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ABSTRACT:
The development of fluted nozzles for oil and gas drilling is reviewed. A new nozzle design is presented and field data are shown to demonstrate that this design can significantly increase bit rate of penetration. Two new fluid mechanics studies are summarized, showing that the fluted nozzles slightly increase the hydraulic impact force and typically double the hydraulic horsepower of the flow.

1. INTRODUCTION

The initial report on the development of new asymmetric nozzles for oil and gas drilling appeared in the Oil and Gas Journal in 1996 (Akin 1996). That article reported on several novel hydraulics aspects of the three-dimensional (3-D) flows through the fluted nozzles. It described an initial interior geometry model, and listed some promising early laboratory and field drilling comparisons of the rate of penetration (ROP) of both polycrystalline diamond compact (PDC) and rotating cone rock bits (RB) with and without fluted nozzles. The main 3-D hydraulic feature of the asymmetric fluted jets noted at that time was the way the asymmetric interior transitional geometry of the jets caused a suction, or less than hydrostatic, pressure to be developed on selected regions of the impingement surface. This "negative pressure" aspect of the fluted jets was explained conceptually, demonstrated with experimental tank tests in water, and it was noted that computational fluid dynamics (CFD) studies had also verified this new hydraulic phenomena that was the basis of the first patent issued on this technology. Initially, it was speculated that the negative impingement pressure was the main reason that improved bit performance was observed when the standard jets were replaced with the fluted jets in both PDC and RB applications. However, it was noted that experimental measurements had shown that the local entrainment of re-circulating fluids increase by about a factor of four which, in turn, may help keep bits cooler and assist in removing cuttings. Now, a few years later, additional studies have proved that the fluted nozzles have additional significant hydraulic behavior as a result of their 3-D flows. Those new findings will be covered in later sections, but first the original fluted design and field performance will be compared with the revised geometry and its rate of penetration (ROP) at PDC drilling sites around the world.

2. INITIAL GEOMETRIES AND PERFORMANCE

The negative pressure fluted nozzles can be constructed from more than one geometrical design. The initial field and laboratory drilling evaluations were carried out using a valentine- or heart-shaped outlet, see Fig. 2 of (Akin 1996). Using that nozzle interior, a number of proof-of-concept field comparisons were carried out at eight different well sites in central Texas where a number of reliable offset well logs were available for ROP comparisons. In that study, PDC bits with fluted nozzles were run on four different manufacturer's bit platforms and compared to the results for the same bits with standard jets. There were 61 runs with standard jets compared to 27 with the fluted jets. The increase in ROP at the eight well sites ranged from a high of 44 % to a low of 11 %, with a weighted average of 28 % as shown in Fig. 1. Higher maximum increases were recorded for a small number of RB runs in Canada (Akin 1997). However, a 10 % change in ROP between runs may sometimes be considered as "noise" level changes even between good offset wells. Thus, some manufacturer's bit designs did not clearly see a ROP increase simply by changing from standard to fluted jets. That could be due to the original hydraulic design of such bits and/or the orientation of the valentine-shaped asymmetric nozzle outlet relative to the bit blade or cone locations.

After the completion of this proof-of-concept phase, the design went into regular production and was used in locations all over the world. The careful gathering of offset data was discontinued since the fluted nozzles were then a standard commodity. However, from field reports it was noted that the bi-lobe shape of the outlet in the original design occasionally made it awkward for field personnel to envision the best orientation of the fluted nozzles that would yield the ROP increases demonstrated in Fig. 1. Thus, after about a year of regular use of the original geometry a redesign study was begun. It considered three approaches that might lead to further improvements in the performance obtainable by using the fluted nozzles. They included:

1. evaluation of alternative fluted interior geometries to improve the hydraulics and to simplify the nozzle orientation in the field,
2. carry out more CFD and analytic fluid mechanics studies of asymmetric nozzle flows, and
3. work with manufacturers in the study of the general hydraulics behavior of their bit designs.

3. CURRENT GEOMETRIC DESIGN

That comprehensive study of field orientation procedures, typical bit geometries, manufacturing processes, CFD studies, and the proven interior transition geometries led to the development of a new fluted nozzle design. It also led to the clear identification of enhanced hydraulic features described later. The new geometry design retains the non-circular interior transition surface, with a fluted channel and sharp interior edges, but it terminates in a circular exit area that is offset to the edge of the nozzle. The new interior surface geometry is shown in Fig. 2. The outflow from the new fluted nozzle retains the important characteristics of the original design. That is, the pressure varies with location over the outlet area, the shear stresses vary with location over the outlet, the velocity vector varies in both its magnitude and direction over the outlet, and the outlet flow has angular momentum with respect to the inlet one-dimensional (1-D) flow axis. The interior edges of the transition surface are designed to shed internal vortices and therefore create an internal angular momentum in the flow. Thus the fluted jet flow differs from the classic standard 1-D flow model which is assumed to have a constant outlet pressure, constant parallel velocity vectors, no angular momentum, and no consideration of the shear stresses. The important flow differences between the fluted and standard nozzles are shown in Fig. 3 for future reference. A 3-D sectional view of the iso-surfaces of velocity from the fluted nozzle is shown in Fig. 4. From that figure it is clear that the fluid velocity is asymmetric and offset from the nozzle inlet centerline. Its surfaces of constant speed differ significantly from those of the standard circular jet. The latter has a "pencil like" tight cluster of cylindrical iso-surfaces of velocity that are axi-symmetric with respect to the nozzle centerline.

The flow through the fluted jets is clearly fully 3-D. One can gain some insight into the interior nozzle flow differences by averaging the velocity over the interior cross-sectional area and comparing those averages to the classic 1-D circular jets for the same flow rate, inlet area, and outlet area. Of course, they must therefore have the same inlet and outlet average velocities, which is confirmed by the non-dimensional velocity versus position graph in Fig. 5.

From that graph we see that the average velocities and velocity gradients of the two designs differ, especially at the outlet. To emphasize the difference a graph of the corresponding non-dimensional velocity gradients versus position is shown in Fig. 6. That graph shows that the fluted jet is designed, in part, to maximize the velocity gradient at the outlet. The shear stresses are defined by the velocity gradient, so this creates high shear stresses at the outlet flow area of the fluted nozzles. The increased shear stresses are important in 3-D calculations of hydraulic power, but drop out of a 1-D approximation of the flow. Another fluid mechanics consideration is that the turbulent energy equation and turbulent momentum equation both involve the product of the velocity and its gradient. That product is zero at the standard nozzle outlet, but is quite large for the fluted design. Thus the fluid exiting the fluted design has more turbulent energy and turbulent momentum.

4. ENHANCED FIELD PERFORMANCE FROM NEW DESIGN

The new design has been in widespread use around the world for more than a year as a standard commodity. To aid in assessing the improved ROP performance of this design an extensive search of site records was carried out to locate off-set runs for PDC bits with the same IADC code, run in the same formations, and run with and without the new fluted jets. One offset comparison group for 8½-inch PDC bits where five standard jet runs and one fluted jet run were found for the Belagim formation in Egypt. The fluted ROP was 12.1 m/hr while the average of the standard runs was 4.9 m/hr, which represents about 146 % increase in ROP. For the Rabbi Field in Gabon there were two primary formation segments that were also drilled with 8½ inch PDC bits. In the limestone, one fluted run at 19.3 m/hr was compared to three standard runs, which averaged 9.7 m/hr. That represented about an average 99 % increase in ROP, but that average value dropped to about 44 % for the salt/shale formation where two fluted runs averaged 21.27 m/hr while eight standard jet runs averaged 14.73 m/hr. Another pair of 8½-inch PDC bit offset runs for the Mungarong shale formation in the Gorgon 6 field produced a standard jet run of 37.9 m/hr, while the fluted jet run had a ROP of 66.3 m/hr (about a 75 % increase). Runs with 12¼-inch PDC bits were compared in the Asab Field in Abu-Dhabi. For rotating operations two fluted jet runs averaged 19.23 m/hr while six standard jet runs had a weighted average of 12.43 m/hr, for an average 55 % increase in ROP. Operations with drilling motors in the same field yielded nine standard jet runs that averaged 17.05 m/hr while the one fluted jet run was at 23.27 m/hr, giving about a 37 % increase in ROP.

Of course, in each group of offset comparison runs one observes a significant deviation from the reported group averages cited above. Combining these 25 offset runs into a single ROP comparison chart for the current fluted nozzle design yields the results in Fig. 7. There we see that the average for all 25 runs gives an 81.5 % increase in ROP. If we delete the highest and lowest results the average is 76.7 %. Keeping only the middle 75 % of the comparison runs we find that the fluted nozzle average ROP increase is 68.4 %. Thus, even though the PDC field data show scatter, one can typically still rely on a significant increase, on average, from utilizing the fluted nozzle technology.

While the statistical base for these comparisons is not as strong as that used in the original proof-of-concept tests in Fig. 1 they clearly show that, on average, the current design exceeds the 28 % average ROP increase found for the original design. Of course, the ROP results will clearly vary with the lithology and other factors so it is not practical to predict the performance increase in every case, but these randomly selected comparisons show that a significant ROP increase can be expected and at times it will be very significant. ROP is not the only consideration of importance in selecting nozzles for a bit run. The state of wear of the bit after the run is somewhat subjective but does determine when a bit can be rerun. Anecdotal comments and supporting photographs suggest that the flows associated with the fluted jets also extend bit life in several formations.

Fluted jet runs with available offset comparisons in Canada are quite small in number and are mainly limited to RB runs. Comparisons were found for the Alderson field about 40 km ESE of Brooks, Alberta, Canada. Using the same IADC 417, 200 mm, RB platform a series of vertical evaluation holes were drilled through shales, sand, and carbonates under similar operating conditions except that different types of nozzles were utilized. Both fluted nozzles and mini-extended nozzles were run and compared to the standard nozzle offset data. In (Akin 1997) the instantaneous ROP values versus depth were plotted for those three nozzle types. The fluted jets were shown to produce ROP values that were higher than the mini-extended and standard jets. For a different type of comparison those data were integrated to produce the corresponding curves of depth versus rotating time. Figure 8 shows that for the top 500 meters (to the top of the Viking formation) the averaged ROP for the fluted jets was about 60 % higher than the standard jet average, and about 20 % higher than the averages for the mini-extended nozzles. The ROP slope scale in the lower right corner illustrates these differences. For the final 300 meters through the Viking, Basal Colorado, and Mississippian formations (to a full depth of about 800 meters) there was not much ROP difference between the three nozzle choices. However, the initial high ROP for the fluted nozzles did reduce the time required to reach the total vertical depth (TVD).

The less than hydrostatic impingement pressure can clearly contribute to increased ROP in some formations, and the large local re-circulation flows may remove more cuttings, keep the bit cooler, and/or reduce balling so as to extend bit life. However, those features alone were not sufficient to explain the ROP improvements found with the initial design. The initial laboratory drilling tests clearly suggested that there were additional hydraulic aspects of the fluted jet flows that were not yet understood. The redesign studies provided proof that the fluted jets have higher hydraulic impact forces and confirmed the suspicion that they very significantly increase the hydraulic horsepower imparted to the exiting fluid. Having those new engineering study results helps explain why a consistent bit performance increase is seen with the use of the fluted nozzles.

5. FLUTED JETS DOUBLE HYDRAULIC HORSEPOWER

An important mechanical aspect of drilling hydraulics is the hydraulic horsepower that a nozzle imparts to its fluid and thus on to the impingement surface. The definition of the power, P, of a force vector, F, at a point is the scalar dot product of the force vector and the velocity vector, V, of the point. The magnitude of the (scalar) power is the product of the force and velocity times the cosine of the angle, say ang, between their two directions: $P = V * F * \cos(ang)$. Thus the power is zero when the velocity and force vectors are perpendicular (ang = 90 deg), and when V = 0 which occurs for a viscous fluid at a fixed solid wall. Otherwise a moving force produces power. At a point in a fluid moving with a velocity, V, a differential force, dF, is present and causes a differential power contribution:

$$dP = V * dF * \cos(ang). \qquad (1)$$

The fluid force, dF, comes from the pressure and shear stresses at a point acting over a differential area, dA. To obtain the total power, P, imparted to the fluid flowing through any nozzle it is necessary to integrate the differential power, dP, over the surface of the nozzle control volume. For a nozzle control volume we only need to consider its inlet and outlet areas when calculating the power because the fluid at the nozzle wall has zero velocity ($V_{wall} = 0$) and produces no power. Consider a cross-section of the nozzle (of area A_n) that is perpendicular to the flow. The volumetric flow rate, Q, is related to the velocity by $Q = A_n * V_n$, where V_n is the component of the velocity, V, that is perpendicular to the cross-section. For a steady flow in a standard circular section the flow is very accurately described by the classical 1-D theory given in beginning engineering courses and hydraulics references (Hughes 1976, Moore 1976). As noted above and in Fig. 3, the 1-D flow has a constant velocity, $V = V_n$, a constant pressure, p, and a constant shear stress, say T. The 1-D theory makes it easy to calculate the power because the value of any constant interated over an area is simply the area times the constant. In the 1-D case the force, dF, in the direction of the fluid flow is simply due to the constant pressure ($dF = p\, dA$) so the power applied at such a section is:

$$P = V_n * (p * A_n) = p * Q, \qquad (2)$$

which is simply the pressure times the flow rate. It is important to note here that in the 1-D theory the shear stress, T, does not appear in the power calculation in Eq. (2) because it is tangent to the plane of the cross-sectional area and thus perpendicular to the total velocity, $V = V_n$. That is, ang = 90 degrees between V and the shear stress part of dF which prevents the shear stress from contributing to the hydraulic power of a standard circular jet. That special case does not occur in the fluted nozzle. For a standard circular nozzle defined by the 1-D theory the total power imparted to the fluid by the nozzle, say P_{std}, is found as the difference in the power at the outlet and inlet sections given by the above equation, and results in the common expression for hydraulic power: $P_{std} = (p_2 - p_1) * Q$, where the subscripts _1 and _2 denote the inlet and outlet sections, respectively. Thus, the net power for a standard jet is simply the pressure drop through the nozzle, $(p_2 - p_1)$, times the flow rate, Q, as seen in typical drilling hydraulics references (Hughes 1976, Moore 1976). For a fluted jet with the same flow rate and pressure drop a higher hydraulic horsepower, say P_f, is created in part by the inclined velocity vectors combining with the increased shear stresses. Still, the pressure drop and flow rate, combined with the geometrical shape of the fluted jet, allow the 3-D fluted jet power to be obtained by multiplying the classical 1-D definition of hydraulic power by a coefficient, say C_f, that accounts for the 3-D nature of its outlet flows: $P_f = C_f * (p_2 - p_1) * Q$ for $C_f > 1$. Typical values of C_f for fluted jets used in drilling applications vary with nozzle size and interior geometry. For the current design shown in Fig. 2 the correction factor has been shown to range from about 1.8 in small sizes (#7) to about 2.1 in large sizes (#16), see (Akin 1998) for details for other sizes.

For fluted nozzles the net power imparted to the fluid is higher because the outlet flow (as shown in Figs. 3, 4, and 5) is 3-D [see (Akin 1998) for full mathematical details]. That 3-D flow causes two velocity components, say V_t, tangent to the outlet cross-sectional area and thus in the directions of the two shear stress components also lying in that plane. The angular momentum due to the interior edges causes the new velocity components perpendicular to the plane of Fig. 3 and the shear stress magnitudes are higher due to the high exit velocity gradients shown in Fig. 6. This means that the shear stresses add significant power to the fluid in the fluted jets, but do not do so in standard jets. Although the inlet power is still given by Eq. (2), all of the 3-D flow quantities vary significantly over the fluted jet outlet. That means one can no longer go from Eq. (1) to Eq. (2) at the outlet by inspection but must rely on an integration of the outlet flow quantities over the outlet area to get the outlet power. While the fluted jet outlet power can not be computed exactly in closed form because of the turbulent 3-D nature of its flow the result can be calculated by at least two completely independent methods. The most accurate value for the flute correction term for a specific fluted geometry is obtained by using a CFD calculation for the outlet flow quantities [see (Akin 1994)], and then numerically integrating those quantities over the outlet area to obtain the outlet power. Those calculations yielded the range in C_f cited above.

An independent estimate of C_f has been given by Dr. H. I. Huang (Huang 1998) where the analogy between turbulent fluid flow and non-linear electrical power systems was used to bound the coefficient, C_f. Huang shows that in general the range is $2 < C_f < 3$. This agrees with various CFD results before corrections (Akin 1998) necessary to cause the same pressure drop were applied to give $1.8 < C_f < 2.1$ cited above. Therefore, it can be proven that the fluted jets will typically double the hydraulic horsepower imparted to the mud, and thus it doubles the hydraulic horsepower per square inch (HSI) computed from the area of the hole.

6. HYDRAULIC IMPACT FORCE FROM FLUTED JETS

A secondary mechanical aspect of drilling hydraulics is the Impact Force applied perpendicular to a rigid surface on which a jet flow impinges. Recall that the Impact Force is developed by writing Newton's second law, $F = d(m\,V)/dt$, in the Impulse-Momentum Law form, which is the integral of $F\,dt = d(m\,V)$. In the above F is the force vector, V the velocity vector, m a point mass, t is time, m V is the momentum vector, F dt is the impulse force vector, and d denotes differentiation. For a steady fluid flow from a jet an Impact Force, $I=F$, develops an impulse of $I * (t_2 - t_1) = m * (V_2 - V_1)$ where the subscripts $_1$ and $_2$ now denote reference values before and after the jet impacts the wall, respectively. The usual assumption is that at impact the fluid stagnates so $V_2 = 0$. Rewriting this single particle impulse-momentum model one obtains the jet velocity, V_1, times the mass flow rate. Thus, a general statement is that the Impact Force vector, I, that a fluid particle creates has a magnitude that is the product of the mass flow rate and the velocity component before impact (perpendicular to the impingement surface). Of course, its direction is opposite to that of the velocity vector. The total value of the Impact Force must be summed, or integrated, over all particles in the jet flow. The mass flow rate is the same for both the standard and fluted nozzles and is given by the mud density, times its volume flow rate, Q. For the standard nozzle (unlike the fluted nozzles) the perpendicular exit velocity, V_n, is also the total velocity, $V_1 = V_n = Q/A_n$. Therefore, for 1-D flow we obtain the common result [see the section "Hydraulic Formulas" of (Hughes 1976)]: $I = \rho * Q * Q / A_n$. This is used to create a set of standard nozzle jet impact force coefficients, say C, that can be tabulated for various nozzle areas and flow rates such that the impact force is obtained from a table as: $I = \rho * C$. See Table 8 of (Hughes 1976) for jet impact force coefficients, C, for typical values of Q and A_n. The Impact Force from the fluted jets, say I_f, can be computed in the same way by including an extra correction coefficient to account for its 3-D flow. A typical value is 1.1, but one must again employ CFD to obtain a precise value for the correction. For the fluted nozzles the exiting flow, V_1, is 3-D, not 1-D, and varies at every point in the exit area. Since it is 3-D it has two tangential components in the exit area as well as one perpendicular to the exit area. To have the same flow rate the fluted jet has the same integral average normal component, say V_{n_ave}, as the standard 1-D jet. That is, $V_{n\_ave} = V_n$. However, the fluted jet also has an average tangential velocity component, say V_{t_ave}, that is not present in the standard jet. Thus, in general we can see that the resultant velocity vector V_1 used in the Impact Force calculation has a magnitude greater than that of the standard jet. We can then conclude that the fluted jets will slightly increase the hydraulic impact force, by about 8 % to 10 %, even though they can be proven to typically double the hydraulic horsepower.

7. CLOSURE

In this review we have outlined recently discovered fluid mechanics aspects of the fluted jets that help demonstrate at least four ways in which their 3-D flows offer improvements over the classical 1-D flow from the circular jets for use in oil and gas drilling. A newer geometric fluted nozzle design that is easier to orient has been documented to clearly cause significant increases in ROP for PDC bits for various lithologies around the world. While thousands of PDC bit runs have been completed only a few RB runs have been reported in detail [see (Akin 1997)]. Anecdotal reports indicate similar RB ROP increases with the new fluted geometry but a statistically meaningful group of off-set data are not yet available.

8. REFERENCES

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NOMENCLATURE

Variables:

A_n	=	outlet area of nozzle
ang	=	angle between vectors
C	=	coefficient
C_f	=	flute correction coefficient, 1.8 < C_f < 2.1
d	=	differentiation
F	=	force vector
I	=	impact force
m	=	point mass
P	=	power
p	=	pressure
Q	=	volume flow rate
rho	=	mass density
T	=	shear stress
t	=	time
V	=	velocity vector

Subscripts:

ave	=	average
f	=	fluted
n	=	normal
std	=	standard
t	=	tangential
x	=	x-direction
y	=	y-direction
z	=	z-direction
1	=	initial
2	=	final

FIGURES

Figure 1: Original Fluted Jet Field Performance

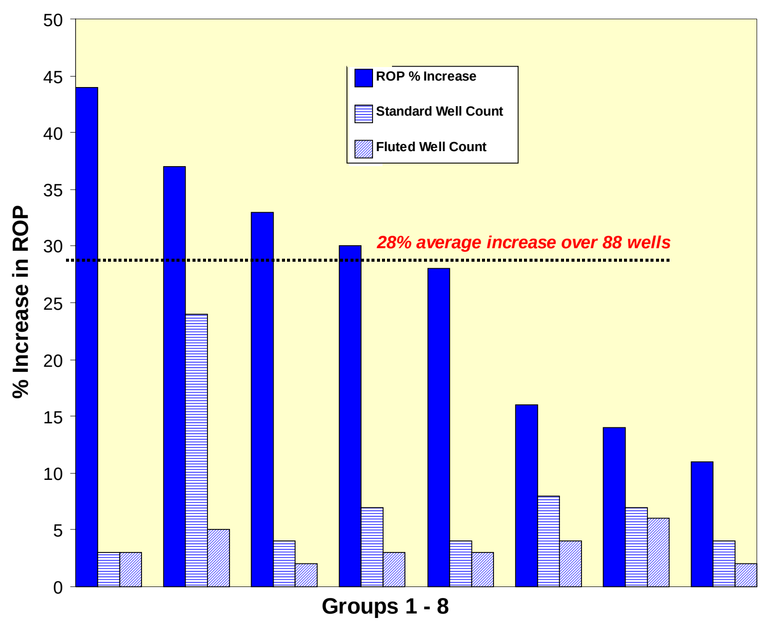

Figure 2: Revised Fluted Nozzle Geometry

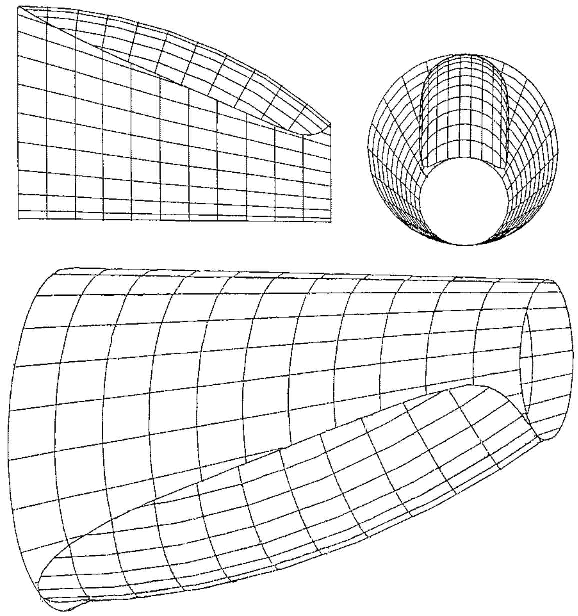

Figure 3: Nozzle Outlet Edge View

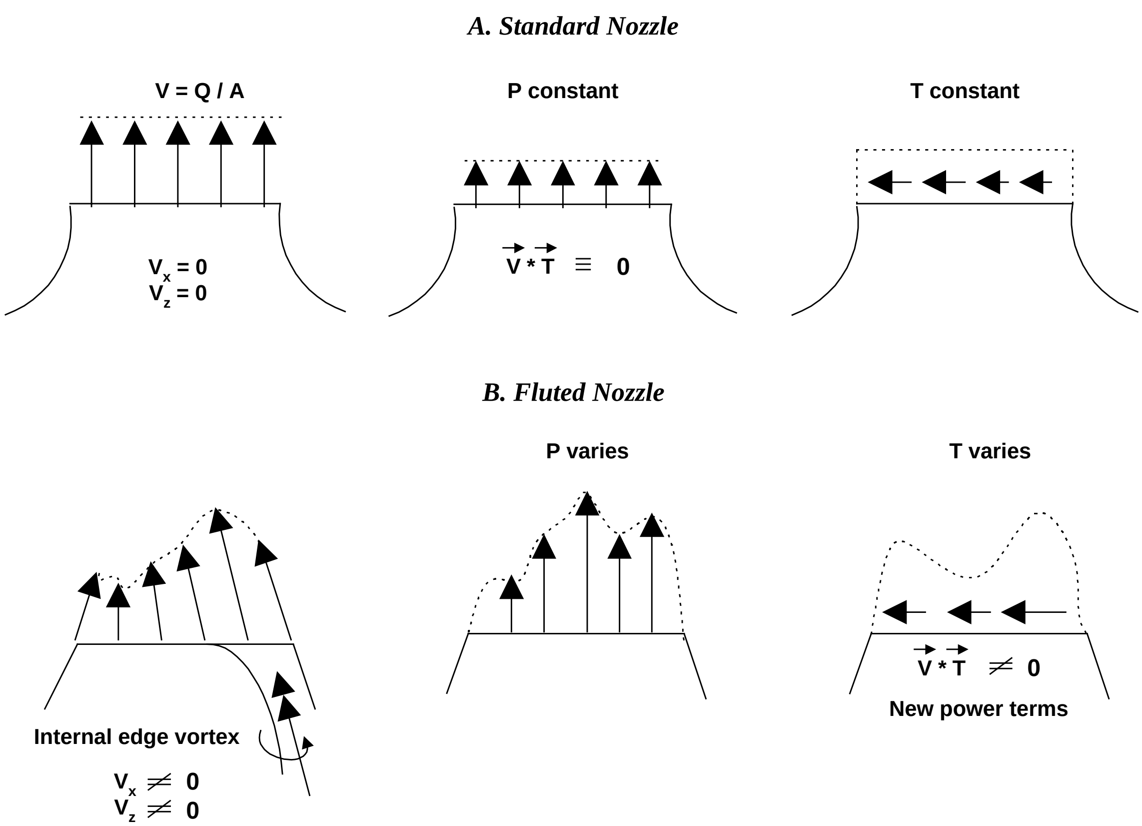

Figure 4: Fluted Jet Velocity Iso-Surfaces

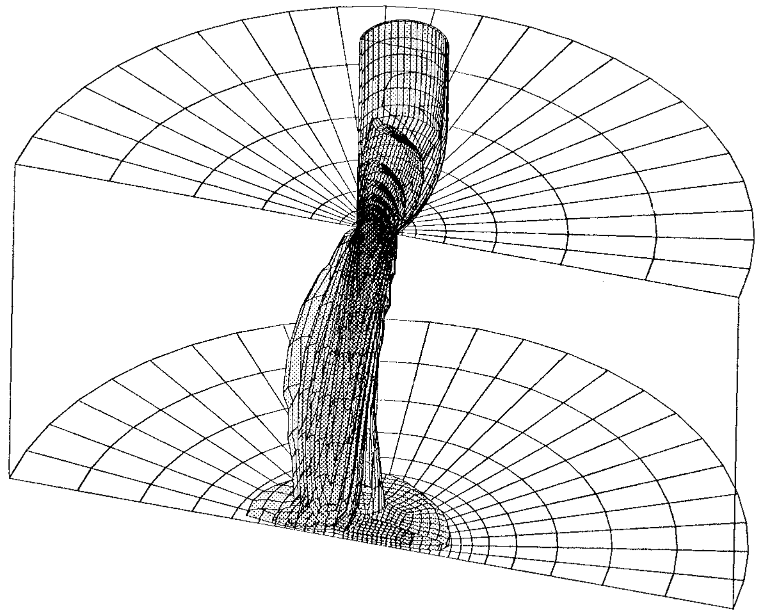

Figure 5: Velocity Along the Nozzle

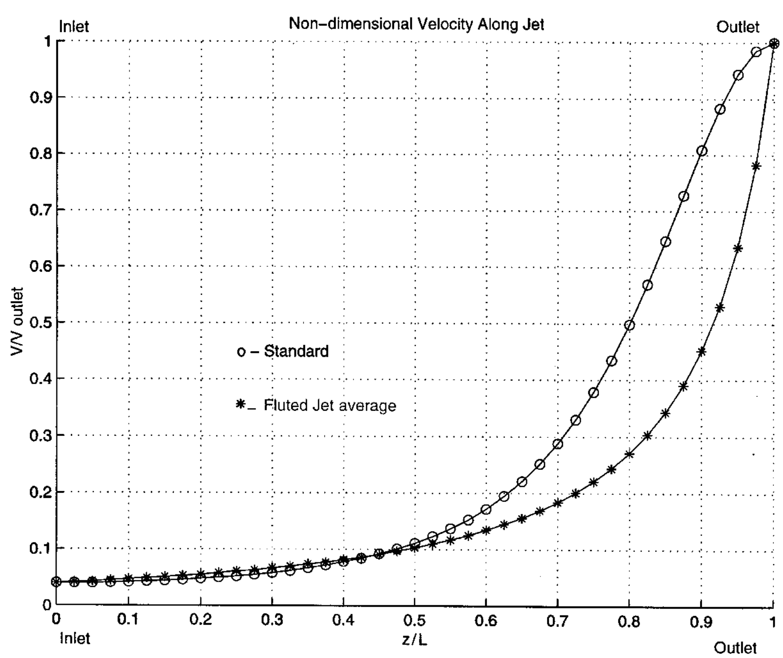

Figure 6: Velocity Gradient Along the Nozzle

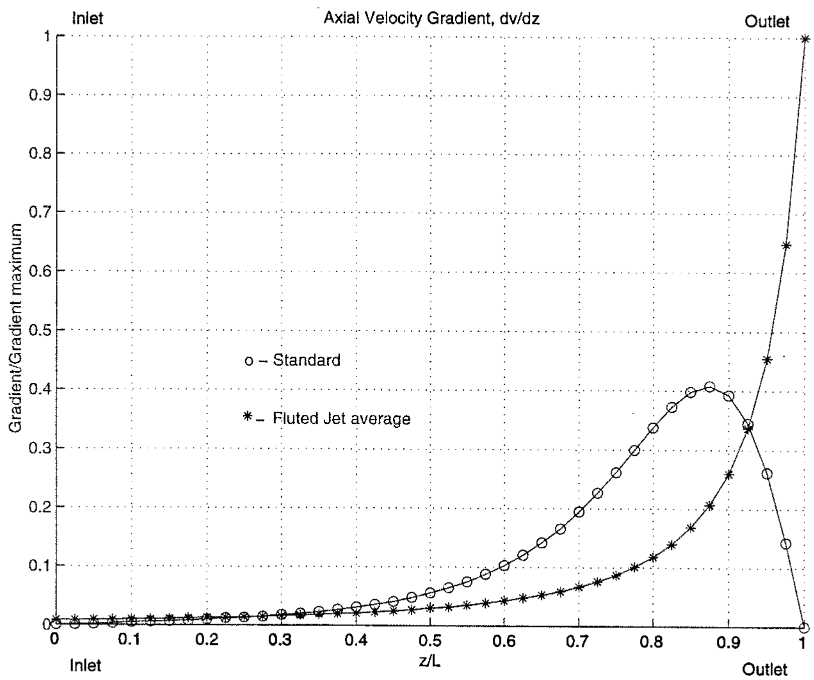

Figure 7: PDC Field ROP, Current Fluted Nozzle Design

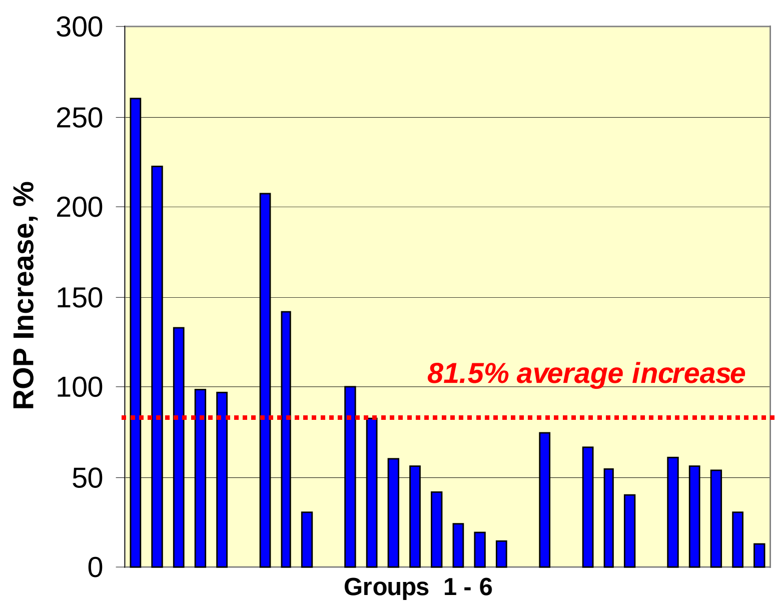

Figure 8 Canadian RB Upper Depth versus Time

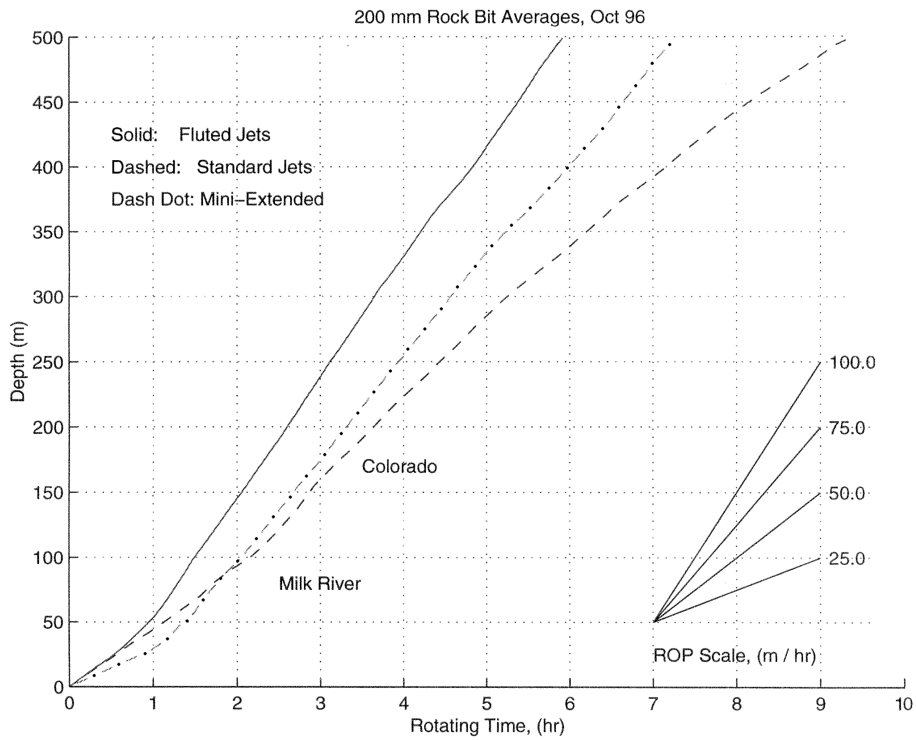

Figure 9 Canadian RB Total Depth versus Time