Effects of Rock/Cutter Friction on PDC Bit Drilling Performance: An Experimental and Theoretical Study.

Ergun Kuru

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Effects of rock/cutter friction on PDC bit drilling performance: An experimental and theoretical study

Kuru, Ergun, Ph.D.
The Louisiana State University and Agricultural and Mechanical Col., 1990
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ABSTRACT

A new mechanistic drilling model for PDC bits was derived from the balance of forces acting at the PDC cutter. The model combined the torque and the drilling rate equations, cutter's geometry and rock properties.

A new understanding of frictionally generated heat between rock and PDC cutter is introduced. The numerical analysis revealed that neglecting the heat generated at the cutting surface area results in underestimation of the actual wearflat temperature by 10% to 530%, depending upon bit dull and downhole hydraulics.

The example bit performance comparison made by calculating the MBP curves showed 18% reduction of drilling rate when the new and more rigorous temperature limitation is used.

A new PDC bit wear model was derived and used for bit performance prediction. The model relates bit life with temperature, weight on bit, rotary speed, and cutter geometry. The predictions showed that the effect of friction dominates bit life, and this effect is greater than the effect of convective cooling.

A new laboratory instrument was constructed and successfully used to measure friction forces between...
sliding surface of a PDC cutter and the rock surface. Results showed that friction coefficient did not change considerably within the range of tested rock and fluid types.

The concept of maximum bit performance (MBP) curve was introduced. The curves represented the maximum values of average drilling rates for various pre-assumed footage values.

A new method for preparing a multi-bit drilling program, the dynamic drilling strategy, was developed. The dynamic drilling strategy provided the best combination of PDC bit runs to achieve the minimum drilling cost for a long borehole interval. The method was numerically compared to the conventional drilling optimization and to the field practices. Considerable savings of 25% and 60% were estimated, respectively.

Based on the drilling model, a new method was developed for the insitu measurements of the PDC bit condition and lithology change detection. The technique was verified by comparing the predicted and measured PDC bit wear and by showing the correlation between rapid formation changes and discontinuities in the diagnostic plots.

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INTRODUCTION

The development of modern rotary drilling bits goes back as early as 1880's, so as drag bit's. Since 1880, drag bits have undergone many improvements. Fishtail bits were among the first ones used to drill oil wells. They were forged from "Black Diamond" steel of 0.3 % carbon content.

The short fish tail or gumbo bit was highly valued by drillers since they could drill faster. Three-blade and four-blade bits provided more cutting edge and lasted somewhat longer. However, they would not drill as fast as the fish tail.

The development of the rock bit in 1909 helped in further improvement of metallurgy of fishtail bits by applying different types of water or oil quenching bit steel.

The use of hardfacing on fishtail bits in early 1922 was one of the outstanding developments for improving bit wear. Much progress had been made through the use of better steels and in tempering bits, but hardfacing provided a simpler way of rebuilding worn bits as well as improving wearing qualities. Bits were built up with oil tempered spring steel using an electric arc.
and then quenched with water at a red heat and allowed to cool. The method was improved upon almost immediately, using a high carbon rod coated with a mixture of tungsten carbide, manganese, chromium and carbon which was applied on the cutting surface.

Later, welded steel blade bits were introduced. They were made by welding individual blades to the bit body. Drag bits had been enhanced further by replacing the alloy steel heat treated blades set with tungsten carbide.

Although the drag bit was undergoing many improvements, its popularity was on decline due to the belief that the action of the bit on bottom contributed to crooked holes, particularly when the bit became dull.

The drag bits have always been considered ideal for the softer drilling areas. It is well known that in top shale and sandy shale formations, the drag type bit always outperformed other types of available oil well drilling bits. However, problem of hole deviation have probably been the main reasons of the reduced demands for this type bit.

The use of drag-type bits had reduced considerably by the end of 1960's. Improved designs and metalurgy, together with a better understanding and solution to hole deviation.
deviation problems had been expected to improve their use in later years.

In 1973, General Electric Company developed a technique to bind small synthetic diamonds together using diamond to diamond bond on top of a tungsten carbide substrate. As a result, a relatively large diamond cutter could be produced. These synthetic diamond cutters are called STRATAPAX or SYNDITE, were developed by General Electric and Dee Beers respectively (Fig. 1). The technique was initially used to manufacture diamond compacts for machining and dressing purposes. The application was successful and these diamond blanks were also introduced into the drilling bit industry. As a consequence, a new type of drill bit called Polycrystalline Diamond Compact (PDC) had been developed (Fig. 2).

Polycrystalline Diamond Bits are a high tech revival of the drag bits. By using the state-of-the-art materials, drag bits have been competitively reintroduced into oil well drilling industry. The PDC bit consists of a solid bit head onto which multiple PDC cutters are attached. The configuration of a PDC bit can vary in shape, size, number and placement of cutters, and hydraulic design. These and other variables determine the performance of a bit in a given application.
Figure 1 - Single PDC cutter, showing front and side view. Circular diamond compact on this cutter is 0.5 inches in diameter.
Figure 2- Collection of PDC bits, showing variations in bit shape and location and number of cutters.
PDC bits have proven to be long lasting bottomhole tools because of the absence of moving parts and the high wear resistance of synthetic diamond. Consequently, in many drilling applications, they were introduced as an alternative to roller bits. The successful application of PDC bits in the field also revealed several problems which created a need for research. It has been recognized that the synthetic diamond compact material deteriorates rapidly (loses strength and abrasion resistance) at temperatures exceeding 350 °C. Therefore, PDC bits failed to drill in hard, abrasive formations such as granite or quartz. Such formations generate too much friction at the rock/cutter interface for the convective cooling of drilling mud to keep operating temperatures within the 350 °C threshold. PDC cutters' behavior were investigated mathematically as well as experimentally.

RESEARCH DEVELOPMENT

This research emerged as a part of the general effort to improve the performance of PDC bits. A flowchart summarizing the research development is given in Fig.3.

This research began in 1985 when the problem of verifying a new optimization theory for a minimum-cost-drilling, dynamic drilling strategy, was realized. In order to show the impact of the new optimization
Figure 3- Flowchart of the research development
concept on drilling cost reduction, a mathematical tool, (drilling model) formulating the bit behavior was needed. Therefore, a new drilling model including the drilling rate, bit life, and the torque relationships was developed. Model development required an understanding of the cutter/rock interaction mechanism through the analysis of forces at the cutter/rock interface. At this early stage, a new field data interpretation method (to detect lithology change and to evaluate bit dull condition) was also verified by using the new drilling model.

Presence of friction forces at the rock/cutter interface suggests that heat is generated at these contact areas. The next step was to determine the effect of frictionally generated heat on the cutter performance. Based on the proposed cutter/rock interaction mechanism, an additional source of heat generation at the cutting surface area was introduced to the model which evaluates the thermal response of a PDC cutter under downhole drilling condition.

Having seen the effect of frictionally generated heat on the bit performance, the next step was to introduce temperature into the model which explains frictional wear of a PDC cutter. The result was a closed form relationship which relates the bit operational
life with weight on bit, rotary speed, bit geometry, drilling fluid flow rate and temperature.

Results of parametric studies using the new bit life model showed that friction has a strong influence on the bit operational life. Therefore, the last part of the research was designed as an experimental study of mechanism of friction induced by PDC cutters during rock cutting.

RESEARCH ORGANIZATION

At first an analysis of PDC drilling mechanics and a mathematical model of drilling with PDC bits will be presented in chapter one. A mechanistic approach is used throughout the derivation of the drilling rate, the torque and the bit life relationships. The model is fully explicit with physical meaning given for all constants and functions. The model is then used to define forces active at the rock/cutter interface and to investigate phenomena of frictionally generated heat and wear.

Previous laboratory studies revealed that a vast amount of heat is generated due to the shearing of the rock surface by a PDC cutter. High temperature encountered by a PDC cutter during drilling destabilizes the diamond structure causing a severe reduction in its operational
life. A review of the previous research on the thermal response of a PDC cutter under downhole drilling conditions is given in chapter two.

Analysis of previous models showed that they didn't consider all possible sources of heat into the cutter. Therefore, they may underestimate the cutter temperatures. Chapter 2 will present a simulation study to evaluate the combined effect of cutting depth (drilling rate) and wear (bit dull) on the thermal response of polycrystalline diamond compact (PDC) cutters under down-hole drilling conditions. A new understanding of frictionally generated heat between rock and PDC cutter will be introduced from the analysis of forces active on the wearflat and the cutting (leading) surfaces of a cutter. Then this new concept will be used to predict PDC bit performance.

The strong influence of frictionally generated heat on the PDC cutter's wear suggested that any realistic bit performance prediction model should include the temperature together with other drilling variables. Experimental data from previous studies show a possibility of an empirical relationship between volumetric cutter wear rate and the cutter wearflat temperature. By introducing this relationship into the bit life equation (derived in chapter one), a modified PDC bit

xxx
wear model for performance prediction can be developed, as presented in chapter three.

The new bit life model relates bit life with temperature, weight on bit, rotary speed, cutter geometry and frictional heat fluxes through wearflat and cutting surfaces. The predictions shows that the effect of friction dominates bit life and this effect is stronger than that of the convective cooling. A parametric study to determine the sensitivity of modified wear model to downhole drilling conditions shows that any possible reduction on the magnitude of friction coefficient (through the enhancement of drilling fluid chemistry) will elongate the PDC cutters' life. In this respect, a better understanding of friction phenomenon together with lubrication by drilling fluid is needed. This issue is pursued in chapter 4.

Chapter four presents a prototype laboratory instrument designed and fabricated to measure friction coefficient between the sliding surface of a PDC cutter and the surface of a instantaneously-cut rock in the presence of drilling mud. The effect of lithology and drilling fluid type on the friction forces will be investigated. Theory of friction at the rock/cutter interface and its related heat generation will be modified
in the light of experimental findings.

In chapter five, the drilling model was used to verify a new drilling optimization theory. The concept of maximum bit performance (MBP) curve was introduced. The MBP curve represents a relationship between any possible footage made by a single drill bit and its maximum drilling rate. It provides a single fundamental relationship necessary for drilling optimization of any kind. A new method for preparing a multi-bit drilling program, the dynamic drilling strategy, was also developed in chapter five. The dynamic drilling strategy provided the best combination of PDC bit runs to achieve the minimum drilling cost for a long borehole interval.

An effort will be made to simplify the drilling model as much as possible in order to make its practical use more convenient without losing the accuracy of model's prediction. Based on the drilling model, a simple procedure will be developed in chapter six to evaluate an instantaneous bit wear as well as to detect rapid changes in lithology of formations drilled. The technique will be verified by comparing the predicted and the measured PDC bit wear from the MWD records in the Gulf Coast area. The importance of new method lies in the MWD software development for the purpose of the in-situ rock detection and PDC bit evaluation and control.
CHAPTER I.

SINGLE-CUTTER MECHANICS APPLIED TO PDC BIT DRILLING
MODEL

ABSTRACT

A new mechanistic drilling model for polycrystalline diamond compact (PDC) bits was presented. The model was derived from the balances of forces acting at the PDC cutter and it is fully explicit with physical meanings given for all constants and functions.

The model combined the torque, the drilling rate and the bit life equations, cutter's geometry and rock properties. The response of the drilling model to weight on bit and cutters removal and the stability of constants were tested using the laboratory drilling data from several research reports as well as the field drilling data collected by the author.

INTRODUCTION

Since 1960 when Galle and Woods presented their mathematical model of rockbits [1], [2], there have been several models developed [3], [4], [5], [6]. Most of the models were purely empirical, based on arbitrarily selected functions and curve fitting. Later developments included the effect of hydraulics on drilling rate [7],
[8], the effect of overburden pressure [4], and the effect of differential pressure on drillability [9]. The number of empirical constants increased from three [1] to eight [4] and both dimensional analysis [10] and multiple regression analysis were used for their estimation. Also, several attempts were made to correlate single tooth rock interaction with rock bit performance [11], [12] but the conclusion were mainly qualitative. The stochastic nature of the rockbit cutting action precluded any attempt to formulate a mechanistic model.

The polycrystalline diamond compact bits do not have the difficult kinematics of the rock bits; in addition, their cutting action is less complex, thus they can be effectively modeled with the mechanistic rather than the empirical approach. To date a few predictive models have been proposed [13], [14],[15],[16],[17],[18].

Ziaja [13] developed a mathematical model of a single PDC cutter penetration assuming a circular cut and absence of cutters interaction. The model was designed for a core bit. The single proportionality constant was used to up-scale from the single cutter load and penetration to the weight on bit and drilling rate. Data from one field run of a core bit was used to verify the model.

Glowka [14] used experimental data from laboratory
drilling in hard rocks and developed a power function correlation between cutter penetration and stress at the wearflat area. His analytical work also included derivation of the single cutter wear equation as a function of penetration per rotation and footage. He used the model to analyze response of a single cutter to various input loads. He also analyzed distribution of wear, penetration and load across the bit face. He suggested that the PDC bit performance should be calculated by integrating performance of all individual cutters.

Warren and Sinor [15], [16] developed a computer program which rigorously calculated individual cutter's performance based on observed drilling rate, cutter position and its geometry. Wear and load were calculated for each cutter, then they were summarized and compared to the observed values of bit weight and torque. To date, program has been successfully verified by using experimental full-scale drilling results. The program seems to be an effective simulator needed by PDC bit manufacturers to investigate various bit design configurations.

In this study, it was aimed to develop a mathematical model for the purpose of optimum drilling programs [17] and interpretation of drilling data [18]. The model requires a small amount of input data regarding
Figure 1 - Definition of the forces at the PDC cutter.
Figure 2- Balance of normal forces - Mechanical model
the bit geometry and its dull condition. Such an approach calls for explicit expressions for all the drilling variables measured on the rig site such as bit weight, rotary speed, drilling rate, torque, and the bit dull. An effort is made to simplify the model as much as possible in order to make its practical use more convenient without losing the accuracy of model's predictions.

MODEL ASSUMPTIONS AND FUNDAMENTAL RELATIONSHIPS

The model is derived from the analysis of forces active at the single cutter, as shown in Fig. 1 and 2. The bit life, the drilling rate, and the bit torque equations are deduced from the static balance of forces for a cutter moving through the rock at a constant angular velocity. The assumptions made throughout the derivation of the model are as follows:

1. Formation rock behaves plastically, i.e. rock deforms without losing its cohesion. In other words, rock resistance to pressing(or cutting) is proportional to the contact area and it does not depend upon cutter's penetration (vertical) or displacement (horizontal).

2. Bottomhole profile is predominantly parallel to the bit profile as a result of the cutters interaction.

3. There is a mechanical similitude between
the single cutter and the entire bit;

a) The normal force acting on a single cutter is proportional to the weight on bit with proportionality constant $k_1$.

b) The drilling rate is proportional to the cutter penetration with proportionality constant $k_3$.

4. The cutting angle, $\alpha_c$, is ignored, i.e. cutter moves in the direction perpendicular to its stud axis.

5. Volumetric wear of a PDC cutter is proportional to the friction work with proportionality constant $k_2$.

6. Inadequate bottomhole cleaning results in a non-linear response of the drilling rate to the rotary speed.

7. Friction at the cutter side surface is ignored.

Assumption 1 is based on the fact that many rocks exhibit ductile behavior at confining pressures corresponding to depths usually reached during drilling [19-21]. Triaxial compression experiments at relatively low confining pressures indicated that many sedimentary rocks can undergo deformations without losing cohesion [22]. In general, the rock failure mechanism is brittle at low confining pressures and it is ductile at high
confining pressures, with transition from predominantly brittle to predominantly ductile behavior. It is also known that the macroscopic rock failure mode is controlled by the nominal effective stresses, i.e. by the difference between confining pressure and pore pressure [23-26]. As the nominal effective stress increases with depth, the rock deformation mode becomes more plastic. In addition, for soft and medium soft formations (shales, carbonates), the angle of internal friction decreases at a higher confining stress, thus increasing the contribution of cohesive resistance to the shear strength [27].

The simplified mechanical model of assumed rock deformation mechanism is shown in Fig. 2. The simplification is supported by the experimental measurements of forces at a single cutter [28]-Fig. 3. It should be noted that the measurements were taken from laboratory drilling experiments performed under atmospheric conditions. The zone of force variation (chipping) represents consecutive cycles of forming new fractures. It is expected that at the higher confining pressure the amplitude of stress fluctuations would reduce even further, therefore, it could be averaged by some constant value as shown in Fig. 4. This value represents rock resistance to cutting or pressing.

Examination of the bottomhole pattern [15] for several different bit types showed that cutters make a
Figure 3- Measured cutter's forces (top figure, after [28]) and their interpretation (bottom figure).
Figure 4- Physical interpretation of plastic rock resistance to cutting and its relation to Mohr's theory.
smooth cut at the boundary and there was no breakout between cuts (assumption 2).

Assumption 3 implies that bit behavior can be inferred from a single cutter behavior. Though commonly made in bit modelling (except for one research [15],[16]), such an assumption is quite controversial, specifically for PDC bits. Cutters placement and the bit profile are, at present, subject to experimentation by bit manufacturers. The most rigorous approach to the modelling of PDC bits is, without doubt, to measure the individual cutters position and orientation and to mathematically integrate individual cutters’ performance across bit face. Such an approach considers the uniqueness of bit geometry ("bit fingerprint") and it can effectively detect differences between behavior of individual bits.

Field applications, however, call for standardization, i.e. selecting features which are common in large group of bits. The recent IADC classification of the fixed cutter bits geometry, using a three by three matrix system, is a good example of generalization made at the expense of accuracy [29].

In addition, there is already some experimental evidence supporting assumption 3. The proportionality between drilling rate and cutter penetration is not just a
Assumption 3A, regarding bit weight, $W$, being linear function of cutter normal force, $F_N$, can be expressed as:

$$W \sim F_N$$

Also, the linear relationship between cutter penetration, $h$, and the normal force, $F_N$, as it is revealed from many laboratory experiments, can be written as:

$$h \sim F_N$$

The example of experimental data supporting theoretical relationship between $h$ and $F_N$ is shown in Fig. 5. Moreover, there is massive laboratory and field data [30] showing a linear relationship between drilling rate and bit weight, similar to that shown in Fig. 6. Thus,

$$R \sim W$$

The three linear correlations above are simply equivalent to the linear relation between drilling rate and cutter penetration as follows;

$$R \sim h$$

or to the assumption 3B. In view of the need for simplicity, assumption 3 is considered the very first
Figure 5- Correlation between thrust force and cutting depth, after [30].
Figure 6- Drilling rate as a function of weight on bit, after [30].
attempt to formulate an explicit model for PDC bits. Extensive verification, sensitivity analyses, as well as further development of the model still remains to be seen.

Assumption 5 regarding proportionality between volumetric wear and the work of friction is commonly made when some grinding mechanism is involved. Specifically for PDC cutters, there is an additional evidence of the frictional nature of their wear. It was experimentally shown that the wear rate of the PDC drill blanks was much greater for dry cutting than for wet cutting [30]. This fact implies that the basic cutting mechanism for PDC cutters is shearing action, and that the friction work predominantly contributes to cutters wear.

Assumption 6 draws from several laboratory and field observation, showing a linear effect of rotary speed on drilling rate [30], [31]. It also implies that single cutter penetration is independent from its linear velocity, when bottomhole cleaning is sufficient. In the full scale tests, however, the effect of rotary speed on drilling rate was often nonlinear. Such discrepancy was usually attributed to poor hole cleaning in field operation as opposed to the laboratory tests.

Assumption 7 is supported by the laboratory experiments which revealed that side forces are in the range of less than 10% of the thrust forces [30]. Thus,
the frictional drag, resulting from the side forces, becomes insignificant.

The equilibrium of forces for the typical PDC cutter geometry (Fig. 1) defines the components of the normal and the tangential forces considered in the model. The normal force is a distributed force across cutting surface area and the wearflat. Its distribution is simplified using the mechanical analog shown in Fig. 2. Plastic behavior of rock under the PDC cutter is assumed. Rock behaves in such a way that its elastic limit equals plastic yield and, when it is exceeded, the rock deforms continuously under the action of the PDC cutter until a new balance is reached. The balance of all interacting forces between the rock and the cutter is shown in Fig. 7-A. For small values of cutting depth, the cutting angle, will be eliminated as shown in Fig. 7-B.

The horizontal and the vertical forces balance equation can be written by using Fig. 1 and 7-A as

\[ F_N = F_c \sin \alpha + F_{fc} \cos \alpha - F_w \cos \alpha_c + F_{fw} \sin \alpha_c \]  
\[ F_T = F_c \cos \alpha - F_{fc} \sin \alpha + F_{fw} \cos \alpha_c - F_w \sin \alpha_c \]  

where

\[ F_c = R_c A_c \] ; \[ F_{fc} = \mu F_c = \mu R_c A_c \]  
\[ F_w = R_p A_w \] ; \[ F_{fw} = \mu R_p A_w \]  

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Figure 7 - Balance of all forces at PDC cutter.
For small values of cutting depth \((h < 0.1 \text{ in.})\) the value of the cutting angle, \(\alpha_c\), is very small. After ignoring the cutting angle and substituting (3) into (1) and into (2), we obtain:

\[
F_N = R_c A_c (\sin \alpha + \mu \cos \alpha) + R_p A_w
\]  

(4)

\[
F_T = R_c A_c (\cos \alpha - \mu \sin \alpha) + \mu R_p A_w
\]  

(5)

The horizontal force at the cutter, \(F_H\), is related to the tangential force as

\[
F_H = F_T \cos \beta^{-1} \equiv F_T
\]  

(6)

because the side rake angle, \(\beta\), is as small as 5 deg.

**DERIVATION OF WEARFLAT AREA EQUATION**

Shape of the cutter wearflat is defined as an ellipse (Fig. 8). Elements of the ellipse are defined in terms of PDC cutter terminology as follow:

\[
2a = \frac{d_c}{\sin \alpha}
\]  

(7)

\[
2b = d_c
\]  

(8)

The distance \(d_1\) gives the amount of worn cutter diameter and is written in terms of linear cutter wear, \(x\), as follow:
The distances $l$ and $t$ are given as follow:

\[ l = \sqrt{d_c \frac{x}{\cos \alpha} - \frac{x^2}{\cos^2 \alpha}} \tag{10} \]

\[ t = a \sqrt{1 - \frac{l^2}{b^2}} \tag{11} \]

The equation of an ellipse is given as:

\[ \frac{t^2}{a^2} + \frac{l^2}{b^2} = 1 \tag{12} \]

Where $t$, $l$, $a$, $b$ are defined in Fig. 8. The wearflat area (shaded part of the ellipse in Fig. 8-B) is geometrically defined as:

\[ A_w = (ab \arccos \frac{t}{a}) - tl \tag{13} \]

By introducing (7), (8), (9), (10), (11), and (12) into (13) wearflat area is finally given as;
Figure 8 - Definition of the cutter wearflat area.
\[ A_w = \frac{d_c^2}{4 \sin\alpha} \sqrt{\frac{1 - \frac{xd_c - x}{\cos\alpha}}{\cos\alpha}} \]

\[ = \frac{d_c}{2 \sin\alpha} \sqrt{1 - \frac{(d_c x - \frac{x^2}{\cos\alpha})}{\cos^2\alpha}} \left( \frac{d_c x - \frac{x^2}{\cos\alpha}}{\cos^2\alpha} \right) \]

Equation (14) is valid for \( x < d_c \cos\alpha \).

Define a dimensionless cutter wear as follow:

\[ w = \frac{x}{d_c \cos\alpha} \] (15)

by introducing (15) into (14).

\[ A_w = \frac{d_c^2}{4 \sin\alpha} \left[ \sqrt{\arccos(1-4(w-w^2))} \right. \]

\[ - 2\sqrt{w - 5w^2 + 8w^3 - 4w^4} \] (16)

Define dimensionless wearflat area as:

\[ A_{wd} = \frac{A_w}{\left( \frac{\pi d_c^2}{4} \right)} \] (17)
Finally, by using (16) and (17), dimensionless wearflat area is given as:

\[ A_{wd} = \frac{1}{\pi} \sin \alpha \left( \arccos \sqrt{1 - 4(w-w^2)} \right) \]

\[ - 2\sqrt{w-5w^2+8w^3-4w^4} \]  

(18)

**DERIVATION OF CUTTING AREA EQUATION**

The model assumes that PDC cutters are interacting such that bit works always on flat bottom, i.e., a complete removal of rock after each turn of the bit is assumed.

Equation for a segment of a circle is given as:

\[ P = \sqrt{(2l)^2 + \frac{16}{3}c^2} \]  

(19)

Area of a segment of a circle is also given as:

\[ A = \frac{1}{2} (Pr - 2l(r-c)) \]  

(20)

where \( r \) is the radius of the circle. Thus, as it is seen from Fig. 9:

\[ A_c' = \frac{1}{2} \left( P_2 \frac{d_2}{2} - 2l_2 \left( \frac{d_2}{2} - \frac{(h + x)}{\cos \alpha} \right) \right) \]

\[ - \frac{1}{2} \left( P_1 \frac{d_1}{2} - 2l_1 \left( \frac{d_1}{2} - \frac{x}{\cos \alpha'} \right) \right) \]  

(21)
Figure 9 - Definition of the cutting surface area.
On the other hand, cutting area is defined as:

\[ A_c = \cos \beta \cos \alpha \ A'_c \]  \hspace{1cm} (22)

or, by combining (21) and (22)

\[ A'_c = \frac{\cos \beta \cos \alpha}{2} \left( \frac{1}{2} \left[ P_2 \frac{d_c}{2} - 2l_2 \left( \frac{d_c}{2} - \frac{(h + x)}{\cos \alpha} \right) \right] \right) 
\]

\[ - \frac{1}{2} \left[ P_1 \frac{d_c}{2} - 2l_1 \left( \frac{d_c}{2} - \frac{x}{\cos \alpha} \right) \right] \]  \hspace{1cm} (23)

where;

For \( (x + h) < 0.5 \ d_c \ \cos \alpha \)

\[ c_1 = \frac{x}{\cos \alpha} \]  \hspace{1cm} (24)

\[ c_2 = \frac{x+h}{\cos \alpha} \]

\[ l_1 = \sqrt{\left( \frac{d_c}{2} \right)^2 - \left( \frac{d_c}{2} - c_1 \right)^2} = d_c \sqrt{(w-w^2)} \]  \hspace{1cm} (25)

\[ l_2 = \sqrt{\left( \frac{d_c}{2} \right)^2 - \left( \frac{d_c}{2} - c_2 \right)^2} = d_c \sqrt{(y-y^2)} \]
\[ P_1 = \sqrt{(2l_1)^2 + \frac{16}{3} c_1^2} = 2d_c \sqrt{w + \frac{1}{3} w^2} \]  
\[ P_2 = \sqrt{(2l_2)^2 + \frac{16}{3} c_2^2} = 2d_c \sqrt{y + \frac{1}{3} y^2} \]  

For 
\[ (x+h) > 0.5 \ d_c \cos\alpha \]

\[ c_1 = d_c - \frac{(x+h)}{\cos\alpha} \]  
\[ c_2 = d_c - \frac{x}{\cos\alpha} \]  

\[ l_1 = d_c \sqrt{y - y^2} \]  
\[ l_2 = d_c \sqrt{w - w^2} \]  

\[ P_1 = 2d_c \sqrt{\frac{4-5y+y^2}{3}} \]  
\[ P_2 = 2d_c \sqrt{\frac{4-5w+w^2}{3}} \]  

and also,

\[ y = w + \hat{h} = \frac{x}{d_c \cos\alpha} + \frac{h}{d_c \cos\alpha} \]  

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Finally, by introducing equations (24) through (30) into (23), cutting area is obtained as follow:

For \((x + h) < 0.5\) \(d_c \cdot \cos \alpha\)

\[
A_c = \frac{\cos \beta \cos \alpha}{2} \frac{d_c^2}{2} \left( \sqrt{y^2 + \frac{1}{3} y^2} - \sqrt{y^2 - 5y^2 + 8y^3 - 4y^4} \right)
\]

\[
-\sqrt{w^2 + \frac{1}{3} w^2} + \sqrt{w^2 - 5w^2 + 8w^3 - 4w^4}
\]

(31)

For \((x + h) > 0.5\) \(d_c \cos \alpha\)

\[
A_c = \frac{\cos \beta \cos \alpha}{2} \frac{d_c^2}{2} \left( \sqrt{\frac{4 - 5w^2 + w^2}{3}} - \sqrt{-w^4 + \frac{5w^2}{4} - w} \right)
\]

\[
-\sqrt{\frac{4 - 5y^2 + y^2}{3}} + 2\sqrt{-y^4 + \frac{5y^2}{4} - \frac{y}{4}}
\]

(32)

**DERIVATION OF WEAR FUNCTION**

Since the function \(A_c\) shows very small non-linearity with cutting depth \(h\), (Fig. 10). We force \(A_c\) to be linear function of cutting depth, \(h\), within the interval \(0 < h < 0.1\) in. Maximum cutting depth is estimated to be \(0.1\) in. by assuming a maximum penetration.
Figure 10- Relationship between cutting surface area and cutting depth.
rate of 100 ft./hr., at a minimum rotary speed of 40 rpm, and for 5 interfering blades. Hence, the function $A_c$ can be formulated as:

$$A_c(h, x, \alpha, d_c) = h \frac{dA_c}{dh}$$  \hspace{1cm} (33)

Since $A_c$ is a linear function of $h$, $\frac{dA_c}{dh}$ becomes independent (i.e., weak function of $h$) from $h$. Therefore, $\frac{dA_c}{dh}$ can be written as a function of $x$ only. That is:

$$\hat{h} = \frac{h + x}{d_c \cos \alpha} \equiv f_1 \left( \frac{x}{d_c \cos \alpha} \right)$$  \hspace{1cm} (34)

Function $f_1$ was calculated for the average values of normal force, $F_N (300-800$ lbs/cutter) and the results are plotted in Fig. 11. The relation between $h$ and dimensionless cutter wear, $w$, is then formulated as follow:

$$\hat{h} = 0.028 \exp(-0.05w)$$  \hspace{1cm} (35)

Consequently, function $y(w)$ is given as:

$$y(w) = \frac{h + x}{d_c \cos \alpha} = w + 0.028 \exp(-0.05w)$$  \hspace{1cm} (36)
Figure 11- Relationship between dimensionless cutting depth and the linear cutting wear.
In order to find the rate of change of cutting area, $A_c$, wrt. cutting depth $h$, take the first derivative of equation (31) (or 32) wrt. $h$ at constant wear condition (ie; $w$=constant). By taking the first derivatives of equations (31) and (32), rate of change of cutting area wrt. cutting depth are given as follow:

For $(x + h) < 0.5$ $d_c \cos \alpha$

$$\frac{dA_c}{dh} = \frac{d_c \cos \beta}{4} \left( \frac{1 + \frac{2y}{3}}{\sqrt{y + \frac{1}{3}y^2}} \right)$$

$$- \frac{(1-10y+24y^2-16y^3)}{\sqrt{y-5y^2+8y^3-4y^4}}$$

(37)

For $(x + h) > 0.5$ $d_c \cos \alpha$

$$\frac{dA_c}{dh} = \frac{d_c \cos \beta}{4} \left( \frac{5-2y}{\sqrt{3(4-5y+y^2)}} \right)$$

$$+ \frac{2(-4y^3+6y^2 - \frac{10}{4}y + \frac{1}{4})}{\sqrt{-y^4 + 2y^3 - \frac{5}{4}y^2 + \frac{y}{4}}}$$

(38)

Finally, using equations (33), (37) and (38), cutting area can be written in the form of:
where;

For \( (x+h) < 0.5 d_c \cos \alpha \)

\[
b_1 = \frac{d_c \cos \beta}{4} \quad (40)
\]

\[
U_D = \frac{\sqrt{(y+\frac{1}{3}y^2)(y-5y^2+8y^3-4y^4)}}{(1+2y)\sqrt{y-2y^2+8y^3-4y^4}} - (1-10y+24y^2-16y^3)\sqrt{y+\frac{1}{3}y^2} \quad (41)
\]

For \( (x+h) > 0.5 d_c \cos \alpha \)

\[
b_1 = \frac{d_c \cos \beta}{4} \quad (42)
\]

\[
U_D = \frac{\sqrt{3(4-5y+y^2)} \sqrt{-y^4+2y^3-\frac{5}{4}y^2+\frac{Y}{4}}}{(5-2y)\sqrt{-y^4+2y^3-\frac{5}{4}y^2+\frac{Y}{4}}} + 2 (-4y^3+6y^2-\frac{10y+1}{4})\sqrt{3(4-5y+y^2)} \quad (43)
\]
DERIVATION OF BIT LIFE EQUATION

It is assumed that drilling parameters such as weight on bit, rotary speed, mud flow rate are constant and also the formation is homogeneous. As we continue drilling under these conditions, cutting depth decreases while the linear wear of PDC drill blank increases. The linear wear of cutter is associated with the work done by the cutter. This work is of friction nature and is a function of:

a-) Cutting path
b-) Bit weight
c-) Coefficient of sliding friction between rock and the PDC cutter.

Cutting path is the helical path that the bit goes through during drilling under the action of rotating drill string and weight on bit. It is slightly longer than the diameter of the bit for average values of rotary speed and weight on bit. For high rotary speed and low weight on bit, cutting depth is smaller but the path becomes more helical. For high weight on bit and low RPM, cutting depth is high, but the path is less helical, thus there is a balance on the length of cutting path.

In order to simplify calculations, cutting path is taken equal to the bit diameter. The basic assumption made is that the volume of worn portion of a PDC cutter is proportional to the work performed by the cutter. Also
assumed that there is a uniform cutting power distribution across the bit face. Power is the work done by each cutter times the rotary speed.

\[
\text{Power} = F_N \mu 2\pi r_c N = \text{constant} \quad (44)
\]

where;

\( r_c \): Radial location of cutter under consideration.

Experimental data, [30], on abrasive wear of drag cutters indicated that the volumetric wear of a cutter per unit cutting length is proportional to the penetrating force. In other words, the depth of wear into the surface of the cutter material per unit cutting distance is proportional to the contact stress [32]. That is;

\[
\frac{dx}{dl} = \frac{F}{A_w} \quad (45)
\]

It is also known that;

\[
\frac{dV}{dx} = A_w \quad (46)
\]

Thus, by using the chain rule;

\[
\frac{dV}{dL} = \frac{dV}{dx} \frac{dx}{dL} \sim \frac{A_w F_d}{A_w} \quad (47)
\]

or,
by introducing a proportionality constant, \( k_2 \):

\[
\frac{dV}{dL} = k_2 F_d \tag{49}
\]

Integral of (49), would give volumetric cutter wear as follow:

\[
V = k F_d L = k_2 * \text{Work} \tag{50}
\]

Since power is equal to work per unit time, volumetric wear \( V \), can be correlated to the power input to the system. Drag force is given as:

\[
F_d = \mu F_N \tag{51}
\]

length of the path, \( L \), is given as:

\[
L = 2\pi r_c N t_b \tag{52}
\]

where,

\( t_b \) : time, hrs.

By combining (50), (51) and (52), volumetric wear is given as:

\[
V = 2\pi k_2 r_c \mu F_N N t_b \tag{53}
\]

Assuming that normal force acting on a single cutter is
proportional to the weight on bit as follows:

\[ W \sim F_N \quad (54) \]

or,

\[ W = k_1 F_N \quad (55) \]

where, \( k_1 \) is the proportionality constant. By introducing (55) into (53);

\[ V = 2 \pi r_c k_2 \mu \frac{W}{k_1} N t_b \quad (56) \]

Volumetric cutter wear can also be calculated from geometrical consideration. That is;

\[ V = \int_0^x A_w(x) \, dx \quad (57) \]

Where;

\[ dx = d_c \cos \alpha \, dw \quad (58) \]

and \( A_w \) is given by equation (16). Hence, by combining (16), (57), and (58)

\[ V = d_c \cos \alpha \int_0^w A_w \, dw \quad (59) \]

or,
\[ V = d_c \cos \alpha \frac{d_c^2}{4 \sin \alpha} \int_0^w \left( \arccos \frac{1}{1-4(w-w^2)} \right) \]
\[ -2 \sqrt{w-5w^2+8w^3-4w^4} \, dw \quad (60) \]

By introducing dimensionless volumetric cutter wear function \( V_0 \):
\[ V = \frac{d_c^3}{4 \tan \alpha} V_0 \quad (61) \]

where:
\[ V_0 = \int_0^w \left( \arccos \frac{1}{1-4(w-w^2)} \right) \]
\[ -2 \sqrt{w-5w^2+8w^3-4w^4} \, dw \quad (62) \]

Values of \( V_0 \) were calculated numerically and given in Fig. 12 for different values of cutter wear, \( w \).

Finally, by combining (56) and (61) and writing out for bit life, \( t_b \):
\[ t_b = \frac{d_c^3}{4 \tan \alpha} \frac{V_0}{2 \pi r_c k_2 \mu \frac{W}{k_1} N} \quad (63) \]

or,
\[ t_b = A_f G_3 \frac{V_0}{W N} \quad (64) \]
Where, \( A_f \) is the abrasiveness factor and is given as:

\[
A_f = \frac{k_1d_2^2}{48 \times 10^6 \pi \mu r_c k_2 \tan \alpha}
\]  

(65)

The physical meaning of the abrasiveness constant, \( A_f \), is the weight-on-bit (in thousands of pounds) necessary to cause a volumetric cutter wear equal to 0.1 volume of a cube with each side equal to the cutter radius \( r_c \), when drilling for 100 hrs. with the rotary speed of 100 RPM. The dimensionless volumetric wear function, \( V_d \), is equal to the ratio of the actual volumetric cutter wear to the volume of a cube with each side equal to the cutter radius.

DERIVATION OF PENETRATION RATE EQUATION

From the normal force balance equation (equation 4), it is known that:

\[
F_N = R_c A_c (\sin \alpha + \mu \cos \alpha) + R_p A_w
\]  

(4)

\( A_c \) can be written as:

\[
A_c = \frac{F_N}{R_c(\sin \alpha + \mu \cos \alpha)} - \frac{R_p A_w}{R_c(\sin \alpha + \mu \cos \alpha)}
\]  

(66)

Combining (39), (40), (41) [or, (42) and (43)] and (66),

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cutting depth, $h$, can be written as:

$$h = \frac{U_d}{b_1 R_c (\sin \alpha + \mu \cos \alpha)} \frac{(F_N - R_p A_w)}{(F_N - R_p A_w)}$$  \hspace{1cm} (67)

Assuming the penetration rate is proportional to the cutting depth, the bit rotates at speed $N$, and the number of interfering cutters is $I$, the drilling rate is:

$$R = k_3 h N^{n_1} I$$  \hspace{1cm} (68)

An assumption on linearity between bit weight and the cutter's normal force, gives:

$$W = k_1 F_N$$  \hspace{1cm} (55)

By combining (67), (68) and (55), drilling rate equation is:

$$R = K G_1 (W - W_o) U_d N^{n_1}$$  \hspace{1cm} (69)

where

$$W_o = G_2 A A_{wn} k_1 R_p$$

$$A = \pi d_c^2 / 4$$

$$A_{wd} = A_w / A$$

$$A_{wn} = A_{wd} / 2.77$$  \hspace{1cm} (70)

and $K$ is the drillability constant defined as:

$$K = \frac{4 k_3 I}{k_1 d_c \cos \beta R_c (\sin \alpha + \mu \cos \alpha)}$$  \hspace{1cm} (71)
DERIVATION OF THE BIT TORQUE EQUATION

By introducing (6), and (66) into (5); horizontal force is found as:

\[ F_h = F_N \left( \frac{1-\mu \tan \alpha}{\mu + \tan \alpha} - \frac{2-(\mu + \tan \alpha)^2}{2(\mu + \tan \alpha)} \right) R_p A_w \] (72)

To convert equation (72) into the bit torque equation, the density of cutters placement is considered as a function of bit radius, \( n(r) \), and the torque is given as:

\[ T_b = \frac{1-\mu \tan \alpha}{\mu + \tan \alpha} \int_0^{r_b} F_N n(r) r \, dr \]

\[- \frac{2-(\mu + \tan \alpha)^2}{\mu + \tan \alpha} R_p \int_0^{r_b} A_w n(r) r \, dr \] (73)

The explicit form of equation (73) requires some knowledge of load distribution and wear distribution across the bit face. Let us assume that the load is evenly distributed along bit radius and that the work performed by cutters is different at each radius (unbalanced bit design), and it is represented by function \( f_2(r) \):

\[ W(r) = \frac{2W}{d_c} = \text{CONST.}; \text{ and } A_w(r) = A_w f_2(r) \] (74)
After substituting (74) into (73) and integrating, the bit torque becomes

\[
T = \frac{W}{d_b} \frac{4(1-\mu \tan \alpha)}{(\mu + \tan \alpha)} - A_w \frac{2-(\mu + \tan \alpha)^2}{\mu + \tan \alpha} e_1 R_p \tag{75}
\]

where constant \( e_1 \) is given as:

\[
e_1 = \int_0^{R_b} f_2(r) n(r) r \, dr \tag{76}
\]

and it represents bit design features.

It is seen from equation (75) that, for a constant weight on bit, torque is solely the function of the wear flat area. Equation (75) indicates that, as the wear increases, the torque should reduce which is supported by the laboratory experiments - Fig. 13 and 14.

Field data collected for this research from the PDC drilling in the Gulf Coast area further support the linear correlation of torque vs. wear as shown in Fig. 15. However, all the field data from seven wells analyzed here consistently showed low sensitivity of torque to bit weight, as compared to the laboratory data and as expected from equation (75).

Theoretically, in homogeneous rock, the torque and the penetration rate should both respond to the bit weight
Figure 13- Effect of PDC bit wear on torque vs. weight on bit relationship, laboratory drilling in Sierra White Granite, after[30].

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Figure 14 - Effect of PDC bit wear on torque vs. weight on bit relationship, laboratory drilling in sandstone, after [33].

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Figure 15 - Effect of PDC bit wear on torque vs. weight on bit relationship, field data.
similarly. In real drilling conditions, however, correlation between drilling rate and torque is much stronger than correlation between each of them and the weight on bit. This phenomenon is well documented in the recent research, [16], and also was qualitatively observed in previous works [34]. Equation (75) indicates that, though absolute value of torque for the worn bit is smaller than that for the new bit, its sensitivity to the bit weight (slope) should not change. This is not quite supported by laboratory experiments, (compare Fig. 13 and 14), and not observed in the field. The most likely reason might be an unequal distribution of the bit weight across the bit face, resulting from uneven distribution of cutters wear. Modelling of these effects requires an individual cutter approach, which makes the model quite complex.

VERIFICATION OF THE DRILLING MODEL

In contrast to the purely empirical drilling models based on the curve fitting of some experimental data, the new model for PDC bits was derived from the theoretical considerations. The physical meaning was given to the constants and functions of the model.

The predictive value of the model was tested by using the actual offshore Louisiana field record (Table-1)
<table>
<thead>
<tr>
<th>Run #</th>
<th>Bit</th>
<th>Out, ft</th>
<th>footage</th>
<th>Time, hr</th>
<th>ROP, ft/hr</th>
<th>WOB, $10^3$ lbf</th>
<th>RPM, 1/min</th>
<th>Wear, W</th>
</tr>
</thead>
<tbody>
<tr>
<td>32</td>
<td>SX-3</td>
<td>13335</td>
<td>165</td>
<td>13</td>
<td>12.7</td>
<td>10</td>
<td>90</td>
<td>0.05</td>
</tr>
<tr>
<td>33*</td>
<td>SX-3</td>
<td>15044</td>
<td>709</td>
<td>45</td>
<td>15.0</td>
<td>15</td>
<td>60</td>
<td>0.10</td>
</tr>
<tr>
<td>35</td>
<td>SX-3</td>
<td>15992</td>
<td>792</td>
<td>31.0</td>
<td>25.5</td>
<td>10</td>
<td>90</td>
<td>0.10</td>
</tr>
<tr>
<td>36*</td>
<td>SX-3</td>
<td>16302</td>
<td>310</td>
<td>30</td>
<td>10.3</td>
<td>15</td>
<td>90</td>
<td>0.2</td>
</tr>
<tr>
<td>38</td>
<td>SX-3</td>
<td>17175</td>
<td>569</td>
<td>33.5</td>
<td>17.0</td>
<td>15</td>
<td>90</td>
<td>0.3</td>
</tr>
</tbody>
</table>

*Bit runs used for prediction.
of drilling a wellbore section from 13,170 to 17,175 ft.
with the 8 1/2 in. Stratapax SX-3 PDC bits. The record of
the two bits was used to predict performance of the other
bit runs. The prediction was based on the calculated
values of constants \( K = 1.464 \times 10^{-4} \) ft/klbf., \( A_2 = 30 \) klbf.,
and \( k_t R_p = 34.5 \) klbf/sq.in. The unit value was assumed
for the RPM exponent \( a \). Because of the low values of the
bit weight, the calculated temperature effect proved
insignificant for most of the bit life, except for the
early stage. The predicted and the actual bit performance
is shown in Table-2. Two cases proved consistent with the
model but in one case the bit performance was clearly
overestimated. Early bit removal from the well on
misjudgement of its wear might have been the reason for
this over-estimation. Table -2 exemplifies the need for a
reliable bit wear record which greatly affects drilling
planning strategies. In all calculations similar to those
in Table -2 , we encountered problems of unreliable bit
wear data.

For verification of the model, an attempt was also
made to use available laboratory drilling data with PDC
bits. First, we looked at the somewhat controversial
issue of the linear effect of the bit weight on drilling
rate. Though very few experiments indicated otherwise,
the prevailing empirical data with PDC bits showed ROP vs.
WOB close to linearity [13], [31], [34] especially at the
### Table 2

**PDC BIT PERFORMANCE PREDICTION WITH PDC BIT MODEL**

<table>
<thead>
<tr>
<th>Run #</th>
<th>Actual Performance</th>
<th>Predicted Performance</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Time, hr</td>
<td>Footage, ft</td>
</tr>
<tr>
<td>32</td>
<td>13</td>
<td>165</td>
</tr>
<tr>
<td>35</td>
<td>31</td>
<td>792</td>
</tr>
<tr>
<td>38</td>
<td>33.5</td>
<td>569</td>
</tr>
</tbody>
</table>

### Table 3

**VERIFICATION OF THE PDC BIT MODEL USING CUTTER REMOVAL TEST DATA**

<table>
<thead>
<tr>
<th>WOB, Carthage Limestone</th>
<th>WOB, Berea Sandstone</th>
</tr>
</thead>
<tbody>
<tr>
<td>10^3 lbf</td>
<td>10^3 lbf</td>
</tr>
<tr>
<td>( I_{2}/I_{1} )</td>
<td>( \text{ROP}<em>{2}/\text{ROP}</em>{1} )</td>
</tr>
<tr>
<td>7.5</td>
<td>10/19</td>
</tr>
<tr>
<td></td>
<td>19/39</td>
</tr>
<tr>
<td></td>
<td>19/27</td>
</tr>
<tr>
<td>10</td>
<td>27/39</td>
</tr>
<tr>
<td></td>
<td>19/39</td>
</tr>
<tr>
<td></td>
<td>19/27</td>
</tr>
</tbody>
</table>

*Worn PDC Bit*
low RPM and in medium hard formations. Showed in Fig. 16 are some results of the laboratory drilling tests [15],[33] performed in medium hard and soft formations with the water-base and oil base drilling fluids. Taking into account inherent data scattering, the linear trends seem acceptable. In soft rock drilling, the theoretical value of the threshold weight-on-bit, $W_0$, often appears to be negative. Though not theoretically justified, the non-zero drilling rate with the zero-weight (WOB nondetected by weight indicator) has been observed while drilling soft formations. In addition, when drilling hard and elastic rocks using new bit, the threshold weight does indicate the limit of surface grinding and an offset of the volumetric chipping. This phenomenon is well known in the rock bit modelling theory [3], [35], [36], it was also observed for PDC bits [32] as shown in Fig. 17.

Further experimental verification was performed by applying the new drilling model to the recent experiments on PDC cutter removal [15], [33]. These experiments showed a peculiar response of PDC bits to reduced number of cutters. The responses can be summarized as follows: While drilling with constant weight and rotary speed, (1) the drilling rate did not respond to the reduced number of cutters for the new PDC bit - Fig. 18, and (2) the drilling rate increased with the decreasing number of
Figure 16 - Linear effect of weight on bit on drilling rate, after [33].
Figure 17—Experimental and theoretical linear effects of cutter normal force on the effective penetration, after [32].
Figure 18 - Cutter removal test; new PDC bit, after [33].
cutters for the worn bit - Fig. 19. The problem was whether the new model for the PDC bit would respond in the same way as the PDC bit tested.

For the new PDC bit (w = 0, U_D = 1), constant drilling variables (W=const., N=const.) and effective cleaning under laboratory conditions (a_i=1), equation (69) becomes:

$$R_1 = (\text{CONST.}) (k_1)_1 I_1$$  \hspace{1cm} (77)

After reducing number of cutters from I_1 to I_2:

$$R_2 = (\text{CONST.}) (k_1)_1 \frac{I_1}{I_2} = R_1$$ \hspace{1cm} (79)

which complies with the experimental results.

In case of the worn PDC bit (w = 1, U_D < 1), the equation (69) can be written as:

$$R = (\text{CONST.1}) k_1 I_1 - (\text{CONST.2}) I_2$$ \hspace{1cm} (80)

and the ratio of drilling rates before and after cutters removal is calculated as:
Figure 19 - Cutter removal test; worn PDC bit, after [33].

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\[
\frac{R_2}{R_1} = \frac{C_4 - (I_2/I_1)}{C_4 - 1} \tag{81}
\]

Where,

\[
C_4 = \frac{\sqrt{3} G_1 W}{3 G_2 R_c R_p A d_c A_w \sin \alpha \cos \beta} \tag{82}
\]

Constant \(C_4\) is a function of PDC bit geometry, rock properties, the bit weight and the bit wear. Its value should be constant in these experiments.

Equation (81) was used for the quantitative interpretation of the worn PDC bit cutters removal test data shown in Fig. 19. The value of \(C_4\), at constant weight, was calculated using the experimental values of drilling rate ratios and the respective ratios of the PDC bit cutters before and after removal. The calculations were performed for two different rocks and the two different values of the weight on bit. The results are shown in Table - 3. The calculated values of the constant \(C_4\) did not display significant variation, thus, provided further verification for the model.

CONCLUSIONS

1. A new mechanistic PDC bit drilling model was derived from fundamental laws using a few simplifying assumptions. The model is physically sound; its
constants and functions have a physical meaning.

2. The new drilling model provides a mathematical tool for the purpose of optimum drilling programming and interpretation of drilling data.

3. Further development in this area may include: (1) the development of a procedure for the PDC cutter placement design; (2) Evaluation of bit wear and load distribution functions.
NOMENCLATURE

English Upper case

\( A \) : Cutter face area, sq.in.
\( A'_c \) : Area defined in figure (1.9), sq.in
\( A_c \) : Cutting area, sq.in.
\( A_f \) : Abrasiveness constant, \( 10^3 \) lbf.
\( A_{wd} \) : Dimensionless cutter wearflat area, unitless.
\( A_{wdn} \) : Normalized dimensionless cutter wearflat area, unitless.
\( C_d \) : Constant defined by equation (1.82).
\( F_c \) : Cutting force, lbf.
\( F_d \) : Drag force, lbf.
\( F_{fc} \) : Friction force effective on the cutting surface area, \( 10^3 \) lbf.
\( F_{fw} \) : Friction force effective on the wearflat area, \( 10^3 \) lbf.
\( F_H \) : Horizontal force at the cutter, \( 10^3 \) lbf.
\( F_N \) : Normal force at the cutter, \( 10^3 \) lbf.
\( F_R \) : Resistance force, \( 10^3 \) lbf.
\( F_T \) : Tangential force, \( 10^3 \) lbf.
\( F_{th} \) : Thrust force, \( 10^3 \) lbf.
\( F_w \) : Component of the normal force effective on the wearflat area, \( 10^3 \) lbf.
\( G_1 \) : Unit conversion constant, \( 0.7576 \times 60 \) min/hr.
\( G_2 \) : Unit conversion constant, \( 2.77 \), unitless.
\( G_3 \) : Unit conversion constant, \( 48 \times 10^6 \) hr/min.
\( I \) : Cutters' interference constant, unitless.
\( K \) : Drillability constant, ft/\( 10^3 \) lbf.
$K_0$ : Drillability at zero depth, ft/$10^3$ lbf.
$L$ : Length of the path that cutter travels, in.
$N$ : Rotary speed, 1/min.
$P$ : Length of the arc defined in figure (1.9A), in.
$R$ : Instantaneous drilling rate, ft/hr.
$R_c$ : Rock resistance to shearing, $10^3$ lbf/sq.in.
$ROP$ : Average drilling rate, ft/hr.
$R_P$ : Rock resistance to pressing, $10^3$ lbf/sq.in.
$S$ : Elastic limit of rock deformation, lbf/sq.in.
$T_b$ : Bit torque, ft-lbf.
$T_r$ : Single cutter torque, in-lbf.
$U_D$ : Dimensionless cutter wear function, unitless.
$V$ : Sliding velocity, ft/sec.
$V_D$ : Dimensionless volumetric wear function, unitless.
$W$ : Weight on bit, $10^3$ lbf.
$W_0$ : Threshold weight on bit, $10^3$ lbf.

English lover case

$a$ : Distance defined in figure (1.8B), in.
$a_1$ : Rotary speed exponent, unitless.
$b$ : Distance defined in figure (1.8B), in.
$b_1$ : Function defined by equation (1.40) or (1.42), in.
$c_1$ : Function defined by equation (1.24) or (1.27), in.
$c_2$ : Function defined by equation (1.24) or (1.27), in.
$d$ : Distance defined in figure (1.8B), in.
$d_b$ : Bit diameter, in.
$d_c$ : Cutter face diameter, in.
$e_1$ : Function defined by equation (1.76), sq.in.
$f$ : Function defined by equation (1.34), unitless.
$h$ : Cutting depth, in.
\( \hat{h} \) : Dimensionless cutting depth, unitless

\( k_1 \) : Proportionality constant between weight on bit and normal force acting on the single cutter, unitless.

\( k_2 \) : Cutter wear constant, unitless.

\( k_3 \) : Proportionality constant between penetration rate and cutting depth, unitless.

\( l \) : Distance defined in figure (1.8C), in.

\( n \) : Number of blades, unitless.

\( n_c \) : Number of cutters, unitless.

\( n(r) \) : Cutters' density, unitless.

\( r \) : Radius, in.

\( r_b \) : Bit radius, in.

\( r_c \) : Radial location of the cutter under consideration, in.

\( t \) : Distance defined in figure (1.8C), in.

\( t_b \) : Bit rotating time, hrs.

\( w \) : Dimensionless linear cutter wear, unitless.

\( x \) : Linear cutter wear, in.

\( y \) : Function defined by equation (1.30), unitless.

**Greek lower case**

\( \alpha \) : Back rake angle, deg.

\( \alpha_c \) : Cutting angle, deg.

\( \beta \) : Side rake angle, deg.

\( \varepsilon \) : Strain (displacement), in.

\( \mu \) : Friction coefficient at rock/cutter interface, unitless.

\( \phi \) : Angle between failure plane and reference (horizontal) plane, deg.

\( \delta \) : Cutting stress, lbf/sq.in.
\( \tau_s \) : Shear stress, lbf/sq.in.

\( \omega \) : Bit rotational speed, sec\(^{-1}\).
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and Exhibition of the SPE, New Orleans, LA (Oct. 5-8, 1986).


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and Exhibition of the SPE, New Orleans, LA, (Oct. 5-8, 1986).


CHAPTER II.

A COMPOUND EFFECT OF CUTTING DEPTH AND BIT DULL ON CUTTERS' TEMPERATURE FOR POLYCRYSTALLINE DIAMOND COMPACT BITS

ABSTRACT

This paper presents a simulation study to evaluate the combined effect of cutting depth (drilling rate) and wear (bit dull) on the thermal response of polycrystalline diamond compact (PDC) cutters under down-hole drilling conditions. A new understanding of frictionally generated heat between rock and PDC cutter is introduced from the analysis of forces active on the wearflat and the cutting (leading) surfaces of a cutter. Then this new concept is used to predict PDC bit performance with the controlled temperature of its cutters.

Previous concepts, largely based on the laboratory drilling tests (with low drilling rate and under atmospheric conditions), recognize only one source of heat - the wearflat surface. However this study, using field data, shows that the heat generated at the cutting surface may significantly contribute to the total heat flux into the cutter. In the result, the distribution of temperature within the cutter is changed which particularly affects the maximum value of temperature at the cutting edge.
A simplified 2-D finite difference numerical code is used to quantify the difference in cutter wearflat temperatures calculated with and without the additional heat flux generated at the cutting surface. The numerical analysis reveals that neglecting the cutting surface effect results in underestimation of the actual wearflat temperature by % 10 to % 530, depending upon bit dull and downhole hydraulics.

Also demonstrated is the actual impact of these findings on field drilling practices. The example comparison is made by calculating the optimal-control procedures for PDC bit calculated with and without the effect of cutting surface. In these procedures, wearflat temperature becomes a mathematical constraint which limits weight on bit and rotational speed. The comparison includes calculation of the maximum bit performance curves which represent maximum drilling rate attainable for a bit to drill a predetermined length of a borehole (footage). The curves show an up to 18% reduction of drilling rate when the new and more rigorous temperature limitation is used. In addition, the example calculations show that the actual temperature of the bit cutters can be 860 °F and exceeds by almost 30 % its maximum acceptable value of 560 °F.

For practical applications, the study reveals that many field failures of PDC bits may have been caused by
lack of understanding of operational limits imposed by heat considerations.

INTRODUCTION

Drag bits are designed to fail the rock by shear rather than by crushing as is the case with roller cone and diamond bits. Since rock requires significantly less energy to fail in shear, more efficient drilling with less weight on bit and higher rate of penetrations is possible. A drag bit operating in a shear failure mode, however, has to have tremendous abrasion resistance to maintain a sharp cutting structure for a reasonable downhole life. The diamond/tungsten carbide compacts, originally developed as cutting tools in machining applications, have provided high abrasion resistance to polycrystalline diamond compact (PDC) drilling bits. In addition, the development of new cutters added a self-sharpening feature to the already more efficient shear cutting. Self-sharpening drag bits have appeared less susceptible to chip holddown effect than roller cone bits and much less than diamond bits [1], [2]. The polycrystalline diamond compact bits provide also higher drilling rates.

The success of PDC bits, however, not only depends on obtaining a sufficient drilling rate, but also on running the bit long enough to make their application economical. Changes in drilling operation parameters (such as weight
on bit; rotary speed) which result in an increased drilling rate, will also result in an increase of the bit wear. Hence, better understanding of wear mechanisms effective on the cutting structure of the PDC cutters is essential to eliminate conditions leading to rapid wear and uneconomic bit life.

WEAR MECHANISMS OF PDC CUTTERS

PDC cutters have been experimentally tested by various investigators using single-cutter experiments [3],[4],[5],[6],[7],[8]; laboratory tests of full-scale prototype bits [4],[8],[9], [10],[11],[12],[13]; and field tests of full-scale bits designed using laboratory data [16],[17],[18].

A PDC cutter life is dependent upon the interrelated wear mechanisms of the tungsten carbide base and the PDC cutting edge that shears the rock surface, protecting the tungsten-carbide substrate from abrasive wear. At the same time, the tungsten-carbide base gives support to the diamond layer which is being subjected to the tensile and shear stresses generated during cutting, absorbing shock loadings and preventing gross failure of the PDC layer [19], [20].

The steady state abrasive wear is normally associated with the development of uniform wearflats and gradual decrease in drilling rate over the bit life. On the other
between the diamond grain boundaries which lead to intergranular cracking and grain boundary failure.

The wear of the tungsten-carbide substrate, to which the sintered diamond compacts are attached, is confined to its wearflat area. Larsen-Basse [22], in surveying the literature of hard metal wear, concludes that in rock cutting with hard metal tools the predominant wear mechanisms are impact spalling, impact fatigue spalling, sliding abrasion, and thermal fatigue.

In the case of PDC cutters, the leading impact is received by the sintered diamond edge. Thus abrasion and thermal fatigue are the principal contributors to the metal wear at the wearflat. Glowka [19] gives an excellent discussion of wear mechanisms effective on a PDC cutting structure for both diamond tip and the tungsten-carbide back-up.

Better understanding of frictional heat generation associated with the PDC cutters action is required in order to control higher temperatures which accelerates the bit wear and thereby makes the bit run uneconomical.

Ortega and Glowka [23] developed an analytical relationship between PDC cutter wearflat temperature and drilling operation variables i.e. normal force effective on the cutter, cutter's velocity, and the wearflat area as
\[ T_w - T_f = \frac{\alpha \mu F_N V}{A_w} f \]  

(A10)

Derivation of equation (A10) is given in Appendix A. The thermal response function, \( f \), must be determined numerically in order to compute mean wearflat temperatures. It is a unique function of the cutter geometry, thermal properties of cooling fluid and cutter material, and cooling fluid rate.

Thermal response function is determined by using numerical models of the cutter's heat transfer based upon assumed cutter/rock interaction. Currently, there are three different approaches available [10], [23], [24].

HEAT EXCHANGE MODELS FOR PDC CUTTER

Glowka and Stone [19] stated that the PDC cutter geometry introduced as a result of wear depends on the type of rock being cut. In soft, ductile rocks, where cutting forces (particularly impact loads) are small, the relative abrasive-wear resistance of tungsten carbide prevails. Owing to high wear rate of tungsten-carbide, a wear angle between tungsten-carbide back-up and rock surface develops due to the action of abrasive rock particles sliding beneath the cutter (Fig. 1-A). Examination of field-worn cutters revealed that this angle
Figure-1  Rock/cutter interaction model for ductile formations

A- After Glowka [26]  B- This study

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is about 5 to 10 degrees [25]. In the effect, only the diamond cutting edge remains in intimate contact with rock leaving a relatively small effective wearflat contact area (self-sharpening effect).

In harder, more brittle rocks, they observed that the wearflat was nearly parallel to the rock surface. They reasoned that the higher cutting forces for hard rocks effectively removed any unsupported diamond protruding beyond the tungsten-carbide backing. Furthermore, a substantial tungsten-carbide backing was required to support the diamond near the rock interface for high impact loading. They concluded that, under these conditions, the relative wear of the two materials reached equilibrium at a smaller wear angle (Fig. 2). The result is an effectively larger cutter area in contact with the rock. Their computational model took both cases into account. Temperature distribution within the cutter was evaluated by assuming heat flux into cutter only through wearflat area (Fig. 3-A). They also considered the difference in size of the wearflat area contacting the rock in each case. The thermal response function, $f$, was first evaluated for hard rock cutting by using a finite difference numerical code [23]. Later, it was calculated by using a finite element numerical code [26].

Ortega and Glowka [23],[26],[27] developed their understanding from their experimental observations during
Figure 2 Rock-cutter interaction model for hard formations, after Glowka [26].
Figure 3. Boundary Conditions of Numerical - Thermal Model for PDC Cutter

laboratory drilling of various rocks under atmospheric condition. In general, brittle failure is dominant under atmospheric condition, and the rock pieces fly away as they are cut. These loose chips do not exert any force on the leading (cutting) surface of the diamond compact. This is also the case of hard rock drilling, when a cutter wears out by impact loading rather than by abrasion. In hard rock drilling the cutter's penetration (cutting depth) is small, particularly for a worn cutter with large wearflat area. Only a minor part of the cutting surface of the diamond compact contacts with the rock. Considering all the above factors, Glowka neglected the friction force acting on the cutting surface area. Therefore the model presented by Glowka (Fig. 2, Fig. 3-A) is valid for hard rock drilling, especially, in geothermal drilling areas.

In oil well drilling, however, most of the rock displays plastic behavior as its deformation characteristics change from brittle to ductile pattern with increasing depth [28], [29], [30]. Therefore, it is believed that the self-sharpening wear mechanism of PDC cutters prevail during deep drilling which means that only a small wearflat area remains in contact with rock. The cutting depth, on the other hand, is much larger when drilling ductile rocks because soft-formation bits have fewer compacts of larger size. Large cutter penetration results in increased cutting surface being in contact with
rock. A balance of forces acting on the single cutter for the ductile rock drilling conditions is presented in Fig. 1-B.

Laboratory studies with single PDC cutters [4],[5],[6] revealed that the force acting on the cutting surface area (cutting force) has the same order of magnitude as the normal force effective on the cutter wearflat area. This cutting force inevitably induces a significant value of sliding friction between cutting surface and rock surface thus adding to the friction effective under wearflat surface.

Fairhurst and Lacabonne [31] analyzed forces effective on conventional drag bits during hard rock drilling. They concluded that, as the bit advances into the rock a contact pressure develops normal to the front cutting face and the cut rock is constrained to move upwards across the face. Such movement, thus develops in the rock a frictional force acting downwards parallel to the bit front face. The opposite and equal reaction force on the bit tends to push the bit up out of the rock. It is this frictional effect that results in the necessity to use heavy axial thrusts in rotary drilling to maintain bit contact with the rock. The friction coefficient is highly variable and assumes high values sometimes greater than unity.
Evidence of the presence of friction forces on the rake surface was also found in the experimental study of Kenny and Johnson into the cutting mechanism with tungsten-carbide tools for abrasive rocks [32].

They measured the amount of wear at the rake face and cutting edge. The magnitude of rake face loss was found to be in the same range as the cutting edge loss. The relative contributions of the wearflat, side face wear, and rake face wear to the total tip wear is given in Fig. 4. The existence of wear on the rake face indicated action of friction forces on this face.

Furthermore, considering friction coefficients on both surfaces to be in the same order of magnitude and assuming that the rock pieces are removed as soon as they are cut one will expect frictionally generated heat per unit area, per unit time (as calculated by equation A1) to be in the same range on both surfaces. Since the cumulative heat going into the cutter is determined by the size of the area in contact with the rock, the additional heat input through the cutting surface may significantly change the temperature profile within the cutter. Glowka and Ortega disregarded possible contribution of heat into the cutter through the interface between cutting surface and the rock surface. Therefore, their model will underestimate cutter temperature distribution for deep drilling.
Figure 4- The relative contributions of the wear flat and the rake and side face wear to the total tip wear.

After Kenny and Johnson [32].

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Zijsling, [10], stressed on the self-sharpening feature of the PDC cutter. He presented an analytical model similar to Glowka's model together with a numerical code to predict temperature distribution in PDC's mounted on a matrix type bit. Two field-worn matrix type PDC bits were tested on a full-scale drilling machine. Detailed inspection of PDC's revealed that the diamond layer shows wear only at the cutting edge, mainly at the front and tapering off towards the backup material. As a result of this shape, the diamond layer attacks the rock at a large (-78 degrees) negative rake angle (Fig. 5-A). Zijsling suggested that this geometry appeared to be independent of the degree of wear of a particular PDC. Average cutting depth in these experiments was 0.0035 in. as imposed by the operating conditions of 15 Klb. weight on bit, 130 RPM rotary speed and the drilling rate of 10 ft/hr.

At elevated rotary speed (turbine drilling), however, geometry of cutter/rock interaction has changed (Fig. 6). In this case, the cutting edge appeared to consist of a part which was worn at a large negative rake angle and a part where the diamond surface was parallel to the direction of the motion of the cutter. Cutting geometry in this case, was attributed to the fact that cutting depth was smaller than the maximum possible height of the cutting edge under a negative rake angle of -78 degrees.

Zijsling also suggested formation of a build-up
Figure-5  Cutter/Rock interaction model (Self-sharpening effect)

A- After Zijsling [10]  B- This study
Figure 5 - Cutting Edge Geometry—High Rotary Speed, After Zijssling [10].
edge consisting of pulverized rock under the cutting edge of the diamond compact. A particular loading pattern on the build-up edge keeps it in a stable position with respect to the cutting tool. Because of the high stress levels at the cutting edge, the build-up edge is composed of rock flour. During drilling, the actual sliding surface under the diamond layer is thus located at the interface between virgin rock and the rock flour of the build-up edge and its plane is parallel to the direction of motion of the PDC. As a result of the build-up edge the actual rake angle is not necessarily equal to -78 degrees observed below the diamond layer. The performance of a cutter for various cutting depths is therefore not only governed by different rake angles but also by different friction coefficients at the sliding interfaces rock/rock flour and rock/diamond.

Based on the suggested rock-cutter interaction mechanism, Zijsling developed his numerical model (Fig. 3-B) to predict cutter-temperature distribution under downhole drilling conditions. He assumed that all heat is generated at the sliding interfaces $S_1$ through $S_4$. The effect of heat generated at the interface $S_4$ on the cutting edge temperature is negligible compared to the other heat sources. The partitioning of the frictional heat at the interfaces $S_2$ and $S_3$ is not affected by heating of the contact materials of interface $S_1$. 

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Frictionally generated heat is estimated according to equation (A1). Owing to the presence of the build-up edge under the diamond layer, the friction coefficient to be used in equation (A1) for computation of the specific heat at the sliding interface $S_1$ should be that for rock/rock flour interface. Since sedimentary rocks and rock matrix material have a low thermal conductivity compared to that of diamond, the amount of heat flowing into the cutting element via interface $S_1$, is reduced as a result of the presence of build-up edge.

In his study, Zijssling emphasized the self-sharpening feature of the PDC cutters. Analyzing the laboratory test results of two field-worn, full-scale bits, he suggested that the frictional heat at the rock/bit interface is generated at the diamond cutting edge of the cutter. His model (Fig. 3-B) employing a large negative back rake angle and a rather small cutting depth, suggested that heat generation is restricted mainly to the sharp cutting edge of the diamond compact and partially to tungsten carbide back-up.

However, small cutting depth associated with low drilling rate (10 Ft/Hr) observed by Zijssling does not reflect the PDC drag bit performance, particularly in soft formations. In the actual field drilling conditions, PDC cutters experience larger cutting depths than what Zijssling observed in the laboratory. Fig. 5-B presents a
cutter-rock interaction model employing a more realistic cutting depth value which was estimated using field data. Comparing to the Zijsling's model (Fig. 5-A), the difference between the size of the cutting area which is in contact with the rock in both case is tremendous. Thus, the contact area at the cutting surface should be considered as an important part of the frictional heat generation process.

Recently, Prakash [24] presented a theoretical model together with a finite element formulation of temperature distribution and heat transfer in the rock and in the cutter for orthogonal cutting of rock. He used orthogonal metal cutting theory to define cutting forces effective on a PDC cutter during drilling. The heat generation along the shear plane due to shearing of the rock and the heat generation by sliding friction along the rake face and wearflat are all considered in this model (Fig. 3-C). The model was verified by using limited data available from experiments of Hibbs et al., [3,4].

The Prakash work gives the most complete model of heat generation and cutter temperature. However, his method suffers from the inherent error of computing the thermal response in a three dimensional element with two-dimensional model. Extensive verification of this model is yet to be seen. Cutting depth value (0.0012 in.) used in his computations did not represent the whole spectrum of
field drilling conditions. Due to the low value of cutting depth, his conclusion was the same as Glowka's and Zijssling's by saying that the largest contribution to the heat generation in the model is from the heat generation due to sliding friction on the wearflat which is overgeneralization of the average field conditions. The degree of heat contribution from different sources depends very much on the cutting depth (drilling rate) as well as cutter geometry and hydraulics.

THERMAL RESPONSE OF PDC CUTTER TO FRICTION AT CUTTING SURFACE

A complete theoretical solution of the problem requires a 3-D numerical model which includes all possible sources of heat generation during rock cutting with PDC bits. This is still a subject for future study. It was not the purpose of this paper to develop a new numerical model. The intention here is to study engineering implications of frictionally generated heat during rock cutting with PDC bits. From the engineering standpoint, an understanding of the frictional heat helps to improve PDC bits' control and to predict their performance.

A wide range of field operational variables were considered in the simulation study (Table-1). MWD data from two wells drilled in the Gulf Coast of Louisiana was
# TABLE - 1

**RANGE OF DRILLING OPERATIONAL VARIABLES**

<table>
<thead>
<tr>
<th>Bit Size</th>
<th>6 3/4 - 9 7/8 in.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bit Type</td>
<td>Fish tail bit</td>
</tr>
<tr>
<td>Number of Blades</td>
<td>3 - 4</td>
</tr>
<tr>
<td>Rotary Speed</td>
<td>80 - 120 rpm</td>
</tr>
<tr>
<td>Rock/Bit Friction Coefficient</td>
<td>0.05 - 0.2</td>
</tr>
<tr>
<td>Bit Dull Conditions</td>
<td>New bit - Worn bit (% 25 Wear)</td>
</tr>
<tr>
<td>Mud Flow Rate</td>
<td>180 - 700 gpm</td>
</tr>
<tr>
<td>Mud Types</td>
<td>Water, Bentonite Mud, KCL-Polymer, Oil-Base Mud</td>
</tr>
<tr>
<td>Convective Cooling Coefficient</td>
<td>0.1761 -1.761 Btu/h-ft-F</td>
</tr>
<tr>
<td>Drilling Rate</td>
<td>20 - 150 ft / hr</td>
</tr>
</tbody>
</table>
used to define the maximum and minimum values of penetration per rotation per cutter under field conditions. A 9 7/8" fish-tail bit drilled from 6358 ft. to 8325 ft. in Well No. 1. Instantaneous penetration per rotation per cutter were calculated and plotted versus depth as shown in Fig.7. Another 9 7/8" fish tail PDC bit drilled from 3100 ft. to 5138 ft. in Well No. 2. Distribution of the instantaneous cutting depth is given in Fig.8. The penetration per rotation per cutter is calculated using the relationship given as follow;

\[
R_h = \frac{R}{5 N n_c}
\]

Where, \( n \) is the number of interfering cutters at any radius along the bit face (or the number of blades). Analysis of field data has shown that cutting depth varies between 0.02 in. and 0.08 in. The difference is enormous, when these numbers were compared with Glowka's [23] 0.006", Zijling's [10] 0.0035 in., and Prakash's [24] 0.0012". Consequently, wearflat area is not necessarily the only source (or dominant source) of frictional heat generation all the time. One should take the cutting surface area into consideration to estimate cutter temperature especially for new bits running at high drilling rates.

The second aspect of the simulation study was to
Figure 7— MWD data from well # 1.
Figure 8— MWD Data from well # 2.
determine the limits of convective cooling of cutters by drilling fluid. Two fish tail bits (6 3/4" and 9 7/8" dia.) and two short taper-concave profile bits (6 1/2" and 9 7/8" dia.) were used to determine typical cooling ranges. Distribution of the cutters along the bit face were known for these four bits. Radial velocities around the cutters were estimated using the flow rates recommended by the manufacturers of these bits (180-700 gal/min). A simple mass balance was used to estimate radial velocities. Knowing the clearance between the bit body and rock surface, the cross-sectional area of cutter exposed to cross-flow and the distribution of the cutters on the bit, the effective flow area was calculated. Maximum possible velocities around the cutter at each radius were then calculated by using known mud flow rate and effective flow area (see Appendix B).

To date, no data are available for convective cooling of PDC cutters with actual drilling fluids and flow rates similar to those used in the oil field. Glowka [33] presented empirical correlations for convective cooling of PDC cutters exposed to a uniform flow of water. He also proposed using the principles of similitude to infer cooling with drilling mud from the results of simplified laboratory measurements with water.

The literature is also very short of data on actual drilling mud thermophysical properties. Corre et al.[34]
performed experiments on several field muds. They presented correlations for KCL-Polymer and oil-base mud. Some other data were presented by Holmes and Swift [35], and Tregesser et al. [36]. Convective cooling coefficients were calculated by using available thermophysical mud properties and similitude theory (see Appendix B). The range of calculated values were found to be from .1761 to 1.761 Btu/h-sq.ft.-°F for all practically attainable field drilling conditions.

A 2-D finite difference code was used to quantify the effect of this additional heat source on the temperature distribution within the PDC cutter. Thermal response function, \( f \), was evaluated for both cases where heat generation was restricted only to the interface between wearflat and rock, and where cutting surface together with wearflat surface contributed to heat generation. Fig.9 shows the maximum and minimum error made by neglecting the heat flux through the cutting surface. The error varies from 70% to 580% within the limits of convective cooling.

The equation (A10) gives the direct relation between thermal response function (estimated by using a numerical model) and the wearflat temperature. Therefore, underestimation of thermal response function will result in underestimation of the wearflat temperatures as well. The error in estimated wearflat temperature varied from 7%
Figure 9 - The Effect of Heat Influx Through Cutting Surface Area on the Thermal Response Function.
The impact of additional heat source on the wearflat temperature is more pronounced for a new cutter and it reduces as the wearflat builds up (Fig. 11). A set of bit operation variables and rock properties presented in Table-2 were used to generate Fig. 11.

EFFECT OF THERMAL CONSIDERATIONS ON THE OPTIMIZED PDC BIT PERFORMANCE

For PDC bits, the maximum cutter's temperature limits its operational parameters and affects bit's performance. Any source of an additional heat will effect the design of a single bit run so that the operational variables namely weight on bit, rotary speed, and mud flow rate have to be adjusted to make the bit run economical i.e. to minimize the drilling cost per foot.

The drilling bit efficiency can be completely described by using maximum bit performance (MBP) curves [37]. In engineering sense, a MBP curve represents a relationship between any possible footage made by a single drill bit and its maximum drilling rate.

In this study, the PDC bit optimization model and computer code for MBP curves, previously developed [37], were modified by introducing thermal response function. In the closed-form formulation, this function describes relationship among the cutter thermal response and the
Figure-10 The Effect of Heat Influx Through Cutting Surface Area on the Wearflat Temperature.
Figure-11 % Change in Estimated Cutter Wearflat Temperature Due to Additional Heat Influx Through Cutting Surface.
### TABLE-2

**DATA USED FOR SIMULATION STUDY**

- **Bit Size**: 9 7/8 in.
- **Bit Type**: Fish tail bit
- **Number of Blades**: 4
- **Rotary Speed**: 130 rpm.
- **Average drilling rate**: 135 Ft./Hr.
- **Mud Flow Rate**: 717 gpm.
- **Mud Type**: Lignosulfanate
- **Convective cooling coefficient**: 1.6 Btu/h-ft-F
- **Total number of cutters**: 27
- **Average Weight on bit**: 10 Klbf.
- **Cutting depth**: 0.052 in./revolution/cutter
- **Drilling fluid temperature**: 122 F
- **Rock thermal conductivity**: 1.1 BTU/h-ft-F
- **Friction coefficient**: 0.15 (Berea Sandstone)
- **Rock thermal diffusivity**: 1.6e-03 sq.in./sec.
- **Back rake angle**: 20 degrees.
- **Diameter of a single cutter**: 0.52 in.
convective cooling coefficient, cutting depth, and dimensionless cutter wear as

\[ F = 1.1 \, w^{0.563} \, h_\infty^{-0.614} + 3.8 \, h_\infty^{-1.1417} \, w^{0.37259} \, h_c \] (2)

A 2-D numerical code was used to generate values of the thermal response function within practical limits of the variables. The limits of first two variables were discussed above. The dimensionless cutting wear varied between 0 to .25, corresponding to a new bit and worn bit respectively. Field practices reveal that PDC bits that have been worn over 25% are not rerun and are taken back to be salvaged.

Having covered all the practical ranges and combinations of these three variables, a set of thermal response function values was generated. Then, a non-linear regression analysis was used to determine a closed-form relationship among the thermal response function and the three variables. Fig. 12 presents the results of regression analysis as compared to the results from numerical analysis. A straight line passing through the origin at 45 degrees angle indicates the existence of a good quality fit.

The theory of maximum bit performance curves [37] provides also an optimal control of weight on bit and rotary speed which would render a maximum drilling rate for a given footage. An example of the optimum bit control
Fig. 12 Comparison of thermal response function values; numerical analysis vs. non-linear regression analysis.
is shown in Fig. 13. Here, wearflat temperature is used as a mathematical constraint which limits weight on bit and rotary speed. The maximum weight-on-bit is calculated by using maximum allowable wearflat temperature (660°F) as a limiting criterion. The optimal bit control requires that the rotary speed be constant at its maximum value of 200 rpm. while the weight on bit be increased gradually, corresponding to increasing wearflat area so that the maximum allowable wearflat temperature is not exceeded.

The example in Fig. 13 indicates that 20% weight-on-bit reduction is required to accommodate the effect of cutting depth on the wearflat temperature. In practice, this means that bit will be overloaded and prematurely damaged if the effect of cutting surface was ignored. At later moments of bit life, the bit wears out and the wearflat surface becomes larger (which reduces the normal stress) so the wearflat temperature does not limit bit control. When the cutting depth effect is ignored, Fig. 13 shows that beyond the 600 ft. footage, the temperature constraint is not effective and the maximum weight can be used. However, in the case where the cutting depth was included in the calculations, the temperature constraint restricted bit control over the whole bit life (900 ft. footage).

The effect of bit penetration per revolution (cutting depth) on the bit performance was investigated by
Figure 13 - Effect of cutting depth on the optimum bit control strategy.
considering the change of the maximum bit performance curve with and without the cutting depth effect. In Fig.14, the solid curve represents the case where cutting depth is not included in the calculation of wearflat temperature, whereas the dashed line shows the expected bit performance with both thermal effect considered. Apparently, by neglecting the contribution of heat generated at the cutting surface, restrictions on operational parameters becomes less severe, and consequently the bit performance is overestimated. In other words, the bit will never achieve the performance predicted by solid line in Fig.14 - it will be prematurely damaged.

The reason for the above suggestion can be seen clearly in Fig. 15. In this figure, the bit temperature is calculated as the bit progresses in the rock. The calculation include the case where cutting depth is included (solid line) and the case when it is not included (dotted line) in the calculation of wearflat temperature. In the latter case, the actual wearflat temperature is truly constant and equal to the maximum allowable temperature until drilling 580 ft. Later, the bit develops considerable wearflat area, and its temperature falls below its maximum value. In case the bit was operated with consideration given to its cutting depth, the temperature by far exceeded its maximum value.
Figure 14- Effect of cutting depth on the maximum bit performance.
Figure 15- Effect of Cutting Depth on the Wearflat temperature.

- Cutting depth not included
- Cutting depth included
and would reach over 800 °F for most of the bit life. Such conditions would surely cause thermal bit damage. Therefore, the bit performance indicated by the solid curve in Fig. 14 by following the erroneous bit control strategy (Fig. 13 solid curve) would never be reached. Moreover, the actual performance would be lower than its maximum possible performance (Fig. 14, dashed line).

Also illustrated is the effect of cooling provided by drilling fluid. As it is seen from Figure-16, drilling rates changed considerably depending on the degree of cooling provided. Therefore, it is very important to design nozzle-cutter orientation as well as mud thermophysical properties properly to provide maximum possible cooling to the bit.

Finally, the effect of rock/bit friction coefficient on the drilling performance of a bit was investigated. Fig. 17 shows two cases where friction coefficients are 0.05 (solid line) and 0.15 (dashed line). The resulting plot clearly indicates that the bit performance can be improved enormously by reducing the friction coefficient which would in turn reduce the frictionally generated heat at the cutters. Therefore, the mechanism of the friction should be further studied to determine necessary means to reduce friction under PDC bits.

Field data used to generate figures 13, 14, and 15,
Figure 16- Effect of convective cooling on the maximum bit performance.
Figure 17- Effect of Rock/Cutter Friction Coefficient on the Drilling Performance.
CONCLUSIONS AND RECOMMENDATIONS

The cutting surface of a PDC cutter appears to be an important part of the frictional heat generation process. Therefore, the previous studies, by neglecting this fact, underestimated the values of thermal response function, and PDC cutter wearflat temperature. Validity of equation A10 describing wearflat temperature in terms of drilling variables is limited by the assumptions inherent in its derivation. By introducing modified values of the thermal response function (cutting depth included) into the wearflat temperature calculation, the results will still be approximation. However, the effect of cutting depth is significant enough to realize that the thermal response function becomes a controlling factor in PDC bit operations.

Comparative study using MBP curves showed a significant change in PDC bit performance when the new and more rigorous temperature limitation is used. The study reveals that many failures in field applications of PDC bits may have been caused by lack of understanding of operational limits imposed by heat considerations.

There is very limited information available about the thermophysical properties of drilling fluids. An extensive investigation is needed to determine thermal
conductivity, and specific heat of drilling fluids in relation to temperature and pressure.
NOMENCLATURE

<table>
<thead>
<tr>
<th>English upper case</th>
</tr>
</thead>
<tbody>
<tr>
<td>$A_{\text{eff}}$ : Effective area open to flow, in$^2$</td>
</tr>
<tr>
<td>$A_{\text{nf}}$ : No flow area, in$^2$</td>
</tr>
<tr>
<td>$A_w$ : Cutter wearflat area, in$^2$</td>
</tr>
<tr>
<td>$C_p$ : Fluid specific heat, Btu/lb$_m$-°F</td>
</tr>
<tr>
<td>$D$ : Characteristic diameter, in</td>
</tr>
<tr>
<td>$F_d$ : Drag force, lb$_f$</td>
</tr>
<tr>
<td>$F_N$ : Normal force acting at the cutter wearflat, lb$_f$</td>
</tr>
<tr>
<td>$L$ : Characteristic length, in</td>
</tr>
<tr>
<td>$L_c$ : Bit body / rock clearance, in</td>
</tr>
<tr>
<td>$L_{\text{wf}}$ : Cutter wearflat length, in</td>
</tr>
<tr>
<td>$N$ : Rotary speed, 1/min</td>
</tr>
<tr>
<td>$Nu$ : Nusselt number, dimensionless</td>
</tr>
<tr>
<td>$Pr$ : Prandtl number, dimensionless</td>
</tr>
<tr>
<td>$R$ : Drilling rate, ft./h</td>
</tr>
<tr>
<td>$Q_c$ : Heat generated per unit area at the cutting surface, Btu/ft$^2$-h</td>
</tr>
<tr>
<td>$Q_{\text{wf}}$ : Heat generated per unit area at the wearflat surface, Btu/ ft$^2$-h</td>
</tr>
<tr>
<td>$S_i$ : Surface section corresponding to the ith length sliding interface, unitless</td>
</tr>
<tr>
<td>$T_r$ : Bulk rock temperature, °F</td>
</tr>
<tr>
<td>$T_s$ : Mean rock surface temperature beneath cutter, °F</td>
</tr>
<tr>
<td>$T_w$ : Mean cutter wearflat temperature, °F</td>
</tr>
<tr>
<td>$V$ : Cutting speed, ft/s</td>
</tr>
<tr>
<td>$V_{\text{ch}}$ : Rock chip velocity, ft/s</td>
</tr>
<tr>
<td>$X_2$ : Rock thermal diffusivity, ft$^2$/h</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>English lower case</th>
</tr>
</thead>
<tbody>
<tr>
<td>$d_c$ : Cutter diameter, in</td>
</tr>
<tr>
<td>$f$ : Thermal response function, ft$^2$-h-°F /Btu</td>
</tr>
<tr>
<td>$h_c$ : Penetration per rotation per cutter, in</td>
</tr>
<tr>
<td>$h_{\text{conv}}$ : Convective heat transfer coefficient, Btu/h-ft$^2$-°F</td>
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</table>

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<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
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</thead>
<tbody>
<tr>
<td>$h_{m}$</td>
<td>Convective heat transfer coefficient for mud, Btu/h-ft$^{-2}$-°F</td>
</tr>
<tr>
<td>$h_{w}$</td>
<td>Convective heat transfer coefficient for water, Btu/h-ft$^{-2}$-°F</td>
</tr>
<tr>
<td>$k_{2}$</td>
<td>Rock thermal conductivity, Btu/h-ft-°F</td>
</tr>
<tr>
<td>$k_{f}$</td>
<td>Rock/cutter friction coefficient, unitless</td>
</tr>
<tr>
<td>$k_{m}$</td>
<td>Thermal conductivity of drilling fluid, Btu/h-ft-°F</td>
</tr>
<tr>
<td>$k_{w}$</td>
<td>Thermal conductivity of water, Btu/h-ft-°F</td>
</tr>
<tr>
<td>$l_{i}$</td>
<td>Length of sliding interface, in (i=1,2,3,4)</td>
</tr>
<tr>
<td>$n_{c}$</td>
<td>Number of interfering cutters at any radius across the bit face, unitless</td>
</tr>
<tr>
<td>$p$</td>
<td>Fluid pressure, psi</td>
</tr>
<tr>
<td>$q$</td>
<td>Heat flux, Btu/ft$^{2}$-h</td>
</tr>
<tr>
<td>$q_{1}$</td>
<td>Fraction of total heat flux which goes into the cutter, Btu/ft$^{2}$-h</td>
</tr>
<tr>
<td>$q_{2}$</td>
<td>Fraction of the total heat flux which goes into the rock, Btu/ft$^{2}$-h</td>
</tr>
<tr>
<td>$q_{f}$</td>
<td>Fluid flow rate, gal./min.</td>
</tr>
<tr>
<td>$r_{c}$</td>
<td>Bit radius, in</td>
</tr>
<tr>
<td>$t$</td>
<td>Time dimension</td>
</tr>
<tr>
<td>$t_{d}$</td>
<td>Diamond layer thickness, in</td>
</tr>
<tr>
<td>$u_{rm}$</td>
<td>Cross-flow velocity of the drilling fluid at any radius across the bit face, ft/s</td>
</tr>
<tr>
<td>$u_{rw}$</td>
<td>Cross-flow velocity of water at any radius across the bit face, ft./s</td>
</tr>
<tr>
<td>$\bar{v}$</td>
<td>Fluid velocity vector, ft/s</td>
</tr>
<tr>
<td>$w_{wf}$</td>
<td>Width of the wearflat, in</td>
</tr>
<tr>
<td>$w$</td>
<td>Dimensionless cutter wear</td>
</tr>
</tbody>
</table>

**Greek lower case**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\alpha$</td>
<td>Rake angle at the cutting edge, degrees</td>
</tr>
<tr>
<td>$\delta$</td>
<td>Cutting depth, in</td>
</tr>
<tr>
<td>$\nabla$</td>
<td>Del operator</td>
</tr>
<tr>
<td>$\gamma$</td>
<td>Global rake angle, degrees</td>
</tr>
<tr>
<td>$\mu$</td>
<td>Viscosity, cp</td>
</tr>
<tr>
<td>$\rho$</td>
<td>Fluid density, lb/gal</td>
</tr>
</tbody>
</table>
Subscripts

\[ f \quad : \quad \text{Field} \\
\[ l \quad : \quad \text{Laboratory} \\
\[ m \quad : \quad \text{Mud} \\
\[ w \quad : \quad \text{Water} \\

Superscripts

\[ * \quad : \quad \text{Indicates dimensionless variable} \]
REFERENCES


APPENDIX A

PDC Cutter Temperature Model — Analytical Solution

Ortega and Glowka, (23), assumed that a PDC cutter is subjected to a uniform frictional heat flux which is stationary with respect to the cutter but which moves at the sliding velocity over the rock surface. Assuming that all the mechanical friction is converted into heat at the interface, the total heat generated per unit wearflat area per unit time is given as;

\[ q = \mu F_N V / A_w \]  \hspace{1cm} (A1)

Fraction of these heat goes into the cutter is given by

\[ q_1 = \alpha q = \alpha \mu F_N V / A_w \]  \hspace{1cm} (A2)

and fraction which goes into the rock is given by

\[ q_2 = (1-\alpha) \mu F_N V / A_w \]  \hspace{1cm} (A3)

Where \( \alpha \) is called partitioning function and defined as a function of thermal properties of the two surfaces and the sliding velocity. The solutions for the temperature field about square and band sources of heat (Fig. A-1) moving
over a semi-infinite constant property solid were given by Jaeger, [38], for the case where no heat is lost from the surfaces not exposed to heating. The mean temperature rise of the contact area, between square heat source and the surface of a semi-infinite slab was given as;

\[ T_S - T_r = \frac{4q_2}{3k_2\sqrt{\pi}} \sqrt{\frac{X_2}{V L_{wf}}} \]  

(A4)

Where \( k_2 \) and \( X_2 \) are thermal properties of the rock, \( L \) is the wearflat length parallel to the cutting direction, and \( V \) is the cutting speed. Due to the intimate contact between the cutter and the rock, it is assumed that the cutter wearflat temperature \( T_w \) is equal to the mean surface temperature of the rock. That is;

\[ T_w = T_S \]  

(A5)

Also assuming the wellbore is cooled down to the drilling fluid temperature;

\[ T_r = T_f \]  

(A6)

By introducing equations (A1), (A2), (A3), (A5), (A6) into equation (A4);
Figure - A1  Simplified frictional slider [38].
Finally, the partitioning function, $\alpha$, is solved from equation (A7) as follows;

$$T_W - T_f = \frac{4(1-\alpha)q_1\alpha}{3k_2^2\sqrt{\pi}} \sqrt{\frac{X_2}{V}}$$  \hspace{1cm} (A7)

Glowka [23] defined the ratio;

$$f = \frac{T_W - T_f}{q_1}$$  \hspace{1cm} (A9)

as thermal response function, $f$. The value of function $f$ depends upon the cutter configuration, thermal properties, and cooling rates. Thermal response function must be determined in order to compute mean wearflat temperatures. Numerical codes [10],[23] were used to estimate the values of function $f$. By substituting equation (A7) into equation (A9), the wearflat temperature is given as follows;

$$T_W - T_f = \frac{\alpha \mu F_N V}{A_w} f$$  \hspace{1cm} (A10)

In equation (A10), $F_N$ represents the normal force acting on the cutter, thereby it is directly related to the applied weight on bit. The sliding velocity $V$ is a
function of rotary speed. $A_w$ represents the cutter wearflat area. The thermal response function varies depending upon the amount of cooling imposed on the cutter by drilling fluid. In summary, equation (A10) is a unique representation of the PDC cutter temperature as a function of controllable drilling operation variables namely, weight on bit, rotary speed, and drilling hydraulics. Consequently, wearflat area (on the contrary to previous researcher's conclusion) is not necessarily the only source (or dominant source) of frictional heat generation all the time. The cutting surface area should be taken into consideration to estimate cutter temperature especially for new bits running at high drilling rates.
Similitude principle will be given in summary here as it is applied in the enhancement of optimization of bit hydraulic configurations. For detailed information reader is referred to Glowka [33], and Holman [39].

Similitude is the principle used in fluid mechanics to simulate flow fields around full-scale geometries with smaller scale models and/or different test fluids. Glowka examined the Navier-Stokes equation for fluid flow to derive this principle. For the flow of an incompressible Newtonian fluid, this equation takes the form:

$$
\rho \left[ \frac{\partial \vec{V}}{\partial t} + (\vec{V} \cdot \nabla) \vec{V} \right] = -\nabla p + \mu \nabla^2 \vec{V} \quad \text{ (B1)}
$$

He disregarded the effects of non-newtonian behavior of the drilling fluid on observable flow field. By introducing the dimensionless variables,

$$
\begin{align*}
\vec{V}^* &= \vec{V}_L \\
\overline{\vec{V}}^* &= \overline{\vec{V}} / u \\
p^* &= p / (\rho u^2) \\
t^* &= tu / L
\end{align*}
$$

Glowka, non-dimensionalized the Navier-Stokes equation as follows;
\[
\frac{\partial \mathbf{v}^*}{\partial t^*} + (\mathbf{v}^* \cdot \nabla^*) \mathbf{v}^* = -\nabla^* p^* + \left(\frac{1}{Re}\right) \nabla^2 \mathbf{v}^*
\]  

(B3)

Here, \( L \) and \( u \) are the characteristic length and velocity of the flow field respectively. The only parameter in this differential equation is the Reynolds number, \( Re = \frac{\rho u L}{\mu} \).

Hence, two flow fields with identical Reynolds number are described by the same non-dimensional governing equation. If the non-dimensional boundary conditions of the flow fields are identical, i.e.; similar geometries and properly scaled inlet velocities, the non-dimensional solutions of equation (B3) are also identical. That is, non-dimensional velocity and pressure distributions, \( v \) and \( p \), are the same in the two flow fields. The implication of this conclusion is that simulation of downhole flow fields may be accomplished in the laboratory using clear water at flow rates significantly lower than mud flow rates used in the field. For similar flow fields, Reynolds numbers become identical as:

\[
(Re)_f = (Re)_l
\]

\[
\left(\frac{\rho u L}{\mu}\right)_f = \left(\frac{\rho u L}{\mu}\right)_l
\]  

(B4)

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The effects of flow field scaling on the cutter cooling rates may be determined with the following analysis. Most forced convection heat transfer data can be correlated in the form [40]:

\[ \text{Nu} \sim \text{Re}^n \text{Pr}^m \]  

(B5)

Where \( n \) is exponent dependent upon the geometry and is an exponent with a value generally in the range of 0.3 to 0.4. Convective heat transfer data generated by Glowka [33] for a PDC cutter exposed to a uniform flow of water takes the form given as follows:

\[ h_w = \frac{k_w}{D} [3 \left( \frac{\rho_w u_w D}{\mu_w} \right)^{0.4} \left( \frac{C_{p_w} \mu_w}{k_w} \right)^{0.3} ] \]  

(B6)

Employing the definitions of \( \text{Nu} \) and \( \text{Pr} \) and using \( m = 0.3 \) gives the results for full-scale bits as follows:

\[ \frac{h_f}{h_1} = \left( \frac{\text{Re}_f}{\text{Re}_1} \right)^n \left[ \frac{k_f}{k_1} \right]^{0.7} \left[ \frac{(C_p \mu)_f}{(C_p \mu)_1} \right]^{0.3} \]  

(B7)

Since \( (\text{Re})_f = (\text{Re})_1 \) for similar flow fields the ratio of the projected field value to the measured laboratory value of
the heat transfer coefficient is simply a function of the fluid properties.

Example problem:

Properties of drilling fluid at 130 F:

<table>
<thead>
<tr>
<th>Type</th>
<th>KCL-Polymer</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density</td>
<td>9 lb/gal</td>
</tr>
<tr>
<td>Specific heat</td>
<td>1.034 Btu/lb-F</td>
</tr>
<tr>
<td>Thermal conductivity</td>
<td>0.774252 Btu/h-ft-F</td>
</tr>
<tr>
<td>Viscosity</td>
<td>5 cp</td>
</tr>
</tbody>
</table>

Properties of water at 130 F:

| Density         | 8 lb/gal    |
| Specific heat   | .9981 Btu/lb-F |
| Thermal conductivity | 0.3749 Btu/h-ft-F |
| Viscosity       | .513 cp     |

A -
Calculate maximum possible radial velocity around the PDC cutters at any radius along the bit face:

Bit type and size : 6 1/2 in. PD-2BT
Cutter diameter : 0.52 in.
Bit body-rock clearance : 0.75 in.
Number of cutters at outer radius : 8
Layout of cutters for 6 1/2 in. PD-2BT bit is given Fig. B-1. Cross-sectional area of cutter exposed to cross flow is given as follows:

\[ A_{nf} = L_c \times d_c = 0.75 \times 0.52 = 0.3575 \text{ sq.in.} \]

Effective area open to flow is given as follows:

\[ A_{eff} = \pi d_b L_c - A_{nf} \]

\[ = 3.14159 \times 6.5 \times 0.75 - 8 \times 0.3575 = 12.415 \text{ sq.in.} \]

Recommended flow rate range for 6 1/2 in. PD-2BT bit is 180 - 240 gpm. Finally radial velocity is calculated as follows:

\[ u = \frac{q_f}{3.117 \times A_{eff}} \]

\[ u_{min} = \frac{180}{3.117 \times 12.415} = 4.65 \text{ Ft./sec.} \]

\[ u_{max} = \frac{240}{3.117 \times 12.415} = 6.2 \text{ Ft./sec.} \]

B-) Calculate corresponding velocity of water having similar flow field as drilling fluid around the cutters:

Reynolds numbers are equal for systems which have the similar flow fields:

\[ (Re)_{mud} = (Re)_{water} \]
Figure - B1 Layout of cutters for a 6 1/2" PD-2BT bit.
where \( Re = \frac{\rho u D}{\mu} \)

In this case, since both fluids experience the same geometry, characteristic length (or diameter) \( D \) will be same for both fluids. Thus, from the equality of Reynolds numbers, water velocity can be written in terms of fluid properties and mud velocity as follows:

\[
u_w = u_m \left[ \frac{\mu_w \rho_m}{\mu_m \rho_w} \right]
\]

\[
u_{w,\text{min}} = 4.65 \times \left( \frac{0.513 \times 9}{5 \times 8} \right) = 0.53 \text{ ft./sec.}
\]

\[
u_{w,\text{max}} = 6.2 \times \left( \frac{0.513 \times 9}{5 \times 8} \right) = 0.71 \text{ ft./sec.}
\]

C-) Calculate convective cooling coefficient for water flowing uniformly around a PDC cutter by using correlation developed by Glowka [33];

\[
h_{\infty,w} = \frac{k_w}{D} \left[ 3 \left( \frac{\rho_w u_w D}{\mu_w} \right)^{0.4} \left( \frac{C_{pw} \mu_w}{k_w} \right)^{0.3} \right]
\]
Take \( D = \) Cutter diameter = 0.52 in.

\[
(h_{\infty, w})_{\text{min}} = \frac{0.3749 (\frac{0.9857 \times 0.53 \times 0.52}{0.513})^{0.4} (\frac{-9981 \times 513}{0.3749})^{0.3}}{0.52}
\]

\[
(h_{\infty, w})_{\text{min}} = 0.4173 \frac{\text{Btu}}{\text{h-ft-°F}}
\]

\[
(h_{\infty, w})_{\text{max}} = 0.4701 \frac{\text{Btu}}{\text{h-ft-°F}}
\]

D-) Calculate convective cooling coefficient for drilling fluid around a PDC cutter:

\[
\frac{(h_{\infty})_m}{(h_{\infty})_w} = \frac{(k_m)^{0.7}}{k_w} \left[ \frac{(C_p \mu)_m}{(C_p \mu)_w} \right]^{0.3}
\]

\[
(h_{\infty})_{m, \text{min}} = (h_{\infty})_{w, \text{min}} \times \frac{(k_m)^{0.7}}{k_w} \left[ \frac{(C_p \mu)_m}{(C_p \mu)_w} \right]^{0.3}
\]

\[
= 2.37 \left( \frac{0.774}{0.3749} \right)^{0.7} \left( \frac{-1.034 \times 5}{0.9981 \times 0.513} \right)^{0.3}
\]

\[
(h_{\infty})_{m, \text{min}} = 1.391 \frac{\text{Btu}}{\text{h-ft-°F}}
\]

\[
(h_{\infty})_{m, \text{max}} = 1.567 \frac{\text{Btu}}{\text{h-ft-°F}}
\]
CHAPTER III.

FRICITION UNDER PDC CUTTERS AND ITS IMPLICATION IN BIT LIFE PREDICTION

ABSTRACT

A new PDC bit wear model was derived and used for bit performance prediction. The model relates bit life with temperature, weight on bit, rotary speed, cutter geometry, and frictional heat fluxes through wearflat and cutting surfaces.

The predictions showed that the friction at the cutter/rock interface has more influence on bit life than convective cooling provided by drilling fluid. Friction could become so severe that under certain conditions, drilling fluid simply would not provide cooling necessary for adequate bit life.

It is also theoretically proven (by using 2-D numerical model simulating heat transfer within the cutter) that convective cooling of a PDC cutter by drilling fluid has physical limitation.

Effect of drilling operation variables namely weight on bit, rotary speed, and drilling fluid flow rate, on the bit life was determined by using the new model. Results showed that many field failures of PDC bits may have been
caused by lack of understanding operational limits imposed by heat considerations.

INTRODUCTION

Because of their aggressive nature polycrystalline diamond compact (PDC) bits can generally drill as fast as or faster than roller bits in any environment. Therefore, economics of a given bit run is primarily determined by the operational life of the bit in the absence of other drilling problems. Lin et al., [1,2], investigated the impact of drilling rate and bit life on geothermal well costs using a drilling cost simulator. It was seen that by increasing the drilling rate and the bit life twice, the total well cost could be reduced by % 10-15. If, on the other hand, the PDC bit life is reduced %50 due to the drilling operation conditions, savings are also cut by % 50. In particularly severe environments, PDC bits can actually increase drilling costs because of the reduced life. In other words, short runs with PDC bits are costly. Hence, in order to take the advantage of higher penetration rates with PDC bits, care should be taken to prolonge their operational life.

Theory of frictional wear of materials [3] suggests that, in the absence of thermal effects, the volumetric wear of a metal specimen sliding on an abrasive surface is proportional to the load, $F_n$, on the specimen and the
sliding distance $l$. That is;

$$V \sim F_N \cdot l$$

or in other words;

$$\frac{1}{F_N} \frac{dV}{dl} = \text{Const.}$$

If the thermal effects were not important, equation (2) indicates that volumetric wear rate per unit load should be constant. A bit life relationship based on this assumption was developed previously [4]. However, both rock cutting and metal cutting experiences do not show this linear trend. The effect of frictionally generated heat on the PDC cutter is such that the cutter experiences an accelerated wear. Therefore, in the presence of thermal effects, the wear mechanism can not be explained only by abrasion and the resultant microchipping. The nature of the dependency of wear rate on the temperature is yet to be understood completely.

EFFECT OF MATERIAL HARDNESS ON THE VOLUMETRIC WEAR RATE

Earlier studies on abrasive wear of various metals revealed that the rate of abrasive wear is primarily determined by metal hardness [5], [6], [7], [8].
Larsen-Basse [5] studied the abrasive wear of cemented carbide composite. His results showed that the rate of abrasive wear of cemented carbides is primarily controlled by the alloy hardness. A transition in mechanism exists when the ratio of abrasive hardness to alloy hardness is about 1.2.

For harder abrasives, the removal rate is rapid and proportional to the hardness of the abrasive. The material removal mechanism is primarily plastic deformation in the formation of craters and grooves in the surface. The alloy hardness is the prime factor controlling abrasion resistance which is defined as the inverse of the volumetric wear rate. Fig.1 shows variation of abrasion resistance wrt. bulk specimen hardness.

Deviation at very high hardness levels is due to the approaching hard-soft transition. At this stage, composite hardness becomes higher than abrasive hardness and the material removal mechanism is totally different (soft-abrasion mode).

For soft abrasives, the mechanism of wear is based on a very localized interaction between the abrasive grit and the constituents of the microstructure. Frictional forces from the abrasives gradually extrude some binder metal, this results relaxation of compressive stresses in the carbide grains and they gradually fragments by fatigue.
Figure 1- Abrasion resistance vs. WC-Co bulk hardness, after [5].
type loading. The removal rate in this region is controlled by binder mean free path.

Fish [6] studied the effect of cutting edge hardness on the wear rate of a WC-Co blade of a drag bit. Results presented in Fig. 2 show that bit life varies exponentially with carbide hardness. He concluded that, to minimize the effect of wear and to extend the operational acceptability of rotary drilling to the widest range of rocks, it is clearly, desirable to use the hardest possible carbide. This however leads to a dilemma, in that, in general, the harder the carbide, the more brittle it is and the more likely it is to fracture under the rotational impact forces.

Data from Latin [7] shows that the hardness itself possibly does not give a singular measure of wear resistance (Fig. 3). There appears also to be influence of WC grain size. For the harder alloys, a fine grained material has somewhat better wear-resistance than a course grained material of the same hardness. For the softer, more ductile alloys, it appears that the opposite may be true, though the difference in this case is small.

Kenny and Johnson [8] presented results of the test with various tool tip materials. Fig. 4 shows that the wear of tool tips depends not only on hardness but also on the grain size. The fine grained material gives a greater wear
Figure 2- Variation of bit life with hardness of carbide, after [6].
Figure 3- Volumetric wear resistance vs. hardness, after [7].
Figure 4- Volume of material removed from the cutting edge for tools with various tip materials of different hardness, after [8].
resistance than a coarse grained material.

By using the experimental evidences given above, it is postulated that the wear rate is inversely proportional to the hardness. Hence, abrasive wear relationship (equation 2) is modified as follows;

\[
\frac{1}{F_N} \frac{dV}{dl} = f(H)
\]  

(3)

An exponential relationship of the form;

\[
\frac{1}{F_N} \frac{dV}{dl} = c_1 e^{-c_2 H^2}
\]  

(4)

is proposed to express the dependency of the wear rate on the hardness mathematically. The data from Fig. 1, [5], were used to verify the equation (4). Abrasion resistance is defined as the inverse of volumetric wear rate per unit load. Equation (4) suggests that the plot of natural logarithm of volumetric wear rate per unit load against temperature should give a straight line with decreasing slope. The data from Fig. 1 were rearranged and presented in Fig. 5 again. The linear trend of the plotted points indicates that the proposed model is satisfactory to explain the nature of the relationship between wear rate and material hardness.
Figure 5- Verification of the proposed correlation between volumetric wear rate and metal hardness.

\[ y = 2.7583 - 4.2052e^{-7}x \]

\[ R^2 = 0.908 \]

Experimental data, after [5].
TEMPERATURE EFFECT ON THE POLYCRYSTALLINE DIAMOND HARDNESS

Lee and Hibbs [9] presented data on the wear of synthetic diamond during rock cutting. Tests showed that the Vickers hardness of sintered diamond drops of about 65% of its room temperature value (Fig. 6). At temperatures about 750 °C, the differential thermal stresses in the microstructure cause failure of the diamond-diamond grains from the compact. PDC's thus have no practical life under conditions that lead to temperatures above 750 °C.

Indentation hardness testing was used to obtain hot hardness values and to observe the deformation characteristics of a polycrystalline sintered diamond compact. The hardness-temperature curve was determined using 5 kg load to amplify the microcracking effect on the hardness. The Vickers hardness for a diamond compact up to 700 °C is given in Fig. 6, together with the published hardness curve of an unspecified diamond as determined by Loladze [10]. A consistent decrease in hardness with increasing temperature reported in both case. The relationship between sintered diamond hardness and temperature (as implied by Fig. 6) can be mathematically expressed as follows;

\[ H = c_3 - c_4 T \]  

(5)

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Figure 6- High temperature hardness of diamonds, after [9].
TEMPERATURE EFFECT ON THE VOLUMETRIC WEAR RATE OF POLYCRYSTALLINE DIAMOND DURING ROCK CUTTING

By analysing the results of previous research, it is reasonable to assume that the microchipping mode of wear for the individual diamond crystal grains in polycrystalline sintered compact is enhanced by the increase in the tool-rock contact temperature.

Since wear resistance varies depending upon the bulk hardness of the specimen, any reduction in hardness due to temperature increase will induce reduction in wear resistance as well.

In conclusion, volumetric wear rate of a polycrystalline diamond compact is in fact, dependent upon its temperature. Or, in other words;

\[
\frac{dV}{F_N \, dl} = f(T)
\]

(6)

The nature of the function \( f(T) \) was inferred from the analysis of the results presented above. Considering the dependency of wear rate on the material hardness and the dependency of material hardness on temperature, the analysis suggested that the relation between volumetric wear rate and temperature is of exponential type. By
introducing equation (5) into equation (4) volumetric wear rate per unit load as a function of temperature can be expressed mathematically as follows:

\[
\frac{1}{F_N} \frac{dV}{dl} = a e^{b'T} + cT^2
\]  

(7)

This conclusion is confirmed by the data presented in previous studies [9], [11].

Lee and Hibbs [9] recorded change in cutter wear-flat temperature with respect to sliding velocity. They observed an accelerated wear of diamond compact cutters with increasing sliding velocity (hence, with increasing temperature) as shown in Fig.7.


Fig.8 shows that the PDC cutter wear rate decreases with temperature until a wear-flat temperature of 350 °C (570 °F) is reached, after which wear rate increases. Glowka [11] attributed the decrease in wear to the thermal healing of defects in the microstructure, and the increase
Figure 7-The rate of diamond tool wear per unit cutting distance as a function of cutting speed, after [9].
Figure 8- Experimental PDC cutter wear rate per unit weight on cutter while cutting hard, abrasive sandstone, plotted against computed wearflat temperature for the same conditions.
to the general softening at the higher temperatures. Similar behavior with the adhesive wear of cemented carbides was reported by Dawihl and Frisch [14].

In order to verify the model, wear rate per unit load values were calculated by using equation (7) (for the same temperature range of Fig. 8) and they were plotted in Fig. 9. The solid curve in Fig. 9 follows the general trend presented by the experimental data. Hence, the proposed model for wear-temperature relationship of polycrystalline diamond compact cutter is verified.

From the practical point of PDC bit drilling, the effect of thermal healing on bit life was ignored and only the region where wear accelerates due to temperature effect is considered for simulation purpose. Fig. 10 shows the correlation between the wear model and the experimental data belong to accelerated wear region (i.e. temperature > 400 °F).

Unfortunately, Fig. 8 provides only available measured values in this matter; so the verification of the model is limited to these data at the moment.

BIT LIFE MODEL

Analytical solution of Ortega and Glowka [13] gives cutter wearflat temperature in terms of drilling operation variables as follows:
Figure 9- Verification of new bit wear model

- Experimental data, after [9].
- Simulation
Figure 10- Correlation used for simulation study
By substituting equation (8) into equation (7) for wearflat temperature and rearranging, the bit life equation is obtained as follows:

\[ t_b = K \cdot f(w) \]  

Where, \( f(w) \) is some complex function of cutter wear, \( w \), and it is evaluated by using a numerical integration technique at any given bit wear. \( K \) is a constant. Derivation of the equation (9) is given in Appendix A.

Practical use of equation (9) calls for predetermination of normal force, \( \frac{F_N}{N} \), distribution across the bit face. A cutter placement criterion is needed to ascertain force distribution. Several criteria has been discussed by Glowka [15]. Rigorous computer codes have been developed based on these criteria to determine cutter wear distribution and force distribution under any drilling condition [16],[17].

A fictitious cutter has to be assigned in order to pursue bit performance prediction by using equation (9). A fictitious cutter is the one which is exposed to most wear. A suggested way to determine the fictitious...
cutter and its location on the bit face is to run the bit under certain drilling condition in a specific rock and check to see wear distribution across the bit face. This information can be provided by bit manufacturers for each bit type which is tested under average drilling operational conditions in a specific rock.

The location of the fictitious cutter is used as a characteristic radius in the bit life relationship. Finally, under any drilling condition, the normal force effective on the predetermined fictitious cutter for a given bit geometry and drilling rate can be calculated by using one of the available simulators [16], [17].

EFFECT OF DRILLING PARAMETERS ON THE BIT LIFE

Simulation studies [15], [18] showed that under downhole drilling conditions, the PDC cutter temperature would easily reach to the point over which cutter wear starts accelerating. Hence, practically, bit experiences thermal healing for only a shortwhile then, thermal deterioration overcomes.

Fig.11 shows an exponential decrease in bit life with increasing rotary speed as it had been observed experimentally elsewhere [9], [12].

The effect of normal force on the cutter life is shown in Fig. 12. Bit life reduces to zero exponentially
Figure 11- Effect of rotary speed on bit life.
Figure 12- Effect of normal force on the bit life.
with the increasing normal force.

Any change in drilling fluid temperature would also effect the cutter life. Fig. 13 shows that, within the practical range of drilling fluid temperature [100-400 °F], the cutter life may be reduced (or prolonged) up to 300%. The drilling fluid temperature is controlled mainly by geothermal gradient. It changes gradually as the bit goes deeper in the well. Mud temperature should be taken into account for the optimum bit run especially in deep-high temperature wells.

The drilling fluid which flows vigorously around an individual cutting element may effect cutter temperature by influencing convective heat removal from the lateral surfaces, by lubricating the contact between wearflat and the rock, and by removing rock chips and obstructions which influence the cutting efficiency. Every bit designer has tried to increase convective cooling in an effort to cool the PDC cutter. Laboratory testing and field experiences have made evolutionary improvements in this area. Further improvement is likely to be only of marginal value because of two physical limits. First, at some point high mud flow rates are counter-productive because they cause unacceptable fluid erosion. Second, there is a cooling limit at which cutter temperatures can not be reduced by any amount of mud flow. At a certain level of
Figure 13- Effect of mud temperature on the bit life.
friction, the tungsten carbide can not conduct heat away from the wearflat to the convecting surfaces of the stud fast enough to limit the temperature of the wearflat. This often happens in hard formations where larger weight on bit or higher rotary speed is required in order to maintain drilling rate. Increasing mud flow to improve the convective cooling does not help much to control temperature because the thermal resistance of PDC layer becomes limiting factor for heat transport.

Ortega and Glowka [13] introduced the thermal response function as a representation of cutter's response to heat influx. Thermal response function has a characteristic shape. It is a function of convective cooling coefficient, and cutter geometry only. A simulation study conducted by using a 2-D numerical model verified the Ortega's conclusion. As it is seen from Fig. 14, for the higher value of convective cooling coefficient, the thermal response function curve levels off. In other words, drilling fluid becomes inefficient in terms of convective cooling at high flow rates.

Perhaps, the most important parameter which controls cutter life is the friction coefficient. Fig.15 shows that by reducing the friction coefficient from 0.15 to 0.1, the bit life can be prolonged three times more.

Equation (9) implies that if the friction
Figure 14- Effectiveness of convective cooling on the thermal response of a PDC cutter.
Figure 15- Effect of friction coefficient on the bit life.
coefficient (friction force) is somehow reduced then, the PDC cutter life can be improved considerably. This effect is much more pronounced than that of a convective cooling. Fig.16 shows that by reducing friction coefficient 30 %, the cutter life can be extended up to 500 %; whereas if we improve the degree of cooling by 400 %, the cutter life can be prolonged only by 250 %. Also shown in Fig.16 is that the degree of cooling becomes insignificant at low friction coefficient. Thus, if it is possible to reduce friction coefficient than, high flow rates of drilling fluid is not needed anymore to provide efficient cooling to the cutters. At high friction coefficient condition however, no matter how high is the cooling rate, the bit life is shortened by tremendous heat generated frictionally.

EFFECT OF CUTTER WEAR CONSIDERATIONS ON THE OPTIMIZED PDC BIT CONTROL

The new bit life model was introduced into the optimized bit control theory. The optimum bit performance can be determined by using maximum bit performance [MBP] curves [4]. In an engineering sense, a MBP curve represent a relationship between any possible footage made by a single drill bit and its maximum drilling rate.

In this study, the PDC bit optimization model and the computer code for MBP curves were modified by
Figure 16- Effect of friction coefficient vs. convection coefficient on cutter life.
Introducing the new bit life relationship. An example MBP curve is presented in Fig. 17. Here, maximum drilling rate was determined for each given footage provided that the bit is completely worn out at the end of each run. A constant (optimum) weight on bit and a constant (optimum) rotary speed were imposed on the bit throughout the run. Also, the bit is exposed to a constant cooling by drilling fluid (i.e. constant flow rate and flow geometry were assumed). Data used for the optimization study is presented in Table-1.

The new bit life equation accounts for the effect of temperature. Consequently, an optimized bit run is controlled by the cutter temperature which varies depending upon the cutter wear and the cutting depth (i.e. drilling rate). A typical temperature profile for a 4500 ft. bit run is given in Fig. 18. The cutter temperature takes a high value at the beginning of run, then it goes down as the bit wears out.

Approach taken here does not include optimization of bit performance from the standpoint of optimum cooling of cutters. Optimization of cooling rate which depends on flow rate, mud properties as well as bit geometry needs further investigation to improve bit performance.
Figure 18- Change in wearflat temperature during 4500 Ft. drilling.
Figure 17- Maximum bit performance curve (PDC bit).
**TABLE-1**

**DATA USED FOR OPTIMIZED BIT RUN SIMULATION**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bit size</td>
<td>9 7/8 in.</td>
</tr>
<tr>
<td>Bit type</td>
<td>Fish tail bit</td>
</tr>
<tr>
<td>Number of blades</td>
<td>4</td>
</tr>
<tr>
<td>Convective cooling coefficient</td>
<td>0.35 Btu/h-ft(^{2})-°F (2 W/m(^{2})-°C)</td>
</tr>
<tr>
<td>Total number of cutters</td>
<td>27</td>
</tr>
<tr>
<td>Rock thermal conductivity</td>
<td>1.1 Btu/hr-ft-°F</td>
</tr>
<tr>
<td>Friction coefficient</td>
<td>0.15 (Berea sandstone)</td>
</tr>
<tr>
<td>Rock thermal diffusivity</td>
<td>0.0016 in(^{2})/s.</td>
</tr>
<tr>
<td>Back rake angle</td>
<td>20 degrees</td>
</tr>
<tr>
<td>Diameter of a single cutter</td>
<td>0.52 in.</td>
</tr>
<tr>
<td>Drillability constant</td>
<td>0.001 ft/lbf</td>
</tr>
</tbody>
</table>

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DISCUSSION

The model is limited with the assumed mechanism of frictional heat generation during rock/cutter interaction. In this case, frictional heat generation model [13] was extrapolated from the metal cutting practice and applied in rock cutting theory. Heat generation mechanism at the sliding interfaces of the two solid body (such as metal-to-metal) may not be representative for rock cutting by PDC cutters. Under the sliding action of the cutter, solid rock will disintegrate and generate a transition zone out of loose grains between itself and the PDC cutter surface. Presence of drilling fluid would even enhance this process further. Therefore, the nature of friction process and the resultant heat generation would be different than the case where two solid body rub on each other.

The frictionally generated heat is equal to the work done by the frictional forces. The results of simulation study showed that if the friction forces are somehow reduced, than the bit life would be improved significantly. However, the mechanisms of possible reduction in friction forces are yet to be understood.

The metal cutting theory suggests that effective lubrication would reduce the friction forces. However, rock cutting practice shows different features than that
of metal cutting practice. In general, much higher normal forces are encountered in rock cutting than metal cutting. Some of the friction mechanisms (such as adhesion and metallic bonding) which are important for metal-to-metal contact are not as important in rock-cutter interaction. Lubrication mechanism is also different at metal-to-metal interface than at the rock/cutter interface. A protective film of lubricant, for example, may not form at rock/cutter interface as effectively as it is formed at metallic interfaces. Therefore, friction process, lubrication and wear should be investigated further under conditions similar to rock cutting by PDC cutters.

CONCLUSIONS and RECOMMENDATIONS

A new PDC bit wear model was derived and used for bit performance prediction. Model relates bit life with temperature weight on bit, rotary speed, cutter geometry, and frictional heat fluxes through wearflat and cutting surfaces.

The new model shows that the effect of friction at the cutter/rock interface has more influence on the bit life than convective cooling provided by drilling fluid.

The convective cooling of a PDC cutter by drilling fluid has physical limitation, under conditions where the friction become so severe that the tungsten carbide can
not conduct heat away from the wearflat to the convecting surfaces of the cutter fast enough to limit the temperature of the wearflat.

Friction process, lubrication and wear should be investigated further under conditions similar to rock cutting by PDC cutters.

The practical use of new bit model calls for the determination of load distribution across the bit face. Further research is necessary to find a systematic way to define distribution of forces among the cutters during rock cutting.

An optimized bit run will not be achieved unless the variations in flow conditions (i.e., flow rate, mud properties) and in bit geometry are taken into account. Further research is also needed to determine drilling fluid velocity around cutters in order to maximize cooling process.
**Nomenclature**

**English upper case**

- **$A_w$** : Cutter wearflat area, sq.in.
- **$F_N$** : Normal force acting at the cutter wearflat, lbf.
- **$H$** : Metal hardness, kg/mm$^2$.
- **$K$** : Constant in bit life equation.
- **$L$** : Cutter wearflat length, in.
- **$T_{fl}$** : Drilling fluid temperature, °F.
- **$T_w$** : Mean cutter wearflat temperature, °F.
- **$V$** : Volumetric cutter wear, cu.in.
- **$V_s$** : Cutting speed, ft/sec.
- **$X_2$** : Rock thermal diffusivity, ft$^2$/hr.

**English lower case**

- **$a, b, c$** : Correlation coefficients, unitless.
- **$c_1, c_2, c_3, c_4$** : Correlation coefficients, unitless.
- **$d_c$** : Diameter of a single cutter, in.
- **$f$** : Thermal response function, ft$^2$-hr-°F/BTU.
- **$f(w)$** : Function defined by equation (A-11), unitless.
- **$h_\infty$** : Convective heat transfer coefficient, BTU/hr-ft$^2$-F
- **$k_2$** : Rock thermal conductivity, BTU/hr-ft-°F
- **$l$** : Distance travelled by a single cutter, in.
- **$r_c$** : Radius at which fictitious cutter is located, in.
\[ t_b \quad : \quad \text{Bit life, hrs.} \]
\[ w \quad : \quad \text{Dimensionless linear cutter wear, unitless.} \]

Greek lower case

\[ \alpha \quad : \quad \text{Rake angle at the cutting edge, degrees.} \]
\[ \mu \quad : \quad \text{Friction coefficient between cutter and rock, unitless} \]
REFERENCES


APPENDIX

Ortega's analytical solution [13] gives functional relationship between cutter wearflat temperature and drilling operation variables as follows:

\[ T_w = T_{fl} + \frac{\mu F_N V_s f}{A_w \left( 1 + \frac{3\pi}{4} k_2 f \sqrt{\frac{V_s}{X_2 L}} \right)} \]  

(A1)

The relationship between volumetric wear rate per unit load and temperature derived from metal cutting theory is given as follows:

\[ \frac{1}{F_N} \frac{dV}{dl} = a e^{bT_w} + cT_w^2 \]  

(A2)

Correlation coefficients \( a, b, c \) were determined by applying equation (A2) to the experimental data presented by Glowka [11]. The values of \( a, b, c \), in this case, were estimated as follows:

\[ a = 4 \times 10^{-13} \]
\[ b = -6.13 \times 10^{-3} \quad \text{(A3)} \]
\[ c = 9.17 \times 10^{-6} \]

and \( dl \) is, the unit distance travelled by the cutter, given as;
Volumetric wear, $V$, is given as:

$$V = d_c \cos \alpha \int_0^1 A_w \, dw$$  \hspace{1cm} (A5)\]

by using chain rule:

$$\frac{dv}{dt} = \frac{dv}{dw} \frac{dw}{dt} = d_c \cos \alpha \frac{A_w}{dt} \hspace{1cm} (A6)$$

by introducing (A4) and (A6) into (A2), we obtain:

$$\frac{d_c \cos \alpha \, A_w}{120\pi r_c N F_N} \, dw = a e^{b T_w} + c T_w^2 \hspace{1cm} (A7)$$

Then, by substituting (A1) into (A7) for $T_w$ and separating variables, we get:

$$\frac{dt}{d_c \cos \alpha} = \frac{A_w}{120\pi r_c N F_N} \frac{A_w}{e^{b (\frac{E_t + T_{f1}}{f_1}) + c (\frac{E_t + T_{f1}}{f_1})^2}} \, dw $$  \hspace{1cm} (A8)

Where:

$$f_1 = A A_w + B A \frac{A_w}{Y_w}$$
A = \frac{0.172}{\mu r_c N f} \quad (A9)

B = 5.16 f k_2 \sqrt{\frac{r_c N \sin \alpha}{d_c X_2}}

Finally by integrating (A8), we obtain bit life equation as follows;

\[ t_b = K f(w) \quad (A10) \]

Where;

\[ K = \frac{d_c \cos \alpha}{120 \pi a r_c N F_N e^{\left( \frac{b T_{t_1} + c T_{t_1}^2}{f_{t_1}} \right)}} \quad (A11) \]

\[ f(w) = \int \frac{A_w}{e^{\left( \frac{c F_{x_2}^2}{f_{x_2}^2} + b F_{x_1} + 2 c \frac{F_{x_1} T_{t_1}}{f_{x_1}} \right)}} \, dw \]
CHAPTER IV.

AN EXPERIMENTAL STUDY OF FRICTION INDUCED BY PDC CUTTERS DURING ROCK CUTTING

ABSTRACT

A new laboratory instrument and a method are described to measure friction between the sliding surface of a polycrystalline diamond compact (PDC) cutter and the surface of a rock.

The design offers a physical separation of two phenomena inherent in the action of a single cutter during drilling. These phenomena include: (1) an instantaneous exposure of the clean rock surface through cutting at the cutter's front and (2) sliding action across the cutter wearflat area. Such design enables direct measurement of the friction force unaffected by the cutting force.

The instrument is unique among other means for studying friction which employed standard machine-shop tools for cutting metals (lathes, milling machines, drills) with rigid imposition of normal force and no instantaneous exposure of the freshly cut rock surface. The measurable variables include rotary speed of the rock, normal and tangential forces, and composition of the circulating liquid. Water, water-base mud, and oil base
mud were used to investigate lubricant effect on the sliding friction forces. Lithology effect was investigated by using samples of Berea Sandstone, Nugget Sandstone, Mancos Shale, and Sierra-White Granite.

Friction forces were insensitive to any change in lithology and lubricants within the tested range of materials. The interaction between borehole fluid and rock seems to control mechanism of friction though the friction coefficient value is little affected. Measured friction coefficients were independent from normal forces, thus satisfying Amonton's law of friction. Friction coefficient either decreased or remained same with increasing sliding speed. The effect of sliding speed is more pronounced for the weaker, coarse grained rocks.

INTRODUCTION

Polycrystalline diamond compact cutters' operation involves frictional forces resulting in heat generation. It is well documented [1] that the high heat induced by a drag cutter greatly reduces its operational life. Although mechanically efficient, the shearing action of the PDC drag bit produce high amount of friction at the cutter/rock interface. Laboratory tests[2] and simulation studies (chapter 3) reveal that friction has more influence over bit temperature than cooling action of drilling fluid.
The drilling fluid which flows vigorously around an individual cutting element may affect cutter temperature by: (1) providing convective heat removal from the lateral surfaces; (2) lubricating the contact between wearflat and the rock, and (3) removing rock chips and obstructions which influence the cutting efficiency.

Every bit designer has tried to increase convective cooling in an effort to reduce temperature of the PDC cutter. Laboratory testing and field experiences have made evolutionary improvements in this area. Further improvement is likely to be small because of two physical limits. First, at some point, high mud flow rates are counter-productive because they cause unacceptable fluid erosion. Second, there is a cooling limit at which cutter temperatures can not be reduced further by any amount of mud flow. This latter point deserves further explanation.

At a certain level of friction the tungsten carbide can not conduct heat away from the wearflat to the convecting surfaces of the stud fast enough to limit the temperature of the wearflat. This often happens in hard formations where larger weight on bit or higher rotary speed is required in order to maintain drilling rate. Increasing mud flow to improve the convective cooling does not help much to control temperature because the thermal resistance of PDC layer becomes limiting factor for heat transport [1]. Consequently, reduction of the
friction at the cutter/rock interface emerges as an alternative method of controlling heat generation. Metal cutting practice [3] showed that higher frictional forces (six-to-seven fold) were required to produce visible hot spots when the surfaces are flooded with liquid.

The principal factors that control the magnitude of the frictionally generated heat include: (1) cutter wear, (2) cutting speed, and (3) the friction coefficient between the rubbing parts (diamond and stud material rubbing against the rock).

PDC cutters have been tested to identify their wear mechanisms by using single-cutter experiments [4], [5], [6], laboratory experiments with full-scale prototype bits [5], [7], [8], and field tests of full-scale bits designed using laboratory data [9], [10], [11]. However, even though the importance of friction on the cutter wear rate is well documented, there are insufficient data indicating the degree of possible reduction in friction by use of various drilling fluids and additives and hence, to improve the cutter life.

Hibbs et al. [4] provided the only available data to date on measured sliding friction coefficients. They used a 54 in. vertical turret lathe. The forces acting on the PDC cutter were measured as the cutter moved many times across the same ridge on the rock surface.
Hibbs' experiments did not simulate the actual behavior of the cutter. In reality, the PDC cutter's drilling action includes an exposure of the new rock surface through cutting followed by a sliding action across the cutter wearflat area. Also in Hibbs' experiments the thrust force was applied by continuous down feeding. Therefore normal force imposition was not controlled directly. Because the normal force was resulted from downward displacement of the cutter, it is very likely that cutting the rock rather than sliding on the rock surface occurred. Therefore, the tangential force measured in their experiments was in fact equal to the combined effect of cutting and friction forces. Finally, their investigation of the effect of different lubricants on the friction coefficient was not conclusive.

The purpose of this research is to evaluate the friction factor experimentally and to investigate the effect of various factors such as, normal force, sliding velocity, type of drilling fluid, and type of rock on the magnitude of the sliding friction coefficient.

EXPERIMENTAL DESIGN

A prototype instrument was designed and constructed to measure sliding friction coefficient between the PDC cutter and rock surface. The design provided a physical separation of two phenomena inherent in the action of a
single cutter during drilling. These phenomena include: (1) an instantaneous exposure of the new rock surface through cutting at the cutter's front and, (2) sliding action across the cutter wearflat area. Such a design enables direct measurement of the friction force unaffected by the cutting force.

Physical principle of the single cutter friction experiment is shown in Fig. 1. A normal force is imposed by the PDC cutters on the both surfaces of a rotating disc of rock surface. The PDC cutters are connected to air cylinders which are mounted on the freely suspending test frame. As the rock turns, it will pull the test frame downward (due to friction forces). The magnitude of these downward forces (friction forces) and the magnitude of the normal forces can be measured. Finally, the friction coefficient is calculated from the ratio of friction force to the normal force.

The experimental set-up consists of several parts operating independently: (1) load frame (Fig. 2-a), (2) prime mover and shaft (Fig. 2-b), (3) rock surface cleaning set-up (Fig. 2-c), (4) rock (Fig. 2-d), (5) closed-loop circulation system (Fig. 3-a), (6) pressure supply and control system (Fig. 3-b), (7) data acquisition system (Fig. 3-c).

The load frame was constructed of black iron.
Figure 1 - Physical principle of single cutter friction experiment
Figure 2- Experimental set-up for friction coefficient measurement, general view.

Figure 3- Experimental set-up for friction coefficient measurement, general view.
(Fig. 4). Two SHEFFER C-20-FF air cylinders (Fig. 4-a) modified with reducing adaptors (Fig. 4-b) for PDC cutter connection were mounted on the iron frame. Air cylinders were used to impose pressure (thrust force) on the cutters. The use of air cylinders also provided flexibility to the normal force imposition. The alignment of the load frame was accomplished by hanging it freely on the main structure.

A 3 HP-DC motor turns 1 7/8" diameter shaft (through a chain drive system) which eventually rotates a core slab fixed onto the shaft (Fig. 5). A gear box mounted on the motor provides variable speed from 20 to 180 rpm (50-500 ft/min linear velocities) which represent rotary speed values in conventional drilling operation.

Two heavy duty steel (end) brushes (Fig. 6) are used to maintain constant rate cutting on both surface of the core slab. Brushes were fixed onto die grinders which were hold by steel barrels and loaded by springs to provide light weight required for effective resurfacing.

A closed-loop hydraulic system delivers drilling fluid onto the sliding surface between rock and cutters (Fig. 7). In the earlier stage of the experiments, a progressive cavity type pump was used. It supplied drilling fluid at a maximum rate of 9.4 gal/hr under maximum 50 psi. discharge pressure. Later, however, this
Figure 4- Load frame.
Figure 5- A 3 HP-DC motor with variable speed gear control box.
Figure 6- Rock surface cleaning set-up.
Figure 7- Closed loop hydraulic system.
pump was broken down and a centrifugal pump was employed. Both pumps have 3/4 in. suction and discharge units. The hydraulic impact of the drilling fluid was improved by reducing the size of the pump discharge (3/4 in. diameter) to the size of pipes (1/2 in. dia.) conducting liquid.

Two SYNDRILL SD1038 PDC cutters were used. The cutter has 0.529 inches diameter and 0.315 inches thickness.

The lithology effect was checked upon by using Berea Sandstone, Mancos Shale, Nugget Sandstone and Sierra-White Granite. The uniaxial compressive strength of tested rock samples are given in Table 1. The core slabs were cut in 10 1/2 in. diameter and 3 1/2 in. thickness with 2 in. diameter concentric hole.

Water, water-base mud and oil-base mud were tested for their effect on friction coefficient. Their properties are shown in Table 2. In addition, a lubricant commercially known as IDLUBE was also tested. IDLUBE is originally used for borehole friction reduction. It is composed of vegetable oil mixed with naturally occurring glycerides.

MEASUREMENTS AND CALIBRATION

The measurable variables included rotary speed of the rock, normal and tangential forces acting on the
## TABLE 1 - ROCK SAMPLES TESTED

<table>
<thead>
<tr>
<th>ROCK TYPE</th>
<th>Bulk Density gr/cc</th>
<th>Porosity %</th>
<th>Permeability md.</th>
<th>UNIAXIAL COMpressive STRENGTH, psi</th>
</tr>
</thead>
<tbody>
<tr>
<td>Berea sandstone</td>
<td>2.1</td>
<td>24</td>
<td>300</td>
<td>8,500</td>
</tr>
<tr>
<td>Nugget sandstone</td>
<td>2.23</td>
<td>10</td>
<td>7</td>
<td>23,000</td>
</tr>
<tr>
<td>Mancos shale</td>
<td>2.54</td>
<td>1.4</td>
<td>-</td>
<td>9,700</td>
</tr>
<tr>
<td>Sierra White Granite</td>
<td>2.65</td>
<td>&lt; 1</td>
<td>-</td>
<td>28,200</td>
</tr>
</tbody>
</table>
Table 2 - DRILLING FLUIDS TESTED

<table>
<thead>
<tr>
<th>Drilling Fluid</th>
<th>Basic Additives</th>
<th>pH</th>
<th>Density (lbm/gal)</th>
<th>Low-Gravity Solids (wt %)</th>
<th>High-Gravity Solids (wt%)</th>
<th>Oil (vol%)</th>
<th>Water (vol%)</th>
<th>Chlorides (ppm)</th>
<th>Methylene Blue Test (cc)</th>
<th>API Filtration (ml/30 min)</th>
<th>Viscosity (cp)</th>
<th>Yield Point (lb/100 sq.ft.)</th>
<th>Gel, 10-sec/10-min (lb/100 sq.ft.)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Water Base Mud</td>
<td>Bentonite, Ca-Ligno.</td>
<td>9</td>
<td>12.2</td>
<td>12</td>
<td>11</td>
<td>0</td>
<td>77</td>
<td>5500</td>
<td>4</td>
<td>10.5</td>
<td>32</td>
<td>16</td>
<td>4/11</td>
</tr>
<tr>
<td>Oil Base Mud</td>
<td>Diesel</td>
<td>6.5</td>
<td>12.2</td>
<td>13</td>
<td>10</td>
<td>50</td>
<td>27</td>
<td>200,000</td>
<td>.</td>
<td>1.5</td>
<td>58</td>
<td>14</td>
<td>15/30</td>
</tr>
</tbody>
</table>
A GSE 5353 Aluminum load cell (Fig. 4-c) with a maximum load capacity of 1000 lb. was used to measure the tensile loads (tangential force acting on the cutter). The equipment has 2 mv / v output at rated capacity with 0.05 % full-scale sensitivity. Its maximum excitation voltage is 10 VDC.

A TRANSMETRICS P21AB (Fig. 8-a) pressure transducer rated for maximum 200 psig. was used to measure the thrust applied on the cutters. The accuracy of the equipment is 0.5 % of a full-scale reading. Its maximum excitation voltage is 18 VDC.

Pressure in the air cylinders (normal force on the rock surface) is controlled manually through the VICTOR 250 BL-555 Nitrogen regulator mounted on the high pressure Nitrogen tank (Fig. 8-b).

All readings (i.e., pressure, and friction force) were accomplished by using HEATH SR-206 dual pen strip chart recorder. The overall sensitivity of the equipment is 0.02 % of full scale maximum on calibrated range (Fig.9).

Before experiments, all measurements were checked and calibrated. The gear box which provided variable rotating speed was calibrated by recording the number of
Figure 8- Pressure control system.
Figure 9- Data acquisition system.
rotation made by rock sample in real time. The strip chart recorder operation was continuously controlled with the variable external voltage supply of known intensity that in turn was checked against a calibrated multimeter tester of 0.04 % basic DC accuracy.

The accuracy of pressure reading by a pressure transducer was checked upon by using a special set-up which provided direct measurements of known weights acting on the known area (hence, direct measurement of pressure) and the corresponding signals from transducer in voltage. The calibration curve is given in Fig. 10. Measured values of pressure is converted to the normal force by using the known cross-sectional area of air cylinder bore. A typical recording of pressure during one experimental run is shown in Fig. 11. Straight line indicates that the pressure (and hence, the normal force) is constant during the experiment. The estimated maximum error in pressure measurements was 1 %.

The accuracy of tangential force readings by a load cell was checked upon by applying known dead weights on the load cell and recording its response in volts. The calibration curve is given in Fig. 12. Tangential force measurement was influenced significantly by vibration and shatter created during an experimental run. The resolution of tangential force recordings varied depending on the alignment of the load frame (and thus, alignment of the
Figure 10- Calibration curve for pressure measurement by using pressure transducer.
Figure 11- An example experimental record
Figure 12- Calibration curve for force measurement by using a load cell.
cutters) and smoothness of the rock surface. Therefore, special interpretation technique was used for tangential force readings. An average value of the lowest and the highest peak was chosen for representative tangential force reading. Fig. 11 is an example of good experimental run with small variation in the tangential force recordings. Fig. 13, on the other hand, shows an experimental run with considerable vibration and shatter as indicated by big variation of the recorded tangential force values. The maximum estimated error in tangential force recordings was 10%. Thus the overall error in calculation of friction coefficient was estimated 11%.

RESULTS

Original mud samples collected from the field provided actual field condition (aging, cutting removal, chemical treatment) and were therefore suitable for the experiment. Four different rock types namely, Berea sandstone, Nugget Sandstone, Mancos Shale, and Sierra White Granite were used during the experiments. The summary of experimental results is given in Table 3.

Preliminary tests have shown that plot friction forces vs. normal force would give a straight line passing through origin with slope equal to the sliding friction coefficient. Therefore, a statistical approach was used to analyse the experimental data. Results of linear regression analysis are given in Table A1 and in Figs. B1.
Figure 13- An example experimental record
### Table 3: Summary of Measured Values of Friction Coefficients

<table>
<thead>
<tr>
<th>ROCK TYPE</th>
<th>FLUID TYPE</th>
<th>FRICTION COEFFICIENT $\times 10^{-3}$</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Water</td>
<td>82</td>
</tr>
<tr>
<td>Berea sandstone</td>
<td>Water-Base Mud</td>
<td>81</td>
</tr>
<tr>
<td></td>
<td>Oil-Base Mud</td>
<td>75</td>
</tr>
<tr>
<td></td>
<td>Water</td>
<td>82</td>
</tr>
<tr>
<td>Nugget sandstone</td>
<td>Water-Base Mud</td>
<td>66</td>
</tr>
<tr>
<td></td>
<td>Oil-Base Mud</td>
<td>80</td>
</tr>
<tr>
<td></td>
<td>Air</td>
<td>110</td>
</tr>
<tr>
<td>Mancos shale</td>
<td>Water-Base Mud</td>
<td>84</td>
</tr>
<tr>
<td></td>
<td>Oil-Base Mud</td>
<td>88</td>
</tr>
<tr>
<td>Sierra White Granite</td>
<td>Water-Base Mud</td>
<td>68</td>
</tr>
<tr>
<td></td>
<td>Oil-Base Mud</td>
<td>73</td>
</tr>
<tr>
<td></td>
<td>Water base mud with lubricant</td>
<td>63</td>
</tr>
<tr>
<td></td>
<td>Air</td>
<td>120</td>
</tr>
</tbody>
</table>
to B12. Eventhough the origin (0-0) point was not input to the regression analysis, the regression straight line passed through (or close to)the origin in all plots. Considering data discrepancy within the experimental error, we conclude that the experiments confirmed the linear relationship between normal force and friction force passing through the origin i.e., constant value of friction coefficient and independent from normal force.

Berea Sandstone and Water

Nineteen preliminary experiments were performed using Berea Sandstone and water to develop a methodology that produced satisfactory results.

It was seen that the rock surface in this case, was extremely softened by water. Hence, it was not necessary to exceed the compressive strength of the rock for the cutter to effectively penetrate (or cut) the rock. Consequently, normal force imposed by the cutters were kept at lowest possible range (1-100 lbf) in order to prevent cutter from making ridges in the rock. Alignment of the cutters were also crucial for a smooth run.

Initially, aluminum oxide grinding wheels were used to resurface the rock. However, it was realized that they were polishing the rock surface rather than cutting it. Therefore, later experiments were performed by using heavy
duty steel brushes to clean the rock surface. Even though, the brushes were not effectively cutting the rock, they were efficient to remove the debris (if there is any due to the cutters' action) and preventing a build-up of mud film which would completely change the nature of the friction phenomena. In addition to continuous brushing, rock surface was further reconditioned by using PDC cutters after each run and prepared for the next run.

The average value of friction coefficient at constant sliding speed of 1.2 ft./sec. was found to be 0.082 for the Berea Sandstone water combination.

The second part of the preliminary test runs was aimed at finding the effect of sliding speed on the friction coefficient. Thirteen experiments were performed. In each case, the run began at lowest rotating speed (1.2 ft/sec.) and motor was speeded up gradually to the highest level (9 ft./sec.). Friction coefficient decreased considerably with in the tested range (1.2-9 ft/sec.), of sliding speed (Fig. 14).

Berea Sandstone and Water-Base Mud

Over twenty tests were performed in this case. It was not possible to have a smooth run in most of these experiments. Relatively high porosity (24%) and clay content (10%) of the Berea Sandstone enhanced invasion of
Figure 14- Effect of sliding speed on friction coefficient
rock pores by this highly dispersing mud. Due to the dispersion of rock matrix particles, the rock structure was disintegrated. Such change in rock structure would lead to severe reduction in the intergranular cohesive forces. If the sliding friction forces between a PDC cutter and rock surface exceeds the magnitude of internal friction forces (already reduced by dispersion), then the rock particles are removed from the surface as the cutter slides on. This phenomena was observed very often in experiments with Berea Sandstone and water-base mud. Cutters built-up a ridge as soon as they were pushed against surface of a rotating rock.

Only five experimental runs were succesful in unbiased determination of friction coefficient which varied from 0.078 to 0.93 in this case. By using the average of these five runs, friction coefficient was found to be 0.081.

Berea Sandstone and Oil-Base Mud

Similar behavior of Berea Sandstone was observed with oil-base mud as in the case of water-base mud. Due to the invasion of pores by oil-base mud, the intergranular structure of the rock changed completely which in turn reduced the shear strength of the surface layers. Consequently, rock particles in the vicinity of the surface lost their coherence and hence, they were easily
removed under even very low shear. Continuous removal of rock particles from the surface made it very difficult to measure any friction forces.

Four experiments were conducted at 1.2 ft/sec sliding speed. Average value of friction coefficient was 0.075.

Nugget Sandstone and Water-Base Mud

Experiments with Berea Sandstone made it clear that a more aggressive, stronger and a less porous rock sample was needed to have more accurate measurements of friction forces. Therefore, Nugget Sandstone was used for the next set of experiments.

Ten runs were performed in this case. All runs were smooth and sliding friction forces were recorded without having any major difficulty. Since effective cutting depth was very small, steel brushes and mud stream were able to clean the rock surface from any cuttings efficiently. All tests were conducted at 1.2 ft/sec sliding speed. The average value of friction coefficient was 0.066.

Nugget Sandstone and Oil-Base Mud

Ten experiments were performed at 1.2 ft/sec sliding speed. All runs were smooth and the friction forces were
accurately measured. Average value of friction coefficient was 0.08.

The effect of sliding speed on the friction force was also investigated in this case. The friction coefficient reduced from 0.085 at 1.2 ft/sec to 0.074 at 6 ft/sec (Fig. 14).

Mancos Shale and Water-Base Mud

Eight experiments were performed in this case. All the tests were run at 1.2 ft/sec sliding speed. The major problem with Mancos Shale was to resurface the rock after each run. Sticky nature of the shale was even more pronounced when it was exposed to the mud. Consequently, balling of both cutters and steel brushes were often encountered.

The wear of Mancos Shale also showed other peculiarities. The rock was chipped away in big pieces which made the rock surface very irregular and the recording of sliding friction forces difficult. Only eight experiments were conducted and the rock had to be abandoned afterwards. Average value of friction coefficient was 0.084.
Mancos Shale and Oil-Base Mud

Although it was in lesser extent, balling of cutters and brushes were also seen in this case. Friction force measurements were also rendered in this case due to big shale particles chipped away by the cutter which made the rock surface rough for cutter to slide on.

Ten experiments were carried on with Mancos Shale and oil-base mud. Average value of friction coefficient was 0.088.

Sierra White Granite and Water-Base Mud

Sierra White Granite is even more aggressive and stronger rock than Nugget Sandstone. All the tests were run smoothly and friction forces recorded without having any difficulty.

Cutters were not able to generate any considerable debris, therefore, cleaning by brushes and mud stream was very efficient. Ten experiments were carried out at 1.2 ft/sec sliding speed. The average value of friction coefficient was 0.068.

Additional five runs were performed by mixing water-base mud with a commercially available lubricant which is originally used for borehole friction reduction. The lubricant is composed of vegetable oil mixed with...
naturally occurring glycerides. Maximum recommended amount (3% by volume) was used in this case. Average value of friction coefficient was 0.063 which is 10% smaller than the case where only water-base mud was used.

Sierra White Granite and Oil-Base Mud

Ten experiments were performed with Sierra White Granite and oil-base mud. All runs were smooth and sliding friction forces were measured accurately.

Additional experiments with Sierra White Granite and oil-base mud were conducted to investigate the effect of sliding speed on the friction forces. Results (Fig. 7) showed that the friction coefficient did not change considerably with increasing sliding speed.

Dry Friction Experiments

Five experiments were conducted with Sierra-White Granite in air at 1.2 ft/sec sliding speed. All runs were smooth and friction forces were measured without having any major vibration and shatter. Average friction coefficient was 0.12.

Five experiments were performed with Nugget Sandsone in air at 1.2 ft/sec sliding speed. Slight wear of rock surface was observed. Average value of friction coefficient was 0.11.
DISCUSSION

The friction coefficient values measured on the laboratory tester showed slight variation with applied normal forces. When the sliding motion was smooth, as in the case of measurements with Sierra-White Granite, the range of variation of friction coefficient with increasing load was very small. Fine grained, stronger texture of granite cultivated a smooth run with minor vibration and no shatter and sign of wear at the interface, whereas, Berea Sandstone, having coarse grained texture with low shear strength, induced considerable vibration and surface wear. Consequently, sliding motion of the cutter on Berea Sandstone was not as smooth as on Sierra-White Granite. Hence, range of variation of friction coefficient with normal force was more pronounced in Berea Sandstone than that of in Sierra-White Granite. This complies with the present friction theory (Appendix C). In general, the friction coefficient was not dependent on the normal force and thus, Amonton's law was satisfied in this respect.

Figures B1 to B12 show the plots of measured values of normal forces versus friction forces giving straight lines with slope equal to the friction coefficient for each tested combination of rock and fluid type.

Effect of sliding speed on the friction coefficient was also investigated for different rock and fluid types.
In all cases but one with Sierra-White Granite, it was seen that the friction coefficient decreased with sliding speed. In Sierra-White Granite with oil-base mud, it was observed that friction coefficient remained constant as the sliding speed increased.

This also agrees with the existing friction theory which explains the effect of sliding speed on the friction process based on the stick-and-slip characteristic (Appendix C). Here again, when the motion is smooth the friction coefficient shows little change with the increasing speed. With intermittent motion, however, the behavior was different. The friction at the "stick" decreases relatively rapidly with the increasing speed. The mean friction during slip decreases more gradually. Consequently, as the speed increases, the friction at the stick approaches the mean friction during slip with corresponding decrease in the size of the fluctuations. Coarse grained, relatively rough surface of the sandstone would induce stick-and-slip motion. Fine grained, stronger texture of the granite, on the other hand, enhance continous motion of the cutter. Consequently, a decreasing friction coefficient was observed in the former where as the friction coefficient was constant in the latter with increasing sliding speed.

Experimental results indicated that friction coefficient did not change considerably within the range
of rock types and lubricants used. However, wet friction coefficient (when water, water-base mud and oil-base mud were used as a lubricant) was lower by about 50 to 100 % than dry friction coefficient (when air was used as a lubricant).

The insensitivity of friction coefficient to the type of lubricants namely, water, water-base mud, and oil-base mud can be explained through the analysis of lubrication mechanism effective during rock cutting by a PDC cutter. It appears that boundary lubrication is the effective lubrication mode during rock cutting with the PDC cutter in the presence of drilling fluid. In boundary lubrication, the rheological properties of the lubricant are of no importance (Appendix D). Therefore, the rheological differences of water, water-base mud, and oil-base mud did not have any influence on the measured friction forces.

The main purpose of a boundary lubricant (Appendix D), is to interpose, between the moving surfaces, a film that is able to reduce the amount of direct solid/solid interaction and that is itself easily sheared. This is best provided by an interfacial film consisting of long chain molecules providing the following properties:

- strong attraction between the molecular chain to resist penetration by surface asperities,
b- low shear strength,
c- high melting point so that it provides solid film protection up to a high temperature.

Last condition is of minor importance under laboratory conditions (although high melting point of solid agent added would be favorable under field drilling conditions, where higher temperatures are encountered). It seems that the first two conditions may be controlling factor in this case. Tested fluids did not show any major difference, in providing strong attraction between their molecules and in their shear strength. Hence, the lubrication quality of the tested fluids, as far as boundary lubrication is concerned, seemed to be at the same level. Consequently, the friction coefficient showed slight variance with the type of fluids used throughout these experiments. Further chemical analysis is needed to verify this observation.

Observations made in the single-cutter experiments also contributed to the understanding of rock/cutter frictional interaction mechanism. When the cutter was sliding on coarse-grained relatively porous rock, i.e., Berea Sandstone, it was seen that, surface layer of rock was disintegrated and formed a loose intermediate zone between the cutter surface and the solid rock. The disintegration occurred mainly due to the reaction between rock particles and drilling fluid which invaded the pores
close to surface. Presence of liquid softened the rock by reducing the intergranular bond strength. Rock particles with reduced cohesion provided weak resistance to the shearing action imposed by the cutter. Consequently, friction forces between cutter and rock became higher than the friction forces between destabilized rock layers and the undisturbed rock. Therefore, removal of rock particles from the surface was observed instead of pure sliding of cutter on the solid rock surface.

The above phenomena was observed to the lesser degree with increasing compressive strength of the rock, and decreasing porosity. For example, in Sierra-White Granite, damage to the surface was insignificant. The insensitivity of friction coefficient to the rock types (at least within range of tested samples) also implies that the measured friction forces were not representative for pure sliding action between rock and PDC cutter. Instead, the measured tangential force included two effects: the intergranular friction within the body of loose grains above the surface of the uncut rock and the friction force between the top of grains and the PDC cutter. The summed value of these two forces appeared to be almost the same for the tested rock types. However, the relative magnitudes of these effects are believed to be different; with no intergranular friction effect for hard rocks.
CONCLUSIONS AND RECOMMENDATIONS

A medium-scale laboratory instrument was constructed and successfully used to measure friction between the sliding surface of a rock and polycrystalline diamond compact cutter.

Measured friction coefficients were independent from normal forces, thus satisfied Amonton's law of friction.

The magnitude of the friction coefficient either decreased or remained constant with the increasing sliding velocity. The effect of velocity is less pronounced for stronger rock with fine grained texture.

No significant effect of lithology and lubricant on the friction coefficient was observed within the range of tested rock and fluid types. Dry friction coefficient, however, was about 50 to 100% higher than friction coefficient measured with water, water-base mud, and oil-base mud.

The interaction between borehole fluid and rock and its speed seem to control mechanism of friction though friction coefficient value is little affected. The size and extent of the relative motion in the disintegrated layer of rock under the PDC cutter is the main factor which controls heat generation.
Further improvement of the instrument is necessary by providing more efficient cutting (resurfacing) mechanism. This could be pursued by employing PDC cutters on the side of the rock opposite to place where sliding cutters located.

Further studies should be concentrated on the effect of reservoir conditions namely pressure, temperature, and the fluid content of the rock on the rock/cutter friction coefficient.

ACKNOWLEDGEMENTS

Financial support from Ministry of Education of Turkey and from Department of Petroleum Engineering of Louisiana State University is greatly appreciated.

Thanks to Marty White, Bob Walen, Keith Reckling, and Del Leggatt for their help in material supply.

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REFERENCES


presented at the 61st Annual Technical Conference and Exhibition of the SPE, New Orleans, LA, (Oct. 5-8, 1986).


### Table A1 - Table of results for all friction experiments

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Table A1- Table of results for all friction experiments (continued)

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<tr>
<td></td>
<td></td>
<td>84</td>
<td>9.4</td>
<td>111</td>
</tr>
</tbody>
</table>
APPENDIX B

Sliding speed : 1.2 ft / sec

Figure B1- Data from experiments with Berea Sandstone and water.

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Figure B2- Data from experiments with Berea Sandstone and water-base mud.
Figure B3- Data from experiments with Berea Sandstone and oil-base mud.
Figure B4- Data from experiments with Nugget Sandstone and water-base mud.
Sliding speed: 1.2 ft/sec

Figure B5- Data from experiments with Nugget Sandstone and oil-base mud
Figure B6- Data from experiments with Mancos Shale and water-base mud.

Sliding speed : 1.2 ft / sec
Figure B7- Data from experiments with Mancos Shale and oil-base mud.
Figure B8- Data from experiments with Sierra White Granite and water-base mud.
Figure B9- Data from experiments with Sierra White Granite, water-base mud and lubricant.
Sliding speed: 1.2 ft/sec

Figure B10- Data from experiments with Sierra White Granite and oil-base mud.
Sliding speed: 1.2 ft/sec

Figure B11- Data from experiments with Sierra White Granite and air.
Figure B12- Data from experiments with Nugget Sandstone and air
APPENDIX C

MECHANISM OF SLIDING FRICTION

A feature common to all processes of motion is the occurrence of effects of "resistance to motion", i.e., the occurrence of friction of some kind or another. The effects of friction are due to physical interactions between bodies or objects moving relatively to each other. Friction is generally classified based on the type of motion involved i.e., sliding friction, rolling friction...etc. as well as based on the type of material involved in the process such as, dry friction, air friction, friction under lubrication ... etc. Since the action of PDC cutter on the rock is simulated as the combination of cutting and sliding, this review will be limited on the sliding friction related to this process.

If, in a dry static situation [12], the load applied tangential to the contact surface exceeds a certain value, gross sliding between the components in contact occurs (Fig. C1). In this situation, the following macroscopic rules have in general been observed experimentally:

a- When tangential motion between two contacting bodies occurs, the friction force always acts in a direction opposite to that of the relative velocity of the surfaces.
Figure C1- A solid body in sliding motion, after [12].
b- The friction force is proportional to the normal force.

\[ F_F = f F_N \]

Through this relationship, it is possible to define a coefficient of friction,

\[ f = \frac{F_F}{F_N} \]

Where:

- \( F_F \): Friction force, lbf.
- \( F_N \): Normal force, lbf.
- \( f \): Friction coefficient

c- The friction force is independent of apparent geometric area of contact.

These rules, known as the "Amonton's-Coulomb Laws" of dry sliding friction, has been used so far as guiding rules in engineering applications.

When two solid surfaces are in contact over a nominal (apparent) geometric area, they are actually touching only in discrete spots (asperities) called micro-contacts. The sum of the micro-contacts forms the real area of contact which carries the normal load.
In the different stages (Fig. C2) of the formation and separation of micro-contact, the following main processes are involved:

a- elastic asperity deformation,
b- plastic asperity deformation,
c- ploughing,
d- shearing of adhesive junctions.

Each of the partial processes involves a tangential force necessary to maintain the relative motion as well as the partial process of energy dissipation.

According to the microscopic model of friction, the different contributions to friction may be classified broadly into two groups:

a-) deformation processes and
b-) adhesion processes.

In general, the processes belonging to these two groups are not independent of each other. Under certain special conditions, however, one process may dominate so that the other contributions are comparatively small and may be neglected to a first approximation.

If a very hard rough surface slides over a soft one, the frictional resistance is then mainly caused by the asperities of the harder surface ploughing through the
Figure C2- Schematic representation of a unit event in the friction process, after [12].
softer. In the case of two relatively soft clean metals, however, adhesion bonding in the micro-contacts occurs depending on the degree of interpenetration of asperities and the surface composition.

One important feature of sliding friction process is the stick-slip motion [13]. If one of the sliding surfaces has a certain degree of elastic freedom, the motion may not be continuous but may intermittent and proceed by a process of "stick-slip". The stick is due to the higher static friction between the surfaces and the slip is due to the lower kinetic friction during the slip itself.

Even when the moving parts are rigid, the surface irregularities may be capable of microscopic elastic deformations of the order of 0.00001 cm., as in the case of measured minute displacement using quartz crystals. In such cases, the elasticity of the surface irregularities themselves, may be sufficient to set up vibration or intermittent motion in the moving parts. Even in the presence of lubricant films, intermittent motion of a similar nature may occur.

The microscopic model of friction (Fig. C2) also indicates that depending on the operating conditions and the atmospheric environment, i.e., the "third body", interacting with the two sliding partners, changes in the surface topography, as well as in the surface composition
and surface strength properties will take place. These changes may influence the friction coefficient markedly.

Although deformation properties are important, the factor that has the largest overall influence on friction is the cleanliness of the surfaces. A small trace of oxygen or some other contaminant can radically reduce the adhesion and friction. As a consequence of this basic observation, lubrication theory and practice was introduced in all tribological processes i.e., contact, friction, and wear processes, related to direct physical interactions between relatively moving surfaces.
APPENDIX D

MECHANISM OF BOUNDARY LUBRICATION

The purpose of lubrication is to separate the surfaces moving relative to each other with a film of material which can be sheared with low resistance without causing any damage to the surfaces.

Depending on the thicknesses of the lubricant film (which may range from some tenth of a millimeter down to to the nm range), the interfacial height distribution of the lubricant film and the degree of geometric conformity, three main lubrication modes can be distinguished [14].

a- Hydrodynamic lubrication
b- Mixed lubrication
c- Boundary lubrication

In hydrodynamic lubrication, the rigid surfaces are separated by a continuous lubricant film, whose thickness is much larger than the combined surface measure of the surfaces. The friction resistance is due to the internal friction of the lubricant. In this regime, the tribological behavior of the system is determined by the rheology of the lubricant and can be calculated or estimated by the methods of fluid mechanics. Since there is no direct physical contact interactions between the surfaces occur, wear processes can not take place in
hydrodynamic mode of lubrication.

If under condition of hydrodynamic lubrication, the lubricant viscosity or the velocity decreases or the load increases, the lubricant film gets "thinner" and the separation of the surfaces decreases. If then the first asperity contact (physical interaction between surfaces) interactions occur, then mixed lubrication is reached. In this regime, the load is carried partly by the fluid film and partly by the contacting surface asperities. Consequently, the friction resistance is due partly shearing of the lubricant film and partly to the asperity interactions.

The boundary lubrication is reached when the amount of asperity interactions within the contact area increases further, and the film thickness decreases down to some monolayers or below, the boundary lubrication is reached. This lubrication mode is characterized by the following features:

- the solid surfaces are so close together that appreciable contact takes place between asperities,

- hydrodynamic effects and influences of the bulk rheological properties of the lubricants are of little or no importance,

- the tribological behavior is determined by
surface interactions between thin layers of boundary lubricants and the solid surfaces.

Under these conditions, the tribological behavior of solid surfaces can be summarized as:

a- the contact mechanics and the elastic and plastic asperity deformation processes,

b- the contact physics and chemistry and the action of surface forces,

c- the friction processes, especially the shearing of adhesive junctions and deformation of asperities,

d- the wear processes which are given in the general case by the superposition of effects of surface fatigue, abrasion, adhesion, and tribochemical reactions.

All these solid/solid interactions are modified through the action of the boundary lubricant, so that the tribological behavior of a boundary lubricated system is determined by the processes at the solid/lubricant/solid interface influenced by the environmental atmosphere.

Concerning the lubricating action of a boundary lubricant, its main purpose is to interpose between the moving surfaces a film that is able to reduce the direct solid/solid interactions and that is itself easily sheared. This is provided best by an interfacial film.
consisting of long chain molecules possessing the following properties:

a- strong attraction between the chains to resist penetration by surface asperities (thus mitigating wear processes),

b- low shear strength to give a low friction,

c- high melting point so that it provides solid film protection up to a high temperature.

Since the mechanisms of boundary lubrication are determined mainly by the physics and chemistry of the solid/lubricant/solid interface, a discussion of the processes of boundary film formation and its tribological behavior would be helpful to select proper lubricant for the particular problem.

The lubricant-solid interactions which lead to the generation of protective boundary film can be classified into three mechanisms:

a- physical adsorption,

b- chemical adsorption,

c- chemical reaction.

Physical adsorption occurs when the molecules of the lubricant are held to the surface by Van der Waals surface forces. Polar molecules, particularly long chain
hydrocarbons, condense on the surface to form a solid film. Many molecules pack in as closely as possible and strengthen the film with lateral cohesive forces. This solid film, adhering to the surface and with the molecular cohering to each other, then has the ability to resist penetration of asperities and thus inhibit metal-to-metal contact.

Chemisorption occurs when the molecules of the lubricant are held to the surface by chemical bonds, i.e., short range surface forces. Chemisorbed films lubricate effectively up to their melting point, whether the film is formed "in-situ" on a reactive surface or spread on an unreactive surface. Boundary chemisorbed films provide lubrication at moderate loads, temperatures and sliding velocities and fail under severe operating conditions.

Chemical reaction between the solid surfaces and the lubricant molecules occurs where there is an exchange of valence electrons and a new-chemical compound is formed. The boundary films are unlimited in thickness (governed by diffusion process through crystalline lattices) and characterized by high activation and bonding energies and irreversibility. Most of the chemically reactive reactive boundary lubricants contain sulphur, chlorine and phosphorus atoms in molecule. Boundary lubricants dependent upon chemical reaction are suitable for high
load, high temperatures, and high sliding speeds and limited to reactive metals.

Concerning the tribological behavior of a boundary-lubrication system, since the boundary films may be considered to be solid and to behave "as a rigid continuation of the solid body", the contact, friction, and wear processes are valid in principle.

In closure, the effect of load and sliding speed on the friction process in the presence of lubricant will be reviewed briefly. Bowden and Tabor [15] conducted experiments with lubricants such as paraffins, alcohols, and fatty acids to investigate the effect of load and sliding speed on the frictional behavior.

They used loads ranging from 500 to 6000 G. Exactly as for unlubricated surfaces, it was found that the main effect of increasing the load is to increase the magnitude of the frictional force. If the motion is smooth, as for fatty acids, the frictional force is directly proportional to the load, so that Amonton's law is accurately obeyed. If the motion is intermittent, the maximum friction, the mean friction, and the size of the fluctuations increased proportionately with the load.

When the motion was smooth, the friction showed little change with increasing speed from 0.001 cm/sec to
2 cm/sec. With intermittent motion, however, the behavior was different. The friction at the "stick", decreases relatively rapidly with increasing speed. The mean friction during the slip decreases more gradually. Consequently, as the speed increases the friction at the "stick" approaches the mean friction during slip with a corresponding decrease in the size of the fluctuations.

If the friction-speed characteristic is flat sliding will always be smooth. If it shows a marked falling characteristic, the motion may be intermittent, if the elastic constants of the moving parts are suitable.
CHAPTER V.

DYNAMIC DRILLING STRATEGY FOR MINIMUM COST

ABSTRACT

The concept of maximum bit performance (MBP) curve was introduced. The curves represented the maximum values of average drilling rates for various pre-assumed footage values. In contrast to elaborate drilling models, the MBP curves are a single, comprehensive correlation representing drilling bit behavior in a formation. For calculating purposes, the curves were normalized and thus, became insensitive to drillability change with depth as well formation abrasiveness. The curves were plotted and analyzed for both rock bits and PDC bits. The simple method for using MBP curves for drilling optimization was presented.

A new method for preparing a multi-bit drilling program, the dynamic drilling strategy, was developed. The dynamic drilling strategy provided the best combination of PDC bit runs to achieve the minimum drilling cost for a long borehole interval. The method was numerically compared to the conventional drilling optimization and to the field practices. Considerable cost-saving potential of 25% and 60%, respectively, were estimated.
INTRODUCTION

The selection and operation of drilling bits have been traditionally considered an art more than science. In spite of drilling models and sophisticated drilling data banks, the trial-and-error method still dominates field practices because of the seeming absence of method for converting massive drilling data into a single, comprehensive technical representation of a drilling bit. The maximum bit performance (MBP) curve introduced in this research is intended to provide a missing link between data acquisition and its utilization. Specifically, in PDC bit technology there is a growing amount of valuable field information regarding these bit applications in various drilling areas [1], [2], [3], [4], [5], [6]. The MBP curve can be useful in quantifying these experiences.

The concept of drilling optimization has been traditionally narrowed to the single bit run, [7], [8]. [9], without consideration to the whole drilling process. Such an approach seems satisfactory when drilling information is inadequate, the rig time is not expensive, and the average bit runs are small. In contrast, the efficient offshore drilling operations with expensive and long-lasting PDC bits and modern data collecting system require minimal non-rotating time, hence there is a need for careful planning of the number of bit runs. The dynamic drilling strategy formulated in this research indicates new progress in the methodology of drilling optimization methods. It also lays out a theoretical basis for development of the software for the MWD feed-back system and for the bit-planning methods.
The dynamic drilling strategy provides a two-level optimization, the bit level and the well level, of drilling process at minimum cost. At the bit level, the optimal-control algorithms for bit operational variables (bit weight, rotary speed, and others) are formulated and compound in the form of one concise function - maximum bit performance (MBP) curve. At the well level, the multiple-bit program is designed to reach a planned depth with minimum drilling cost. The program considers full utilization of bit performance (MBP curves are program input) and maximum reduction of non-productive tripping time.

The resulting drilling program provides distribution of bit runs along the well paths, depths of planned tripping operations, bit-control algorithms for all bits, and optimum number of bits per well. In principle, this methodology secures that all drilling bits will be fully utilized, and their number will be the minimum possible to drill the hole at lowest cost.

**MAXIMUM BIT PERFORMANCE (MBP) CURVES**

The new concept of the maximum performance curve for a drilling bit, though derived from optimization theory, has a very practical appeal. In an engineering sense, the MBP curve represents a relationship between any possible footage made by a single drill bit and its maximum drilling rate. Thus the maximum bit performance curve implies the minimum bit rotating time, given footage or the maximum possible footage, given time. It can be proven that the maximum bit performance curve provides a single, fundamental relationship necessary for drilling
optimization of any kind. The MBP curve represents the most efficient drilling bit control procedure within a particular rock.

In a strict mathematical sense, each point on the MBP curve represents a unique solution to the following optimization problem:

To find a minimum bottomhole time:

\[ t_b(W,N,w,A_f) = \text{MIN;} \]  
(1)

for given footage:

\[ F = \int_0^{t_b} R(W,N,w,K,...) \, dt = \text{Const.} \]  
(2)

within the following constraints:

\[ W_0 \leq W \leq W_{\text{max}} \]  
(3)
\[ N_{\text{min}} \leq N \leq N_{\text{max}} \]

The constraints (3) represent either torque limitations (small rigs), temperature limitations (diamond bits, PDC bits), or simply the mechanical strength of a drilling bit and rotary transmission limitations. The formulas (1), (2), (3) imply all available knowledge about the drilling bit (mathematical model) resulting from the field data, laboratory experiments, and drilling theory. All this knowledge, however, is synthesized in the form of a single MBP curve. The curve is easy to use by drilling field personnel to determine the best drilling strategy.

The optimization problem depicted above was solved in this work using the conventional rockbit model by Galle and Woods [7], the
Bourgoyne and Young model [8], and the PDC bit model [10, 11]. The optimization procedure and computer program algorithm are described in the previous work [10]. Similar curves can be plotted for all other existing drilling models. The examples of the maximum bit performance curves are shown in Fig. 1. The MBP curve for the rockbit shows decreasing maximum drilling rate for increasing footage. Small bit runs can be made most effectively without using the entire bit. Longer footages, on the other hand, can be made by reducing weight on bit and rotary speed to the minimum values at which the maximum footage for the rockbit can be achieved.

The response of a PDC bit is entirely different; as shown in Fig. 1, the maximum drilling rate increases with increasing footage. This means that a somewhat worn PDC bit drills faster than a new one. This surprising behavior is merely a result of the wearflat temperature restriction.

The inherent characteristics of PDC bits include high-frictional heating concentrated at the cutters' wearflat areas. Recent research [12] shows that the rate of wear increases by order of magnitude when wearflat temperatures exceed a critical value of 662 °F. In other research [13] an analytical expression was presented describing the wearflat temperature as a function of weight on bit, rotary speed, and wearflat area. The expression indicates that wearflat temperature is directly proportional to the bit weight-wearflat area ratio, and it is also positively affected by rotary speed. The expression was rearranged and used in these calculations for the bit weight-rotary speed limitation.
FIGURE 1. MAXIMUM BIT PERFORMANCE CURVES FOR THE CONVENTIONAL ROCK BIT AND THE PDC BIT

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In the MBP curve calculations, when short footage is assumed, bit weight and rotary speed are suppressed by the temperature restriction over the entire bit run, hence drilling rate is low and PDC bit is not finally worn out ($w < w_{\text{max}}$). Any attempt to ignore the temperature restriction and to apply high bit weight would result in increased drilling rate but at the expense of an excessive wear.

The practical conclusion is that PDC bits should not be used to make short runs because: (1) if operated properly—they give low drilling rates; and, (2) if operated for higher drilling rates—they are wasted.

Fig. 2 shows an example of the optimal PDC bit control for the assumed footage 1700 ft. A steady increase in the maximum weight is noticed corresponding to the increase of the bit wear and temperature restriction. The beginning of the maximum weight value of 40 klbf indicates a moment when the PDC cutter becomes insensitive to the temperature limitation. Optimum rotary speed, on the other hand, is constant over the bit life and equal to its maximum value of 200 RPM. Such control of the rotary speed agrees qualitatively very well with conclusions drawn from the laboratory and field experiments [14] that PDC bits should be operated at the highest possible rotary speeds.

Furthermore, as shown in Fig. 1, the MBP curve for PDC bits indicates a wide plateau with highest values of drilling rate. This corresponds to the range of footage when the temperature restriction becomes invalid before the bit run is completed. Drilling is continued with
constant values of the maximum weight and rotary speed, but the bit is pulled out not finished.

The right end of the plateau indicates such a bit run during which bit weight and rotary speed are reached and held at their maximum values and the bit was fully worn out \((w = w_{\text{max}})\). Longer footage may be achieved by reducing rotary speed (with weight maintained at its maximum) until no more footage can be made by the PDC bit.

**OPTIMIZED BIT PROGRAM**

The objective of this section is: (1) to define optimum bit program; (2) to show a conventional bit program based on bit-after-bit optimization and to demonstrate application of the MBP curves in such optimization; and (3) to present the dynamic drilling strategy for multi-bit program.

When preparing a bit program for a well, a drilling engineer attempts to (1) select the type of drilling bits; (2) specify their operational parameters \((W, N)\); and (3) estimate the number of drilling bits required. Traditionally, all the above are approached using some rough estimates of drilling bits performance, the bit manufacturer's recommendation, and personal experience. However, for an optimized drilling program, an attempt is made to choose the combination of consecutive bit runs that gives the minimum drilling cost defined as

\[
C = \sum_{n=1}^{N_b} \left( C_b^n + C_R t_n + C_b t_n \right) = \text{MIN}
\]

\(N_b\)
with the limitation
\[
D_i - D_t = \sum_{n=1}^{N_b} F_n \tag{5}
\]
Therefore, the optimized bit program requires: (1) the mathematical model of a drilling bit behavior to calculate \(t_{b_n}\) and \(F_n\); (2) the optimization method to find a minimum value of the sum (4).

**Bit-after-Bit Optimization**

The conventional way of solving this problem assumes that the minimum of the summation (4) will be achieved for the minimum values of the expression

\[
C_F = \frac{C_{b_n} + C_R(t_{t_n} + t_{b_n})}{F_n} = \text{MIN} \tag{6}
\]
a well known cost-per-foot formula. This means that by minimizing cost-per-foot for each consecutive bit run, the total drilling cost will be kept at a minimum. However, this is theoretically not true.

The maximum bit performance curves provide a simple method for optimizing a single bit run. The equation (6) can be written as,

\[
C_F = \frac{C_{b_n}}{F_n} + t_{t_n} + \frac{1}{R} \tag{7}
\]
and its minimum value can be reached only when
Thus, the second component of the summation is a reciprocal of the maximum bit performance curve. The graphic interpretation of the method is shown in Figs. 3 and 4. There is a distinctive minimum of the cost-per-foot for a conventional rockbit in Fig. 3.

Fig. 3 also demonstrates the assumption that the single-bit optimization for all bit runs gives minimum drilling cost per well. It is believed that the cost savings due to reduction in the number of drilling bits are always smaller than cost increase caused by extended bit runs. The steep U-shaped minimum of the cost-per-foot plot in Fig. 3 seems to support such belief.

In the case of PDC bits, however, the cost-per-foot plot in Fig. 4 shows little sensitivity to the footage over a wide range of values—the U-shape is fairly flat. Therefore, there is a strong possibility that the drilling cost obtained from the bit-after-bit optimization might be much greater than the actual minimum of the expression (4).

**Dynamic Drilling Strategy**

The dynamic drilling strategy is a design method for a multi-bit drilling program in which the number of drilling bits, bit footage, and bits operational variables are simultaneously optimized to obtain a minimum drilling cost of a long borehole section.
FIGURE 3. CONVENTIONAL DRILLING OPTIMIZATION (ROCK BIT)
USING MBP CURVE

\[ \frac{C_R}{C_B} = \frac{C_F}{C_R} \]

\[ C_{F_{\text{min}}} = \$80/\text{ft} \]

\[ t_f = 12 \text{ hrs} \]
\[ C_r = \$\,2000/\text{hr} \]

\[ C_b = \$\,12,000 \]

\[ t_t = 12 \text{ hrs} \]

\[ C_r = \frac{C_F}{C_R} \]

\[ C_{F_{\text{min}}} = \$\,50/\text{ft} \]

**FIGURE 4. CONVENTIONAL DRILLING OPTIMIZATION (PDC BIT) USING THE MBP CURVE**

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Most typically such section will represent a hole drilled below a casing seat down to the next casing's setting depth (surface, intermediate, and production holes). It is probable that the summation of the minimum-cost bits' program for these sections will give a minimum value for the well drilling cost.

Mathematically, the optimum drilling strategy represents a solution to the optimization problem depicted by equations (4), (5). The theory of operations research provides a numerical technique, dynamic programming for solving problems of this nature [15], [16]. In the method used in this research, drilling operations were considered a multi-stage decision process with an unknown number of stages (bit runs) as shown in Fig. 5. At each PDC bit run, the decision was made concerning the control of the bit \((W, N, t_b, F)\) in order to minimize the drilling cost of the whole borehole section given by equ. (4). By using the maximum bit performance curves, this multi-variable problem can be significantly reduced to the simple problem of the optimal distribution of the consecutive bit runs \((F_1, F_2, ..., F_n)\). The method used in the computer program is described in Appendix, and its algorithm was presented in the previous work [10]. The recurrent formula was derived, based on the optimality principle of the dynamic programming [15]. The formula yielded the optimal distribution of the bits runs:

\[
\text{MIN } [C(D_n)] = C_R \text{ MIN } \left\{ [C_b/C_R + t(D_n) + \frac{F}{[R]_{\text{max}}} \right\} 
\]

\[+ \text{ MIN } \left[ \sum_{j=1}^{n-1} C(D_j) \right] \]  \hspace{1cm} (9)

where
Figure 5- Multi-run drilling strategy.
\[ F = F_n \exp[a\xi(D_n - D_i)] \] (10)

and

\[ n = 1, 2, ..., \frac{D_n - D_i}{\Delta D} \]

Such distribution indicated the optimum number of bits and their footage. Through the use of the MBP curves the footage were matched with the corresponding bit control algorithms, thus providing a complete and optimized bit drilling program.

**COMPARISON STUDY**

The objective of this part of the research was to use computer-simulated drilling conditions and to evaluate the technical effect and the economic potential of the dynamic drilling strategy as compared to the bit-after-bit optimization and to the average field practices.

The algorithm simulating average field practices was based on the typical drillers' tendency to maintain a constant and "acceptable" rate of drilling. This objective is achieved by systematic increase of the bit weight (to counteract the effect of bit dull) while keeping the rotary speed constant. (On most drilling rigs the slow increase of the bit weight is more feasible than change of the speed of rotation.) Of course, as the bit wears out, the drilling rate cannot be kept constant. Thus, at the late stage of the bit run, the slow reduction of the drilling rate is observed while applying constant maximum values of weight on bit and rotary speed.
The simulated drilling conditions included a 4100 ft. section of a medium hard, medium abrasive formation between 12,000 and 16,100 ft as well as the same formation located between 4000 and 8100 ft. The effect of a drilling depth was implied in the drillability constant as

$$K(D) = K(D_i) \exp \left[a_2(D - D_i)\right]$$  \hspace{1cm} (27)

The tripping time was calculated as

$$t_t = 0.001 D$$ \hspace{1cm} (28)

Data pertaining to the cost calculations were chosen in view of matching the most typical drilling conditions in the offshore area of the Gulf of Mexico [10]. A land drilling operation, at low operational cost ($C_R = 200 \$/hr) was also considered. The data used for calculations are shown in Table 1.

Shown in Table 2 is the example of the detailed calculations for the land drilling of a deep well. The comparison of the bit run distributions exemplifies the way in which the dynamic drilling strategy provides cost reduction. Though the number of bits for the two optimized bit programs is the same (four bits), their footage are more evenly distributed for the dynamic strategy-- resulting in 18% cost reduction. The significant (over 50%) savings, with respect to the field practice, is caused by its erroneously selected average drilling rate of 30 ft/hr as a bit control criterion. (Such operation leads to the excessive bit-wear rate/drilling rate ratio and results in increased number of bits.)

Table 3 shows a summary of example calculations using two optimized and one non-optimized bit drilling programs for land, and off-shore drilling of deep or shallow wells. The results consistently indicate
TABLE 1

DATA USED FOR COMPARISON OF PDC DRILLING STRATEGIES

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<thead>
<tr>
<th>Depth In, Ft.</th>
<th>12,000 (4,000*)</th>
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<tr>
<td>Depth Out, Ft.</td>
<td>16,100 (8,100*)</td>
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<td>Average Drilling Rate, Ft./Hr.</td>
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<td>Maximum Weight On Bit, Klb.</td>
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<tr>
<td>Average Rotary Speed, 1/min.</td>
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<td>Compaction Coefficient, (a_2)</td>
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<tr>
<td>Rig Cost, $</td>
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</tr>
<tr>
<td>Bit Cost, $</td>
<td>1,000 (12,000***)</td>
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* Shallow Drilling
** Offshore Drilling
*** PDC Bit
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<th>Bit-By-Strategy</th>
<th>Dynamic Drilling Strategy</th>
<th>Cost Per Foot ($/FT)</th>
<th>ROP (FT/HR)</th>
<th>Cumulative Drilling Time (Hr)</th>
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Field Practice:

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<th>Dynamic Drilling Strategy</th>
<th>Cost Per Foot ($/FT)</th>
<th>ROP (FT/HR)</th>
<th>Cumulative Drilling Time (Hr)</th>
<th>Cumulative Cost ($)</th>
</tr>
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<td>1</td>
<td>13568</td>
<td>1300</td>
<td>28.15</td>
<td>76.5</td>
<td>22.15</td>
<td>18830</td>
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<td>16100</td>
<td>2</td>
<td>14851</td>
<td>15000</td>
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<td>34.2</td>
<td>51.73</td>
<td>39427</td>
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<tr>
<td>5000</td>
<td>3</td>
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<td>22.2</td>
<td>24.3</td>
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<table>
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<tr>
<th>Depth Out (FT)</th>
<th>Bit No.</th>
<th>Bit-By-Strategy</th>
<th>Dynamic Drilling Strategy</th>
<th>Cost Per Foot ($/FT)</th>
<th>ROP (FT/HR)</th>
<th>Cumulative Drilling Time (Hr)</th>
<th>Cumulative Cost ($)</th>
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<td>85.06</td>
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<td>137.97</td>
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TABLE 3
COMPARISON OF DRILLING STRATEGIES, SUMMARY

<table>
<thead>
<tr>
<th>Drilling Conditions</th>
<th>Drilling Strategy</th>
<th>Number of Bits</th>
<th>Total Drilling Time, hrs.</th>
<th>Total Cost, $</th>
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<tbody>
<tr>
<td>Offshore Operations;</td>
<td>DDS</td>
<td>4</td>
<td>71</td>
<td>235,755</td>
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<tr>
<td>Shallow Drilling</td>
<td>BBS</td>
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<td>72</td>
<td>239,744</td>
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<td></td>
<td>FP</td>
<td>5</td>
<td>320</td>
<td>749,300</td>
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<tr>
<td>Offshore Operations;</td>
<td>DDS</td>
<td>4</td>
<td>71</td>
<td>299,755</td>
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<tr>
<td>Deep Drilling</td>
<td>BBS</td>
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<td></td>
<td>FP</td>
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<td>813,300</td>
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<td>Land Operations; Deep</td>
<td>DDS</td>
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<td>92</td>
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<tr>
<td>Drilling</td>
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<td>Land Operations; Shallow</td>
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<tr>
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<td></td>
<td>FP</td>
<td>5</td>
<td>320</td>
<td>128,930</td>
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DDS = Dynamic Drilling Strategy
BBS = Bit-By-Bit Strategy
FP = Field Practice
the lowest costs are obtained with the dynamic drilling strategy. They also show a strong correlation between cost reduction and number of PDC bits used; when this number was the same—savings were minimal.

Also, an 18% cost reduction with respect to the bit-after-bit optimization was not the maximum savings possible. The analysis of the MBP curves for PDC bits showed more theoretical potential in cost reduction than that calculated above. Also, in the previous simulation study [10], the calculated cost reduction was 30%. (In the same study [10] the comparison with the field practice based on constant bit weight and rotary speed for the whole bit run yielded up to 300% cost reduction.)

LIMITATIONS STUDY

The objective of this part of the research was to evaluate the maximum possible savings for the dynamic drilling strategy in comparison to the bit-after-bit optimization method. The study was based on computer-simulated drilling conditions.

Observations made in the previous study showed a strong effect of the number of bits used for drilling a hole upon potential cost savings of the dynamic drilling strategy. Since this number of bits depends upon the hole length, a concept of dimensionless footage was introduced and used as a correlating variable.
The dimensionless footage was defined as a ratio of the maximum footage for a bit to the length of a hole. If its value is one, the hole can be drilled theoretically using one bit; the value of 0.2 shows that at least five bits have to be used to drill the hole. The advantage of using the dimensionless footage is that it combines the length of the hole and number of bits in one magnitude providing a mean for comparing PDC bits and rock bits.

The results of this study are summarized in Fig. 6. The maximum possible cost saving for PDC bits is about 40%, and it can be achieved when the hole length is within reach of one bit. (For PDC bits it can be up to 4000 ft or even more, while for rock bits it is usually less than 1000 ft) As the length of the hole increases, the potential saving decreases. For the four-bit region, (0.25) maximum savings are around 22%, which favorably complies with the results obtained in the study above the four bit region. Conditions are likely to occur for PDC bit drilling when the whole borehole section can be drilled with four bits or less. For rock bits, however, the hole length is most likely within the range of over ten bits; the potential savings will be very small.

Fig. 6 also shows that the cost effectiveness of the dynamic drilling strategy is very sensitive to small changes of the hole length. The cost savings fluctuates from zero (no-savings) to maximum value for each range of the dimensionless footage. Therefore the conclusion is such that in order to avoid uncertainty of missing the potential savings, the dynamic strategy should be used rather than the bit-after-bit optimization.
Figure 6. Sensitivity analysis for bit program selection.
DISCUSSION

The research presented here indicates that the dynamic drilling strategy is particularly suitable for optimized PDC bit programs. It seems that the true potential of this method can be used while drilling long sections of intermediate holes in medium-hard formations where the PDC bits have already outdrilled all other types of drill bits.

Such conditions exist in most drilling areas where the PDC bits have already been tested. For example: (1) the Tuscaloosa Trend formations in South Louisiana comprise a 3500-ft sequence of shale and chalk from 14,500 ft to 18,000 ft [3]; (2) the Austin Chalk Trend in East Central Texas includes about a 3000 ft sequence of shale and limestone encountered at various depths from 6000 to 11000 ft [6]; (3) the shallow section of the Frio formations in South Texas is composed of 8000 ft of soft sand and shale drilled without protective casing from 1500 ft to 9500 ft [5]; (4) the geology of the offshore formations of the Miocene Trend in the Gulf of Mexico requires drilling 9000 ft of the soft shale from about 4000 ft to about 13000 ft. Since the PDC bits application resulted in long footage, only a few bits were required to drill the whole interval with at least one PDC bit only partially worn out.

Pulling out a non-used tool is a characteristic problem of PDC bit applications. The analysis of 88 PDC bit runs [5] showed that 32 bits (36%) were re-run at least once. These bits were eliminated from the research because their performance was difficult for quantitative eval-
uation. In addition, the re-running of PDC bits create problems in evaluating their economics. One controversial concept was to consider the PDC bit price only for the first run and to use a zero price for the next runs. Of course, such an approach does not really reflect the actual PDC bit performance.

The results of this research suggest that re-running PDC bits can be significantly reduced by using the MBP curves in conjunction with the dynamic drilling strategy.

CONCLUSIONS

In this research, a new dynamic technique for oil well drilling program was introduced. The research revealed several facts summarized below.

1. The new drilling optimization method, the dynamic drilling strategy, proved cost effective. Sample calculations show up to 25% savings in comparison with conventional drilling optimization and up to 60% savings with respect to the non-optimized, typical drilling practices (i.e., constant drilling rate-practice). The dynamic drilling strategy method can be easily simplified and adopted to the field application.

2. The concept of the maximum bit performance (MBP) curve provides a simple practical way of using drilling model for optimization of any kind. Not only can the MBP
curves be theoretically predicted but they can also be statistically generated from drilling data acquisition systems. In this way, the MBP curves may become a means of drilling technology transfer from one oilwell to another or from the PDC bit manufacturers to the oilwell operators.

3. The optimum dynamic drilling strategy is especially well suited to match a steady progress in drilling bits technology, i.e. stronger, long lasting, and more expensive PDC bits. Drilling economics with these new bits, especially at the offshore locations with very high rig costs, calls for minimum bit runs. The results of this research show that the smaller the number of bit runs, the greater the cost savings for this new method. It is believed that the implementation of the dynamic drilling strategy with careful planning of the bit run distributions, will result in the cheapest method of drilling an expensive well.

4. The new bit footage-controlled optimization method contributes to the 30 year drilling optimization theory. Moreover, it generalizes all other methods by introducing the MBP curves which can be used for conventional drilling optimization as well as the dynamic optimization of the multibit drilling programs.
NOMENCLATURE

\( A_f = \text{abrasiveness constant, } 10^3 \text{ lbf [kg]} \)

\( a_2 = \text{drillability vs. depth exponent, unitless} \)

\( C = \text{total drilling cost, } \$ \)

\( C_F = \text{cost per foot, } \$/\text{ft [}/\$\text{/m.]} \)

\( C_R = \text{rig cost, } \$/\text{hr} \)

\( C_b = \text{bit price, } \$ \)

\( D = \text{depth, ft [m]} \)

\( D_i = \text{initial drilling depth, ft [m]} \)

\( D_t = \text{total drilling depth, ft [m]} \)

\( F = \text{footage, ft [m]} \)

\( K = \text{drillability constant, ft/10^3 \text{ lbf [m/10^3 kg]} } \)

\( N = \text{rotary speed, 1/min} \)

\( N_b = \text{total number of bits, unitless} \)

\( R = \text{instantaneous rate of drilling, ft/hr [m/hr]} \)

\( \bar{R} = \text{average drilling rate, ft/hr [m/hr]} \)

\( (R)_{max} = \text{maximum bit performance curve} \)

\( t_b = \text{bit rotating time, hr} \)

\( t_t = \text{trip time, hr} \)

\( W = \text{weight on bit, lbf x } 10^3 \text{ [kg x } 10^3 \text{]} \)

\( \Delta D = \text{depth increment used in iterations, ft [m]} \)

REFERENCES

presented at the Rocky Mountains Regional Meeting of the SPE, Billings, MT, (May 19-21, 1982).


APPENDIX

Calculation of the Dynamic Drilling Strategy for PDC Bits

The problem of the optimal drilling strategy can be expressed in the dynamic programming terminology as an equipment replacement problem [16], [17], [18]. Habitual to the dynamic programming, the calculating procedure began at the end of the drilling process, at the final depth \(D^t\). It proceeded backwards, with the increments \(\Delta D\) (stages) until it reached the initial depth \(D_i\), as shown in Fig. 5. At each depth \(D_n\); state variable), there was a certain number of possible runs (decision variables) made by a PDC bit that may have been drilled from depth \(D_n\). Each of the alternative bit runs was associated with certain cost of drilling from depth \(D_n\) to the final depth \(D^t\) (return function) calculated as

\[
C(D_n) = C_b + t_b(D_n) + \left[t_b(F)\right] + \min \left[\sum_{j=1}^{n-1} C(D_j)\right] \quad (A1)
\]

where

\[
F = F_n \exp \left[a_2(D_n - D_i)\right] \quad (10)
\]

and \(F_n\) represents a roster of possible bit runs from depth \(D_n\). The only bit runs considered were those which gave a minimum cost (accumulated return), thus

\[
C(D_n) = \min[C_b + t_b(D_n) + t_b(F)] + \min \left[\sum_{j=1}^{n-1} C(D_j)\right] \quad (A2)
\]

or, by using the maximum bit performance curve, we obtain

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Considering the principle of optimality, equus. (A2) or (A3) are equivalent to the expression (minimum total return)

\[ C(D_n) = \min \left\{ C_b + t(D_n) + \left[ t_b \right]_{\min} + \min \left[ \sum_{j=1}^{n-1} C(D_j) \right] \right\} \]  

(A4)

Equation (A4) is identical to equ. (9).

Expression (A4) constituted an iterative procedure of dynamic programming. When the last step was made (stage \((D_t-D_i)/\Delta D)\) and \(D_n = D_i\), the value of expression (A4) was the absolute minimum cost of drilling from \(D_i\) to \(D_t\). The last component of equ. (A4) gave the optimum distribution of the bit runs and, consequently, the optimum number of PDC bits.
ABSTRACT

A new method was developed for in-situ measurements of the PDC bit condition and lithology change detection. In this technique, a diagnostic plot is made by correlating two dimensionless groups containing measured values of torque, weight on bit, rotary speed, and penetration rate. Several laboratory and field data (presented in the study) confirmed the linearity of such a plot. The diagnostic plot is a unique indicator of the bit/rock interaction and it is independent from bit operational variables. Moreover, the instantaneous wear of a PDC bit can be computed from coordinates of the straight line points. This method is feasible for a graphical use supplemented with a computer program.

The technique was further verified by comparing the predicted and the measured PDC bit wear from the MWD records in the Gulf Coast area. Also provided were the examples of a correlation between rapid formation changes and discontinuities in the diagnostic plots.

The new method contributes to the PDC bit drilling
theory. Its importance lies in the MWD software development for the purpose of the in-situ rock detection and the PDC bit evaluation and control.

INTRODUCTION

Polycrystalline diamond compact bits have proven to be long lasting bottomhole tools because of the absence of moving parts and the high wear resistance of synthetic diamond. However; PDC bits are extremely sensitive to formation properties and operating conditions [1]. Studies on PDC drilling performance in the harsh environment such as geothermal and hard rock drilling showed an average two-fold increase of penetration rate and bit life resulted in cost reduction 10 to 15%. However, in the case where the PDC bit life was reduced 50%, cost savings were cut in half [2], [3]. Therefore, there is a strong incentive to improve bit life in any potential application of PDC bits, in order to take advantage of their high penetration rates. Early detection of lithology changes and appropriate adjustment of operational variables are very important measures to save the bit. Such detection can be made possible with the instantaneous drilling data acquisition (MWD) system. Development of data acquisition systems has been in progress for many years in the petroleum industry. However, without having an appropriate data processing tool (a drilling model) some of the information will become
invaluable.

Successful application of a drilling model is not only a result of model's precision but it is also dependent upon the quality of the input data. Accuracy of drilling measurements, specifically regarding the used bit condition, is low. Recently an effort has been made to improve the bit dull evaluation [4]. However, the quantitative measurement of bit dull is still to be developed. Furthermore, there is no measurement of bit condition while drilling, which makes the verification of drilling models unreliable.

The conventional drilling models consisted of the drilling rate and bit wear formulas. The measured variables included weight on bit, rotary speed, and drilling rate. With the development of the MWD, the bit torque has become another important variable measured at the bit, hence improving verification of drilling models. Recently, the roller bit torque model was developed and studied [5]. It was also proved experimentally that drilling efficiency, as measured by specific drilling energy, did not correlate with weight on bit [6]. It was concluded that the torque equation should be developed from the balance of forces rather than from the specific energy concept [5].

Burgess and Lesso, [7], used simple relations
between bit weight, tooth penetration, and wearflat area together with the torque equation to infer roller bit dull from the MWD measurements. This interesting concept was not supported by the actual measurements of the bit dull. However, qualitative correlations between torque, drilling rate, and bit condition were observed for roller bits.

Experimental studies of PDC bits [8], [9] showed that there was a strong correlation between the bit torque and bit wear. It was suggested that rotary torque could be used as a guide for selecting the bit weight, and rotary speed for the best penetration.

In a recent study, Wojtanowicz and Kuru [10] concluded that the main controlling variable in drilling is weight on bit (given constant bit dull). On one hand, bit weight controls the torque through the mechanisms of shear, friction, and rolling resistance (in case of roller bits). On the other hand, the bit weight determines the level of stress at which the rock is destructed. Specific energy depends upon this stress. Therefore for each weight-on-bit value, the drilling efficiency (measured by the energy output-input ratio) is different. Also, the bit dulling process changes stress under the bit cutting structure which results in a change of drilling efficiency. Torque equation represents energy added to the drilling process, thus, providing important energy balance evaluation to the drilling model. They also
presented further development of previous model, [11], by using an expression for PDC bit torque and bit wear. The torque and drilling rate equations were then combined to give a diagnostic drilling model. Based on the model, a simple procedure was developed for evaluation of an instantaneous bit wear, as well as for detecting rapid changes in lithology of formations drilled.

DEVELOPMENT OF THE DIAGNOSTIC MODEL

The torque insensitivity to the bit weight can be compensated by introducing the drilling rate into bit torque relationship,[11], given as:

\[
T = W \frac{4(1-\mu t g \alpha)}{d_b (\mu + t g \alpha)} - \frac{a_w}{e_1 R_p} \left(\frac{2-(\mu + t g \alpha)}{\mu + t g \alpha}\right)^2
\]  

This provides another measured variable, proving a high level of correlation with the torque and the bit wear. The concept employed here is to improve determination of a non-measured variable (bit wear) by using all the four other measured variables (W, N, T_b, R). This will reduce a scatter of data caused by factors not considered by the model.

By using the drilling rate equation,[11], given as:
\[ R = K G_1 \left( W - k_1 R_p A_w \right) U_D N^{a_1} \] \hspace{1cm} (2)

The wearflat area, \( A_w \), in equation (1) can be written as a function of drilling rate, rotary speed, wear, and weight on bit as follows:

\[ A_w = \left( \frac{W - \frac{R}{K G_1 U_D N^{a_1}}} {R_p k_1} \right) \hspace{1cm} (3) \]

By introducing (3) into (1), dividing both sides by \((W*d)\) and by standardization, we obtain:

\[ T_D = E_1 + E_2 F_D \] \hspace{1cm} (4)

where,

\[ E_1 = \frac{48k_1(1 - \mu \tan \alpha) - e_1 d_b (2 - (\mu + \tan \alpha)^2)}{d_b^2 k_1 (\mu + \tan \alpha)} \] \hspace{1cm} (5)

\[ E_2 = \frac{12 e_1 (2 - (\mu + \tan \alpha)^2)}{d_b^2 k_1 K G_1 (\mu + \tan \alpha)} \cdot \frac{R_{st}}{(W)_{st} N_{st}^{a_1}} \]

\[ T_D = \frac{12 \frac{T_D}{W}}{d_b} \] \hspace{1cm} (6)

\[ F_D = \left( \frac{12 \frac{R_{st}}{R}}{W_{st}} \right) / \left( U_D \frac{W}{W_{st}} \frac{N}{N_{st}}^{a_1} \right) \] \hspace{1cm} (7)
equation (4) constitutes a diagnostic drilling model. It indicates a linear relationship between the dimensionless groups $T_D$ and $F_D$ while drilling a homogeneous formation. The linearity of equation (4) was verified, using results obtained in laboratory drilling tests as well as the field drilling data.

An extensive study on the PDC drilling by Hibbs and Sogoian, [8], included results from the full scale drilling test conducted at the University of Tulsa. These data were analyzed here, using the diagnostic plot $T_D$ vs $F_D$. The results obtained for various bit types and different rocks confirmed the linearity of the plot. The example plot for the 8.75 in. bit run in Carthage marble is shown in Fig. 1.

The field verification was made using data from seven wells drilled with PDC's in the Gulf Coast area. In two of these wells the drilling data were provided by the MWD. In the other wells the drilling process was monitored at the surface. Five variables (drilling rate, weight on bit, rotary speed, torque, and drilling depth) were recorded for each one foot interval. Fig. 2 and 3 show typical examples of drilling logs made by plotting bit weight, torque, and drilling rate versus depth. In both figures, bit weight correlates very well with torque, though a significant fluctuation of data can be noticed. However, drilling rate in Fig. 2 indicates poor
Figure 1 - Diagnostic plot, laboratory drilling in Carthage marble.
Figure 2 - Drilling records from surface measurements.
Figure 3 - Composite drilling records from MWD (weight on bit, torque) and from surface measurement (drilling rate).
correlation with torque and bit weight, unlike Fig. 3 where this correlation is good. The reason is that measurements in Fig. 2 were made from surface and, therefore, they are more susceptible to distortion. In Fig. 3, however, all data except drilling rate are from the MWD bottomhole record, and thus chances are better that we have good correlation despite the fact that drilling rate was recorded at the surface. This comparison stresses the importance of having a consistent bottomhole drilling data for proper interpretation.

The two-step procedure was followed, to improve the quality of the data and hence the interpretation technique. At first, the logs of all drilling variables involved depth, were prepared by using MWD data (Fig. 3). The logs were smoothed, to eliminate the noise (or scatter), by following the general trend of each curve. Care was taken to correlate the characteristic points of the logs as shown in Fig. 3. The readings of drilling variables were made, using the smoothing curves. Then, they were used to calculate the dimensionless groups, $T_D$ and $F_D$, and to make plots.

Ten plots were made using early data from various PDC bit runs (sharp bits). Typical examples of the plots are shown in Fig. 4 and 5. There was still a significant data scatter observed, however the linear trend was dominant in all plots made. The values of the
Figure 4- Field verification of diagnostic plot, well #3.
Figure 5- Field verification of diagnostic plot, well #2.
linear correlation coefficient were 0.6624 and 0.69608 for the new bit and for the used bit, respectively. The linear trends for the new and the worn bits (the straight lines in Fig. 4 are close; thus, providing additional evidence that the diagnostic plot $T_D$ vs. $E_D$ does not depend upon anything but rock properties.

DEVELOPMENT OF THE MWD DATA INTERPRETATION METHOD

The concept of the method is based on the diagnostic drilling model - equation (4). In this equation, the dimensionless groups $T_D$ and $E_D$ are functions of four measured drilling variables (bit weight, rotary speed, drilling rate, and torque) and one non-measured variable - bit wear (function $U_D$). At the same time, the mechanical properties of drilled rock and the PDC bit geometry are exclusively combined in constants $E_1$ and $E_2$. These constants represent the straight line intercept and the slope, respectively. That simply means that, at least theoretically, for the same bit and the same formation there will be only one straight line $T_D$ vs. $E_D$, and it will be independent from the bit wear. The interpretation procedure is as follows:

1. Calculate dimensionless values of $T_D$ and $E_D$ by using equations (6) and (7). Bit wear function, $U_D$, is equal to unity for the new bit. Hence, $E_D$ and $T_D$ values can be estimated by using the data from the
initial period of drilling.

2. Make a plot $T_D$ vs. $F_D$. Verify linearity of the plot and draw the straight line. The position and direction of this line ($E_1$, $E_2$) represent the formation drilled.

3. Use a continuous record of the MWD to detect any changes of the straight line's trend. A consistent change in direction will indicate formation changes.

4. Use the straight line data to infer instantaneous value of the bit wear as follows:

A. Calculate instantaneous value of $T_D$ using Eq. (6).

B. Enter the straight line plot ($T_D$ vs. $F_D$) with the calculated $T_D$ value [or use Eq. (4)] and determine corresponding $F_D$ value.

C. Solve Eq. (7) for $U_D$.

D. Solve instantaneous bit wear, $w$, from dimensionless wear function equations,[11], which are given as:

For $(x+h) < 0.5 d_c \cos \alpha$

$$U_D = \frac{\sqrt[3]{(y+\frac{1}{3}y^2)(y-5y^2+8y^3-4y^4)}}{(1+2y)\sqrt{y-y^2+8y^3-4y^4} - (1-10y+24y^2-16y^3)\sqrt{y+\frac{1}{3}y^2}}$$  (8)
For $(x+h) > 0.5d_c \cos \alpha$

$$U_D = \frac{\sqrt{3(4-5y+y^2)}}{(5-2y)\sqrt{-y^4+2y^3+5y^2+y + \frac{1}{4}}}
\sqrt{-y^4+2y^3+5y^2+y + \frac{1}{4}}
+ 2(-4y^3+6y^2-10y+1)\sqrt{3(4-5y+6y^2)}$$

The function $y(w)$ is defined as:

$$y(w) = \frac{h + x}{d_c \cos \alpha} = w + 0.028 \exp(-0.05w)$$

The above algorithm is a general outline of the data processing procedure, to be followed by a computational device installed on-line with the MWD data acquisition center. However, the most important issue here is whether practical applications of the method will provide meaningful results.

**DETECTION OF LITHOLOGY CHANGES**

The MWD drilling data from the investigated wells were collected above and below the transition depths at which a distinctive change in lithology was present. Then, the diagnostic plot was made and a change of the straight line trend was sought and correlated with the transition depth.
Examples of the lithology change detection using the method are shown in Fig. 6, 7, 8, and 9. The data from well No. 1 were used in analysis. There were two PDC bit runs in this well. The first one was a PD-11 9.875 in. bit. The bit was run from 3,100 ft. to 5138 ft. Formation was primarily shale, layered with sand. The sequence of sands and shales was carefully analyzed using the dual induction resistivity logs. The analysis showed that there was a rapid lithology change at depth 3893 ft. -Fig. 6. The dimensionless plot ($T_D^* \text{ vs. } F_D^*$) within the hole section from 3880 to 3910 ft. is shown in Fig. 7. The distinctive change of slope indicated that there was a transition from shale to sand at the same depth of 3893 ft. It was further detected that the sand layer was about 9 ft. thick.

The second run was also with the PD-11, 9.875-in. PDC bit. Fig. 8 shows the logging data, and Fig. 9 shows the diagnostic plot for the well section from 8009 ft. to 8027 ft. The change of slope was even more apparent than for the previous case. The lithology change was clearly detected at depth 8021 ft. The sand layer thickness was about 6 ft.

BIT WEAR VERIFICATION

In field drilling, there are only two measurements of the bit wear: initial ($w = 0$), and final, after the bit
Figure 6- Change in lithology detected by well logs, example 1.
Figure 7- Change in lithology detected by diagnostic plot, example 1.
has been pulled out. Therefore, the verification method has to consider these limitations. The method used in this research included plotting the diagnostic straight line \( T_D \) vs. \( F_D \) from the early record of drilling with new bit. Then the measurement of the bits final wear was used together with the drilling data, recorded just before the bit was pulled out of the well, to calculate several values of \( T_D \) and \( F_D \). New points \((T_D, F_D)\) were added to the plot. These points were associated with the worn bit. Theoretically, for the same rock, these points should fall on the new bit straight line. Therefore, the deviation of the worn-bit points from this line indicates an integrity of the diagnostic model.

The field drilling data collected in this work included detailed information about the bit type, manufacturer and the final bit dull. In two cases; bit dull conditions were reported by using the latest IADC fixed cutter bit dull code, together with the pictures of the dulled bit. In all other cases, the bit dull was assessed according to the bit vendors' standards. A 9.875-in. RP19 bit was run in well No. 3 from 6209 ft. to 10276 ft. The bit dull was graded 2, both for the inner and the outer rows, by using the IADC dull grading code.

Initially, the dimensionless plot of \( T_D \) vs. \( F_D \) for the new bit was prepared in Fig. 4. Then, using the measured final wear, the value of wear function, \( U_D \), was
Figure 8- Change in lithology detected by well logs, example 2.
Figure 9- Change in lithology detected by diagnostic plot, example 2.
calculated. Several $T_D$ and $F_D$ values were calculated, using the same $U_D$ value and the drilling record for the worn bit just before the bit was pulled out. The points corresponding to the late performance of the worn bit were plotted on the same graph, to check whether they fall on the straight line. As it is seen from Fig. 4, these points were reasonably close to the straight line.

The second example was from the 9.875-in. PDC bit run from 6538 ft. to 8325 ft. in well No. 2. The dull of this bit was also graded 2, but close examination of the bit condition, and information from the bit manufacturer, indicated that in their system, the total wear was considered 0.5 of compact face. Therefore, the same wear in the IADC system would be 1 instead of 2.

As in the first case, the values of $F_D$ and $T_D$, corresponding to the final wear, were calculated and plotted on the diagnostic graph. Fig. 5 shows the final points being fairly close to the new-bit straight line.

CONCLUSIONS

1. A new method was developed for the in-situ measurements of the PDC bit condition and for lithology change detection. The diagnostic model, equation 4, constitutes a single linear relation between two dimensionless groups containing four
measured drilling variables. The model allows simultaneous analysis of ever fluctuating values of these variables, at any particular instant of time.

2. In the absence of any direct measurements of bit wear at the hole bottom, the proposed method gives only available option for instantaneous control of drilling parameters to prevent early bit failure.

3. Analysis of the field drilling data showed a potential for this method to detect rapid formation changes, and to assess bit condition while drilling ahead. The latter application, however, requires more precise definition of the bit dull, as well as an analysis of the method's resolution.

4. The method constitutes a theoretical basis for development of a software, necessary to make full use of the MWD data for the optimized drilling control.

ACKNOWLEDGMENTS

The authors wish to express their appreciation to Professor A.T. Bourgoyne of LSU for his help in collecting field data.

An appreciation is given to Mr. Bill Cortlang and Mr. Rick Graff of the Tenneco Oil Company, and to the personnel of Tenneco Real Time Data Center in Lafayette.

We would also like to thank Mr. L.E. Hibbs, Jr. for his assistance in providing laboratory data.

Financial support from the Ministry of Education of Turkey, is greatly appreciated.

NOMENCLATURE

English upper case

\( A_w \) : Cutter wearflat area, sq.in.

\( E_1 \) : Constant in diagnostic drilling model, defined by equation 3, in.

\( E_2 \) : Constant in diagnostic drilling model, defined by equation 3, unitless.

\( F_D \) : Dimensionless drilling rate group.

\( G_1 \) : Unit conversion constant, 0.7576* 60 min./hr.

\( K \) : Drillability constant, \( ft/10^3 \) lbf.

\( N \) : Rotary speed, 1/min.

\( N_{st} \) : Standard rotary speed, 1/min.

\( R \) : Instantaneous drilling rate, ft./hr.

\( R_P \) : Rock resistance to pressing, \( 10^3 \) lb/sq.in.

\( R_{st} \) : Standard penetration rate, ft/hr.

\( T_b \) : Bit torque, in-lbf.

\( T_D \) : Dimensionless torque group.
\[ U_D \] : Dimensionless cutter wear function.
\[ W \] : Weight on bit, \(10^3\) lbf.
\[ W_0 \] : Threshold weight on bit, \(10^3\) lbf.
\[ W_{st} \] : Standard weight on bit, \(10^3\) lbf.

**English lower case**

\[ a_1 \] : Rotary speed exponent, unitless.
\[ d_b \] : Bit diameter, in.
\[ d_c \] : Cutter diameter, in.
\[ e_1 \] : Load distribution function, [11], sq.in.
\[ h \] : Cutting depth, in.
\[ k_1 \] : Proportionality constant between weight on bit and normal force acting on the single cutter, unitless.
\[ w \] : Dimensionless linear cutter wear, unitless.
\[ x \] : Linear cutter wear, in.
\[ y(w) \] : Function defined by equation 10, unitless.

**Greek lower case**

\[ \alpha \] : Back rake angle, deg.
\[ \mu \] : Friction coefficient, unitless
REFERENCES


paper presented at the ASME 1977 Energy Sources Technology Conference and Exhibition, Houston, TX (September 18-20, 1977).


CHAPTER VII.

CONCLUSIONS AND RECOMMENDATIONS

The main contributions of this research can be summarized as:

1. A new mechanistic drilling model for polycrystalline diamond compact (PDC) bits was presented. The model was derived from the balances of forces acting at the PDC cutter and it is fully explicit with physical meanings given for all constants and functions.

2. A new understanding of frictionally generated heat between rock and PDC cutter is introduced from the analysis of forces active on the wearflat and the cutting surfaces of a cutter. The simulation study considering the combined effect of cutting depth (drilling rate) and wear (bit dull) on the thermal response of polycrystalline diamond compact (PDC) cutters under down-hole drilling conditions showed that the cutting surface of a PDC cutter appears to be an important part of the frictional heat generation process. Comparative study using maximum bit performance curve indicated a significant change in PDC bit performance when the new and more rigorous temperature limitation is used. The study reveals that many failures in field applications of PDC bits may have been caused by lack of understanding of operational limits imposed by heat considerations.
3. A new PDC bit wear model was derived and used for bit performance prediction. The model relates bit life with temperature, weight on bit, rotary speed, cutter geometry, and frictional heat fluxes through wearflat and cutting surfaces. The new model shows that the effect of friction dominates bit life and this effect is greater than convective cooling.

4. A medium-scale laboratory instrument was designed, constructed and successfully used to measure friction between the sliding surface of a rock and polycrystalline diamond compact cutter. The prototype instrument offers a practical way to test PDC cutters for their wear performance and temperature response during rock cutting in the presence of lubricant. No significant effect of lithology and fluid types on the magnitude of the friction coefficient was observed.

5. A new method for preparing a multi-bit drilling program, the dynamic drilling strategy, was developed. The dynamic drilling strategy provided the best combination of PDC bit runs to achieve the minimum drilling cost for a long borehole interval. The method was numerically compared to the conventional drilling optimization and to the field practices. Considerable cost-saving potential of 25 % and 60 %, respectively, were estimated.
6. A new method was developed for the insitu measurements of the PDC bit condition and for lithology change detection. The method constitutes a theoretical basis for development of a software, necessary to make full use of the MWD data for the optimized drilling control.

Further investigations, concerning this research could include:

1. Evaluation of bit wear and load distribution functions across the bit face.

2. Constructing maximum bit performance curves for variable weight on bit and rotary speed conditions.

3. There is a very limited information available about the thermophysical properties of drilling fluids such as thermal conductivity, and specific heat. Further investigation in this respect is recommended.

4. The effect of reservoir conditions namely pressure, temperature, and the fluid content of the rock on the rock/cutter friction coefficient should be investigated.

5. Development of the MWD software based on the presented theory would provide a practical tool for field application.
APPENDIX A

EXAMPLE FIELD DATA EVALUATION WORKSHEET USED IN CHAPTER VI

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FIELD DATA FORMAT - MWD RECORDING

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EXAMPLE FIELD DATA EVALUATION WORKSHEET

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MAXIMUM DRILLING RATE : 200 ft/hr
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January 9, 1990

Ergun Kuru
Louisiana State University
College of Engineering
Division of Engineering Services
Baton Rouge, Louisiana 70803-6417

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"A Method for Detecting In-Situ PDC Bit Dull and Lithology Change",
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324–339
APPENDIX D

Reprint:

Kuru, Ergun, and Wojtanowicz, Andrew K.,
"A Compound Effect of Cutting Depth and Bit Dull on Cutters' Temperature for Polycrystalline Diamond Compact Bits",
A COMPOUND EFFECT OF CUTTING DEPTH AND BIT DULL ON CUTTERS’ TEMPERATURE FOR POLYCRYSTALLINE DIAMOND COMPACT BITS

E. Kuru and A. K. Wojtanowicz
Louisiana State University
Baton Rouge, Louisiana

ABSTRACT

This paper presents a simulation study to evaluate the combined effect of cutting depth (drilling rate) and wear (bit dull) on the thermal response of polycrystalline diamond compact (PDC) cutters under down-hole drilling conditions. A new understanding of frictionally generated heat between rock and PDC cutter is introduced from the analysis of forces active on the wearflat and the cutting (leading) surfaces of a cutter. Then this new concept is used to predict PDC bit performance with the controlled temperature of its cutters.

Previous concepts, largely based on the laboratory drilling tests (with low drilling rate and under atmospheric conditions), recognize only one source of heat—the wearflat surface. However, this study, using field data, shows that the heat generated at the cutting surface significantly contributes to the total heat flux into the cutter. In the result, the distribution of temperature within the cutter is changed which particularly affects the maximum value of temperature at the cutting edge.

A simplified 2-D finite difference numerical code is used to quantify the difference in cutter wearflat temperatures calculated with and without the additional heat flux generated at the cutting surface. The numerical analysis reveals that neglecting the cutting surface effect results in underestimation of the actual wearflat temperature by 10 to 50%, depending upon bit dull and downhole hydraulics.

Also demonstrated is the actual impact of these findings on field drilling practices. The example comparison is made by calculating the optimal-control procedures for PDC bit calculated with and without the effect of cutting surface. In these procedures, wearflat temperature becomes a mathematical constraint which limits weight on bit and rotational speed. The comparison includes calculation of the maximum bit performance curves which represent maximum drilling rate attainable for a bit to drill a predetermined length of a borehole (footage). The curves show an up to 18% reduction of drilling rate when the new and more rigorous temperature limitation is used. In addition, the example calculations show that the actual temperature of the bit cutters can be 660°F and exceeds by almost 30% its maximum acceptable value of 660°F.

For practical applications, the study reveals that many field failures of PDC bits may have been caused by lack of understanding of operational limits imposed by heat considerations.

NOMENCLATURE

English upper case

\( A_{\text{eff}} \) : Effective area open to flow, in²
\( A_{nf} \) : No flow area, in²
\( A_w \) : Cutter wearflat area, in²
\( C_{p} \) : Fluid specific heat, Btu/lb·°F
\( D \) : Characteristic diameter, in
\( F_d \) : Drag force, lb
\( F_N \) : Normal force acting at the cutter wearflat, lb
\( L \) : Characteristic length, in
\( L_{bc} \) : Bit body/rock clearance, in
\( L_{wf} \) : Cutter wearflat length, in

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INTRODUCTION

Drag bits are designed to fail the rock by shear rather than by crushing as is the case with roller cone and diamond bits. Since rock requires significantly less energy to fail in shear, more efficient drilling with less weight on bit and higher rate of penetrations is possible. A drag bit operating in a shear failure mode, however, has to have tremendous abrasion resistance to maintain a sharp cutting structure for a reasonable downhole life. The diamond/tungsten carbide compacts, originally developed as cutting tools in machining applications, have provided high abrasion resistance to polycrystalline diamond compact (PDC) drilling bits. In addition, the development of new cutters added a self-sharpening feature to the already more efficient shear cutting. Self-sharpening drag bits have appeared less susceptible to chip holddown effect than roller cone bits and much less than diamond bits [1], [2]. The polycrystalline diamond compact bits provide also higher drilling rates.

The success of PDC bits, however, not only depends on obtaining a sufficient drilling rate, but also on running the bit long enough to make their application economical. Changes in drilling operation parameters (such as weight on bit, rotary speed) which result in an increased drilling rate, will also result in an increase of the bit wear. Hence, better understanding of wear mechanisms effective on the cutting structure of the PDC cutters is essential to eliminate conditions leading to rapid wear and uneconomic bit life.

WEAR MECHANISMS OF PDC CUTTERS

PDC cutters have been experimentally tested by various investigators using single-cutter experiments [3], [4], [5], [6], [7], [8]; laboratory tests of full-scale prototype bits [4], [8], [9], [10], [11], [12], [13]; and field tests of full-scale bits designed using laboratory data [16], [17], [18].

A PDC cutter life is dependent upon the interrelated wear mechanisms of the tungsten carbide base and the PDC cutting edge that shears the rock surface, protecting the tungst-
en-carbide substrate from abrasive wear. At the same time, the tungsten-carbide base gives support to the diamond layer which is being subjected to the tensile and shear stress generated during cutting, absorbing shock loadings and preventing gross failure of the PDC layer [19], [20].

The steady state abrasive wear is normally associated with the development of uniform wearflats and gradual decrease in drilling rate over the bit life. On the other hand, the dynamic loading, caused by the drill string vibration or by bit wobbling results in the wear identified as chipped, broken, or lost cutters [18].

The mechanism of abrasive wear of the diamond layer has been already modeled [18], [21]. Also, relatively accurate predictions of PDC cutter wear have been made for the field bit runs. During the steady-state abrasive wear, the diamond-diamond grain boundaries are sufficiently strong to sustain severe stress and abrasion conditions so that each grain acts like a single diamond cutter and there is no massive pull-out. In the absence of the thermal effects diamond layer wears ideally by micro-chipping with the chip size much smaller than grains themselves. Intensity of the micro-chipping increases with sliding speed, presumably due to the increased temperature associated with higher speeds.

The inherent characteristics of drag bits impose high frictional heating on PDC cutters. A major drawback to PDC cutters, however, is their limited temperature stability. Rate of microchipping increases dramatically with temperature above 660 °F. A suggested explanation [4] is a decrease in the diamond grain fracture strength with increasing temperature. At 1350 °F, the wear mode changes from microchipping to more severe thermal deterioration and whole grain pull-out. This is caused by stresses resulting from differential thermal expansion between the diamond grain boundaries which lead to intergranular cracking and grain boundary failure.

The wear of the tungsten-carbide substrate, to which the sintered diamond compacts are attached, is confined to its wearflat area. Larsen-Basse [22], in surveying the literature of hard metal wear, concludes that in rock cutting with hard metal tools the predominant wear mechanisms are impact spalling, impact fatigue spalling, sliding abrasion, and thermal fatigue.

In the case of PDC cutters, the leading impact is received by the sintered diamond edge. Thus abrasion and thermal fatigue are the principal contributors to the metal wear at the wearflat. Glowka [19] gives an excellent discussion of wear mechanisms effective on a PDC cutting structure for both diamond tip and the tungsten-carbide back-up.

Better understanding of frictional heat generation associated with the PDC cutters action is required in order to control higher temperatures which accelerates the bit wear and thereby makes the bit run uneconomical.

Ortega and Glowka [23] developed an analytical relationship between PDC cutter wear temperature and drilling operation variables i.e. normal force effective on the cutter, cutter's velocity, and the wearflat area as

$$T_w = T_f - \frac{S_n V}{A} f$$

Derivation of equation (1) is given in Appendix A. The thermal response function, f, must be determined numerically in order to compute mean wearflat temperatures. It is a unique function of the cutter geometry, thermal properties of cooling fluid, and cutter material, and cooling fluid rate.

Thermal response function is determined by using numerical models of the cutter's heat transfer based upon assumed cutter/rock interaction. Currently, there are three different approaches available [10], [23], [24].

HEAT EXCHANGE MODELS FOR PDC CUTTER

Glowka and Stone [19] stated that the PDC cutter geometry introduced as a result of wear depends on the type of rock being cut. In soft, ductile rocks, where cutting forces (particularly impact loads) are small, the relative abrasive-wear resistance of tungsten carbide prevails. Owing to high wear rate of tungsten-carbide, a wear angle between tungsten-carbide back-up and rock surface developed due to the action of abrasive rock particles sliding beneath the cutter (Fig. 1-A). Examination of field-worn cutters revealed that this angle is about 5 to 10 degrees [25]. In the effect, only the diamond cutting edge remains in intimate contact with rock leaving a relatively small effective wearflat contact area (self-sharpening effect).

In harder, more brittle rocks, they observed that the wearflat was nearly parallel to the rock surface. They reasoned that the higher cutting forces for hard rocks effectively...
removed any unsupported diamond protruding beyond the tungsten-carbide backing. Furthermore, a substantial tungsten-carbide backing was required to support the diamond near the rock interface for high impact loading. They concluded that, under these conditions, the relative wear of the two materials reached equilibrium at a smaller wear angle (Fig. 2). The result is an effectively larger cutter area in contact with the rock. Their computational model took both cases into account. Temperature distribution within the cutter was evaluated by assuming heat flow into cutter only through wearflat area (Fig. 3-A). They also considered the difference in size of the wearflat area contacting the rock in each case. The thermal response function, \( f \), was first evaluated for hard rock cutting by using a finite difference numerical code (24). Later, it was calculated by using a finite element numerical code (26).

Ortega and Glowka (23), (26), (27) developed their understanding from their experimental observations during laboratory drilling of various rocks under atmospheric conditions. In general, brittle failure is dominant under atmospheric conditions, and the rock pieces fly away as they are cut. These loose chips do not exert any force on the leading (cutting) surface of the diamond compact. This is also the case of hard rock drilling, when a cutter wears out by impact loading rather than by abrasion. In hard rock drilling the cutter’s penetration (cutting depth) is small, particularly for a worn cutter with large wearflat area. Only a minor part of the cutting surface of the diamond compact contacts with the rock. Considering all the above factors, Glowka neglected the friction force acting on the cutting surface area. Therefore the model presented by Glowka (Fig. 2, Fig. 3-A) is valid for hard rock drilling, especially, in geothermal drilling areas.

In oil well drilling, however, most of the rock displays plastic behavior as its deformation characteristics change from brittle to ductile pattern with increasing depth (28), (29), (30). Therefore, it is believed that the self-sharpening wear mechanism of PDC cutters prevail during deep drilling which means that only a small wearflat area remains in contact with rock. The cutting depth, on the other hand, is much larger when drilling ductile rocks because soft-formation bits have fewer compacts of larger size. Large cutter penetration results in increased cutting surface being in contact with rock. A balance of forces acting on the single cutter for the ductile rock drilling conditions is presented in Fig. 1-B.

Laboratory studies with single PDC cutters [4], [5], [6] revealed that the force acting on the cutting surface area (cutting force) has the same order of magnitude as the normal force effective on the cutter wearflat area. This cutting force inevitably induces a significant value of sliding friction between cutting surface and rock surface thus adding to the friction effective under wearflat surface.

Fairhurst and Lacabonne (31) analyzed forces effective on conventional drag bits during hard rock drilling. They concluded that, as the bit advances into the rock a contact pressure developed normal to the front cutting face and the cut rock is constrained to move upwards across the face. Such movement, thus developed in the rock a frictional force acting downwards parallel to the bit front face. The frictional force and equal reaction force on the bit tends to push the bit up out of the rock. This frictional effect that results in the necessity to use heavy axial thrusts in rotary drilling to maintain bit contact with the rock. The friction coefficient is highly variable and assumes high values sometimes greater than unity.

Evidence of the presence of friction forces...
on the rake surface was also found in the experimental study of Kenny and Johnson into the cutting mechanism with tungsten-carbide tools for abrasive rocks [32].

They measured the amount of wear at the rake face and cutting edge. The magnitude of rake face loss was found to be in the same range as the cutting edge loss. The relative contributions of the wearflat, side face wear, and rake face wear to the total tip wear is given in Fig. 4. The existence of wear on the rake face indicated action of friction forces on this face.

Furthermore, considering friction coefficients on both surfaces to be in the same order of magnitude and assuming that the rock pieces are removed as soon as they are cut one will expect frictionally generated heat per unit area, per unit time (as calculated by equation A1) to be in the same range on both surface. Since the cumulative heat going into the cutter is determined by the size of the area in contact with the rock, the additional heat input through the cutting surface may significantly change the temperature profile within the cutter. Glowka and Ortega disregarded possible contribution of heat into the cutter through the interface between cutting surface and the rock surface. Therefore, their model will underestimate cutter temperature distribution for deep drilling.

Zijsling, [10], stressed on the self-sharpening feature of the PDC cutter. He presented an analytical model similar to Glowka's model together with a numerical code to predict temperature distribution in PDC's mounted on a matrix type bit. Two field-worn matrix type PDC bits were tested on a full-scale drilling machine. Detailed inspection of PDC's revealed that the diamond layer shows wear only at the cutting edge, mainly at the front and tapering off towards the backup material. As a result of this shape, the diamond layer attacks the rock at a large (-78 degrees) negative rake angle (Fig. 5-A). Zijsling suggested that this geometry appeared to be independent of the degree of wear of a particular PDC. Average cutting depth in these experiments was 0.0035 in. as imposed by the operating conditions of 15 Klb. weight on bit, 130 RPM rotary speed and the drilling rate of 10 ft/hr.

At elevated rotary speed (turbine drilling), however, geometry of cutter/rock interaction has changed (Fig. 6). In this case, the cutting edge appeared to consist of a part which was worn at a large negative rake angle and a part where the diamond surface was parallel to the direction of the motion of the cutter. Cutting geometry in this case, was attributed to the fact that cutting depth was smaller than the maximum possible height of the cutting edge under a negative rake angle of -78 degrees.

Zijsling also suggested formation of a build-up edge consisting of pulverized rock
under the cutting edge of the diamond compact. A particular loading pattern on the build-up edge keeps it in a stable position with respect to the cutting tool. Because of the high stress levels at the cutting edge, the build-up edge is composed of rock flour. During drilling, the actual sliding surface under the diamond layer is thus located at the interface between virgin rock and the rock flour of the build-up edge and its plane is parallel to the direction of motion of the PDC. As a result of the build-up edge and its plane is parallel to the direction of motion of the PDC. As a result of the build-up edge and the rock flour of the build-up edge being polished, the contact area at the build-up edge increases. Thus, the contact area at the build-up edge increases.

The performance of a cutter for various cutting depths is therefore not only governed by different rake angles but also by different friction coefficients at the sliding interfaces rock/rock flour and rock/diamond.

Based on the suggested rock-cutter interaction mechanism, Zijjsling developed his numerical model (Fig. 3-B) to predict cutter-temperature distribution under downhole drilling conditions. He assumed that all heat is generated at the sliding interfaces $S_1$ through $S_3$. The effect of heat generated at the interface $S_1$ on the cutting edge temperature is negligible compared to the other heat sources. The partitioning of the frictional heat at the interfaces $S_2$ and $S_3$ is not affected by heating of the contact materials of interface $S_4$.

Frictional heating is estimated according to equation (11). Owing to the presence of the build-up edge under the diamond layer, the frictional heat generated to be used in equation (11) for computation of the specific heat at the sliding interfaces should be that for rock/rock flour interface. Since sedimentary rocks and rock matrix material have a low thermal conductivity compared to that of diamond, the amount of heat flowing into the cutting element via interface $S_2$ is reduced as a result of the presence of build-up edge.

In his study, Zijjsling emphasized the self-sharpening feature of the PDC cutters. Analyzing the laboratory test results of two field-worn, full-scale bits, he suggested that the frictional heating at the build-up interface is generated at the diamond cutting edge of the cutter. His model (Fig. 3-B) employing a large negative rake angle and a rather small cutting depth, suggested that heat generation is restricted mainly to the sharp cutting edge of the diamond compact and partially to tungsten carbide back-up.

However, small cutting depth associated with low drilling rate (10 ft/hr) observed by Zijjsling does not reflect the PDC drag bit performance, particularly in soft formations. In the actual field drilling conditions, PDC cutters experience larger cutting depths than what Zijjsling observed in the laboratory. Fig. 5-B presents a cutter-rock interaction model employing a more realistic cutting depth value which was estimated using field data. Comparing this model (Fig. 5-B) with what Zijjsling observed in the laboratory (Fig. 3-C), the difference between the size of the cutting area where the cutting takes place at the rock in both cases is tremendous. Thus, the contact area at the cutting surface should be considered as an important part of the frictional heat generation process.

Recently, Prakash [24] presented a theoretical model together with a finite element formulation of temperature distribution and heat transfer in the rock and in the cutter for orthogonal cutting of rock. He used orthogonal metal cutting theory to define cutting forces effective on a PDC cutter during drilling. The heat generation along the shear plane due to shearing of the rock and the frictional heat generation along the rake face and wearflat are all considered in this model (Fig. 3-C). The model was verified by using limited data available from experiments of Hibbs et al. [24].

The Prakash work gives the most complete model of heat generation and cutter temperature. However, his method suffers from the inherent errors of computing the thermal response in a three-dimensional element with two-dimensional rock drilling conditions. Due to the low value of cutting depth, his conclusion was the same as Glowka's and Zijjsling's by saying that the largest contribution to the heat generation in the model is from the heat generation due to sliding friction on the wearflat which is over generalization of the average field conditions. The degree of heat contribution from different sources depends very much on the cutting depth (drilling rate) as well as cutter geometry and hydraulics.

**THERMAL RESPONSE OF PDC CUTTER TO FRICTION AT CUTTING SURFACE**

A complete theoretical solution of the problem requires a 3-D numerical model which includes all possible sources of heat generation during rock cutting with PDC bits. This is still a subject for future study. It was not the purpose of this paper to develop a complete numerical model. The intention here is to study engineering implications of frictionally generated heat during rock cutting with PDC bits. From the engineering standpoint, an understanding of the frictional heat helps to improve PDC bits' control and to predict their performance.

A wide range of field operational variables were considered in the simulation study [24]. NMD data from two wells drilled in the Gulf Coast of Louisiana was used to define the maximum and minimum values of penetration per rotation per cutter under field conditions. Another 9 7/8" fish-tail bit drilled from 5126 ft. to 5725 ft. in Well No. 1. Instantaneous penetration per rotation per cutter were calculated and plotted versus depth as shown in Fig. 7. Another 9 7/8" fish-tail PDC bit drilled from 5160 ft. to 5138 ft. in Well No. 2. Instantaneous penetration of the instantaneous cutting depth is given in Fig. 8. The penetration per rotation per cutter is calculated using the relationship between the penetration depth and the bit rotation speed given as follow:
Where, \( n \) is the number of interfering cutters at any radius along the bit face (or the number of blades). Analysis of field data has shown that cutting depth varies between 0.02 in. and 0.08 in. The difference is enormous, when these numbers were compared with Glowka's [23] 0.006", Zijlstra's [10] 0.0035", and Prakash's [24] 0.0012". Consequently, wearflat area is not necessarily the only source (or dominant source) of frictional heat generation all the time. One should take the cutting surface area into consideration to estimate cutter temperature especially for new bits running at high drilling rates.

**TABLE 1**

<table>
<thead>
<tr>
<th>Range of Drilling Operational Variables</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bit Size: 6 3/4 - 9 7/8 in.</td>
</tr>
<tr>
<td>Bit Type: Fish Tail Bit</td>
</tr>
<tr>
<td>Number of Blades: 3-4</td>
</tr>
<tr>
<td>Rotary Speed: 80 - 120 rpm.</td>
</tr>
<tr>
<td>Rock/Bit Friction Coefficient: 0.01-0.2</td>
</tr>
<tr>
<td>Bit Dull Condition: New bit - Horn bit (65 wear)</td>
</tr>
<tr>
<td>Mud Flow Rate: 180-700 gpm.</td>
</tr>
<tr>
<td>Mud Type: Water, Bentonite Mud, KCL-Polymer, Oil Base Mud</td>
</tr>
<tr>
<td>Convective Cooling Coefficient: 0.3761 - 1.761 Btu/hr-ft²F</td>
</tr>
<tr>
<td>Drilling Rate: 10-150 ft./hr.</td>
</tr>
</tbody>
</table>

The second aspect of the simulation study was to determine the limits of convective cooling of cutters by drilling fluid. Two fish tail bits (6 3/4" and 9 7/8" dia.) and two short taper-concave profile bits (6 1/2" and 9 7/8" dia.) were used to determine typical cooling ranges. Distribution of the cutters along the bit face were known for these four bits. Radial velocities around the cutters were estimated using the flow rates recommended by the manufacturers of these bits (180-700 gpm). A simple mass balance was used to estimate radial velocities. Knowing the clearance between the bit body and rock surface, the cross-sectional area of cutter exposed to cross-flow and the distribution of the cutters on the bit, the effective flow area was calculated. Maximum possible velocities around the cutter at each radius were then calculated by using known mud flow rate and effective flow area (see Appendix B).

To date, no data are available for convective cooling of PDC cutters with actual drilling fluids and flow rates similar to those used in the oil field. Glowka [23] presented empirical correlations for convective cooling of PDC cutters exposed to a uniform flow of water. He also proposed using the principles of similitude to infer cooling with drilling mud from the results of simplified laboratory measurements with water.

The literature is also very short of data on actual drilling mud thermophysical properties. Corre et al. [14] performed experiments on several field muds. They presented correlations for KCL-Polymer and oil-base mud. Some other
data were presented by Holmes and Swift [35], and Treqesser et al. [36]. Convective cooling coefficients were calculated by using available thermophysical and properties and similitude theory (see Appendix B). The range of calculated values were found to be from 0.1761 to 1.761 Btu/h-ft-°F for all practically attainable field drilling conditions.

A 2-D finite difference code was used to quantify the effect of this additional heat source on the temperature distribution within the PDC cutter. Thermal response function, f, was evaluated for both cases where heat generation was restricted only to the interface between wearflat and rock, and where cutting surface together with wearflat surface contributed to heat generation. Fig. 9 shows the maximum and minimum error made by neglecting the heat flux through the cutting surface. The error varies from 70% to 580% within the limits of convective cooling.

The equation (1) gives the direct relation between thermal response function (estimated by using a numerical model) and the wearflat temperature. Therefore, underestimation of thermal response function will result in underestimation of the wearflat temperature as well. The error in estimated wearflat temperature varied from 7% to 50% (Fig. 10). The impact of additional heat source on the wear-flat temperature is more pronounced for a new cutter and it reduces as the wearflat builds up (Fig. 11). A set of bit operation variables and rock properties presented in Table-2 were used to generate Fig. 11.

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TABLE-2

<table>
<thead>
<tr>
<th>Bit Size</th>
<th>7/8 in.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bit Type</td>
<td>PDC tail bit</td>
</tr>
<tr>
<td>Number of Blades</td>
<td>4</td>
</tr>
<tr>
<td>Rotary Speed</td>
<td>120 rpm</td>
</tr>
<tr>
<td>Average drilling rate</td>
<td>125 Ft./Hr.</td>
</tr>
<tr>
<td>Mud Flow Rate</td>
<td>717 gpm</td>
</tr>
<tr>
<td>Mud Type</td>
<td>Longcircinate</td>
</tr>
<tr>
<td>Convective cooling coefficient</td>
<td>1.6 Btu/h-ft-°F</td>
</tr>
<tr>
<td>Total number of cutters</td>
<td>27</td>
</tr>
<tr>
<td>Average weight on bit</td>
<td>10 lbs.</td>
</tr>
<tr>
<td>Cutting depth</td>
<td>0.002 in./revolution/cutter</td>
</tr>
<tr>
<td>Drilling fluid temperature</td>
<td>125°F</td>
</tr>
<tr>
<td>Rock thermal conductivity</td>
<td>1.1 Btu/h-ft-°F</td>
</tr>
<tr>
<td>Friction coefficient</td>
<td>0.15 (Berea Sandstone)</td>
</tr>
<tr>
<td>Rock thermal diffusivity</td>
<td>1.4×10^-5 in.^2/sec.</td>
</tr>
<tr>
<td>Rake angle</td>
<td>10 degrees</td>
</tr>
<tr>
<td>Diameter of a single cutter</td>
<td>0.52 in.</td>
</tr>
</tbody>
</table>

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EFFECT OF THERMAL CONSIDERATIONS ON THE OPTIMIZED PDC BIT PERFORMANCE

For PDC bits, the maximum cutter's temperature limits its operational parameters and affects bit's performance. Any source of an additional heat will affect the design of a single bit run so that the operational variables namely weight on bit, rotary speed, and mud flow rate have to be adjusted to make the bit run economical i.e. to minimize the drilling cost per foot.

The drilling bit efficiency can be completely described by using maximum bit performance (MBP) curves [37]. In engineering sense, a MBP curve represents a relationship between any possible footage made by a single drill bit and its maximum drilling rate.

In this study, the PDC bit optimization model and computer code for MBP curves, previously developed [37], were modified by introducing thermal response function. In the closed-form formulation, this function describes relationship among the cutter thermal response and the convective cooling coefficient, cutting depth, and dimensionless cutter wear as:

\[ f = 1.1 \cdot 0.569 \cdot b - 0.61 + 3.8 \cdot n^{-0.1417} \]

(2)

A 2-D numerical code was used to generate values of the thermal response function within practical limits of the variables. The limits of first two variables were discussed above. The dimensionless cutting wear varied between 0 to 0.25, corresponding to a new bit and worn bit respectively. Field practices reveal that PDC bits that have been worn over 0.25 are not rerun and are taken back to be salvaged.

Having covered all the practical ranges and combinations of these three variables, a set of thermal response function values was generated. Then, a non-linear regression analysis was used to determine a closed-form relationship among the thermal response function and the three variables. Fig. 12 presents the results of regression analysis as compared to the results from numerical analysis. A straight line passing through the origin at 45 degrees angle indicates the existence of a good quality fit.

The theory of maximum bit performance curves [27] provides also an optimal control of weight on bit and rotary speed which would render a maximum drilling rate for a given footage. An example of the optimum bit control is shown in Fig.13. Here, wearflat temperature is used as a mathematical constraint which limits weight on bit and rotary speed. The maximum weight-on-bit is calculated by using maximum allowable wearflat temperature (660°F) as a limiting criterion. The optimal bit control requires that the rotary speed be constant at its maximum value of 300 rpm, while the weight on bit be increased gradually, corresponding to increasing wearflat area so that the maximum allowable wearflat temperature is not exceeded.

The example in Fig. 13 indicates that 20% weight-on-bit reduction is required to accommodate the effect of cutting depth on the wearflat temperature. In practice, this means that bit will be overloaded and prematurely damaged if the effect of cutting surface was ignored. At later moments of bit life, the bit wears out and the wearflat surface becomes larger (which reduces the normal stress) so the wearflat temperature does not limit bit control. When the cutting depth effect is ignored, Fig. 13 shows that beyond the 600 ft. footage, the temperature constraint is not effective and the maximum weight can be used. However, in the case where the cutting depth was included in the calculations, the temperature constraint restricted bit control over the whole bit life (900 ft. footage).

The effect of bit penetration per revolution (cutting depth) on the bit performance was investigated by considering the change of the maximum bit performance curve with and without the cutting depth effect. In Fig.14, the solid curve represents the case where cutting depth is not included in the calculation of wearflat temperature, whereas the dashed line shows the expected bit performance with both thermal effect considered. Apparently, by neglecting the contribution of heat generated at the cutting surface, restrictions on operational parameters becomes less severe, and consequently the bit performance is overestimated. In other words, the bit will never achieve the performance predicted by solid line in Fig.14 - it will be prematurely damaged.

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Also illustrated is the effect of cooling provided by drilling fluid. As it is seen from Fig. 16, drilling rates changed considerably depending on the degree of cooling provided.

Also illustrated is the effect of cutting depth on the maximum bit performance. The reason for the above suggestion can be seen clearly in Fig. 15. In this figure, the bit temperature is calculated as the bit progresses in the rock. The calculation includes the case where cutting depth is included (solid line) and the case when it is not included (dotted line) in the calculation of wearflat temperature. In the latter case, the actual wearflat temperature is truly constant and equal to the maximum allowable temperature until drilling 580 ft. Later, the bit developed considerable wearflat area, and its temperature falls below its maximum value. In case the bit was operated with consideration given to its cutting depth, the temperature far exceeded its maximum value and would reach over 800°F for most of the bit life. Such conditions would surely cause thermal bit damage. Therefore, the bit performance indicated by the solid curve in Fig. 14 by following the erroneous bit control strategy (Fig. 13 solid curve) would never be reached. Moreover, the actual bit performance would be lower than its maximum possible performance (Fig. 14, dashed line).

Therefore, it is very important to design nozzle-cutter orientation as well as mud thermophysical properties properly to provide maximum possible cooling to the bit.

Finally, the effect of rock/bit friction coefficient on the drilling performance of a bit was investigated. Fig. 17 shows two cases where friction coefficients are 0.05 (solid line) and 0.15 (dashed line). The resulting plot clearly indicates that the bit performance can be improved enormously by reducing the friction coefficient which would in turn reduce the frictionally generated heat at the cutters. Therefore, the mechanism of the friction should be further studied to determine necessary means to reduce friction under PDC bits.

Field data used to generate figures 13, 14, and 16 were presented in Table-1 and in Table-2.
CONCLUSIONS AND RECOMMENDATIONS

The cutting surface of a PDC cutter appears to be an important part of the frictional heat generation process. Therefore, the previous studies, by neglecting this fact, underestimated the values of thermal response function, and PDC cutter wear flat temperature. Validity of equation A10 describing wear flat temperature in terms of drilling variables is limited by the assumptions inherent in its derivation. By introducing modified values of the thermal response function (cutting depth included) into the wear flat temperature calculation, the results will still be approximation. However, the effect of cutting depth is significant enough to realize that the thermal response function becomes a controlling factor in PDC bit operation.

Comparative study using MBP curves showed a significant change in PDC bit performance when the new and more rigorous temperature limitation is used. The study reveals that many failures in field applications of PDC bits may have been caused by lack of understanding of operational limits imposed by heat considerations.

There is very limited information available about the thermophysical properties of drilling fluids. An extensive investigation is needed to determine thermal conductivity, and specific heat of drilling fluids in relation to temperature and pressure.

REFERENCES


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APPENDIX A

PDC Cutter Temperature Model - Analytical Solution

Ortega and Glowka, (23), assumed that a PDC cutter is subjected to a uniform frictional heat flux which is stationary with respect to the cutter but which moves at the sliding velocity over the rock surface. Assuming that all the mechanical friction is converted into heat at the interface, the total heat generated per unit wearflat area per unit time is given as:

\[ q = \mu F^V/A \]  

Fraction of this heat goes into the cutter is given by

\[ q_1 = \eta q = \eta \mu F^V/A \]  

And fraction which goes into the rock is given by

\[ q_2 = (1 - \eta) \mu F^V/A \]  

Where \( \eta \) is called partitioning function and defined as a function of thermal properties of the two surfaces and the sliding velocity. The solutions for the temperature field about square and band sources of heat (Fig. A-1) moving over a semi-infinite constant property solid were given by Jaeger, (38), for the case where no heat is lost from the surfaces not exposed to heating.

The mean temperature rise of the contact area, between square heat source and the surface of a semi-infinite slab was given as:

\[ \Delta T = \frac{4qL}{\pi \kappa} \]  

Where \( \kappa \) and \( L \) are thermal properties of the rock, \( L \) is the wearflat length parallel to the cutting direction, and \( V \) is the cutting speed. Due to the intimate contact between the cutter and the rock, it is assumed that the wearflat temperature \( T_w \) is equal to the mean surface temperature of the rock. That is:

\[ T_w = T_f \]  

Also assuming the wellbore is cooled down to the drilling fluid temperature:

\[ T_f = T_r \]  

By introducing equations (A1), (A2), (A3), (A4) into equation (A6):

\[ T_w - T_f = \frac{4q}{\pi \kappa} \sqrt{\frac{V}{v_w}} \]  

Finally, the partitioning function, \( \eta \), is solved from equation (A7) as follows:

\[ \eta = \left[ 1 + \left( \frac{T_w - T_f}{Q_1} \right)^{\frac{4}{\kappa}} \right]^{\frac{1}{2}} \]  

Glowka [21] defined the ratio:

\[ f = \frac{T_w - T_f}{Q_1} \]  

as thermal response function, \( f \). The value of function \( f \) depends upon the cutter configuration, thermal properties, and cooling rates. Thermal response function must be determined in order to compute mean wearflat temperatures.

Numerical codes (10), (23) were used to estimate the values of function \( f \). By substituting equation (A7) into equation (A8), the wearflat temperature is given as follows:

\[ T_w - T_f = \frac{4qL}{\pi \kappa} f \]  

In equation (A10), \( Q_0 \) represents the normal force acting on the cutter; thereby it is directly related to the applied weight on bit. The sliding velocity \( V \) is a function of rotary speed, \( \eta \) represents the cutter wearflat area. The thermal response function varies depending upon the amount of cooling imposed on the cutter by drilling fluid. In summary, equation (A10) is a unique representation of the PDC cutter temperature as a function of controllable drilling operation variables namely, weight on bit, rotary speed, and drilling hydraulics.

Consequently, wearflat area (on the contrary to previous researcher's conclusion) is not necessarily the only source (or dominant source) of frictional heat generation all the
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The cutting surface area should be taken into consideration to estimate cutter temperature, especially for new bits running at high drilling rates.

The effects of flow field scaling on the cutter cooling rates may be determined with the following analysis. Most forced convection heat transfer data can be correlated in the form [39]:

\[ N_u = \frac{h D}{k_p^0.2} \]  

where \( n \) is exponent dependent upon the geometry and is an exponent with a value generally in the range of 0.3 to 0.4. Convective heat transfer data generated by Clowka [33] for a PDC cutter exposed to a uniform flow of water takes the form given as follows:

\[ h_w = \frac{k_p}{D} \left( \frac{V w}{u} \right)^{0.4} \frac{C_p u v}{k_p} \]  

Employing the definitions of \( N_u \) and \( Pr \) and using \( n = 0.3 \) gives the results for full-scale bits as follows:

\[ N_u = \frac{h_w D}{k_p^{0.7}} \]  

Since (Re) = (Re) for similar flow fields, the ratio of the projected field value to the measured laboratory value of the heat transfer coefficient is simply a function of the fluid properties.

Example problem:

Properties of drilling fluid at 130°F:

- Type: KCL-Polymer
- Density: 9 lb/gal.
- Specific heat: 1.034 Btu/lb°F
- Thermal conductivity: 0.7742 Btu/h-ft°F
- Viscosity: 5 cp.

Properties of water at 130°F:

- Density: 1 lb/gal
- Specific heat: 0.9981 Btu/lb°F
- Thermal conductivity: 0.3749 Btu/h-ft°F
- Viscosity: 0.513 cp.

1. Calculate maximum possible radial velocity around the PDC cutters at any radius along the bit face:

2. Properties of drilling fluid at 130°F:

- Type: KCL-Polymer
- Density: 9 lb/gal.
- Specific heat: 1.034 Btu/lb°F
- Thermal conductivity: 0.7742 Btu/h-ft°F
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A) Calculate maximum possible radial velocity around the PDC cutters at any radius along the bit face:

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Properties of water at 130°F:

- Density: 1 lb/gal
- Specific heat: 0.9981 Btu/lb°F
- Thermal conductivity: 0.3749 Btu/h-ft°F
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A) Calculate maximum possible radial velocity around the PDC cutters at any radius along the bit face:

B) Calculate maximum possible radial velocity around the PDC cutters at any radius along the bit face:

C) Calculate maximum possible radial velocity around the PDC cutters at any radius along the bit face:

D) Calculate maximum possible radial velocity around the PDC cutters at any radius along the bit face:

E) Calculate maximum possible radial velocity around the PDC cutters at any radius along the bit face:

F) Calculate maximum possible radial velocity around the PDC cutters at any radius along the bit face:

G) Calculate maximum possible radial velocity around the PDC cutters at any radius along the bit face:

H) Calculate maximum possible radial velocity around the PDC cutters at any radius along the bit face:

I) Calculate maximum possible radial velocity around the PDC cutters at any radius along the bit face:

J) Calculate maximum possible radial velocity around the PDC cutters at any radius along the bit face:

K) Calculate maximum possible radial velocity around the PDC cutters at any radius along the bit face:

L) Calculate maximum possible radial velocity around the PDC cutters at any radius along the bit face:

M) Calculate maximum possible radial velocity around the PDC cutters at any radius along the bit face:

N) Calculate maximum possible radial velocity around the PDC cutters at any radius along the bit face:

O) Calculate maximum possible radial velocity around the PDC cutters at any radius along the bit face:

P) Calculate maximum possible radial velocity around the PDC cutters at any radius along the bit face:

Q) Calculate maximum possible radial velocity around the PDC cutters at any radius along the bit face:

R) Calculate maximum possible radial velocity around the PDC cutters at any radius along the bit face:

S) Calculate maximum possible radial velocity around the PDC cutters at any radius along the bit face:

T) Calculate maximum possible radial velocity around the PDC cutters at any radius along the bit face:

U) Calculate maximum possible radial velocity around the PDC cutters at any radius along the bit face:

V) Calculate maximum possible radial velocity around the PDC cutters at any radius along the bit face:

W) Calculate maximum possible radial velocity around the PDC cutters at any radius along the bit face:

X) Calculate maximum possible radial velocity around the PDC cutters at any radius along the bit face:

Y) Calculate maximum possible radial velocity around the PDC cutters at any radius along the bit face:

Z) Calculate maximum possible radial velocity around the PDC cutters at any radius along the bit face:

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Cross-sectional area of cutter exposed to cross flow is given as follows:
\[ A_{nf} = L_c \cdot d_c = 0.75 \cdot 0.52 = 0.3875 \text{ sq.in.} \]

Effective area open to flow is given as follows:
\[ A_{eff} = \pi d_c^2 - A_{nf} = 3.14159 \cdot 0.52^2 - 0.3875 = 12.415 \text{ sq.in.} \]

Recommended flow rate range for 6 1/2 in. PD-2BT bit is 180 - 240 gpm. Finally radial velocity is calculated as follows:
\[ u = \frac{Q_g}{A_{eff}} \]
\[ u_{max} = 180 / (3.117 \cdot 12.415) = 4.65 \text{ ft./sec.} \]
\[ u_{min} = 240 / (3.117 \cdot 12.415) = 6.2 \text{ ft./sec.} \]

B) Calculate corresponding velocity of water having similar flow field as drilling fluid around the cutters:

Reynolds numbers are equal for systems which have the similar flow fields:
\[ (Re)_{mud} = (Re)_{water} \]
where \( Re = \frac{u_{d}D}{\nu_{d}} \)

In this case, since both fluids experience the same geometry, characteristic length (or diameter) \( D \) will be same for both fluids. Thus, from the equality of Reynolds numbers, water velocity can be written in terms of fluid properties and mud velocity as follows:
\[ u_{w} = u_{d} \left( \frac{\nu_{w}}{\nu_{d}} \right) \]
\[ (u_{w})_{min} = 4.65 \cdot ((0.513 \cdot 9) / (5 \cdot 8)) = 0.53 \text{ ft./sec.} \]
\[ (u_{w})_{max} = 6.2 \cdot ((0.513 \cdot 9) / (5 \cdot 8)) = 0.71 \text{ ft./sec.} \]

C) Calculate convective cooling coefficient for water flowing uniformly around a PDC cutter by using correlation developed by Glowa [33]:
\[ N_{w} = \frac{N_{p} A_{w}}{\Delta} \cdot c \cdot (\frac{\nu_{w}}{\nu_{p}}) \cdot 0.6 \cdot \sqrt{\frac{\nu_{w}}{\nu_{p}}} \]

Take \( D \) = Cutter diameter = 0.52 in.
\[ (h_{w})_{min} = 0.4173 \text{ W/\(\text{m}^2\)-C} \]
\[ (h_{w})_{max} = 0.4701 \text{ W/\(\text{m}^2\)-C} \]

D) Calculate convective cooling coefficient for drilling fluid around a PDC cutter:
\[ (h_{f})_{min} = (h_{p})_{min} \cdot \frac{C_{p}}{C_{f}} \]
\[ (h_{f})_{max} = (h_{p})_{max} \cdot \frac{C_{p}}{C_{f}} \]

\[ (h_{f})_{min} = 2.37 \cdot 0.7 \cdot \frac{0.9981}{0.513} \]
\[ (h_{f})_{min} = 1.391 \text{ Btu/(hr \cdot ft \cdot \text{F})} \]
\[ (h_{f})_{max} = 1.567 \text{ Btu/(hr \cdot ft \cdot \text{F})} \]
VITA

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