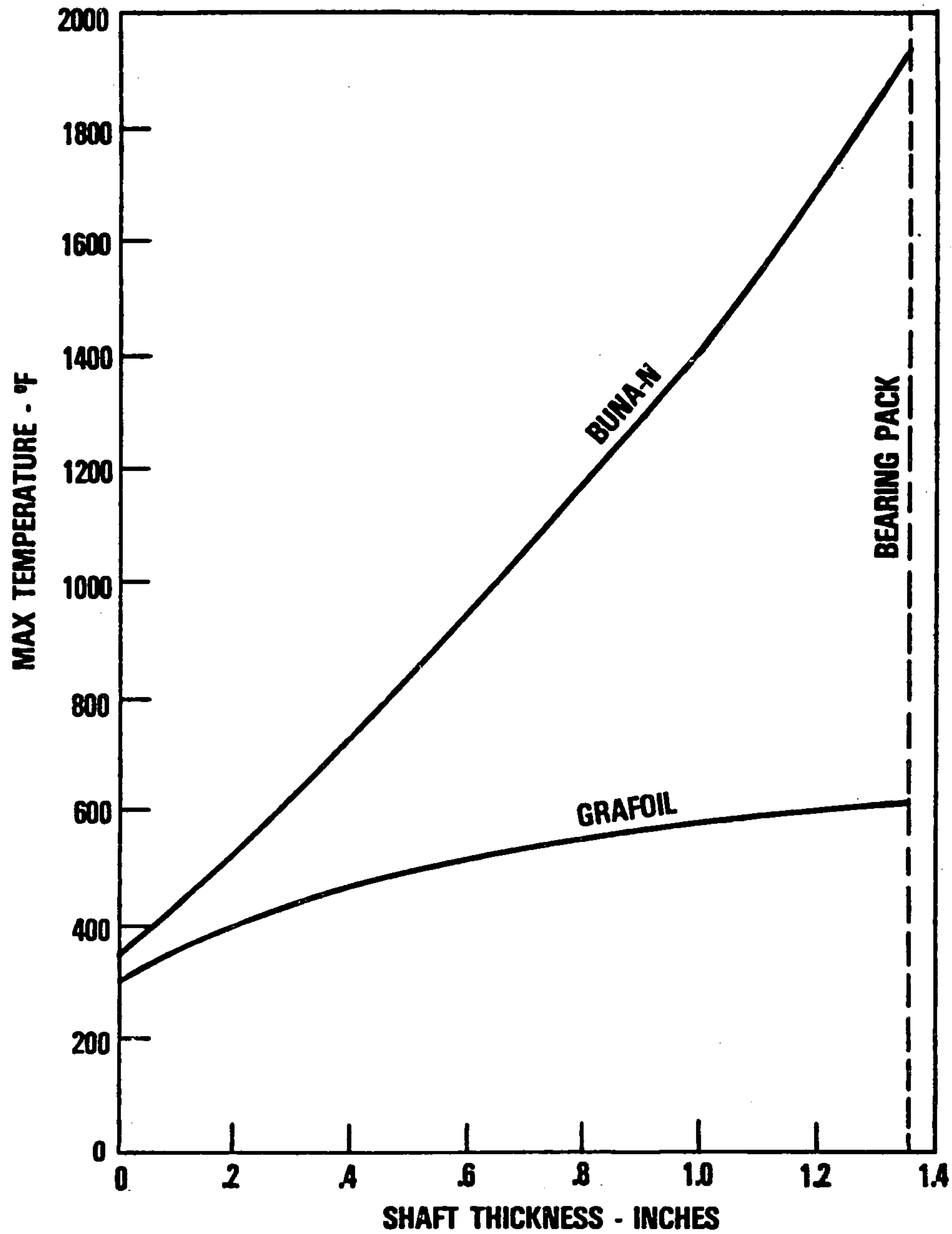

Figure 11 Effect of Shaft Thickness on Bearing Pack Peak Seal Temperatures (50 ft-lbs, 212 rpm)

Effect of Ambient Temperature

Ambient temperature chosen for the previous calculations is 180°F. Changes in ambient temperature have the effect of shifting the entire temperature profile by the incremental temperature change; the temperature profile is not distorted. For a given rate of heat generation, all temperature gradients will be the same for each ambient temperature condition. This point is illustrated by the computer data plotted in Figure 12.

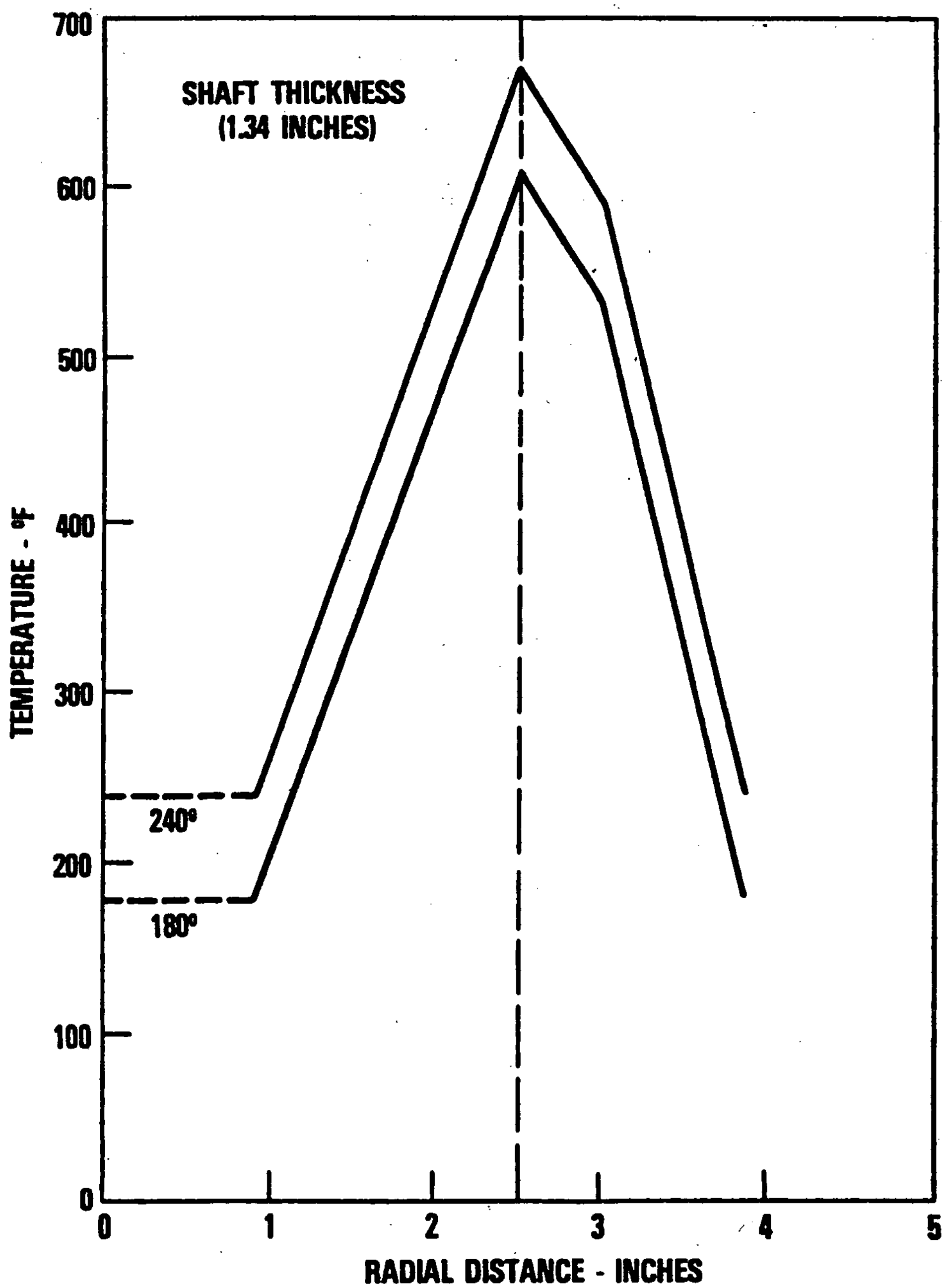

Figure 12 Effect of Mud Temperature on Bearing Pack Grafoil Seal Temperatures

Effect of Rotary Speed

The rate at which heat is generated at the sliding surface is proportional to rotary speed and friction torque (Equation 1). Torque is proportional to seal/shaft contact pressure and coefficient of friction (Equation 2). Calculations show that peak temperature increases linearly with each of these parameters.

For example, the maximum temperature in a Grafoil seal increases linearly from 180°F to 1,380°F as the rotary speed is increased from 0 to 600 RPM (Figure 13). At zero rotary speed, temperature is uniform through the seal assembly. These curves show why Buna-N seals have never been successful in high-speed drilling motors (300 to 1000 RPM).

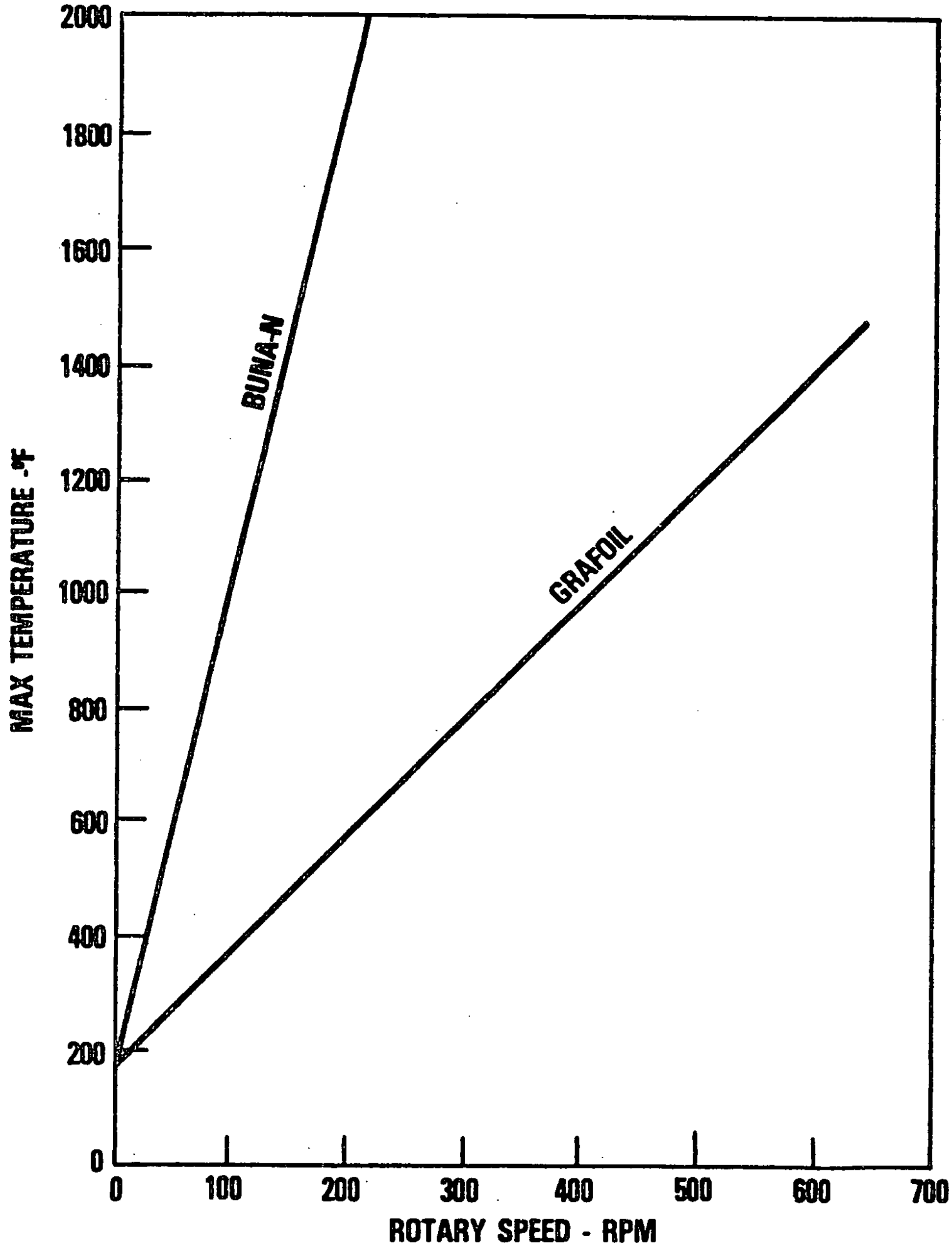

Figure 13 Effect of Rotary Speed on Bearing Pack Seal Temperatures

Effect of Pressure Drop

Seal friction torque is directly proportional to the contact pressure, p_c, between the seal and the rotating sleeve (Equation 2). Experience has shown that roller bit seals and other downhole rotary seals undergo essentially no wear due to the hydrostatic pressure in the wellbore and that the seal wear is directly related to the pressure drop across the seal. The assumption is therefore made that the contact pressure, p_c, between the seal and wear sleeve varies as:

$$p_c \approx \Delta p \qquad (6)$$

where Δp is the pressure drop across the bit. It was shown earlier that 1,500 psi pressure drop produced a torque of 50 ft-lbs. The friction torque at lower pressure drops will therefore approximately equal:

$$T \approx \frac{50}{1500} \Delta p = \frac{\Delta p}{30} \text{ (ft-lbs)} \qquad (7)$$

This relationship has been used to generate the curves in Figures 14 - 16. These curves show that reducing the pressure drop across the Buna-N seals from 1,500 psi to 375 psi reduces the peak seal temperature from 1,950°F to 622°F (Figure 14). Reducing the pressure drop across the Grafoil seals from 1,500 psi to 375 psi reduces the temperature from 607°F to 285°F (Figure 15). These curves show why motors utilizing Buna-N seals are currently limited to bit pressure drops of 100 to 300 psi.

In Reference 2, p_c was assumed to be 1,000 psi when the pressure drop across the seal was 1,500 psi ($p_c = 2/3\ \Delta p$).

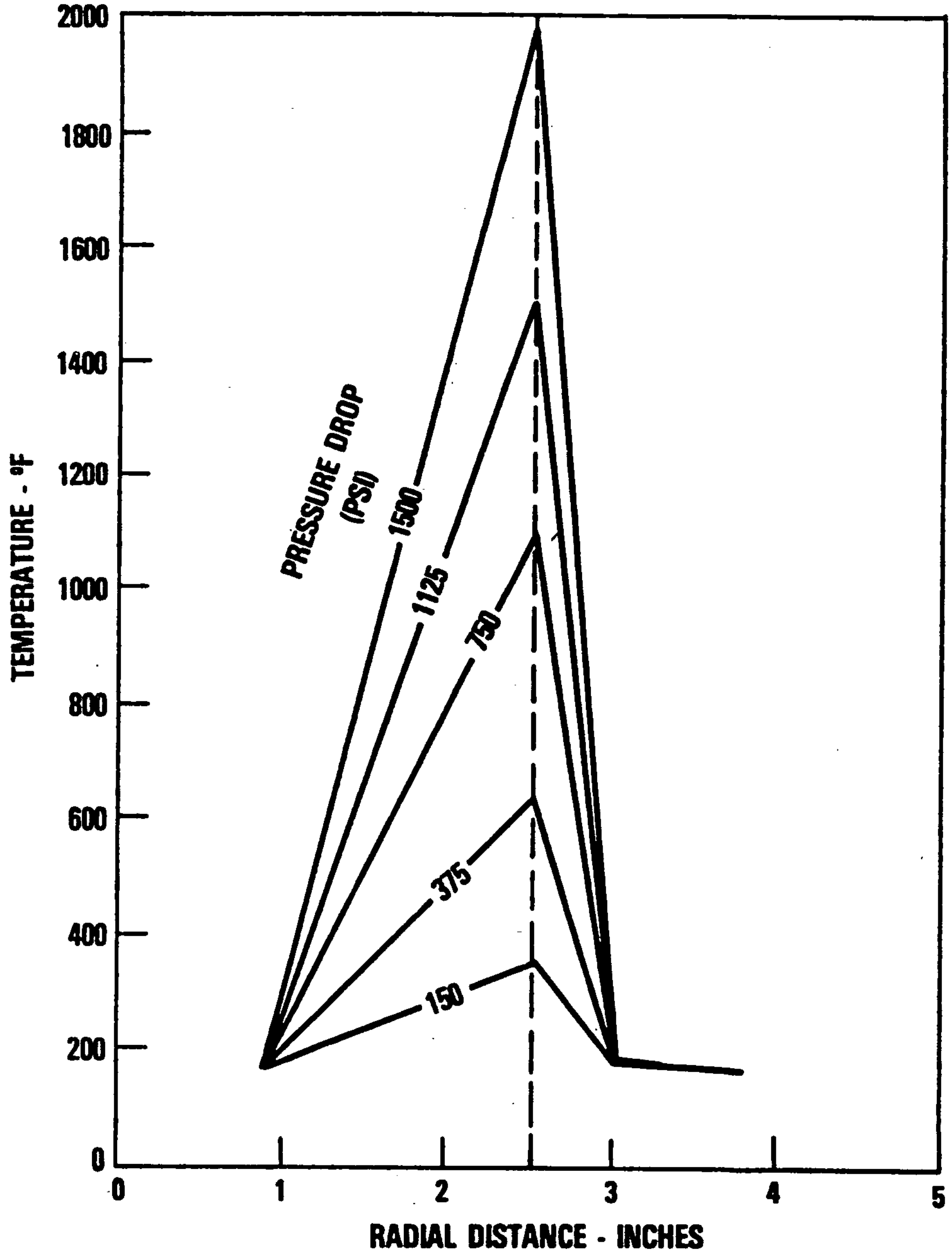

Figure 14 Effect of Pressure Drop on Buna-N Seal Temperature

This contact pressure satisfies Equation 2 when $\mu = 0.0695$ and the friction torque is 50 ft-lbs. As stated above, we are assuming in this report that the contact pressure and the pressure drop across the seal are equal (Equation 6). This corresponds to a coefficient of friction of 0.0463 with 50 ft-lbs torque. The temperature profiles are identical in both reports because the friction torques were equal.

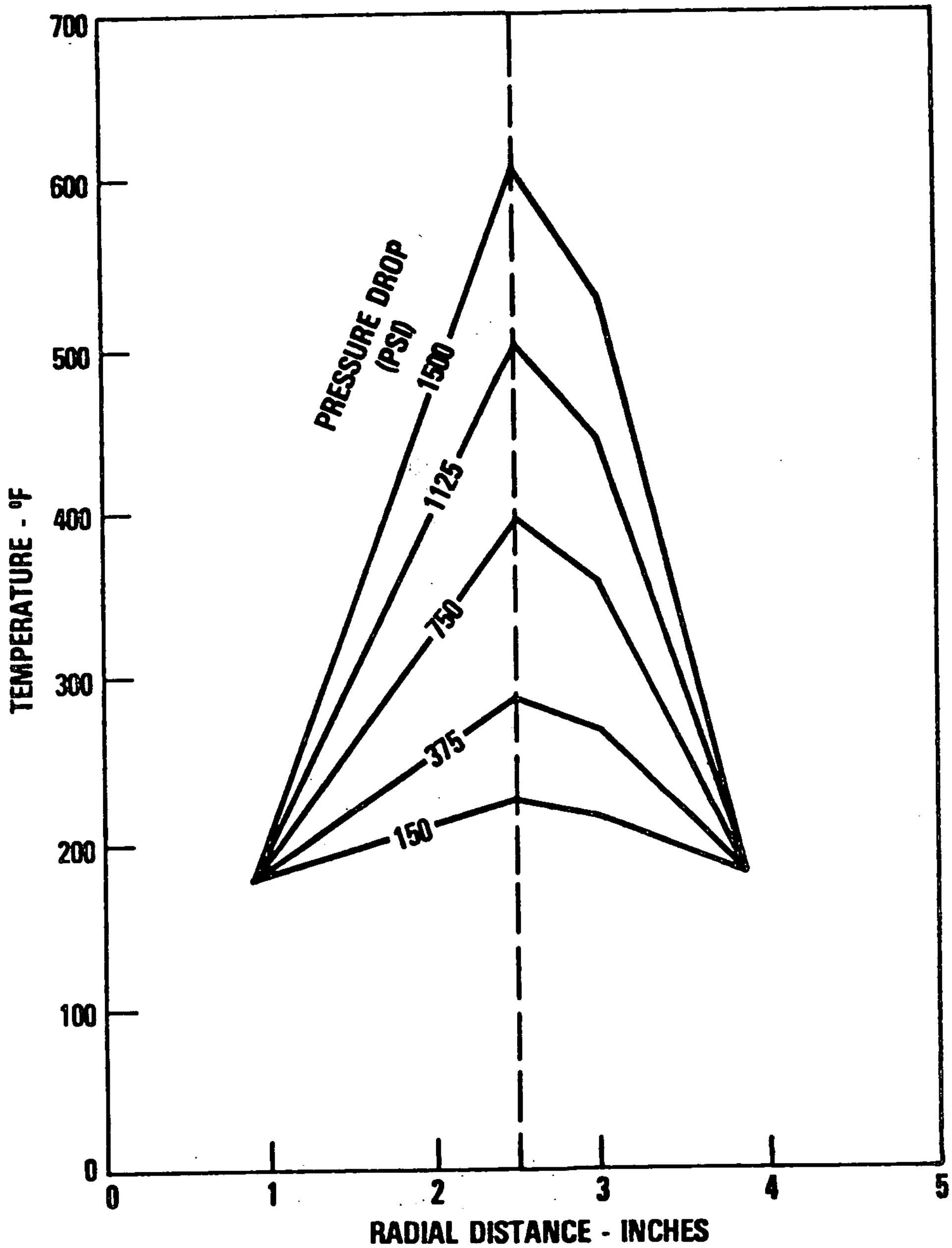

Figure 15 Effect of Pressure Drop On Grafoil Seal Temperature

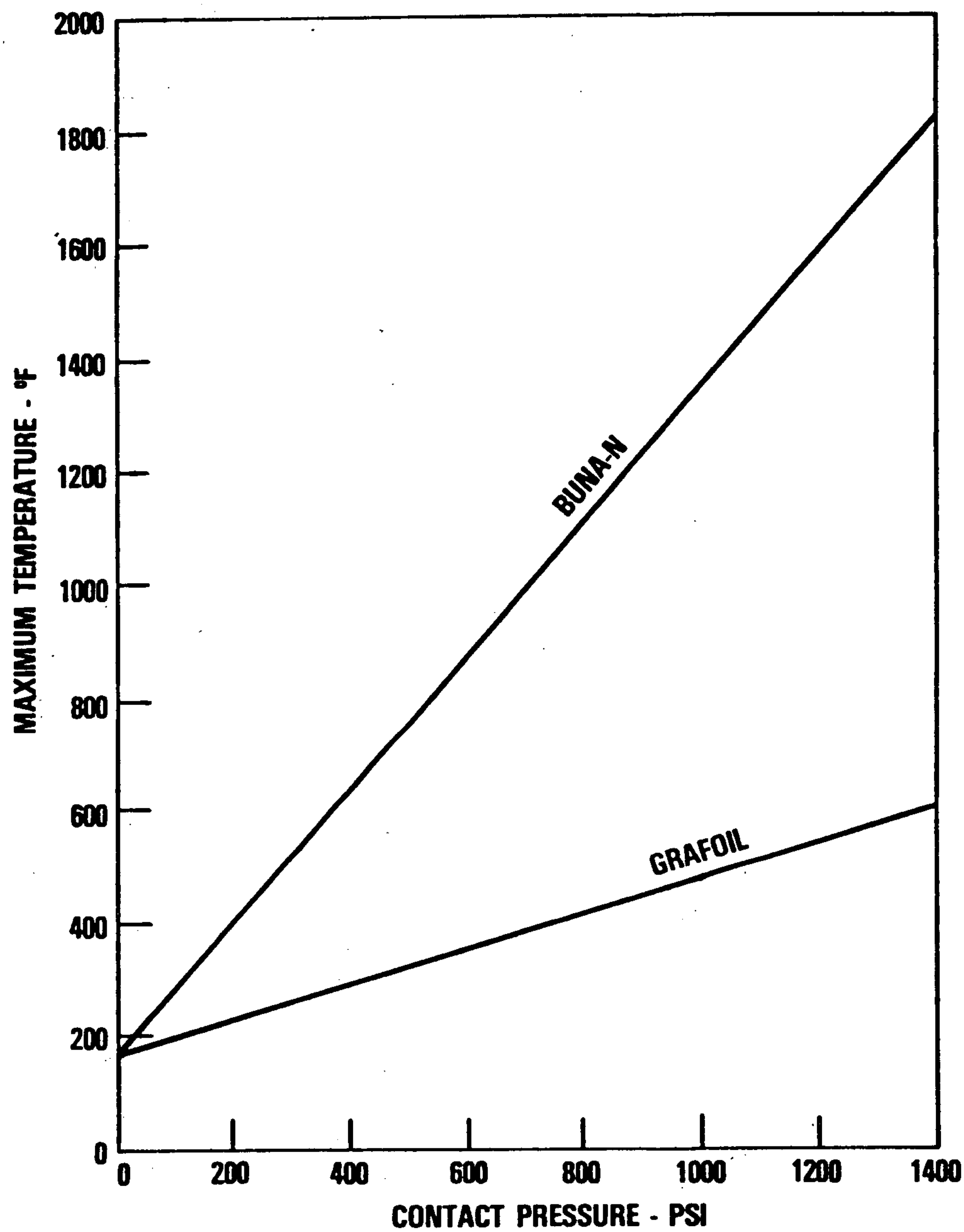

Figure 16 Effect of Pressure Drop On Bearing Pack Seal Temperature

SIMULATING DOWNHOLE CONDITIONS WITH SEAL TESTER

Our calculations show that the maximum temperature in rotary seal occurs at the seal/shaft sliding interface. This is also the location where shear stresses are generated and is no doubt the point where seal failure originates.

In order to simulate downhole operation conditions on a seal with the seal tester, it is important to establish similar interface temperatures and friction torques. For example, the temperature profile with a Grafoil seal corresponding to the bearing seal assembly (Figure 4) is shown in Figure 17 along with the temperature profile for the same rotary speed and friction torque in the Terra Tek seal tester (Figure 3). The maximum temperatures at the sliding interface are different for the two seal configurations because boundary temperatures and wall thicknesses are different. However, the Terra Tek tester can create the downhole maximum temperature by increasing rotary speed in the tester from 212 rpm to 410 rpm. The dashed profile in Figure 17 shows that the interface temperature in the laboratory test will be the same as the interface temperature in the bearing seal assembly. This simulation depends on the friction torque being fairly independent of rotary speed. Seal wear is not modeled by this simulation. Local shear stress and temperatures are.

The seal tester was designed with a heating element around the housing for the purpose of maintaining a fixed temperature in location T_3 (Figure 3). The thermocouple T_3 near the outside surface of the seal is maintained at a given level. The control temperature at T_3 has been set at 250°F for the test runs.

A simulation test has meaning only with regard to Grafoil seals. In which case it will be necessary to uncouple the temperature control mechanism so that the temperature at location, T_3, can seek its own level. The outside surface of the housing can be exposed to room temperature.

Alternatively, the outside surface of the housing can be elevated with the heating element in order to reduce the rotary speed required to generate the downhole maximum seal temperature in the seal tester.

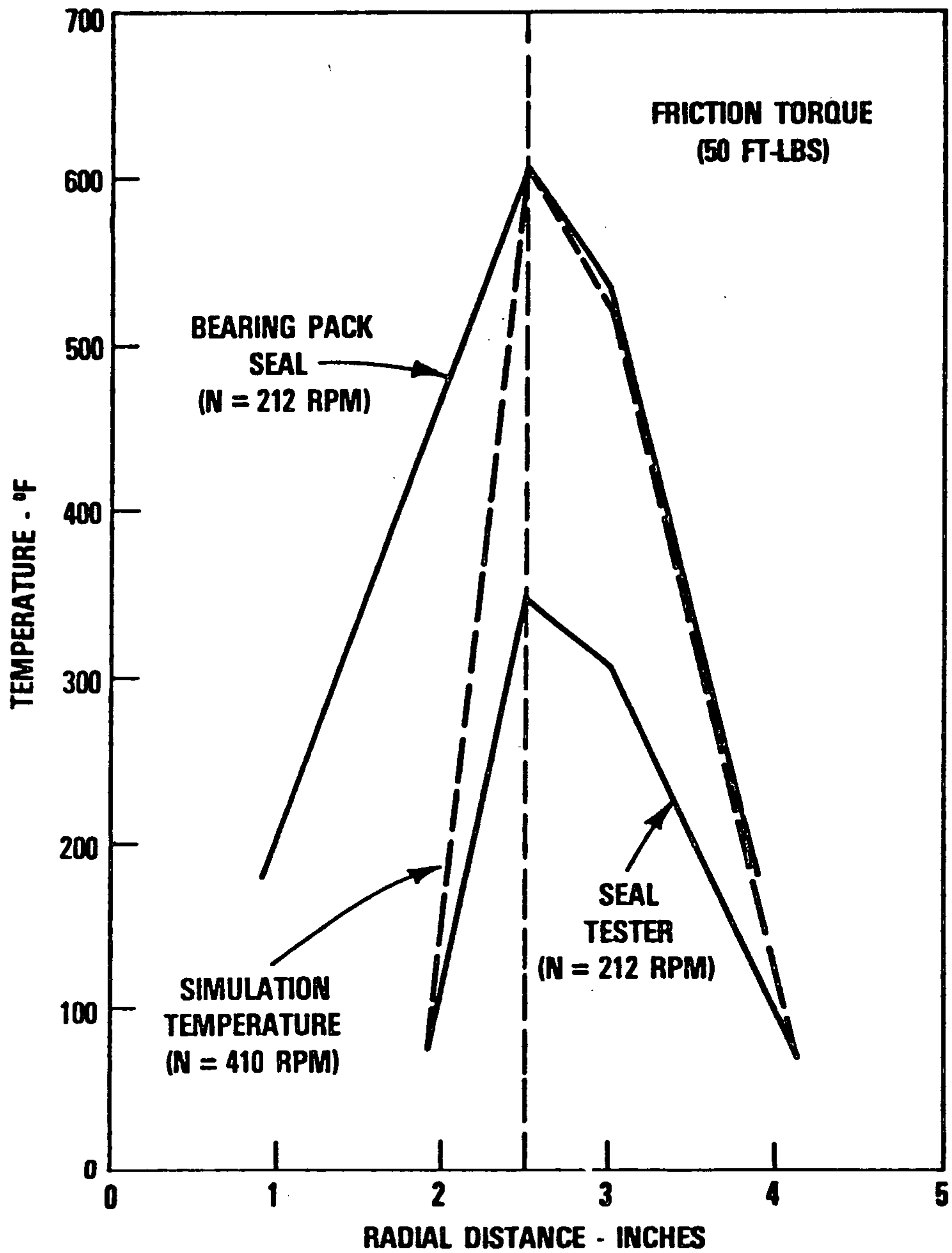

Figure 17 Simulating Bearing Seal Assembly Temperatures in Seal Tester

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RECOMMENDED BEARING PACK DESIGN

The previous section showed that peak temperatures are smallest when shaft wall thickness is zero. However, the 0.25 inch thick sleeve alone would not be strong enough to transmit the required torque and thrust to the drill bit. It is possible to achieve the same heat flow conditions through the sleeve while maintaining an inner shaft by channeling the drilling fluid to the sleeve as shown in Figure 18. This concept was developed by Dave Nagel with Maurer Engineering. It would be necessary to insert a choke or restriction in the shaft to develop a small pressure drop (e.g., 5 to 10 psi) across the channel to force fluid flow past the sleeve. Torque and bit weight would be carried by the shaft, allowing the sleeve to accommodate the sliding friction and heat flow. The temperatures produced with this design would correspond to the zero shaft thickness dashed curves in Figure 9 and 10.

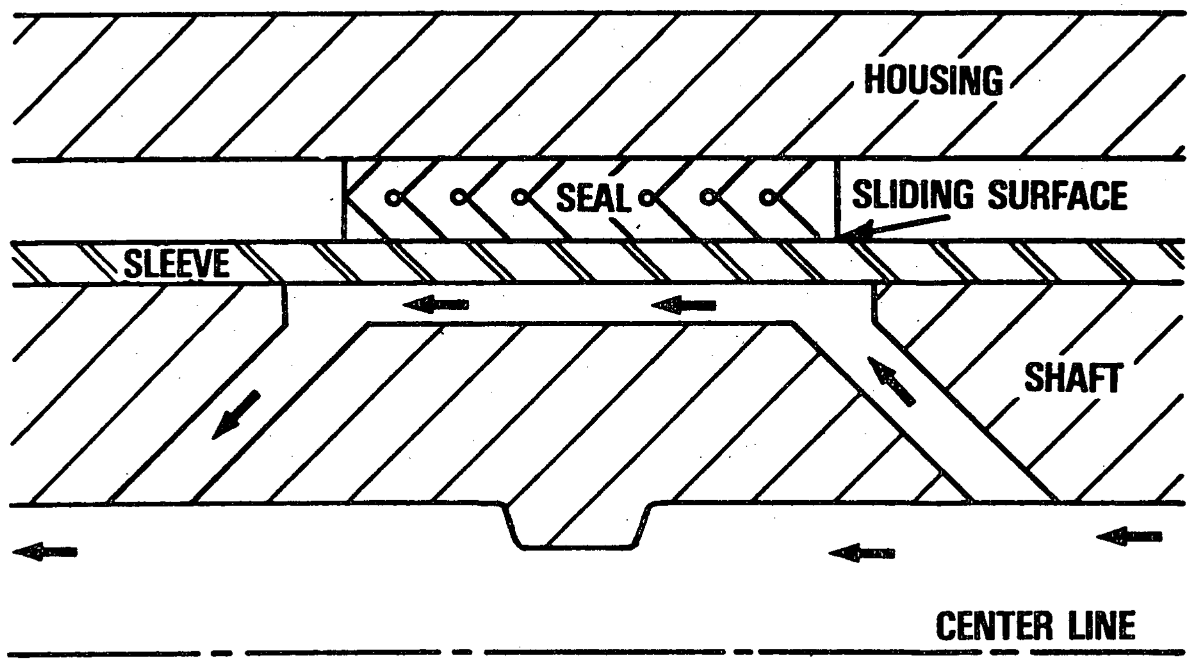

Figure 18 Recommended Bearing Pack Design

The peak seal temperatures could be further reduced by using a material with high thermal conductivity for the sleeve (Figure 19). The earlier curves all correspond to steel shafts and steel sleeves. While there is some reduction in peak temperatures with the higher thermal conductive sleeves, the improvement does not appear to be sufficient to warrant substituting phosphor bronze or beryllium copper for steel in the sleeves. Figure 13 also shows that if brass is selected as bushing material there is little difference between peak temperatures for Buna-N and Grafoil because nearly all of the heat flows to the drilling fluid inside the shaft through the sleeve and very little escapes to the outside through the housing.

A highly conductive sleeve (e.g., beryllium copper) used with the sleeve and shaft arrangement shown in Figure 4 would not substantially reduce peak temperatures because the thermal conductivity of the 1.34 inch thick steel shaft dominates the overall thermal conductivity of the sleeve/shaft combination.

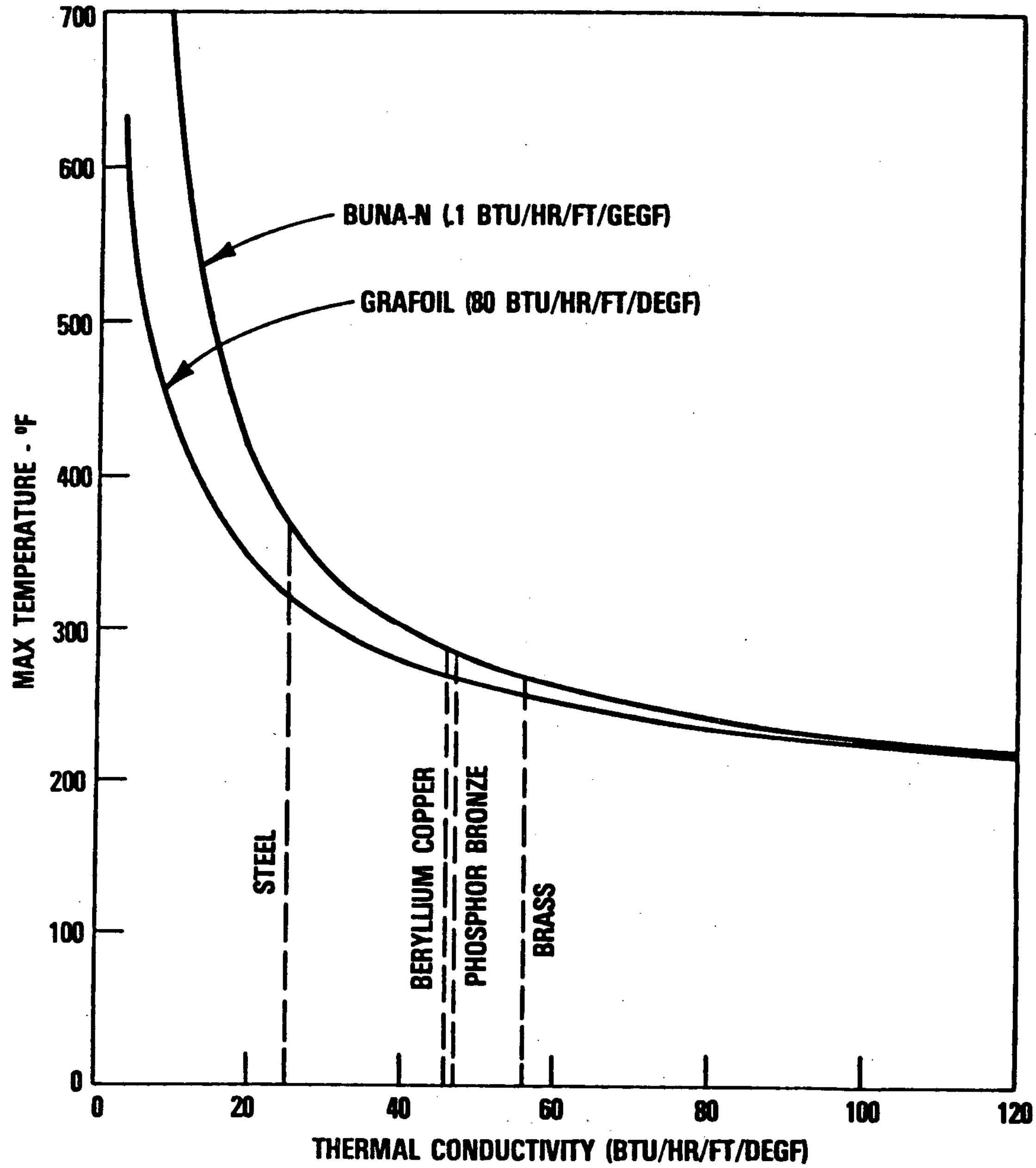

Figure 19 Effect of Sleeve Thermal Conductivity on Bearing Pack Seal Temperature (Zero Shaft Thickness)

CONCLUSIONS & RECOMMENDATIONS

The following conclusions were reached as a result of this study:

1. Rotay seals containing Buna-N rubber will not work in downhole motors because the seal temperatures exceed the 300 to 350°F temperature limitation of the Buna-N rubber.

2. Seal temperatures in the bearing pack will be much higher than those in the seal tester because the bearing pack shaft severely restricts heat flow from the seal.

3. Seal temperatures in Grafoil seals are much lower than in Buna-N seals because the higher thermal conductivity of the Grafoil seal allows more heat to flow through the seal to the mud in the wellbore annulus. At 212 rpm and 1,500 psi pressure drop, the bearing pack Grafoil seal peak temperature is 607°F compared to 1,950°F for the Buna-N seal.

4. Seal temperature increased rapidly with increased rotary speed and with increased pressure drop across the seal. For example, the Grafoil seal temperature increased from 784°F to 1,388°F as the rotary speed is increased from 300 to 600 rpm and from 393°F to 607°F as the pressure drop is increased from 750 to 1,500 psi.

5. Allowing drilling fluid to flow between the shaft and wear sleeve will reduce the temperature of the Grafoil seal from 607°F to 321°F when operating at 212 rpm and 1,500 psi pressure drop.

6. Future emphasis should be placed on developing Grafoil seals because of their high-temperature capabilities and good thermal conductivity.

7. The existing bearing pack should be redesigned to allow drilling mud to flow adjacent to the wear sleeve in order to reduce seal temperature.

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