ADVANCED

OILWELL

DRILLING

ENGINEERING

HANDBOOK

&

computer

programs

MITCHELL

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Dr. Bill Mitchell
MITCHELL ENGINEERING
12299 West New Mexico Place
Lakewood, CO 80228, USA
Email: bmitchel@mines.edu
Tel#: 303 986 7453 USA

Printed in the United States of America.

For additional copies contact:
The Society of Petroleum Engineers of the AIME, PO Box 833836, Richardson, Texas, 75083-3836, USA.
Tel#: 972 952 9393
Fax#: 972 952 9435
Email: spedal@spelink.spe.org

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ADVANCED OILWELL DRILLING ENGINEERING HANDBOOK

CHAPTER I

TUBULAR DESIGN AND USE

GENERAL

The axiom of tubular design is that the loads placed on a tube by natural phenomena must be offset by its strengths. There are many natural phenomena which could dictate a particular tubular design. Also, there are many theories for determining the strengths of a tube. The tubular designer must therefore derive practical design equations from the theories and phenomena. These equations represent the "criteria for tubular design".

COMMON FAILURE THEORY ASSUMPTIONS

The most common simplifying assumptions with regard to tubular strengths are that the failure theory known as the MAXIMUM STRAIN ENERGY OF DISTORTION THEORY[1] applies only to tubular collapse strengths and that only biaxial[2] loads are considered within the theory. Thus tensile loads and burst loads are thought to be uniaxial[3] and strengths are rationalized with the MAXIMUM PRINCIPAL STRESS THEORY OF FAILURE.[4] Design factors are usually based on experience.

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1 This theory predicts failure of a specimen subjected to any combination of loads when the portion of the strain energy per unit volume producing change of shape (as opposed to change of volume) reaches a failure determined by a uniaxial test. Refer to Strengths of Materials, by S. Timonshenko, reprint 1976, Krieger Publishing Company.

2 Biaxial loads are those which result in the material of a structure being subjected to the simultaneous action of tension or compression in two perpendicular directions. Reference same as above.

3 Uniaxial loads are those which result in the material of a structure being subjected to the action of tension or compression in one direction only. Reference same as above.

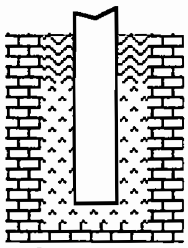
4 This theory predicts failure of a specimen subjected to any combination of normal and shear stresses when the maximum principal stress, which is the maximum normal stress acting on a set of perpendicular planes which have no shear stress acting on them, reaches a failure value determined by a uniaxial test. Reference same as above.

TUBULAR END CONDITIONS

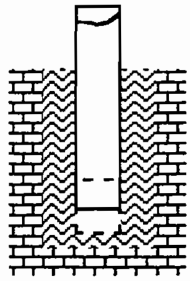
The ends of tubulars (top and bottom of the casing) may either be fixed or free. The bottom end is usually free until cemented and the top end is free until the wellhead slips are set. These conditions are tubular end conditions. The common practice in tubular design today is to design tubulars as if both ends were free. Of course, in stability analysis for buckling and loads, the critical conditions are when both ends are fixed. This condition only occurs after the tubular has been cemented and set in the wellhead.

NAMES OF CASINGS

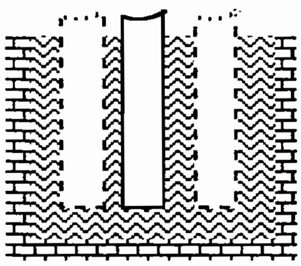
The name given to a casing is closely related to its primary purpose. Some popular names are



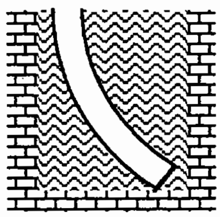
Cemented Casing



Up and Down



Side to Side



Bending

NAME	PURPOSE
Drive	returns mud to elevated pits
Conductor	returns mud to elevated pits and supports the weight of the other casings
Consolidation	prevents caving of hole (sand and gravel)
Surface	seals off fresh water zones
Intermediate	blocks formations and their fluids, commonly set in top of a pressure transition zone
Protection	protects against high pressure formation fluids
Production	conduit for fluid transport
Oil or Gas	conduit for fluid transport
Liner	casing terminated down hole rather than at the surface
Injection	conduit for fluid transport to a formation
Test	for testing a well

LOADS

The following is a list of loads which could be critical with a brief discussion of those loads.

1. Wellhead and BOPE are supported by casing in most designs.

2. When slips are set, the weight of a tubular is transferred from the hook to the wellhead. The wellhead in turn places this new load onto previously set casing.

3. Gravitational loads are the weights of tubulars and fluid pressure forces.

4. Friction loads act in the axial direction of the tubular if torsion is not present. These loads are derived from either an inclined hole or a dogleg. Doglegs cause a two fold friction problem: (1) the forces which bend the tubular around the dogleg also clamps the tube against the wall of the hole causing the highest of drag frictions, and (2) because all of the sections of a dogleg can not be vertical, the tubular, because of gravity, will be lying on the low side of the hole, producing drag. Also Drag is produced in inclined holes only because gravity pulls the tube to the low side of the hole. A load placed on a tubular, such as a collar hanging on a ledge, is not drag. It is contact load.

5. Contact loads are those placed on a tubular by objects within the hole. Common objects are ledges, other pipe, bridges, and even the bottom of the hole.

6. Formation loads can cause hoop loads to levels near those of the overburden. Movement of salt zones are an example. It is also possible for trapped brine in a salt zone to develop full overburden pressure and hydraulically collapse casing.

7. Applied loads at the surface often lead to tubular failure. Common loads are the following

   internal and external pressures
   pick-up and slack-off of the casing
   changing the weight of the internal or external fluids
   evacuating the casing

   Most human errors which directly cause the failure of tubulars are those for which combination loads are overlooked. For example, internal pressure and pick-up both add tension to a tubular.

8. Temperature changes produce loads.

9. Tubular corrosion and erosion are not loads, but both reduce the strength of tubulars. Tubular erosion (reduction of wall thickness) by drill collar or tool joint wear while drilling out of casing cannot be reliably quantified at this time; however, some designers add a measure of wall thickness to account for anticipated erosion and/or corrosion.

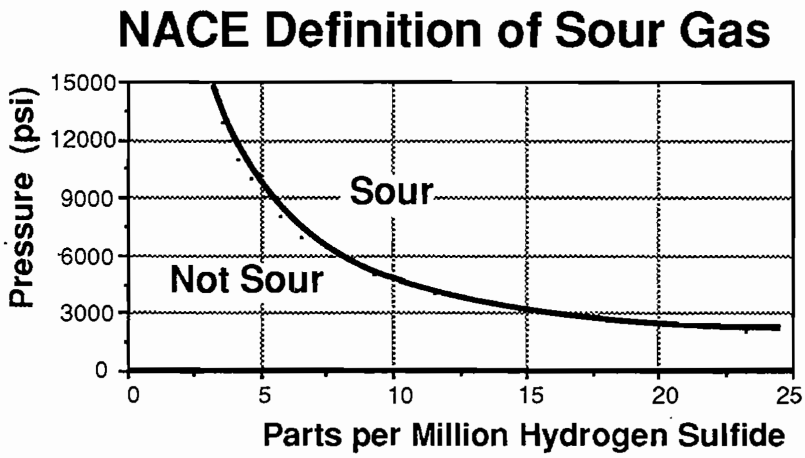
HYDROGEN SULFIDE AND STEEL

SOUR SERVICE - (Hydrogen Sulfide)

NACE Material Requirement MR-01-75 defines gas as "sour" if the partial pressure of Hydrogen Sulfide is 0.05 psia or more.

At 10000 psi, this translates to 5 parts per million (ppm) or 0.0005 Mol% of hydrogen sulfide.

MR-01-75 does not address pressures in excess of 10000 psi or partial pressures less than 0.05 psia. The NACE definition of "sour" is in Figure 1.



Sulfide-stress cracking of a particular steel depends on the amount of Hydrogen Sulfide present and also on the amount of tensile stress in the steel.

Steel at low stress can tolerate more Hydrogen Sulfide than it can at high stress. The "threshold stress" is the maximum stress that the steel can tolerate without brittle fracture.

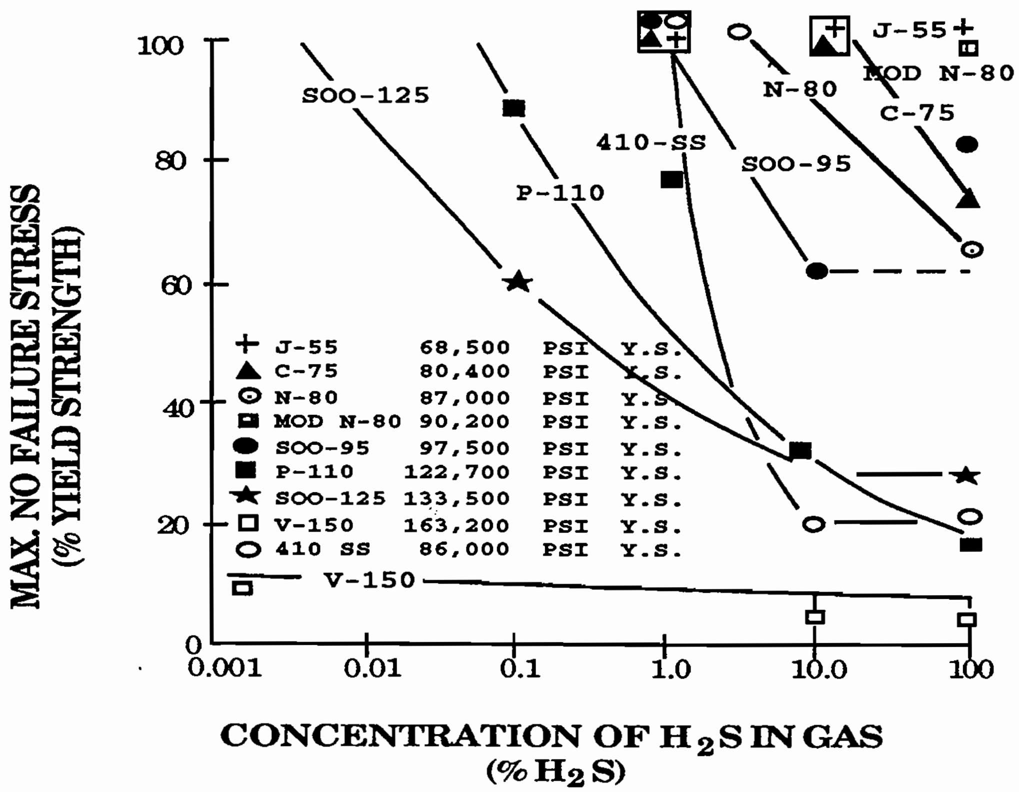
The Threshold Stress decreases as the amount of Hydrogen Sulfide increases.

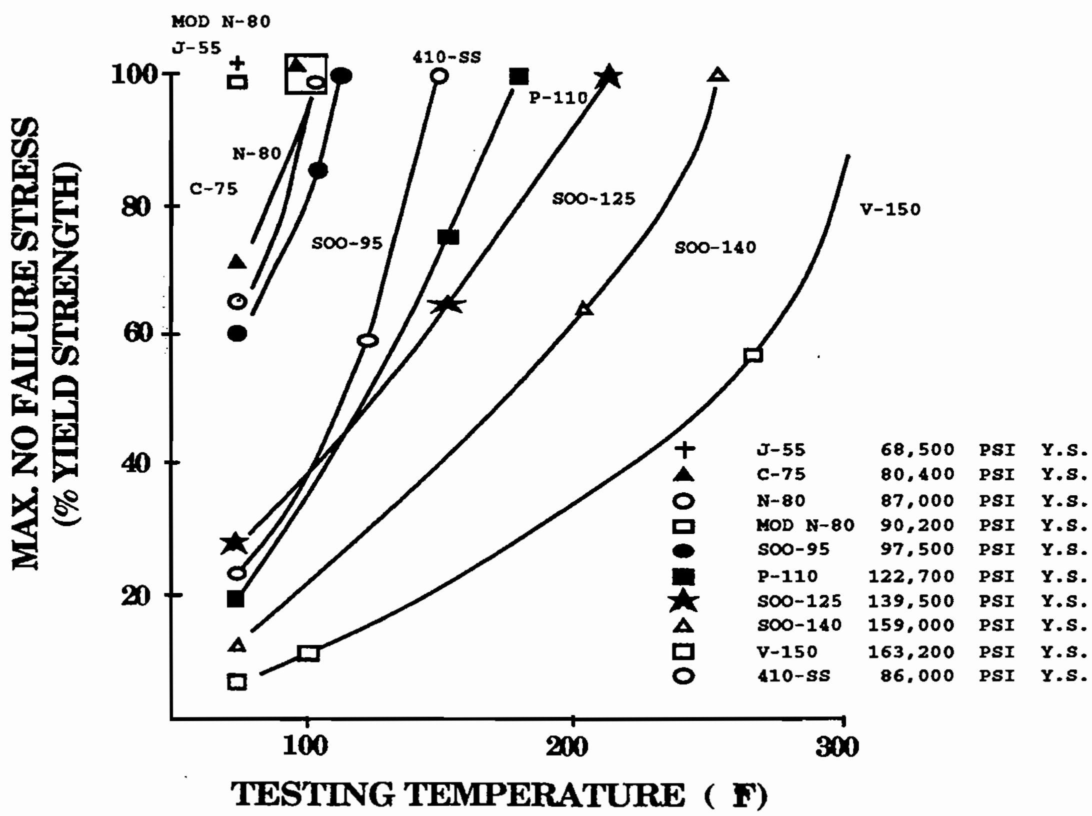
Similarly, steel at high temperature can tolerate more Hydrogen Sulfide than it can at lower temperatures.

Manufacturers usually classify a steel as "sour service" if the minimum Threshold Stress is 80% or more of the yield stress in tests at room temperature.

The next charts, for example, show that steel of grade P-100 should not be used for sour service unless temperatures are 175°F or more while H_2S is in contact with the steel.

STEEL FAILURE IN H_2S





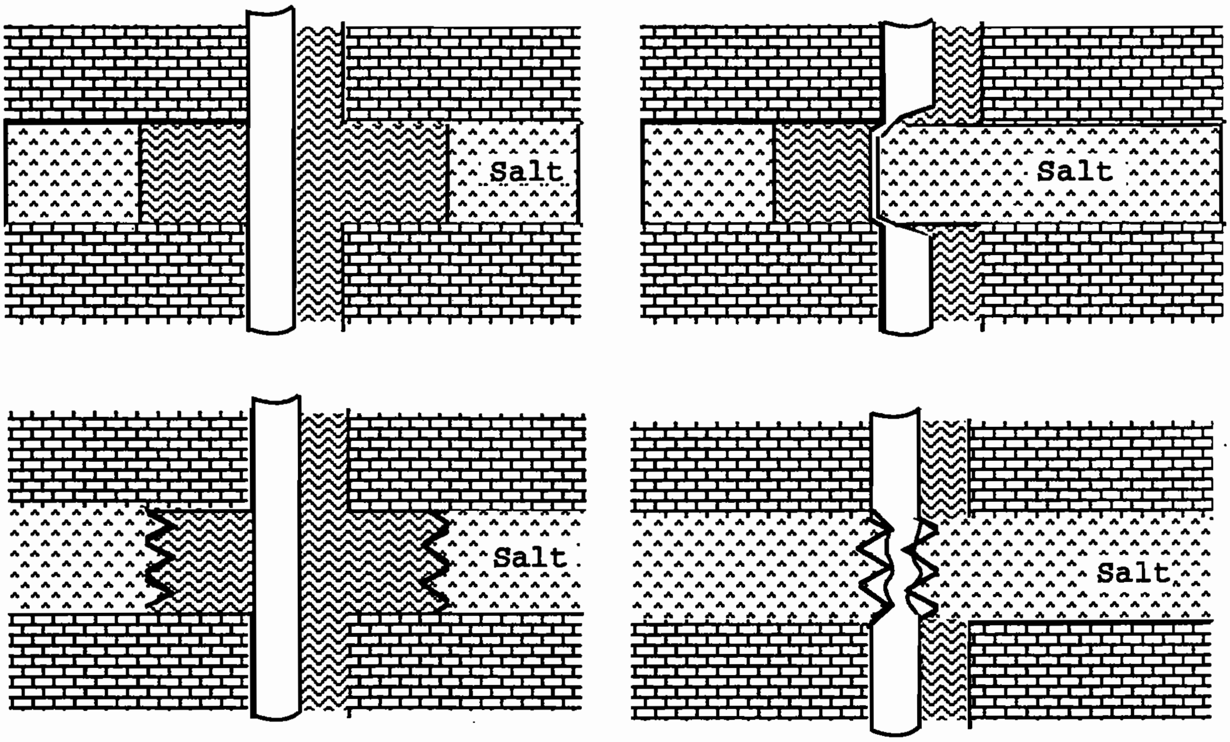
SALT AND DIAPERIC SHALE

The basic guidelines for preventing collapse caused by plastic salt are as follows in order of decreasing priority:

1. Drill a gauge hole through the salt.
2. Get a good cement job.
3. Run heavy wall pipe.

Two completely different mechanisms may responsible for casing collapse. A third is indirectly responsible, but may be the more prevalent cause of failure. Two of the mechanisms are shown in the sketches:

1. Salt Sheer Loading
2. Salt Point Loading



A good sheath of cement reduces the point loading and makes the external load more like hydrostatic pressure. The best results are obtained by increasing wall thickness, rather than yield strength. If point loading occurs, it is unlikely that even the heaviest pipe which is practical to run will be strong enough. A minimum collapse rating corresponding to a load of 1 psi/ft or more is usually required.

CASING DESIGN CRITERIA

MANAGEMENT'S GUIDELINES

Satisfying management's guidelines is one step in the process of choosing casing. Their guidelines fit into the scheme of design after the objectives and loads are reconciled and before the creation of specific criteria into which the casing must fit. The total process of selecting casing involves these steps.

1. Decide on rational objectives to be attained by the casing.
2. Identify the loads to which the casing will be subjected during its life.
3. Satisfy **management's guidelines**. Management's guidelines will be broad and present a balance between risk and cost.
4. Create specific criteria in the form of equations and charts for the well.
5. Make the computations and draw the charts.
6. Select casing.

Management's guidelines are based on risk and cost analysis. When addressing guidelines it should be kept in mind that the primary objective of design is not to eliminate failures; but to provide an optimum balance between materials costs and risk costs.

It is the designers function to explain to management that his design satisfies their guidelines.

Major guideline topics for management's considerations are

Inspection of casing

Whether to run casing empty or filled

Considerations for running casing through doglegs

Margin of overpull for pulling casing

Casing wear by drilling operations

Run new or used casing

Loss of fluid level within the casing (lost circulation)

Gas column to surface versus gas kick bubble, versus water column for surface burst

Displacement of cement plugs with mud or water

Cement back to the surface

Corrosion protection of the casing

H_2S considerations

Salt zone diaperic shale considerations

Permit buckling or not

Permit yielding or not

Considerations of formation bearing stress versus contained fluid pressures

Permit yielding of the steel in the casing within cemented sections

Selection of casing test pressures after setting the casing

The potential for actually having design loads on the casing and the consequence of failure

CRITERIA AND EQUATIONS FOR CASING DESIGN

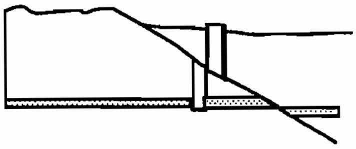
Design criteria are different from management's guidelines in that the criteria are specific to loads and casing strengths. Mathematical equations are derived from criteria, not precepts of risk and cost. Every designer must envision the maximum tensile loads and their backups, maximum burst loads and their backups, and maximum collapse loads and their backups, for every foot over the length of the casing. Then the designer must match those loads with sufficient strength casing. All equations must be in the following format.

BASIC EQUATION

Strength => Design Factor * (Load - Backup)

MAXIMUM LOADS

The governing design load is the maximum which may be reasonably expected. It may or may not be reasonable to expect a fault to slip as shown in the sketch. However, prevalent selected maximum loads for casing design are the following.



1. The maximum burst load for a tubular occurs if all the mud in the hole is displaced by gas.

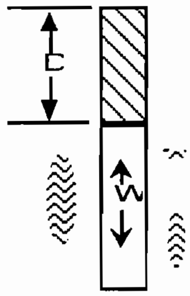
2. The maximum collapse load occurs if the pipe is fully evacuated and

a. the full formation fluid pressure acts on the casing
b. the formation's rock bearing stresses act on the casing.

3. Maximum tensile loads occur during running casing through doglegs or after cementing during stability loadings.

Tensile failures and collapse can be tolerated, but a burst failure, particularly if it occurs at the surface, may be disastrous. A discussion and comparison of various drilling burst load conditions is presented below.

The following criteria and equations are popular in the industry.



TENSION

IN WORD FORMAT

The design tensile load is the weight of the steel in the casing below the depth for which the casing is being designed. The backup is the buoyancy of the casing below the point. Dogleg bending loads are included and are to be computed with Lubinski's modified equation.

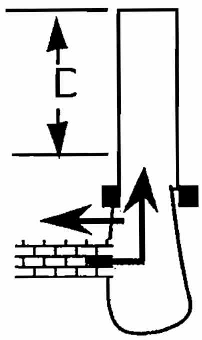
IN EQUATION FORMAT

$$S_t = DF\,[w_b * (TVD - D) + F_{LUB}]$$

S_t	=	tensile strength of the tube or joint; lb
DF	=	design factor of tube or joint; lb
w_b	=	buoyed weight per foot of the casing; lb/ft
TVD	=	total vertical depth of the hole; ft
D	=	design depth; ft
F_{LUB}	=	dogleg bending load with Lubinski's equation; lb

BURST

IN WORD FORMAT



The design burst load is the pressure at any depth placed upon the casing by a column of methane gas which extends from the formation containing gas which will produce the highest pressure at the surface or from the shoe of the casing to the surface.

The gas pressure at the shoe can not exceed the formation's fracture strength at the shoe. The formation which will produce the highest pressure at the shoe must be reasoned or found by comparing all probable formations. In any case their are two basic equations.

The backup load is the pressure of the mud outside the casing at all depths.

IN EQUATION FORMAT

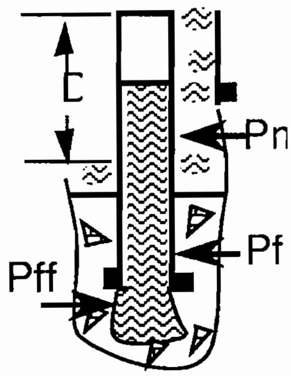
$$S_b = DF * [(P_f - ß * (TVD - D)) - \frac{MW\ D}{19.25}]$$ if formation pressure controls

$$S_b = DF * [(P_{ff} - ß * (TVD - D) - \frac{MW\ D}{19.25}]$$ if fracture strength controls

S_b	=	casing burst strength; psi
DF	=	burst design factor
P_f	=	formation pressure; psi
P_{ff}	=	formation fracture resistance; psi
ß	=	gas gradient; psi/ft
TVD	=	depth of hole; ft
D	=	depth of design; ft
MW	=	mud weight; ppg

COLLAPSE

IN WORD FORMAT



The collapse load is the pressure of the mud outside the casing above the top of the cement and formation fluid pressure below the top of the cement.

For production casing, there is no backup load. For intermediate and surface casing, the casing contains a column of water equivalent to the formation fracture strength at the casing shoe.

IN EQUATION FORMAT

$$S_{c\ bix} = DF * [D * MW - 0]$$ in the evacuated space above the water column

$$S_{c\ bix} = DF * [D * MW - WG * (D - EL)]$$ in the water column

S_c	=	casing collapse strength; psi
bix	=	API method of derating collapse strength for tension
$S_{c\ bix}$	=	derated collapse strength of casing; psi
DF	=	design factor
D	=	depth of design; ft
MW	=	mud weight; ppg
WG	=	water gradient; psi/ft
EL	=	evacuated length; ft

POPULAR DESIGN FACTORS

Design factors for tension loading call for a 1.6 design factor for the pipe body and connection, and a 2.0 design factor for skinny connections. Higher design factors should be used for clearance connections, such as a flush liner connection, because these connections are weaker and prone to failure.

TABLE OF DESIGN FACTORS

CASING	BURST	COLLAPSE	TENSION§	STABILITY	TORSION	VON MISES
CONDUCTOR	1.0	1.0	1.6	1.25	1.5	1.25
SURFACE	1.0	1.0	1.6	1.25	1.5	1.25
INTERMEDIATE	1.0	1.0	1.6	1.25	1.5	1.25
PRODUCTION	1.25	1.0	1.6	1.25	1.5	1.25
LINER	1.25	1.2	1.6	1.25	1.5	1.25
WORK & FISHING	1.25	1.25	1.6	1.25	1.5	1.50

§ Body and joint unless joint skinny; then use 2.0 rather than 1.6.

DRILLING BURST CRITERIA

Design of intermediate casing for burst is controversial because the range in risk and cost can be great and there are many burst criteria. However, they may be placed into four fundamental categories.

1. Gas to surface. This is the most expensive design. It assumes the casing is filled only with gas.

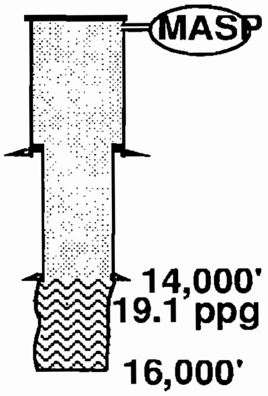
2. Water to surface. This is intermediate in cost. It assumes losses at the casing shoe and the casing is kept full (or partially full) of water by pumping into the casing at the surface.

3. Kick design. This is a lower cost. It assumes taking a gas kick of a specified volume.
4. No control. This permits the least expensive design. It assumes no gas, oil, or water will enter the casing.

The four criteria are compared in the chart for an intermediate casing set at 14000 ft. The external pressure gradient (backup) is 0.468 psi/ft (9 ppg).

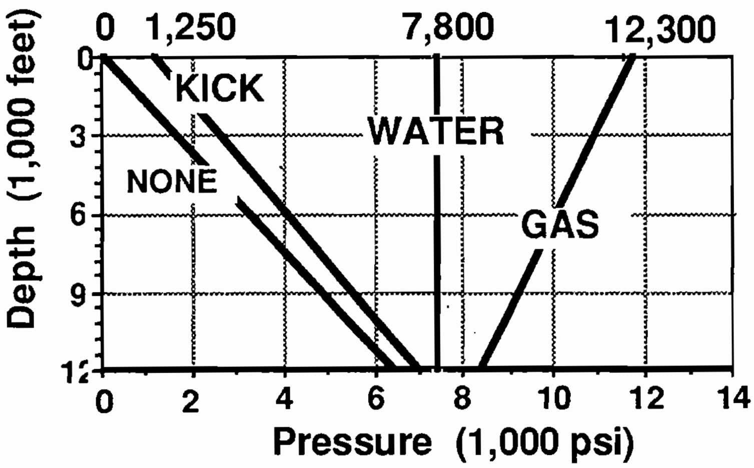
For the kick criterion, a kick sufficient to fracture the shoe is assumed. The Maximum Anticipated Surface Pressure (MASP) for the three cases are

1. Gas column = **12300 psi**
2. Water column = **7800 psi**
3. Kick at surf = **1250 psi**



It should be noted that the three load lines intersect at the deepest casing shoe since all assume fracture at the shoe. All are intended to represent an underground blowout, i.e., the well is flowing at the deepest casing shoe.

Intermediate String Burst Load Lines



The high burst ratings required are difficult to obtain for the large diameter casing required to allow setting multiple liners. Additionally, the gas to surface criterion would require 15000 psi blowout preventers. Even the water to surface burst condition is difficult to achieve.

GAS TO SURFACE

There are two ways "gas to surface" can occur. A gas bubble can be allowed to rise to the surface with the well shut in; or, with a drillpipe kick, the drillpipe parts or is sheared near the surface. An underground blowout will occur if the formations fracture pressure at the shoe is less than the pressure of the gas in the zone from which the gas came; such is usually the case.

WATER TO SURFACE

The assumption of "water to the surface" also may produce an underground blowout. However, it is assumed that preparations can be made to pump water into the well before running out of mud. This does not account for drillpipe failure or pumping equipment failure. The problem of drillpipe kicks can be minimized by running a float. For this case, the pump pressure required is about 7800 psi. Even cementing equipment is not rated for extended service at this pressure; therefore, considerable redundancy is mandatory. If the pumping equipment fails, the casing will burst if not designed for the gas to surface load, leaving a blowout.

KICK

The assumption of a kick also represents an underground blowout if the kick exceeds the available kick tolerance (as is the case here). The shoe is broken down and the permeable gas zone will flow if the fracture pressure at the shoe is below formation pressure in the permeable zone. This does not account for drillpipe failure, pumping equipment failure or running out of mud. A float minimizes the risk from drillpipe kicks. There is usually sufficient pumping equipment redundancy for pumping mud at the low pressures and rates required.

AN OVERVIEW OF CASING SELECTION

Tubular designs are now based on real loads and more realistic tubular strengths and limitations. Design factors and tubular strength derating factors are no longer allowing for the less exacting and arbitrary design. The personal computer is responsible for the new revelation in tubular design. Without a personal it is improbable that a casing can be designed for realistic collapse loading in which buckling and the new API collapse strength equations are applied.

THE "API" DESIGN METHOD

The old way of casing design is out. The old method is known, and incorrectly so, as the API method. ***API does not have a method or a procedure for the design of casing or tubing.*** API has tubular strengths and size tolerances. The old method considers four fictitious loads and the required casing strengths to overcome these loads:

1. The body tensile strength at any point in the casing had to be 1.8 times the hanging weight in air of the casing below that point.

2. The joint tensile strength at any point in the casing had to be 2.0 times the hanging weight in air of the casing below that point.

3. The burst strength of the casing at all points had to be equal to the highest bottom hole pressure created by the mud while drilling.

4. The collapse strength, after derating it for combined loading, of the casing at a selected point had to be 1.125 times the pressure created by the mud at that point. The tensile load used in the combined loading was the buoyed weight of the casing below the point of interest. The combined load derating factor was biaxial and was extracted from von Mises 'Maximum Strain Energy of Distortion Theory at Yield' failure theory.

This set of criteria is not adequate for the design of shallow casings or deep casings. In deep holes casing can not be manufactured of sufficient strengths to overcome the unrealistic loads. In shallow casings the loads considered are incomplete. For example in a real situation the casing must be lowered into the hole one joint at a time and the real load associated with this action is call 'Casing

Running Loads'; however, the above criteria do not contain rules for running casing. The big and important area of 'Buckling' and 'Wellhead Loads' after cementing is also not addressed.

ITERATIVE CASING DESIGN

The new way considers fictitious loads only to initiate an iterative process which produces the final casing design.

The iterative steps are the following:

1. Casing of sufficient strengths are selected to overcome the fictitious loads of
    a. Hanging weight of the casing before cementing
    b. Burst by formation pressures before cementing
    c. Collapse by mud or formation pressures before cementing

2. Adjust casing strengths to account for running loads and dogleg running loads if required.

3. Adjust casing strengths to account for buckling and wellhead loads during and after cementing.

4. If there were changes in steps 2 and 3, then return to step #1 and continue through step #4.

5. If there were no changes, your design is finished.

MINIMUM TUBULAR STRENGTHS

It is recommended to tubular designers to use the minimum strengths as published by the API. It may be noted that the values of tubular strengths as published by the API have no built in safety factors.

API publishes the following strengths for tubulars

1. internal yield (burst)
2. collapse
3. body tension
4. joint tension

TUBULAR FAILURE CRITERIA

It is recommended to tubular designers to use the API procedure for derating casing for collapse. The API procedure incorporates the "Maximum Strain Energy of Distortion Failure Theory at Yield" in a biaxial loading manner. It has been reasonably argued that triaxial equations from the same theory are more valid in theory and less practical in application. To compensate designers frequently use buoyed weight rather than actual loads to initiate their designs. Biaxial loads are loads which act simultaneously in only two perpendicular directions within the wall of the tubular. 'MSEDFTY' predicts failure of tubulars based on the energy needed to change its shape (as opposed to change of volume).

DESIGN FACTORS AND LOADS

A tube design factor must not have a value of less than one (1.0). The best policy is that design tubular strengths must be at least equal to loads. If a design factor can be reasonably assigned a value which is less than one (1.0), then the load needs to be reexamined, because this means that the load is sufficient to fail the tubular. Design factors are often call "ignorance factors", because design factors should be higher as less is known about the loads which will be placed on the tubulars.

FAILURE MODE: ELASTIC, PLASTIC, AND YIELD

Tube failure means that a casing can no longer serve its purpose. In casing design failure means that casing stresses have met or exceeded the API yield point of the steel in the casing. It should be kept in mind that the casing may not show one indication of failure. Yield may occur long before a tube breaks into two pieces, splits, or collapses.

Elastic failure can occur during bending and stability loadings, for example. A large diameter casing may be bent to such a degree in a dogleg that its internal diameter may diminish in size to the point that it does not allow the passage of tools or perhaps a bottom hole assembly. If upon pulling the casing from the dogleg, the casing regains its nominal internal diameter, one could think that the casing had failed, and had failed elastically.

One well known plastic design problem in which the casing is failed if judged by the yield point criterion; but, it will not have failed if failure is thought of as the inability of the casing to continue to serve its function. The problem occurs in steam injection wells and is caused by the cyclic injection of steam. The problem is this: After cementing the casing and locking it in place, injected steam heats the casing to the point that the casing wall is placed in a compressive stress state which is greater than the yield of the steel. Thus, the casing is failed according to the yield point criterion; however, in practice, the casing will continue to serve its purpose and in fact will not be failed. The reason it is not failed is that the casing's shape and dimensions can change because the casing does not have a place in which to deform. However, if the cement is bad and the hole is enlarged then deformation of the casing can cause failure. During a cooling cycle, steam is not being injected, the casing can cool and put itself into tension and under those conditions may be capable of pulling itself into two pieces or more.

TRIAXIAL VS. BIAXIAL TUBE DESIGN

From time to time interest in triaxial stress design of tubulars in the oil patch surfaces. The API supports biaxial stress design in their bulletin 5C3, page 7, February 1, 1985, only in regard to the affect of axial loading on collapse. The usual concern is that triaxial design should be more accurate; and therefore, biaxial must contain some inherent error. Designers have used buoyed weight rather than tension in tubular design to compensate for the 'known' error in biaxial design.

As indicated by its name, triaxial stress design assumes that every cube of steel has three stresses acting on its surfaces. These are axial, tangential, and radial. Biaxial designs set the radial stress to zero, because it is usually the smallest. Other than this assumption, biaxial and triaxial equations are identical.

API BIAXIAL EQUATION DERIVATION

von Mises's failure theory is the beginning point for deriving the biaxial equation. It assumes that all stresses placed on a volume of steel within the wall of a tube consumes part of the strength of the steel. His equation is the following

$$2\ YP^2 = (S_a - S_r)^2 + (S_r - S_t)^2 + (S_a - S_t)^2$$

YP	=	API yield point of the steel in the tube; psi
S_a	=	axial stress, psi
S_r	=	radial stress, psi
S_t	=	tangential stress; psi

The axial stress is either the load or design load divided by the minimum corss-sectional area of the wall of the tube. The radial stress is set to zero. The resulting equation is then solved for the tangential stress, S_t. API has chosen to name the tangential stress, 'axial stress equivalent grade' and has assigned it the symbol "YP_a'. In reality, this is nothing more than the remaining strength of the wall of the tube for resisting collapse pressures.

It may be easily determined with von Mises' equation that negative axial loads (compression loads) will increase the collapse resistance of tubes and positive axial loads (tension loads) will increase the internal yield resistance.

TRIAXIAL EQUATION DERIVATION

The radial and tangential stresses are defined with Lame's equation.

Radial stress $$S_r = P_i(\frac{A_i}{A})(1-(\frac{D}{b})^2) - P_o(\frac{A_o}{A})(1-(\frac{d}{b})^2)$$

Tangential stress $$S_t = P_i(\frac{A_i}{A})(1+(\frac{D}{b})^2) - P_o(\frac{A_o}{A})(1+(\frac{d}{b})^2)$$

S_r	=	radial stress, psi
S_t	=	tangential stress, psi
A_i	=	pi d^2 , in^2
Ao	=	pi D^2 , in^2
A	=	$A_o - A_i$

b	=	diameter of stress investigation; in
d	=	internal diameter of the tube; in
D	=	external diameter of the tube; in
P_i	=	internal pressure; psi
P_e	=	external pressure; psi

It may be shown with Lame's tangential stress equation that setting the parameter 'b' equal to 'd' gives the maximum value of tangential stress for all pressure conditions. Therefore, 'b' is set equal to 'd'.

Following conventional schemes and for the above reason, Lame's radial stress equation reduces to the following, rather simple, equation:

$$S_r = - P_i$$

Substitution of ' - Pi' into Von Mises' equation and then solving for St, in a manner similar to the biaxial case, gives the triaxial 'axial stress equivalent grade'. Thereafter, if one so desires, collapse strengths of tubes may be derated with triaxial stress considerations rather than with API's biaxial considerations.

API's BIAXIAL STRESS EQUATION

$$YP_a = YP \left\{ \left[1 - \frac{3}{4} \left(\frac{S_a}{YP}\right)^2 \right]^{\frac{1}{2}} - \frac{1}{2} \frac{S_a}{YP} \right\}$$

TRIAXIAL STRESS EQUATION

$$YP_a = YP \left\{ \left[1 - \frac{3}{4} \left(\frac{S_a - Pi}{YP}\right)^2 \right]^{\frac{1}{2}} - \frac{1}{2} \frac{S_a}{YP} \right\}$$

EXAMPLE TRIAXIAL & BIAXIAL PROBLEM

7" 29 ppf L80 casing is run to 15,000' in 15 ppg mud and cemented with 15.2 ppg cement from 15,000' to 10,000'. While the mud is displaced with completion fluid, the temperature of the casing above the cement drops by an average of 126 Deg F. What is the minimum weight completion fluid which must reside within the 7" casing to prevent it from collapsing at a depth of 10,000'?

This problem is most easily solved by a trial and error technique in which several weights of completion fluids are estimated. Begin with a 3.31 ppg completion fluid.

The axial stress in the casing at 10,000' is found with a free body of the bottom 5,000' of the casing. Assume a check valve is at the shoe of the casing and that the cement is displaced with 15.0 ppg mud. The forces acting on the free body are the following:

Weight of steel $= 29 * 5000$ **= 145,000 lb**

External pressure area force

$= .052 * (10000 * 15 + 5000 * 15.2) * .7854 * 7^2$ **= 452,271 lb**

Internal pressure force area

$= .052 * (15000 *15) * .784 * 6.184^2$ **= 351,411 lb**

The resultant force at 10,000' because of the cementation

$= 145{,}000 - 452{,}271 + 351{,}411$ **= 44,140 lb** (real tension)

Note that the buoyed weight of the casing at 10,000' is the following (and is not equal in value or meaning to the real tension)

$= 10{,}000 * (29 + .0408 * 6.184^2 - .0408 * 7^2)$

= **285,611 lb** (buoyed weight)

The stability load caused by the temperature drop after displacing the mud

$= 60 * 29 * 126$ **= 219,240 lb**

The stability load caused by changing the 15.0 ppg mud to the 3.31 ppg completion fluid

$= .0122 * 6.184^2 * 10{,}000 * (3.31 - 15)$ **= 54,540 lb**

Total tension after cementation and stability considerations

= 208,840 lbs

The cross-sectional area of the casing

$= .7854 * (7^2 - 6.184^2)$ **= 8.449 sq in**

The axial stress at 10,000'

= 208,840/8.449 **= 24,716 psi**

The external pressure at the depth of 10,000'

$= .052 * 15.0 * 10{,}000$ **= 7,800 psi**

The internal pressure at the depth of 10,000'

$= .052 * 3.31 * 10{,}000$ **= 1721 psi**

BIAXIAL

Biaxial axial stress equivalent grade (See API BULLETIN 5C3)

= 64,275 psi

The biaxial collapse resistance of the casing which is also the allowable differential pressure across the wall of the casing at 10,000'

= 6,077 psi

Thus, an internal column pressure must be precisely equal to the difference between the external mud pressure at 10,000' and the collapse resistance of the casing at 10,000'. The density of the fluid to achieve this pressure

$$= \frac{7800 - 6077}{.052 * 10000}$$ **= 3.31 ppg**

This is the correct density value, because this is also the estimated value. A trial and error procedure of more steps is normally required.

TRIAXIAL

Begin by estimating that a 3.26 ppg completion fluid will be required and complete the above computations. It will be found that the axial stress is 24,609 psi and that the internal pressure is 1,695 psi.

= 65,171 psi

Compute the triaxial axial stress equivalent grade

The triaxial collapse resistance of the casing which is also the allowable differential pressure across the wall of the casing at 10,000'

= 6,107 psi

The required density of internal completion fluid

$$= \frac{7800 - 6107}{.052 * 10000}$$ **= 3.26 ppg**

This is the correct density value, because this is also the estimated value. A trial and error procedure of more steps is normally required.

Note that the percent difference between the biaxial and triaxial strengths of casing relative to the biaxial strength expressed in internal completion fluid densities as set forth in this problem is

$$\% \text{ difference} = \frac{3.31 - 3.26}{3.31} * 100$$ **= 1.5%**

REAL GAS SURFACE PRESSURE

Estimating gas surface pressures to which tubulars are often subjected is critical in tubular design.

It is popular to design the required burst strengths of casing, for example, with ideal gas law column equation.

A better estimate may be made with the procedure described here.

The real gas law is

$$\rho = \frac{P}{Z \frac{R}{MW} T}$$

ρ = pressure gradient of the gas; psi/ft

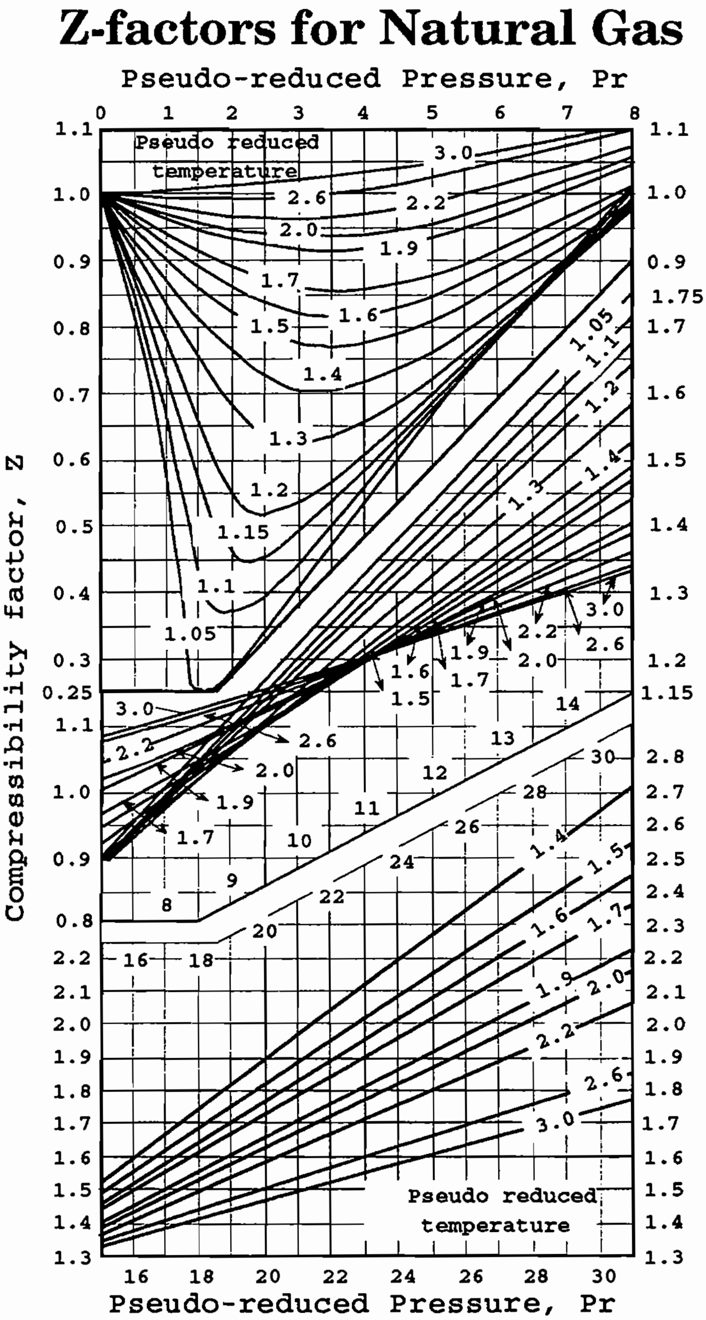
P = pressure of the gas; psia

Z = gas deviation factor

R = gas constant = 1544 ft-lb/mole-R

MW = molecular weight of the gas (methane = 16.04 lb/lb-mole)

T = temperature of the gas; Rankine



Noting that Z is a function of P and T and the fact that the pressure and temperature changes are small in a static gas column suggests that if Z is computed with the known bottom hole temperature and pressure and is thereafter constant, then a reasonable surface pressure for casing design may computed.

Thus, the gas gradient for a methane column which will give the highest surface pressure of all hydrocarbon gases becomes

$$\rho = \frac{P_{BH}}{Z_{P_{BH}T_{BH}} \frac{1544}{16} T_{BH}}$$

Finally, the surface gauge pressure is

$$P_{surf} = P_{BH} - \rho * DEPTH - P_{atmospheric}$$

EXAMPLE GAS COLUMN PRESSURE

If methane gas is drilled at 14,000 feet and the expected bottom hole gas pressure is 9,000 psig at a temperature of 205°F, what is the expected static shut-in pressure?

The absolute bottom hole pressure and temperatures are

$P_{BH} = 9000 + 14.7$ **= 9014.7 psia**

$T_{BH} = 460 + 205$ **= 665 R**

The critical pressure and critical temperature of methane are

$P_c = 673$ psia $\quad T_c = 343$ R

The reduced pressure and temperature of methane are

$P_r = \frac{9014.7}{673}$ **= 13.4**

$T_r = \frac{665}{343}$ **= 1.94**

The Z factor is taken from the chart where a value of 1.32 was read.

$\rho = \frac{9014.7}{1.32 \frac{1544}{16} 665}$ **= 0.1064 psi/ft**

The surface pressure is

$$P_{surf} = P_{BH} - \rho * DEPTH - P_{atmospheric}$$

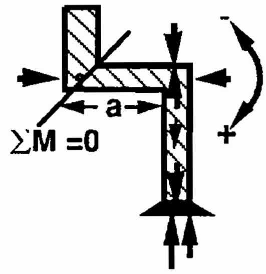
$P_{surf} = 9014.7 - 0.1064 * 14000 - 14.7$ **= 7,510 psig**

FUNDAMENTALS OF TUBULARS

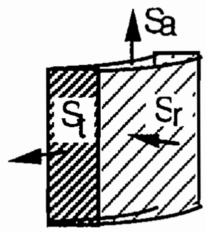
DEFINITIONS AND CONCEPTS

The following is a review of only that part of mechanics which is required for a full understanding of the thoughts put forth in this manual. The section begins with meanings of terms.

1. **Neutral points**: There are five named neutral points.

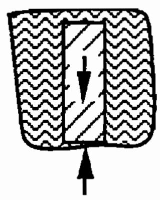


   a. Neutral point of bending (NPB) is the location within a tube at which the sum of the moments are equal to zero. The tube is not bent above this point.

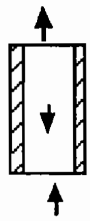


   b. Neutral point of distortion (NPD) is the location within a tube at which one-half the sum of the radial and tangential stresses are equal to the axial stress if mud weight inside and out are equal and no surface pressure. Von Mises stress has a value of zero. The tube at this point is subject to dilatation but not distortion.

$$S_a = \frac{S_t + S_r}{2}$$



   c. Neutral point of buoyancy (NPB) is the location within a tube at which the buoyed weight of the tube hanging below the location is equal to an applied force at its bottom end.

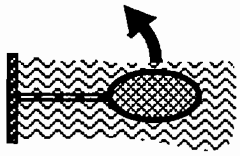


   d. Neutral point of tension and compression is the location within a tube at which the sum of the axial forces are equal to zero.

   e. Neutral point of pressure area is the location within a tube at which the sum of the pressure area force acting on its ends and the weight of the tube are equal to axial tension at the location.

It may be shown that the neutral points 'a', 'b', and 'c', are exactly equal and that points 'd' and 'e' are exactly equal. The thoughts conveyed by 'a', b' and 'd' are most useful in tubular design.

2. **Buoyancy**: **Buoyancy is not a force.** There are three recognized percepts of buoyancy and all three have their place.

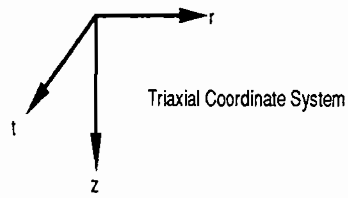


   a. Buoyancy is the weight of displaced fluid. This precept solves most problems but has limitations.

   b. Buoyancy is the difference between fluid pressure area forces acting on the top and bottom of a tube. This precepts also has limitations.

   c. Buoyancy is the sum of the moments acting on a tube or any portion of a tube. This precept is most <u>powerful</u>.

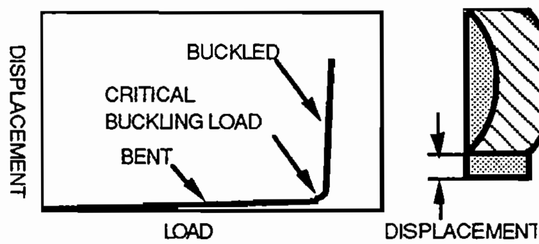
3. **Triaxial** and biaxial loads and coordinate systems:



   a. Triaxial loads are loads in an orthogonal coordinate system in which the loads are axial, tangential, and radial.

   b. Biaxial loads are loads in an orthogonal coordinate system in which the loads are axial and tangential. The radial load is neglected (omitted).

4. **Buckling**: The precept is that buckling is severe bending; however, buckling in and of itself is not necessarily bad.



   a. Buckled is a bent condition in which bending increases with little increase in the load which caused the bending. The steel in the tube may not be yielded. The load exceeds the critical buckling load.

   b. Bent is a condition in which the bending increases proportionally with load.

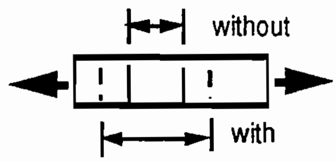
5. **Bending**: There are two recognized equations for computing axial stresses derived from bending of tubes within curved holes (doglegs).

   a. Lubinski's equation is used if the tube has connections which prohibit the wall of the tube from contacting the wall of the hole along the tube's body; i.e., only the connections touch the wall of the hole.

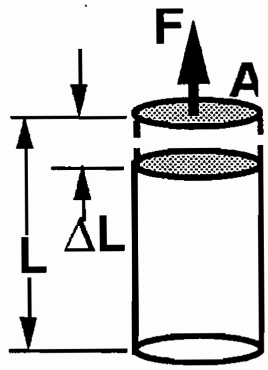
b. The beam equation is used if all of the tube contacts the wall of the hole throughout the dogleg; i.e., the connections do not provide standoff.

c. Equivalent bending load is the bending stress as computed with either of the above equations multiplied by the cross-sectional area of the steel in the wall of the tube. This was invented because designers had rather work with loads than stresses.

6. **Stress and Strain:**



Force is an entity which causes mass to accelerate. There are only three real forces, not including nuclear physics, contact, mass attraction, and magnetism and electrical. Examples of contact forces are a liquid pushing on the end of a tube, a drillpipe pulling on a bottom hole assembly, and gas pushing on the wall of a tube. Examples of mass attraction is the force holding a man on the surface of the earth and the force pulling a tube down a drill hole. A fishing magnet holding a piece of steel is an example of magnetism. **Note that buoyancy is not a force. All real forces are found with Newton's laws of motion. These are $\sum M = 0$ and $\sum F = 0$ if a body is static.**

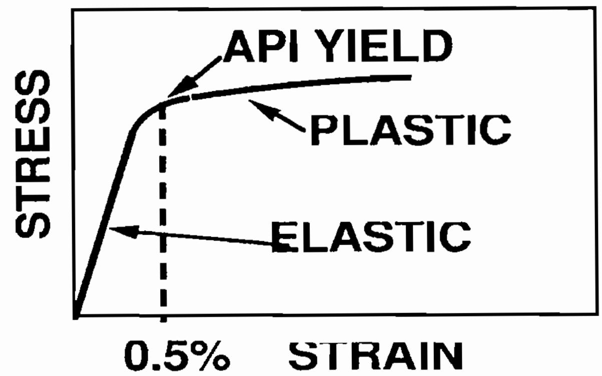


a. Stress is a force divided by an area perpendicular to the force. It acts within solids and it has a direction. By convention, a negative stress is usually associated with compression.

b. Strain is the change in the value of a dimension divided the original value of the dimension. Two common strains in tubes are the change in its length divided by its original length and the change in its circumference divided by its original circumference. A third is the change in wall thickness.

c. Yield occurs if the stress is equal to or exceeds the yield point of the steel.

d. Pressure is exerted by a fluid. It acts in all directions equally in magnitude. A pressure must be positive. Differences in pressures may be negative.



e. von Mises stress determines if steel is yielded and is computed with von Mises equation.

f. Axial stress is real tension divided by the cross-sectional area of the tube.

g. Radial stress is computed with Lame's equation.

h. Tangential stress is computed with Lame's equation.

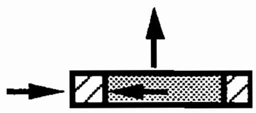
7. **Loads and Tension:**

a. Tension causes two marks which were previously drawn on a tube to separate; i.e. become farther apart.

b. Real tension is found with Newton's law of motion (F = M * a) and free bodies.

c. A free body is sketch of a portion of a tube with all of the loads in a chosen direction (usually vertical or horizontal) drawn on the sketch.

d. Effective tension is the real tension at a location less the internal pressure at that location multiplied by the internal area of the tube plus the external pressure at that location multiplied by the outside area of the tube. It, within von Mises equation, controls yield of tubes.



e. Load is a generic word which could mean force, tension, compression, bending, pressure, or weight.

f. Strength is a generic word which means a tubes resistance to a load.

g. Buoyed weight at a location is the weight of a tube and its contents hanging below a location less the buoyancy of the that section of tube and its contents hanging below the location.

h. Weight in mud is the buoyed weight.

i. Air weight is the 'nominal API' weight and is as listed in the API tables.

8. **End Conditions**

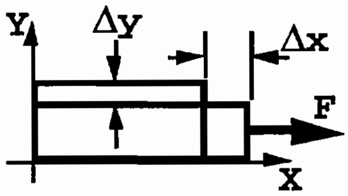
a. Fixed end means that the implied end of a tube can not move in any direction and especially vertically.

b. Free end means that the implied end of a tube can move vertically.

c. Moments at the ends of tubes are not considered in this manual.

d. Lateral displacements at the ends of tubes are not considered in this manual.

9. **Poisson's ratio** is the negative ratio of the contraction of steel perpendicular to the load divided by the extension of the steel in the direction of the load.



10. **Young's modulus** is the slope of the stress-strain plot within the elastic region.

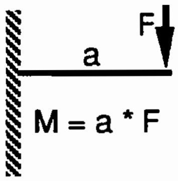
11. **Elastic deformation** is a deformation which results from a stress which is less than the yield of the steel.

12. **Plastic deformation** is a deformation which results from a stress which exceeds the yield of the steel.

13. The **change** in a variable is the last value less the previous value of the variable.

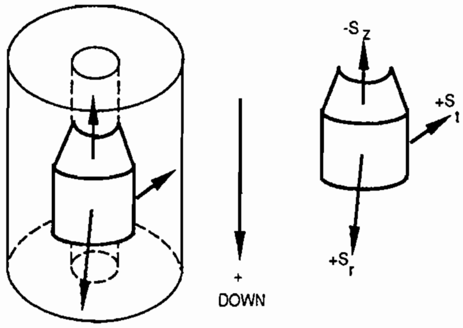
14. **Pressure-Area force** is the force which results from a pressure acting on an area. Its direction is normal to the area and toward the body which has the exposed area.

15. A **moment** is the product of a force and a lever arm.



STRESS ANALYSIS

In order to perform a stress analysis of casing or tubing, we need to understand how the stresses and displacements in a tube are affected by internal and external pressure, axial forces and temperature.



In particular we need to know how stresses and displacements change as we change internal or external pressure, temperature or apply an axial force to the end.[1]

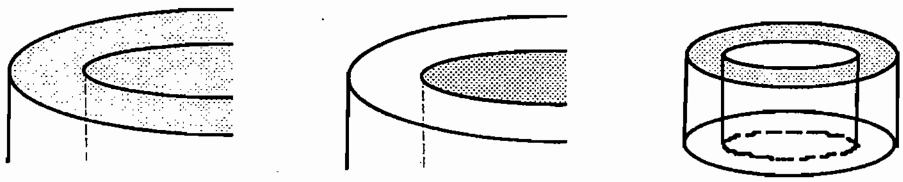
Areas

External area $$A_o = \frac{\pi}{4} D^2 \qquad (1)$$

Internal area $$A_i = \frac{\pi}{4} d^2 \qquad (2)$$

Area of steel $$A = A_o - A_i$$

$$A = \frac{\pi}{4} (D^2 - d^2) \qquad (3)$$



A_o A_i A

If the outside diameter, D, and the inside diameter, d, are in inches, then the area is in square inches (in^2).

Moments of Inertia

Second moment of the area $$I = \frac{\pi}{64} (D^4 - d^4) \qquad (4)$$

Polar moment of inertia $$J = 2\,I \qquad (5)$$

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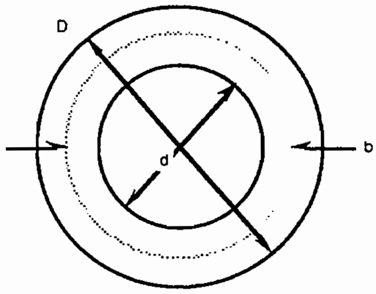
[1]Note: Triaxial Coordinates

I is used in bending equations and J is used in torsion equations.



Stresses

Lame's equations for stresses in a tube, given an internal pressure, P_i, and an external pressure, P_o, are

Radial stress $$S_r = P_i(\frac{A_i}{A})(1-(\frac{D}{b})^2) - P_o(\frac{A_o}{A})(1-(\frac{d}{b})^2) \quad (6)$$

Tangential stress $$S_t = P_i(\frac{A_i}{A})(1+(\frac{D}{b})^2) - P_o(\frac{A_o}{A})(1+(\frac{d}{b})^2) \quad (7)$$

The variable 'b' ranges from d at the inside wall to D at the outside wall.

It is useful to note that the sum

$$S_r + S_t = \text{constant} = \frac{2}{A}(P_iA_i - P_oA_o) \quad (8)$$

This fact will be used in determining the axial stress S_z or the axial strain e_z. The stresses S_r and S_t do not depend on S_z.

$$S_z = \frac{T_{real}}{A} \quad (9)$$

Axial stress

If the tension T is in pounds and the area A is in in^2, then S_z is in psi.

Bending stress

The bending equation developed by Lubinski is

$$F_{LUB} = \frac{3385*D*C*\sqrt{T_{eff}}*\sqrt{\frac{D^2-d^2}{D^2+d^2}}}{\tanh\left[0.2*\sqrt{\frac{T_{eff}}{D^4-d^4}}\right]} \quad (10)$$

$$\tanh(x) = \frac{e^{2x}-1}{e^{2x}+1} \quad (11)$$

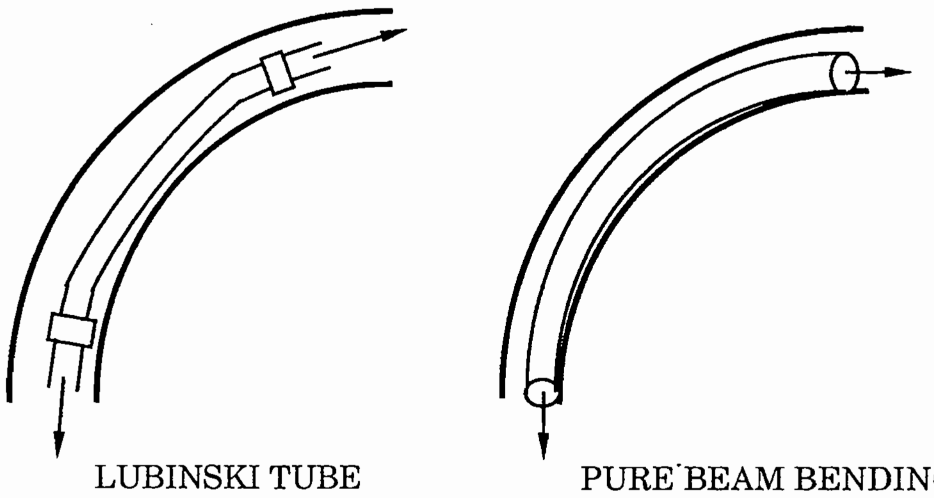
C = tube curvature; = °/ foot

The pure beam bending equation is

$$F_{BEAM} = 17{,}135 * D * C * (D^2\text{-}d^2) \tag{12}$$

The bending stress is

$$S_b = \frac{F_{LUB}}{A} \text{ or } \frac{F_{BEAM}}{A} \quad \text{which ever is larger} \tag{13}$$



Hooke's law for stress-strain relations

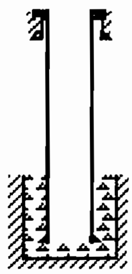
Radial strain $$e_r = \frac{S_r - \mu(S_t + S_z)}{E} + ß\,(\Delta t) \tag{14}$$

Tangential strain $$e_t = \frac{S_t - \mu(S_z + S_r)}{E} + ß\,(\Delta t) \tag{15}$$

Axial strain $$e_z = \frac{S_z - \mu(S_r + S_t)}{E} + ß\,(\Delta t) \tag{16}$$

E	= Young's modulus = 30 x 10^6 psi for steel
μ	= Poisson's ratio = 0.3 for steel (dimensionless)
ß	= coefficient of thermal expansion 6.9 x 10^{-6} °F^{-1} for steel
Δt	= temperature increase in °F.

The equations also apply to changes in strain caused by changes in stress. Changes in stress ΔS_r, ΔS_t, ΔS_z and temperature Δt, give changes in strain Δe. Equation (16) with equations (8) and (9) gives the form that is needed to calculate length changes of tubes caused by tension, pressure, and temperature.



$$\Delta e_z = \frac{\Delta T - 2\mu\,(\Delta P_i\,A_i - \Delta P_o\,A_o)}{AE} + ß(\Delta t) \qquad (17)$$

The change in length of a tube of length L caused by changes in tension, pressure and temperature, is

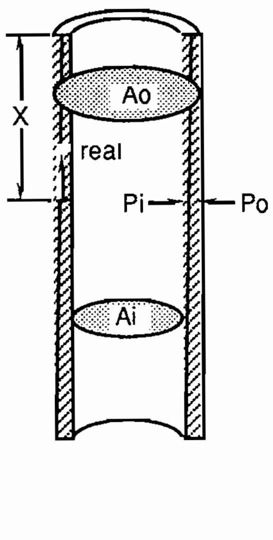
$$\Delta L = L(\Delta e_z)_{avg} \qquad (18)$$

where $(\Delta e_z)_{avg}$ is the average value of Δe_z over the length L. For an axially restrained tube (e.g., cemented casing), Δe_z=0 and we may solve for ΔT from Equation (17)

$$\Delta T = 2\mu\,(\Delta P_i\,A_i - \Delta P_o . A_o) - A\,E\,ß\,(\Delta t). \qquad (19)$$

Equations (17), (18) and (19) are all that are needed to solve problems involving straight tubes.

EFFECTIVE TENSION



$$T_{eff_x} = T_{real_x} - (P_iA_i)_x + (P_oA_o)_x \qquad (20)$$

= "effective tension" at one single depth, 'x'.

Effective tension is the sum of the moments at a depth in a tube. It is called tension only for convenience. In the below sketch and development, the lever arm, a, is assigned a value of unity for convenience, and because its true value can not be known or computed. Thus, effective tension is the force component of a moment. T_{real} is ascertained with a free body and the setting of the sum of forces acting on the free body to equal zero.
Whereas, T_{eff} is ascertained with a free body and the setting of the sum of the moments acting on the free body equal to zero and then setting the lever arm equal to unity.

With the sketches T_{real} is

$$-T_{real} + W + P_{bi}A_i - P_{bo}A_o - WOB = 0$$

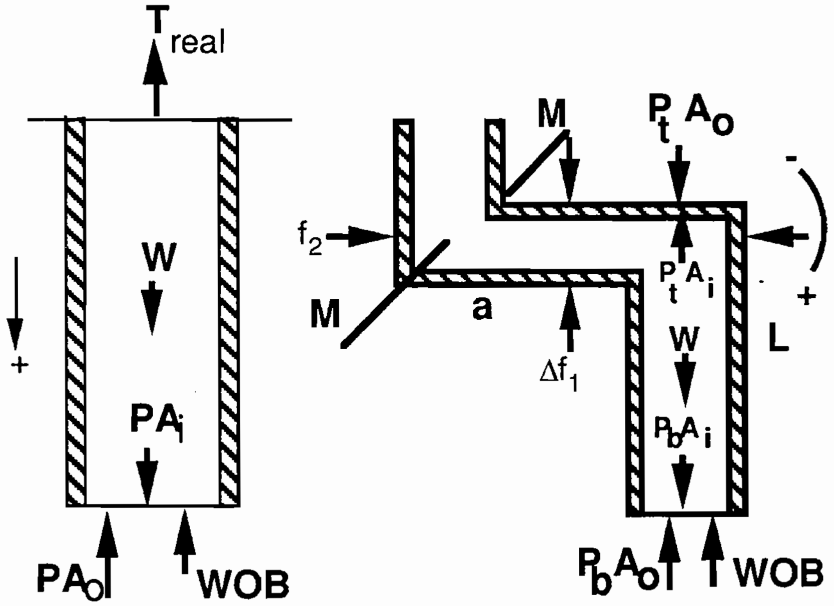
and also with the sketches M is

$$M = + W\,a + [P_{bi}\,A_i - P_{bo}\,A_o\,]\,a + [P_{to}\,A_i - P_{ti}\,A_o]\,a - WOB\,a$$

Considering Δf_1 to be negligible, setting a = 1.0, and T_{eff} = M, then

$$T_{eff} = T_{real} - P_{ti}A_i + P_{to}A_o$$

which is the effective tension at the top of the section. Note that tensions, pressures, and areas must be computed at one single depth. For example, 8,984 feet could be the chosen depth.



BUOYED WEIGHT

Note that if w_i is the weight of the fluid inside the pipe per unit length, w_o is the weight of the fluid displaced by the pipe, and w_s is the weight of the steel per unit length in the wall of the pipe, then the effective weight of the pipe is

$$w_{eff} = w_s + w_i - w_o = \text{"effective weight"} \tag{21a}$$

$$w_{eff} = w_s + 0.0408 * (\,MW_i * d^2 - MW_o * D^2) \tag{21b}$$

All variable in oil field units: lb/ft, lb/gal, inches.

This is also called "buoyant weight" or "weight in mud", etc.

The buoyed weight of pipe at a specified depth is the buoyed weight per unit length of the pipe multiplied by the length of pipe below the depth.

YIELDING OF AN IDEAL TUBE

The calculated stresses in a tube is compared with some maximum allowable stress in order to determine whether or not the tube is adequate to carry the loads. It is usually desirable to avoid yielding the pipe. In the following, the formulation for an ideal tube is developed. Note that "real" pipe is not round and the wall thickness is not uniform.

The yield stress YP is the maximum tensile stress which may be applied without causing permanent deformation of the steel. The pipe body yield

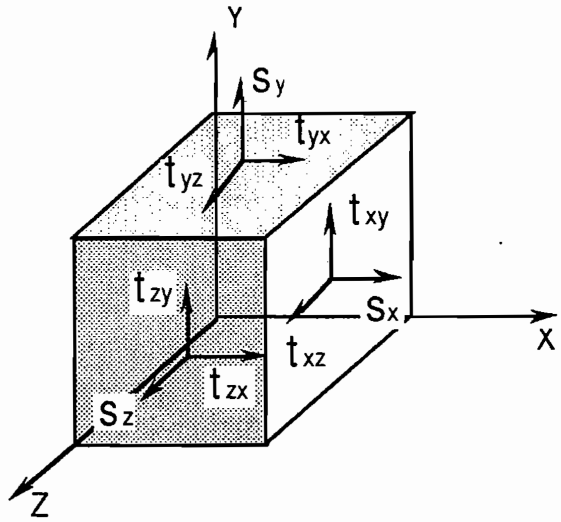
$$T_{yield} = YP * A_s \qquad (22)$$

is the maximum tension that the tube can withstand. If YP is in psi and A_s is in square inches, then T_y is in pounds. The pipe body yield strength listed in API Bulletin 5C2 and most catalogs is in 1000's of pounds.

The yield behavior caused by combined loads, such as tension and pressure, is a bit more complicated. The yield condition attributed to **von Mises** is

$$(S_r - S_t)^2 + (S_r - S_z)^2 + (S_t - S_z)^2 + 6\,(T_r{}^2 + T_t{}^2 + T_z{}^2) = 2\,YP^2 \qquad (23)$$

S_r, S_t, S_z are the normal stresses
T_r, T_t, T_z are the shear stresses



Shear stresses arise from torsion and shearing forces on the pipe. The torsional shear stress is significant for drillpipe but is usually neglected for casing. The other shear stresses are often negligible. For casing under tension and pressure loads, the yield condition becomes

$$(S_r - S_t)^2 + (S_r - S_z)^2 + (S_t - S_z)^2 = 2\,YP^2 \qquad (24)$$

A major assumption in von Mises theory of yield behavior is that a hydrostatic pressure, such as the pressure acting on a bar submerged in a fluid, cannot cause yielding. The left side of Equation (23) is a mean square measure of shear stress. The average stress, $(S_r + S_t + S_z)/3$, does not affect the value of the left side of Equation (23).

Define the "von Mises stress" or "equivalent stress" as

$$2\,V^2 = (S_r - S_t)^2 + (S_r - S_z)^2 + (S_t - S_a)^2 + 6\,(T_r{}^2 + T_t{}^2 + T_z{}^2) \qquad (25)$$

The yield condition is then

V < YP the material is elastic

V = YP the material is yielded.

For a tube under tension and pressure loads, the expression for the von Mises stress V can be written as follows. Substituting the expressions for S_r, S_t, and S_z from Equations (6), (7) and (9) into Equation (25), we obtain (following a little algebraic manipulation)

$$V^2 = 3\left(\frac{A_o}{A}\right)^2 \Delta P^2 + \left(\frac{T_{eff}}{A}\right)^2 \qquad (26)$$

$$\Delta P = P_i - P_o$$

$$T_{eff} = T_{real} - P_iA_i + P_oA_o$$

The Von Mises stress is a maximum at the inside wall of a tube. Equation (26) has been solved to give the von Mises stress at the inside wall of the tube. Incidentally, this form of the equation is only good for the inside wall of the tube. A tube with pressure and tension loads only always yields at the inside wall, both for burst and collapse loads. Further, it is the effective tension T_{eff} which governs the tensile yield behavior of a tube in the absence of shear stress.

Bending stresses may be accommodated in Equation (26) with the principle of superposition. The bending stress S_b is in the axial (z) direction and adds to the axial stress S_z. Equation (26) becomes

$$V^2 = 3\left(\frac{A_o}{A}\right)^2 \Delta P^2 + \left(\frac{T_{eff}}{A} \pm S_b\right)^2 \qquad (27)$$

The ± in Equation (27) appears because the bending stress is positive on one side of the pipe and negative on the other. To obtain the maximum von Mises stress 'V' choose the maximum value of Equation (27) by choosing + or - as appropriate.

Since the expression for the von Mises stress is the value at the inner wall, substitute d for D in Equation (12) to obtain the bending stress. This is the usual practice for tubing stress analysis. However, for sufficiently high bending stresses (or low pressure loads), the maximum stress occurs at the outer wall; therefore, the von Mises stress at the outer wall must be calculated as well.

The usual practice for casing stress analysis is to compute the bending stress at the outer wall as in Equation (12) and use that value in Equation (27); this is conservative.

A useful form of the Von Mises equation is

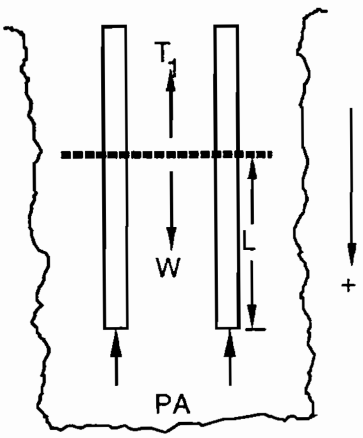
$$V^2 = 3 * \left(\frac{D^2}{D^2-d^2}\right)^2 * (P_i - P_o)^2 + \left[\frac{T_{eff}}{A} \pm \frac{4310*D*C*\sqrt{T_{eff}}}{\sqrt{D^4-d^4} * \tanh\left[0.2\sqrt{\frac{T_{eff}}{D^4-d^4}}\right]}\right]^2 \qquad (28$$

$$\frac{T_{eff}}{A} = \frac{T_{real}}{\frac{\pi}{4}(D^2-d^2)} - P_i\left[\frac{d^2}{D^2-d^2}\right] + P_o\left[\frac{D^2}{D^2-d^2}\right] \qquad (29)$$

FREE BODIES

Free bodies are constructed for the purpose of ascertaining real tension, "T_{real}." There are 4 cases which often occur

1. Find the real tension at any depth and let T_{real} = T

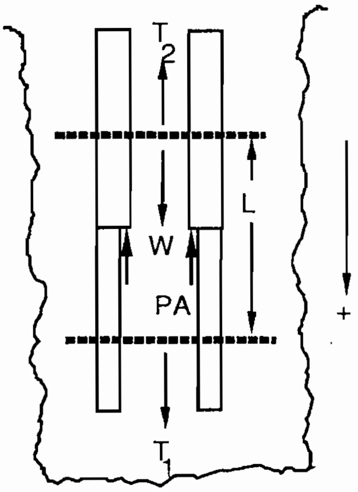


$$\Sigma Fv = 0$$

$$-T_1 + W*L - PA = 0$$

$$T_1 = W*L - PA$$

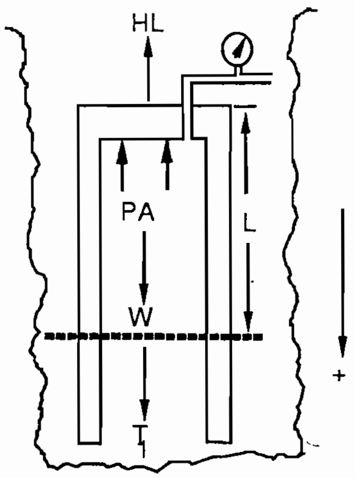
2. Find the real tension in a tube if the real tension at a second location is known ($T_{real} = T$)



$$\Sigma Fv = 0$$
$$-T_2 + W*L + T_1 - PA = 0$$
$$T_2 = T_1 + W*L - PA$$

3. Find the real tension with internal surface pressure ($T_{real} = T$):

HL (Hook Load) = Buoyant weight of steel in all of the tube (w_{eff} * total length) **OR** Weight Indicator less block & tackle.



$$\Sigma Fv = 0$$
$$+T_1 - HL - PA + W*L = 0$$
$$T_1 = HL + PA - W*L$$

4. Find the Hook Load, "HL"

Case A: stuck tube, resting on bottom of hole or ledge, or supported by wall friction.

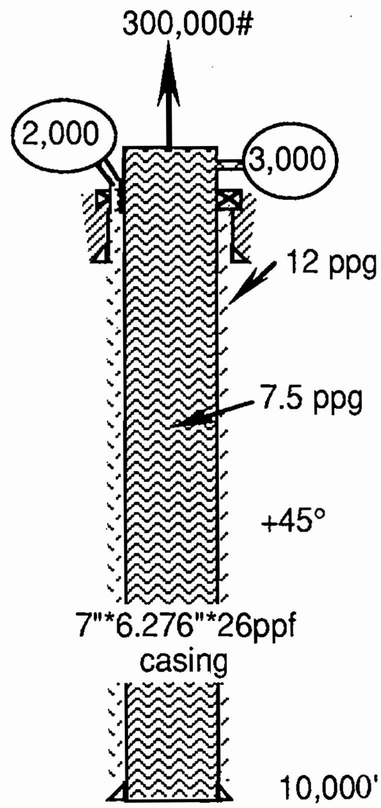
HL = Weight Indicator - Block & Tackle

Case B: freely suspended pipe.

$HL = w_{eff} * L$

EXAMPLES OF THE FUNDAMENTALS OF TUBES

The examples which follow assume the following data: 10,000' of 7" * 6.276" * 26 PPF * L80 tube with 3000 psi internal pipe surface pressure, 2000 psi external pipe surface pressure, 7.5 ppg internal pipe mud weight, 12.0 ppg external pipe mud weight, "b" at midpoint of wall, real tension equal 300,000 lbs at the surface, tube curvature is 0.05°/ft, and the temperature rises +45°F.



In terms of our variables this data would be

L	= 10,000 ft	D	= 7"
d	= 6.276"		
w	= 26 ppf	P_i	= 3000 psi
P_o	= 2000 psi		
T	= 300,000 lbs	C	= 0.05°/ft
Δt	= +45°F		
MW_i	= 7.5 ppg	MW_o	= 12.0 ppg
Y	= 80,000 psi		

Also from the given data: $b = \frac{D+d}{2} = \frac{7+6.276}{2}$ **= 6.638 in**

External area $A_o = \frac{\pi}{4} D^2 = \frac{\pi}{4}(7)^2$ **= 38.485 sq in** (1)

Internal area $A_i = \frac{\pi}{4} d^2 = \frac{\pi}{4}(6.276)^2$ **= 30.935 sq in** (2)

Area of steel $A = \frac{\pi}{4}(D^2 - d^2)$

$= \frac{\pi}{4}(7^2 - 6.276^2)$ **= 7.549 sq in** (3)

Second moment of the area $I = \frac{\pi}{64}(D^4 - d^4)$

$= \frac{\pi}{64}(7^4 - 6.276^4)$ **= 41.703 in^4** (4)

Polar moment of inertia $J = 2\,I$

$= 2 * 41.703$ **= 83.406 in^4** (5)

Radial stress at the surface

$$S_r = P_i(\frac{A_i}{A})(1-(\frac{D}{b})^2) - P_o(\frac{A_o}{A})(1-(\frac{d}{b})^2)$$

$$S_r = 3000(\frac{30.935}{7.549})(1-(\frac{7}{6.638})^2) - 2000(\frac{38.485}{7.549})(1-(\frac{6.276}{6.638})^2) = \mathbf{-2{,}459\ psi} \quad (6)$$

Tangential stress at the surface

$$S_t = P_i(\frac{A_i}{A})(1+(\frac{D}{b})^2) - P_o(\frac{A_o}{A})(1+(\frac{d}{b})^2)$$

$$S_t = 3000(\frac{30.935}{7.549})(1+(\frac{7}{6.638})^2) - 2000(\frac{38.485}{7.549})(1+(\frac{6.276}{6.638})^2) = \mathbf{6{,}655\ psi} \quad (7)$$

$$S_r + S_t = \text{constant}$$

$$S_r + S_t = \frac{2}{A}(P_iA_i - P_oA_o)$$

$$S_r + S_t = \frac{2}{A}(3000 * 30.935 - 2000 * 38.485) = \mathbf{4{,}196\ psi} \quad (8)$$

Axial stress at the surface

$$S_z = \frac{T_{real}}{A}$$

$$S_z = \frac{300000}{7.549} = \mathbf{39{,}740\ psi} \quad (9)$$

$$T_{eff} = T_{real} - P_iA_i + P_oA_o$$

$$T_{eff} = 300000 - 3000*30.935 + 2000*38.485 = \mathbf{284{,}163\#} \quad (20)$$

$$F_{LUB} = \frac{3385*D*C*\sqrt{T_{eff}}*\sqrt{\frac{D^2-d^2}{D^2+d^2}}}{\tanh\left[0.2*\sqrt{\frac{T_{eff}}{D^4-d^4}}\right]}$$

$$F_{LUB} = \frac{3385*7*.05*\sqrt{284163}*\sqrt{\frac{7^2-6.276^2}{7^2+6.276^2}}}{\tanh\left[0.2*\sqrt{\frac{284163}{7^4-6.276^4}}\right]} = \mathbf{208{,}520\ lb} \quad (10)$$

Bending stress at the surface

$$S_b = \frac{F_{LUB}}{A} \quad \text{or} \quad \frac{F_{BEAM}}{A} \qquad \text{Whichever is larger}$$

$$S_b = \frac{208520}{7.549} \text{ or } \frac{57645}{7.549}$$

$$S_b = 27{,}625 \text{ psi or } 7{,}636 \text{ psi (choose largest)} \qquad \mathbf{= 27{,}625\ psi} \qquad (13)$$

Radial strain at the surface

$$e_r = \frac{S_r - \mu(S_t + S_z)}{E} + ß\,(\Delta t)$$

$$e_r = \frac{-2459 - .3(6655 + 39740)}{30 \times 10^6} + 6.9 \times 10^{-6}\,(45) \qquad \mathbf{= -2.35 \times 10^{-4}\ in/in} \quad (14)$$

Tangential strain

$$e_t = \frac{S_t - \mu(S_z + S_r)}{E} + ß\,(\Delta t)$$

$$e_t = \frac{6655 - .3(39740 + (-2459))}{30 \times 10^6} + 6.9 \times 10^{-6}\,(45) \qquad \mathbf{= 1.60 \times 10^{-4}\ in/in} \quad (15)$$

Axial strain

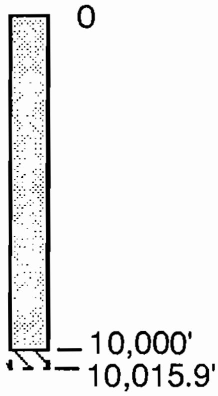
$$e_z = \frac{S_z - \mu(S_r + S_t)}{E} + ß\,(\Delta t)$$

$$e_z = \frac{39740 - .3(-2459 + 6655)}{30 \times 10^6} + 6.9 \times 10^{-6}\,(45) \qquad \mathbf{= .00159\ in/in} \quad (16)$$

Change in pipe length as would be measured at the surface

$$\Delta L = L\,(\Delta e_z)_{avg} \text{ or } L * e_z$$

$$\Delta L = 10000'\,(.00159) \qquad \mathbf{= 15.9\ ft} \qquad (18)$$



Effective weight of pipe for all depths

$$w_{eff} = w + 0.0408 * (\ MW_i{*}d^2 - MW_o{*}D^2)$$

$$w_{eff} = 26 + 0.0408 * (\ 7.5{*}6.276^2 - 12{*}7^2)$$

$$\mathbf{= 14.062\ ppf} \quad (21b)$$

Pipe body yield tension for all depths

$$T_{yield} = YP * A$$

$$= 80{,}000 * 7.549 \qquad \mathbf{= 603{,}929\ lbs} \quad (22)$$

von Mises equivalent stress at the surface

$$V^2 = 3\,(\frac{A_o}{A})^2\,\Delta P^2 + (\frac{T_{eff}}{A})^2$$

$$V^2 = 3\,(\frac{38.485}{7.549})^2\,1000^2 + (\frac{284163}{7.549})^2 \qquad \mathbf{= 38{,}664\ psi} \quad (26)$$

von Mises stress including bending stress at the surface

$$V^2 = 3\,(\frac{A_o}{A})^2\,\Delta P^2 + (\frac{T_{eff}}{A} \pm S_b)^2$$

$$V^2 = 3\,(\frac{38.485}{7.549})^2\,1000^2 + (\frac{284163}{7.549} \pm 27625)^2$$

$$V \qquad \mathbf{= 65{,}861\ psi} \quad (27)$$

Also solve for the von Mises stress at the surface with the expanded form of the equation

$$V^2 = 3 * (\frac{D^2}{D^2 - d^2})^2 * (P_i - P_o)^2 + \left[(\frac{T_{eff}}{A} \pm \frac{4310{*}D{*}C{*}\sqrt{T_{eff}}}{\sqrt{D^4 - d^4} * \tanh\left[0.2\sqrt{\frac{T_{eff}}{D^4 - d^4}}\right]}\right]^2$$

$$\frac{T_{eff}}{A} = \frac{T_{real}}{\frac{\pi}{4}(D^2-d^2)} - P_i\left[\frac{d^2}{D^2-d^2}\right] + P_o\left[\frac{D^2}{D^2-d^2}\right]$$

$$\frac{T_{eff}}{A} = \frac{300000}{\frac{\pi}{4}(7^2-6.276^2)} - 3000\left[\frac{6.276^2}{7^2-6.276^2}\right] + 2000\left[\frac{7^2}{7^2-6.276^2}\right]$$

$$\frac{T_{eff}}{A} = \mathbf{37{,}642\ psi} \qquad (29)$$

$$V^2 = 3\left[\frac{7^2}{7^2-6.276^2}\right]^2\left[3000-2000\right]^2 + \left[\frac{T_{eff}}{A} \pm \frac{4310*7*.05*\sqrt{284163}}{\sqrt{(7^4-6.276^4)}*\tanh\left[0.2\sqrt{\frac{284163}{(7^4-6.276^4)}}\right]}\right]^2$$

$$V^2 = 77965379 + \left\{37642 \pm \frac{804135}{29.109}\right\}^2$$

$$V = \mathbf{65{,}861\ psi} \qquad (28)$$

STRETCH, OD, AND ID, WALL STRAINS

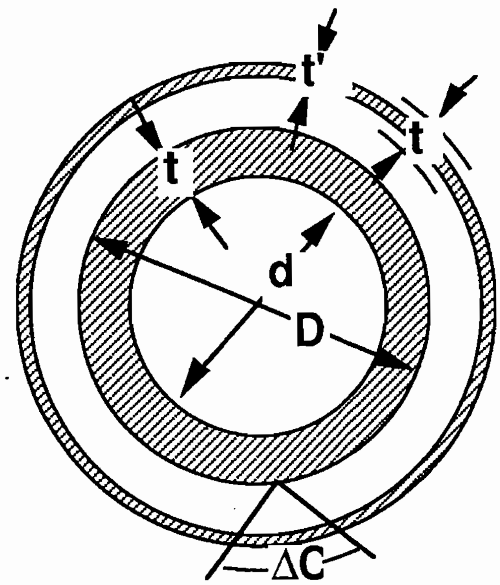
The following five example problems further demonstrate the fundamentals set forth in the stress analysis section.

All problems use the same tube and associated data as in the previous section:

10,000' OF 7" * 6.276" * 26 PPF * L80 CASING

Also, assume as before, that the average change in temperature over the entire length of the casing is 45°F. Other conditions are given as required.

CHANGE IN THE DIAMETER OF A TUBE



The change in the circumference of a tube because of tangential strain is

$$\Delta C_t = C * e_t$$

which leads to

$$\pi \Delta D_t = \pi D * e_t$$

and

$$\Delta D_t = D * e_t \qquad (30)$$

The change in the wall thickness caused by radial strain is

$$\Delta t = t * e_r$$

Because the thickness of the wall of a tube is equally likely to change dimensions on the inside as on the outside of the wall, the affect is to change the OD by half the change in the total thickness $(\frac{\Delta t}{2})$. However, a tube at any diameter has two walls. Thus, the equation for a change in diameter caused by radial strain is

$$\Delta D_r = 2 \left(\frac{\Delta t}{2}\right) \qquad (31)$$

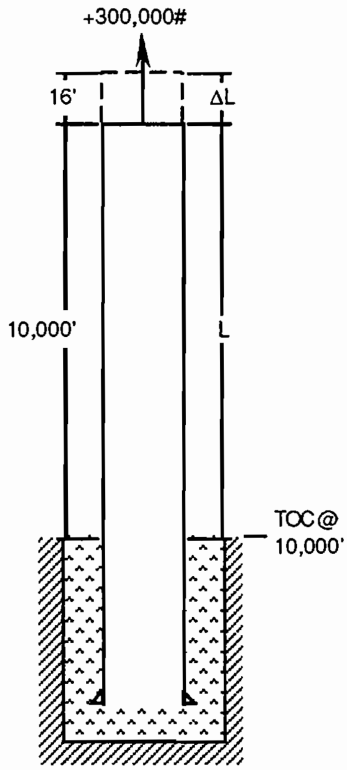
The total change in OD caused by tangential and radial strains is

$$\Delta D = \Delta D_t + \Delta D_r$$

or

$$\Delta D = D\, e_t + t\, e_r \qquad (32)$$

EXAMPLE #1



Find the new length of the casing after pulling 300,000 pounds above the buoyed weight of the casing.

The area of the pipe is

$$A = \frac{\pi}{4}(D^2 - d^2) = \frac{\pi}{4}(7^2 - 6.276^2) \quad \mathbf{= 7.549\ sq\ in} \quad (3)$$

The only load which has changed is the tension in the casing which affects the axial stress. Since there is no internal or external pipe pressure S_r and S_t are equal to zero.

$$S_z = \frac{T_{real}}{A}$$

$$S_z = \frac{300000}{7.549} \quad \mathbf{= 39{,}740\ psi} \quad (9)$$

Hooke's Law relates the strain with the stress

$$e_z = \frac{S_z - \mu(S_r + S_t)}{E} + \beta\,(\Delta t)$$

$$e_z = \frac{39740 - .3(0 + 0)}{30 \times 10^6} + 6.9 \times 10^{-6} \quad \mathbf{= .00164\ in/in} \quad (16)$$

Every foot of the casing has changed in length by the same amount

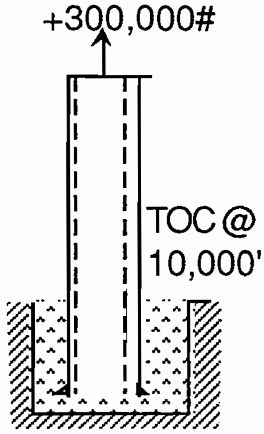
$$\Delta L = L * e_z$$

$$\Delta L = 10000' * 0.00164 \quad \mathbf{= 16.4\ ft} \quad (18)$$

The new length is the change added to 10,000'

$$L_{new} = L + \Delta L = 10000 + 16.4 \quad \mathbf{= 10{,}016\ ft}$$

EXAMPLE #2



What is the new outside diameter of the casing after pulling 300,000 pounds above its buoyed weight?

This is the same problem as #1, only now the concern is with the radial and tangential strains. Again, the only load which has changed is the tension in the casing which affects the axial stress. As in problem #1, since there is no internal or external pipe pressure, S_r and S_t are equal to zero.
As in Example #1

$$A = \frac{\pi}{4}(D^2 - d^2) = \frac{\pi}{4}(7^2 - 6.276^2) \qquad \mathbf{= 7.549\ sq\ in} \quad (3)$$

$$S_z = \frac{T_{real}}{A}$$

$$S_z = \frac{300000}{7.549} \qquad \mathbf{= 39{,}740\ psi} \qquad (9)$$

Hooke's Law relates the strain with the stress

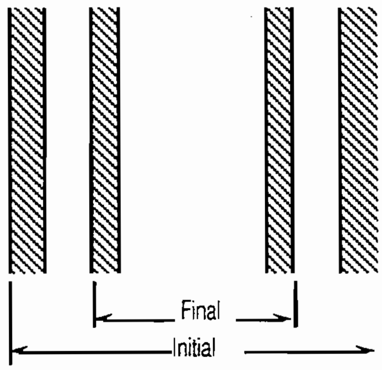
$$e_r = \frac{S_r - \mu(S_t + S_z)}{E} + ß\ (\Delta t) \qquad (14)$$

$$e_r = \frac{0 - .3(0 + 39740)}{30 \times 10^6} + 6.9 \times 10^{-6} \qquad \mathbf{= -8.69 \times 10^{-5}\ in/in} \quad (45)$$

$$e_t = \frac{S_t - \mu(S_z + S_r)}{E} + ß\ (\Delta t)$$

$$e_t = \frac{0 - .3(39740 + 0)}{30 \times 10^6} + 6.9 \times 10^{-6}$$

$$\mathbf{= -8.69 \times 10^{-5}\ in/in} \qquad (15)$$



It is important to note that even though the radial and tangential **stresses** are zero, the corresponding radial and tangential **strains** are **NOT**.

The change in the circumference is

$$\Delta C = \pi D * e_t = \pi * 7 * -8.69 \times 10^{-5} \qquad \mathbf{= -.00191\ in}$$

and the change in the diameter caused by e_t is

$$\Delta D_t = \frac{\Delta C}{\pi} = \frac{-.00191}{\pi} \qquad \mathbf{= -6.08 \times 10^{-4}\ in}$$

The change in the wall thickness is

$$\Delta t = t * e_r = \frac{7-6.276}{2} * -8.69 \times 10^{-4} \qquad \mathbf{= -3.15 \times 10^{-5}\ in}$$

and OD will change by an amount equal to Δt, and is

$$\Delta D_r = \Delta t \qquad \mathbf{= -3.15 \times 10^{-5}\ in}$$

The total change in the OD is then the sum of the partial changes

$$\Delta D = \Delta D_t + \Delta D_r = -6.08 \times 10^{-4} + -3.15 \times 10^{-5} \qquad \mathbf{= -6.40 \times 10^{-4}\ in}$$

The new OD is the change added to 7"

$$D_{new} = D_{old} + \Delta D = 7 + (-6.40 \times 10^{-4}) \qquad \mathbf{= 6.999\ in}$$

EXAMPLE #3

What is the new length of an open end casing after an internal pressure of 3,000 psi is applied? (For this problem let change in temperature = zero.)

The various areas are:

$$A_o = \frac{\pi}{4} D^2 = \frac{\pi}{4} (7)^2 \qquad \mathbf{= 38.485\ sq.\ in.} \quad (1)$$

$$A_i = \frac{\pi}{4} d^2 = \frac{\pi}{4} (6.276)^2 \qquad \mathbf{= 30.935\ sq.\ in.} \quad (2)$$

$$A = \frac{\pi}{4} (D^2 - d^2) = \frac{\pi}{4} (7^2 - 6.276^2) \qquad \mathbf{= 7.549\ sq\ in} \quad (3)$$

The only loads which are changing are the Sr and St. Sz is equal to zero since there is no tension.

$$S_r + S_t = \frac{2}{A} (P_i A_i - P_o A_o)$$

$$S_r + S_t = \frac{2}{7.549}(3000 * 30.935 - 0 * 38.485) \qquad \mathbf{= 24{,}587\ psi} \qquad (8)$$

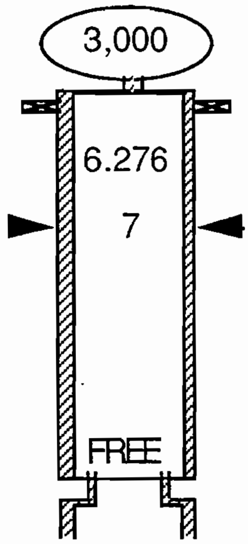
Now we solve for the axial strain

$$e_z = \frac{S_z - \mu(S_r + S_t)}{E} + ß\,(\Delta t)$$

$$e_z = \frac{0 - .3(24587)}{30 \times 10^6} + 6.9 \times 10^{-6}\,(0) \qquad \mathbf{= -2.46 \times 10^{-4}\ in/in} \quad (16)$$

Again, note that even though the axial stress is zero the corresponding axial strain is not.

Every foot of the casing has changed length by the same amount.



$$\Delta L = L * e_z$$

$$\Delta L = 10000' * -2.46 \times 10^{-4} \qquad \mathbf{= -2.46\ ft} \quad (18)$$

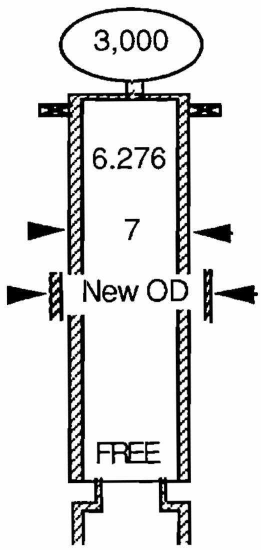
The new length is the change added to 10,000'

$$L_{new} = L + \Delta L = 10000 + (-2.46) \quad \mathbf{= 9{,}998\ ft}$$

An alternate solution is to use eqn (16) rather than eqn (8) and eqn (15) to determine the axial strain.

EXAMPLE #4

What is the new diameter of the casing after an internal pressure of 3,000 psi is applied? (For this problem let the change in temperature = zero.) Lower end is free and not capped.



The change in the axial tension for every foot of casing is

$$\Delta T = \Delta P_i * A_i \quad \text{and} \quad S_t = \Delta T/A$$

The change in the radial stress at the outside wall of the casing is zero

$$S_r = P_i(\frac{A_i}{A})(1-(\frac{D}{b})^2) - P_o(\frac{A_o}{A})(1-(\frac{d}{b})^2)$$

$$S_r = 3000(\frac{30.935}{7.549})(1-(\frac{7}{7})^2) - 0(\frac{38.485}{7.549})(1-(\frac{6.276}{7})^2) \quad \mathbf{= 0\ psi} \quad (6)$$

The change in the tangential stress at the outside wall of the casing is (b=D)

$$S_t = P_i(\frac{A_i}{A})(1+(\frac{D}{b})^2) - P_o(\frac{A_o}{A})(1+(\frac{d}{b})^2)$$

$$S_t = 3000(\frac{30.935}{7.549})(1+(\frac{7}{7})^2) - 0(\frac{38.485}{7.549})(1+(\frac{6.276}{7})^2) \quad \mathbf{= 24{,}587\ psi} \quad (7)$$

The change in the radial strain is

$$e_r = [\frac{S_r - \mu(S_t + S_z)}{E} + ß(\Delta t)]$$

$$e_r = [\frac{0 - .3(24587 + 0)}{30 \times 10^6} + 6.9 \times 10^{-6}(0)] \quad \mathbf{= -2.46 \times 10^{-4}\ in/in} \quad (14)$$

The change in the circumferential strain is

$$e_t = \frac{S_t - \mu(S_z + S_r)}{E} + ß(\Delta t)$$

$$e_t = \frac{24587 - .3(0 + 0)}{30 \times 10^6} + 6.9 \times 10^{-6}(0) \quad \mathbf{= 8.20 \times 10^{-4}\ in/in} \quad (15)$$

The change in the OD is

$$\Delta D = D\, e_t + t\, e_r$$

$$\Delta D = 7 * 8.20 \times 10^{-4} + .362 \times -2.46 \times 10^{-6} \qquad \mathbf{= 5.65 \times 10^{-3}\ in} \qquad (32)$$

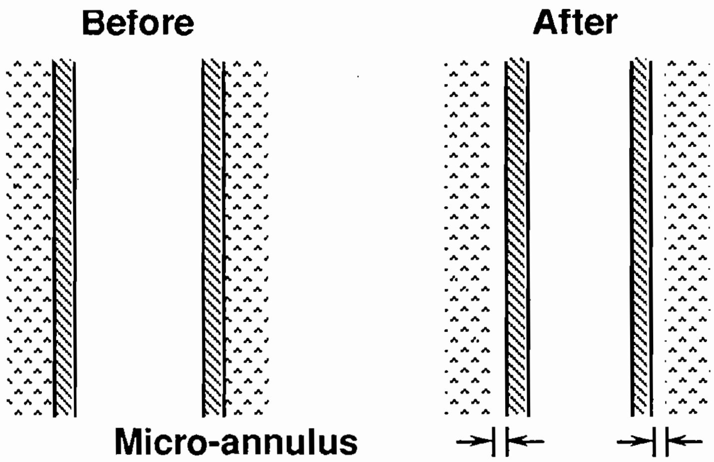
The new OD is

$$OD_{new} = OD_{old} + \Delta D$$

$$= 7 + 5.65 \times 10^{-3} \qquad \mathbf{= 7.00565\ in}$$

EXAMPLE #5

Find the new diameter of the casing after an internal pressure of 3,000 psi is removed? Further, assume that the casing is cemented in place and the diameter of the pipe while it is pressured at 3000 psi is 7". This is the **micro-annulus** problem.



The casing is cemented, thus it is assumed that it can not change length.

$$e_z = 0 \qquad (17)$$

Algebra and leaping forward a few steips yeilds (T is T_{real})

$$\Delta T = 2\mu(\Delta P_i A_i - \Delta P_o A_o) - A\ E\ ß(\Delta t)$$

$$\Delta T = 2*.3(-3000*30.935 - 0*38.485) - 7.549*30 \times 10^6*6.9 \times 10^{-6}\ (0)$$

$$\Delta T = -55684\ \# \qquad (19)$$

and the change in the axial stress is

$$\Delta S_z = \frac{T}{A} = \frac{-55684}{7.549} \qquad \mathbf{= -7376\ psi} \qquad (9)$$

The change in the tangential stress at the OD is:

$$\Delta S_t = \Delta P_i(\frac{A_i}{A})(1 + (\frac{D}{b})^2) - \Delta P_o(\frac{A_o}{A})(1 + (\frac{d}{b})^2)$$

$$\Delta S_t = -3000\ (\frac{30.935}{7.549})(1 + (\frac{7}{7})^2) - 0(\frac{38.485}{7.549})(1 + (\frac{6.276}{7})^2) \qquad \mathbf{= -24587\ psi} \qquad (7)$$

The tangential strain is

$$e_t = \frac{S_t - \mu(S_z + S_r)}{E} + \beta\,(\Delta t)$$

$$e_t = \frac{-24587 - .3(-7376 + 0)}{30 \times 10^6} + 6.9 \times 10^{-6}\,(0) \qquad \mathbf{= -7.46 \times 10^{-4}\ in/in} \quad (15)$$

The the radial strain is

$$e_r = \frac{S_r - \mu(S_t + S_z)}{E} + \beta\,(\Delta t)$$

$$e_r = \frac{0 - .3(-24587 + (-7376))}{30 \times 10^6} + 6.9 \times 10^{-6}\,(0) \qquad \mathbf{= 3.20 \times 10^{-4}\ in/in} \quad (14)$$

The **micro-annulus** which is the change in the outside diameter is

$$\Delta D = D\,e_t + t\,e_r$$

$$\Delta D = 7 * (-7.46 \times 10^{-4}) + .362 * 3.20 \times 10^{-4} \qquad \mathbf{= -5.28 \times 10^{-3}\ in}$$

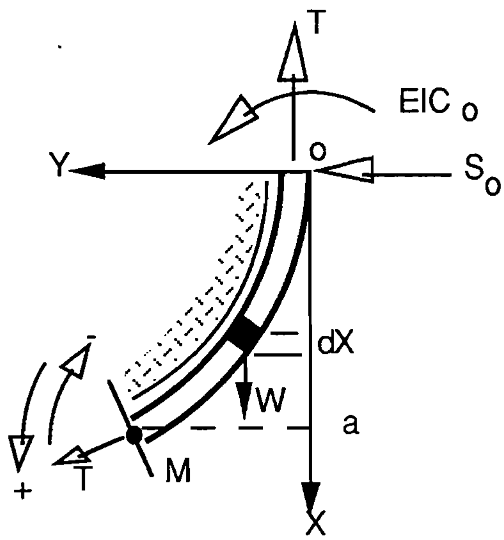
The new diameter is

$$OD_{new} = OD_{old} + \Delta D = 7 + (-5.28 \times 10^{-3}) \qquad \mathbf{= 6.995\ in}$$

BENDING STRESS IN DOGLEGS

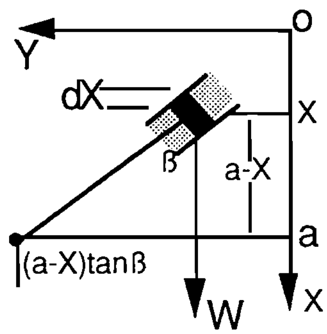
Stresses caused by the bending of tubulars within doglegs is computed with one of either two equations. Lubinski's bending equation resolves the stresses in which the connections contact the wall of the hole but the wall of the tube does not contact the wall of the hole. His equation not only requires a standoff at the wall but that the tube not have bending moments acting on it (positive effective tension or negative buckling tendency).

The second equation is the beam bending equation of mechanics. The equation requires that the wall of the tube fully contacts the wall of a hole which is smooth and circular; for example, slimline tubing in casing. Partial contact near the midpoint of a joint of pipe is possible and in which case a value for the bending stress would lie between the values computed with the two equations.



LUBINSKI BENDING STRESS

Suppose in the above sketch that the plane curve OA represents a free body of a section of a tube between connections but does not include the connections. Further suppose that as a result of connections that the free body does not contact the wall of the hole. The section weighs W pounds per foot in mud. The forces and moments acting on the free body are the buoyed weight of the tube hanging below the section and within the section, B, the bending moment created by the tubes connected free body and at each end of the free body, EIC_0, and the shear at each end by the tube, S_0.



Summing the moments around the point X at the lower end of the free body, setting the positive direction of the moments to be counter clockwise, and limiting angles to small values gives the following equation.

$$\Sigma M = EIC_0 + S_0 X + T Y - \int_0^X W X \sin\beta \, dX$$

The integral is explained in the sketch. Note that the lever arm of the buoyed weight is (a-x)sinß if tanß = sinß and the force is W dx.

The sum of the moments is equal to the second derivative of the elastic line of the tube. After integrating the weight term, the equation is

$$\frac{d^2Y}{dX^2} - \frac{T\,Y}{EI} = C_o + \frac{S_o X}{EI} - \frac{W \sin\beta}{2EI} X^2$$

The solution of the differential equation which satisfies the requisite boundary conditions is

$$Y = \frac{1}{K^2}\left[(C_o - q)(\cosh KX - 1) + s_o(\sinh KX - KX) + \frac{q}{2}(KX)^2\right]$$

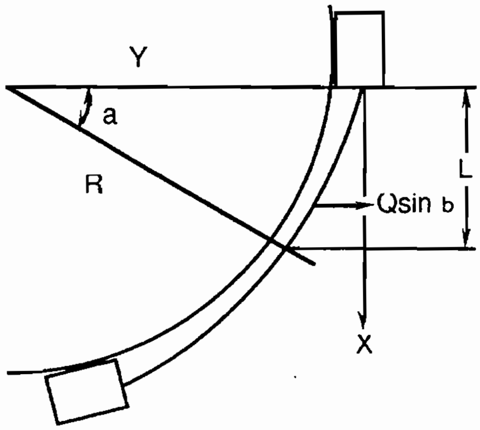
$$K = \sqrt{\frac{T}{EI}}$$

$$\frac{d}{dx}\cosh x = \sinh x$$

$$\frac{d}{d_x}\sinh x = \cosh x$$

$$q = \frac{1}{K^2}\,\frac{Q \sin b}{EI}$$

$$s_o = \frac{1}{K}\,\frac{S_o}{EI}$$



Suppose drill pipe is subject to conditions shown in the figure. If 2 L is the distance between tool joints, a is the change in wellbore angle in the length L, C is the curvature of the dogleg, then the pipe is parallel to the wellbore at L and a boundary condition for the elastic line of the pipe is

$$\left(\frac{dy}{dx}\right)_{X=L} = a = LC$$

If the pipe is not contacting the wellbore wall then shear cannot exist at L and a secondary boundary condition is

$$\left(\frac{d^3y}{dx^3}\right)_{X=L} = 0$$

$$\frac{d^3y}{dx} = (C_o - q)\sinh(KL) + s_o\cosh(KL) = 0$$

$$(c_o - q)\frac{\sinh(KL)}{\cosh(KL)} + s_o = 0$$

This boundary condition yields

$$s_o = -(C_o - q)\ \tanh KL$$

and the first boundary condition in combination with the last equation yields

$$(\frac{dy}{dx})_L = \frac{1}{K^2}[(C_o - q)\,K \sinh(KX) + s_o K \cosh(KX) - K\,s_o + 2\,\frac{q}{2}K^2X = LC$$

$$C = \frac{1}{K^2}(C_o - q)\frac{K}{L}\tanh(KL) + q = C$$

$$C = (\frac{2\,s_b}{ED} - q)\,\frac{\tanh(L\sqrt{T/EI})}{L\sqrt{T/EI}} + q$$

In practical problems the weight of the pipe over the length of the dogleg is insignificant and the length of the tube is 40 feet; therefore

$$S_b = \frac{4310\,D\,C\sqrt{T_{eff}}}{\sqrt{D^4 - d^4}\,\mathrm{TANH}\left[0.2\sqrt{\frac{T_{eff}}{D^4 - d^4}}\right]}$$

For the convenience of designers, the term equivalent bending force, F_B, was invented and defined to be the bending stress acting over the entire cross-sectional area of the pipe. This allows designers to work with loads rather than stresses. The equivalent bending force in conjunction with **Lubinski's equation** is

$$F_{LUB} = \frac{3385\,D\,C\sqrt{T_{eff}}\sqrt{\frac{D^2 - d^2}{D^2 + d^2}}}{\tanh\left[0.2\sqrt{\frac{T_{eff}}{D^4 - d^4}}\right]}$$

S_b	= bending stress in the outer fiber of the wall; psi
F_{LUB}	= equivalent bending force; lb
C	= curvature of the pipe; °/ft
T_{eff}	= effective tension in the pipe at the dogleg; lb
D	= OD of the pipe; inch
d	= ID of the pipe; inch

If the pipe is continuously in contact with the wall of the hole and collars do not provide standoff, the equivalent bending force equation for beams applies and is

$$F_{beam} = 17135\ D\ C\ (D^2 - d^2)$$

EXAMPLE

A string of 7" * 6.094" * 32 #/ft * P-110 casing has been run to 16,000 feet in 14.0 ppg mud. A dogleg with a severity of 6°/100 feet exists where the effective tension in the casing is 200,000 lbs. If there is no wall contact except at the collars, what is the equivalent bending force acting on the casing in the dogleg?

$$F_{LUB} = \frac{3385 * 7 * .06 \sqrt{200000} \sqrt{\dfrac{7^2 - 6.094^2}{7^2 + 6.094^2}}}{\tanh\left[0.2 \sqrt{\dfrac{200000}{7^4 - 6.094^4}}\right]} \qquad = \mathbf{237{,}623\ lb}$$

If continuous wall contact were the case, then

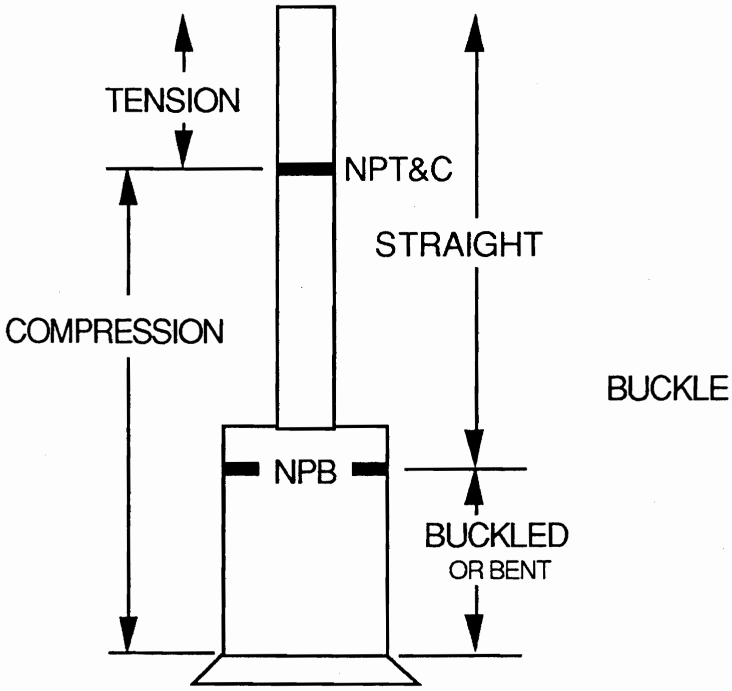
$$F_{beam} = 17135 * 7 * .06\ (7^2 - 6.094^2) \qquad = \mathbf{85{,}376\ lb}$$

BUCKLING versus TENSION & COMPRESSION

A long tubular may be bent or buckled and at same location and time be in tension.

Also, a tubular may be straight and be in compression at the same location and time. There are many so called, "neutral points."

The buckling (or bending) and tension situation for a normal drill string is shown in the sketch. Note that the bottom part of the drillpipe is in compression and straight and that only the BHA is buckled (or bent).



Buckling is caused by moments while tension is caused by stress.

In the drilling literature, the sum of all moments acting at a location within a tube is known as the **buckling tendency**, the **effective tension**, or the **fictitious force**.

The table summarizes the possibilities of moments and stresses, and the experiments which demonstrate them.

TABLE OF MOMENT AND STRESS EXPERIMENTS

TENSION & STRAIGHT	BY	PULLING ON A RUBBER BAND
TENSION & BUCKLING	BY	PRESSURING A TUBE FIXED ONLY AT ITS ENDS
COMPRESSION & BUCKLING	BY	PUSHING ON A RUBBER BAND OR RULER
COMPRESSION & STRAIGHT	BY	SUCKING A NOODLE INTO THE MOUTH

CRITICAL BUCKLING EVENTS OF CASING

Post cementation loads are called stability loads and/or buckling and wellhead loads.The wellhead load is the real tension in the casing just below the wellhead. The buckling load is the moment in the casing at the top of the cement according to the equations of W. R. Cox as published in the paper, 'Key Factors Affecting Landing of Casing,' *DRILLING AND PRODUCTION PRACTICES,* API, (1957), P. 225. Several text books contain his equations; i.e., Craft, Holden, & Graves, *WELL DESIGN: DRILLING AND PRODUCTION*, Prentice-Hall, Inc., 1962, and Mitchell, *DRILLING HANDBOOK,* Mitchell Engineering, 1974.

Cox's equations only compute the real tension load in the casing at the wellhead and the buckling load at the top of the cement. To find the real tension at any other location it is only necessary to subtract the air weight of the casing from the surface to the location. To find the buckling load at any other location it is only necessary to subtract the buoyed weight of the casing between the top of the cement and the location. Buoyed weight is the sum of the weights of the steel and fluids contained in the casing less the weights of the fluids displaced by the casing.

During the life of a casing string there will occur a set of loading conditions which will peak the loading on the casing while lowering its strengths to a minimum. This condition is called the critical load condition. For example, it is well known that axial compression reduces internal yield (burst) strength; therefore, pressuring the casing with hot oil may create a ***critical buckling load condition***.

It has been found that one of the listed ten sets of loadings (as given on the following page) produces a critical load condition. Within the table the '+' means an increase in the important variable or a resulting increase in buckling and/or tension. A minus indicates a decrease.

For example, if the casing is tensioned after the cement sets but before inserting the casing slips, (this is called pre-tension), then the important variable is tension and it does increase and the effect is that buckling decreases and tension increases. In practice if the casing is buckled or if the tension is excessive then steps must be taken to alleviate the situation.

PRETENSIONING OF CASING

Casing is pretensioned for the purpose of preventing buckles during the life of the casing. Pretension is the additional tension put in the casing above its natural hook load at the time that the cement sets. Casing may be pretensioned by two methods.

FIRST METHOD of PRETENSIONING THE CASING

The most popular method of casing pretensioning is to stretch it with rig hoisting equipment after the cement has set and before the casing slips are set.

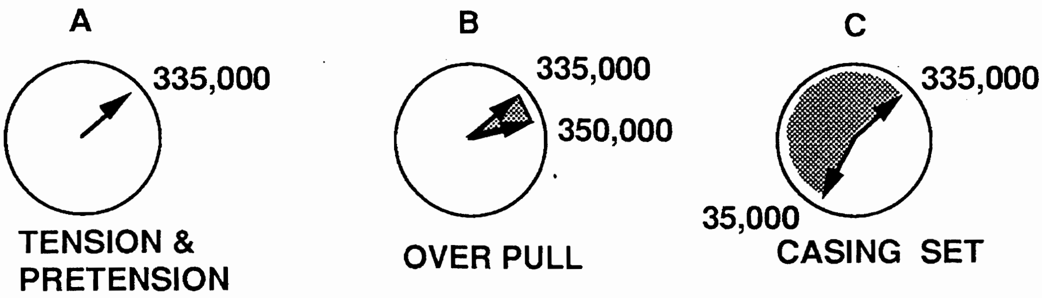
TABLE OF CRITICAL LOAD PRODUCING OPERATIONS

OPERATION	IMPORTANT VARIABLES	BUCKLING	TENSION
Pre-tension	Tension(+)	-	+
Casing pressure test	Internal surface pressure(+)	+	+
Leak off test	Internal surface pressure(+)	+	+
Drilling deeper	Temperature(+)	+	-
	Mud weight (+)	+	+
Fracturing	Internal surface pressure(+)	+	+
	Temperature(-)	-	+
	Internal fluid density(-)	-	-
Kick	Internal surface pressure(+)	+	+
	Temperature(+)	+	-
	Mud weight(-)	-	-
Blowout	Internal surface pressure(-)	-	-
	Temperature(+)	+	-
	Internal fluid density(-)	-	-
Close in on Blowout	Internal surface pressure(+)	+	+
	Temperature(+)	+	-
	Internal fluid density(-)	-	-
Production	Internal surface pressure(-)	-	-
	Temperature(+)	+	-
	Internal fluid density(-)	-	-
Braden head squeeze			
	Inner casing		
	External surface pressure(+)	-	-
	External fluid density(-)	-	-
	Outer casing		
	Internal surface pressure(+)	+	+
	Temperature(-)	-	+

A major problem is that the casing slips do not set instantaneoulsy upon slack-off of the casing in the casing bowl. Because of this, additional load must be pulled above the pretension load. The pretension that is put into the casing may be ascertained with the rig weight indicator.

Let us suppose that the casing weighs 200,000 lbs just after circulating cement and the pretension load is to be 100,000 lbs. The block and lines weigh 35,000 lbs. Now, when the slips bite the casing and set, the weight indicator should show 335,000 lbs. To accomplish this setting weight, it may be required to initially pull 350,000 lbs.

Regardless of what the pull, the final setting weight will be when the weight indicator's hand falls rapidly during the setting process, figure 'C'. The hand will not fall quickly before the slips bite, figure 'B'.



SECOND METHOD

The second method is to internally pressure the casing prior to the setting of the cement. The amount of tension put into the casing is the internal pressure multiplied by the internal end area of the casing.

Suppose that it is desired to pretension a casing which has an internal cross-sectional area of 31 sq. in., A_i, to a value of 100,000 lbs. Then, the pretensioning pressure, pP, required is given by the following equation.

$$pP = \frac{\text{Pretension}}{A_i} = \frac{100000}{31} = \mathbf{3{,}226\ psi}$$

The lower portion of the casing and the bottom end must be free so that the pressure can stretch the casing downward during the pressuring of the casing; if not, the casing will not be pretensioned.

If the casing is internally pressured prior to the setting of the slips and if the weight indicator does not change during the lowering of the casing to set the slips, then the casing will be pretensioned as planned.

BUCKLING TENDENCY & WELLHEAD LOAD

Euler defined the buckling of a column as the continuous lateral displacement of the column with constant axial loads. An oil field tubular is a column. It buckles into a helix if contained in hole. BUCKLING IS MOMENT DEPENDENT AND IS INDEPENDENT OF TENSION OR COMPRESSION. Recall that a moment is a lever arm multiplied by a force. A tubular in a hole can buckle while in either tension or compression. A tubular can be buckled while in either tension or compression. Those who speak of 'buckling loads' or 'buckling tendencies' are simply referring to only the force factor in the moment.

Buckling

Important phenomena which may create buckles in oil tubulars are the following:

(1) Temperatures
(2) Surface pressures
(3) Fluid densities
(4) Evacuations
(5) Slack-off

NORMAL BUCKLING ANALYSIS OF OIL TUBULARS REQUIRES THAT BOTH ENDS OF THE TUBULAR BE FIXED. This is also true for the equations written in this section. Those who have not readily comprehended the effects of the phenomena, most often have falsely assumed that the change in the length of the tubular is also the change in the distance between the fixed ends. A tubular may increase in length between two fixed ends by buckling.

Temperature

Increasing the temperature of steel causes the steel to expand. After it expands in length to a critical length, it will buckle; because, a longer length of tube must occupy the constant distance between its fixed ends.

Surface Pressure

The volume which a tubular can contain, must increase with an increase in internal pressure. Its containment volume can be increased by either an increase in its diameter, or its length, or both. If the length of the tube and internal pressure are sufficient, its length must increase and buckling will occur.

Fluid Densities

The effect of fluid densities is similar to surface pressures, in that, an increase in densities raises the internal pressure.

Evacuation

The effect of evacuation is similar to surface pressures, in that, an evacuation is a decrease in internal pressure.

Slack-off

Slack-off is the lowering of the top of the tubular into the hole. This adds to the length of the tube between the constant distance fixtures at the ends. Thus, the lengthening of the tube causes it to buckle.

Wellhead Loads

The wellhead load is the force placed on the hanger assembly by the tubular. It is equal to the load in the top of the tubular just below the hanger.

The wellhead load equation is a summation of the forces acting on the steel of the tubular. The forces are the attraction of the steel by the earth and pressures acting on cross-sectional areas of the tubular.

Equations

The four equations, two for buckling and two for loads, were modified from Cox's derivations. Equations number 1 and 3 (on the following pages) apply to a tubular which has free ends. This is the end condition of a tube before cement sets or before a packer is set. This is often called the "as cemented" condition. Thereafter, the ends are fixed and equations number 2 and 4 must be used. Final buckling and wellhead loads are evaluated by algebraically adding any changes to the free end solutions ("as cemented").

$$B' = B + \Delta B$$

$$W' = W + \Delta W$$

If the value of B' is positive, then the tubular is buckled. If the value of B (as cemented) is positive, the tubular is buckled after cementing and it will float out of the hole. If the value of B is negative and cement has been circulated to the surface, the tubular can never buckle. The reason is that lateral displacement of the tubular and not take place. If cement is not circulated to the surface and buckling occurs, the buckles begin at the top of the cement and diminish toward the surface. The value of B or B' only applies at the top of the cement which is also the bottom end of the free tubular. During the life of a tubular many operations will be performed with or within it One of the operations in the following list could cause maximum buckling.

EVENTS	VARIABLES
1. "as cemented	u, h, v, w
2. test casing	Δp
3. FPIT (leak-off test)	Δp

4. drilling deeper — Δt, Δw
5. circulating out a kick — Δt, Δp
6. during a blowout — Δt, n
7. shuting in on a blowout — Δt, Δp, n
8. hydraulic fracturing — Δt, Δp
9. production — Δt, Δp, n

Buckling 'as cemented' at the top of cement

$$B = - h\,u + .0408\, h\, (D^2 c - d^2 w) \tag{1}$$

Change in buckling (after cement sets) at the top of cement

$$\begin{aligned}\Delta B = \;& + .0408\, D^2 (-.7\, H\, \Delta v + (v + \Delta v)(.4\, m + \frac{.3\, m^2}{H}) \\ & + .0408\, d^2 (+.7\, H\, \Delta w - (w + \Delta w)(.4\, n + \frac{.3\, n^2}{H}) \\ & + .314\, (d^2\, \Delta p - D^2\, \Delta P) + 60\, u\, \Delta t - \Delta s\end{aligned} \tag{2}$$

Wellhead load 'as cemented'

$$W = + u\, L - .0408\, D^2 (h\, c + H\, v) + .0408\, d^2\, w\, L + .7854\, (d^2\, p - D^2\, P) \tag{3}$$

Change in wellhead load after cements sets

$$\begin{aligned}\Delta W = & -.0122\, H\, (D^2\, \Delta v - d^2\, \Delta w) \\ & + .0245\, D^2\, (v + \Delta v)\, (m - \frac{.5\, m^2}{H}) \\ & - .0245\, d^2\, (w + \Delta w)\, (n - \frac{.5\, n^2}{H}) \\ & + .471\, (d^2\, \Delta p - D^2\, \Delta P) - 60\, u\, \Delta t + \Delta S\end{aligned} \tag{4}$$

Wellhead load at 'Z' depth down from the wellhead

$$W'' = W' - u\, Z$$

Buckling at 'Y' feet up from the top of the cement

$$B'' = B' - Y\, (u + .0408\, (D^2\, v - d^2\, w))$$

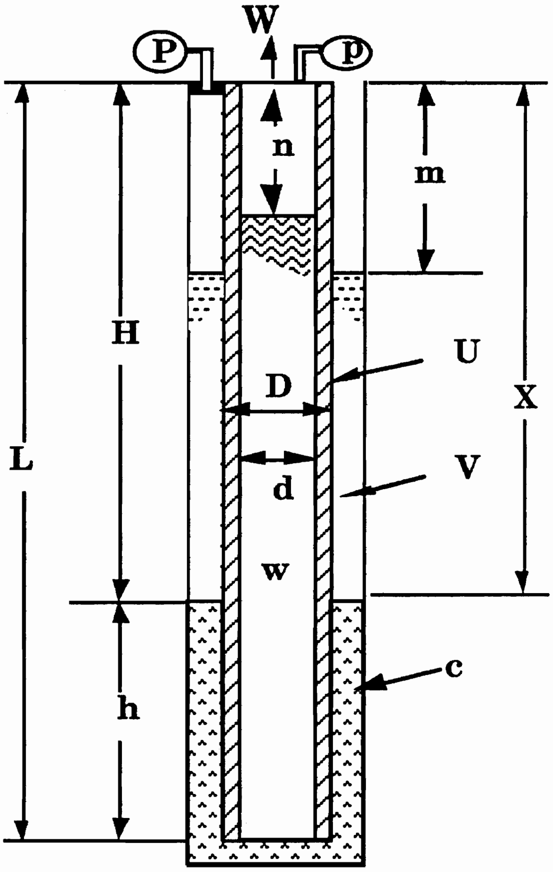
W = wellhead load; lbf

ΔS = pickup load (-slack-off); lbf

P = surface pressure outside pipe during the setting of the cement; psig

ΔP = change in surface pressure outside pipe after cement sets; psi

p = surface pressure inside pipe during the setting of the cement; psig



Δp = change in surface pressure inside pipe after cement sets, psi

m = change in fluid-level outside pipe after cement sets, (-rise in level)

n = change in fluid-level inside pipe after cement sets, (-rise in level)

Δt = average change in pipe temperature above cement top after cement sets;°F

v = fluid weight outside pipe when cement sets; ppg

Δv = change in fluid weight outside pipe after cement sets; ppg

w = fluid weight inside pipe after cement sets, ppg

Δw = change in fluid weight inside pipe after cement sets, ppg

D = outside diameter of pipe, in.

d = inside diameter of pipe; in.

u = weight of pipe; ppf

c = cement slurry weight; ppg

H = distance from cement top to hanger; ft.

h = distance from pipe shoe to cement top; ft

B = buckling tendency load, lb

EXAMPLE

10,000' of 7" * 6.276" * N-80 casing is run in 12 ppg mud and is cemented with 16 ppg cement. Surface pressures were 1000 psig in the annulus and 1500 psig within the casing during the setting of the cement. Cement plugs were displaced with 9 ppg mud.

The buckling of the casing "as cemented" is

$$B = -2000 * 26 + .0408 * 2000\,(7^2 * 16 - 6.276^2 * 9)$$

$B = -52{,}000 + 35{,}048$ **= -16,952 lb** (not buckled)

The wellhead load of the casing " as cemented" is

$$W = +\,26\,(2000 + 8000) - .0408 * 7^2\,(2000 * 16 + 8000 * 12)$$

$$+.0408 * 6.276^2 * 9\,(8000 + 2000) + .785\,(6.276^2 * 1500 - 7^2 * 1000)$$

$W = +\,260{,}000 - 255{,}898 + 144{,}633 + 7{,}915$ **= +156,650 lb**

After the surface pressures are bled off, this is also called the "as cemented" condition:

$\Delta B = +.314\,(-1500 * 6.267^2 - (-1000 * 7^2))$ **= -3,166 lb**

$$\Delta W = +.471\,[6.276^2 * (-1500) - 7^2 * (-1000)]$$

$W = +\,156{,}650 - 4749$ **= +151,901 lb** as cemented load

$B = -16{,}952 - 3{,}166$ **= -20,118 lb** as cemented buckling

Later, 25,000 lbs are slacked off; the annular fluid density is changed to 15 ppg and drops 1700'; the casing fluid is changed to 7 ppg and drops 6000'; the casing is pressured with gas (negligible density) to 4000 psig; and the average temperature of the free casing from the top of the cement to the wellhead drops by 15°F.

$$\Delta B = +.0408 * 7^2\,[-.7 * 8000 * 3 + (12 + 3)\,(.4 * 1700 + \frac{.3 * 1700^2}{8000})]$$

$$+ .0408 * 6.276^2\,[.7 * 8000 * (-2) - (9 - 2)\,(.4 * 6000 + \frac{.3 * 6000^2}{8000})]$$

$$+ 60 * 26 * (-15) - (-25{,}000) + .314\,(4000 * 6.276^2 - 0)$$

$\Delta B = -9{,}945 - 60{,}184 - 23{,}400 + 25{,}000 + 49{,}472$ **= -19,057 lb**

$B + \Delta B = -\,20{,}118 - 19{,}057$ **= -39,175 lb**

$$\Delta W = -.0122 * 8000\,[7^2 * 3 - 6.276^2 * (-2)]$$
$$+ .0245 * 7^2 * (12 + 3)(1700 - \frac{.5 * 1700^2}{8000})$$
$$- .0245 * 6.276^2 * (9 - 2)\,(6000 - \frac{.5 * 6000^2}{8000})$$
$$+ .471\,[6.276^2 * (4000) - 7^2 * (0)] - 60 * 26 * (-15) + (-25{,}000)$$

$$\Delta W = -22{,}036 + 27{,}360 - 25{,}332 + 74{,}207 + 23{,}400 - 25{,}000 \quad = \mathbf{+52{,}599\ lb}$$

$$W + \Delta W = +\ 151{,}901 + 52{,}599 \quad = \mathbf{+204{,}500\ lb}$$

Thus, the final buckling is -39,175 lbs and the wellhead load is +204,500 lbs. The casing is not buckled and is in tension at the wellhead.

INTERMEDIATE CASING DESIGN

An intermediate casing design must consider two factors which promote the buckling of the casing: temperature and/or mud weight changes. It seems reasonable that buckled casing, and therefore crooked casing, will wear at a higher rate, by drill sting rotation, than unbuckled casing. The most critical of the two factors is the rise in the average temperature of the casing while hot mud is circulated from deeper and deeper depths. Normally, mud weight increases as drilling progresses and casing buckles with higher mud weights.

EXAMPLE

Calculate the Hook Load required to prevent the buckling of the following described intermediate casing while drilling at the final total depth. Select a casing grade to run across the dogleg.

Casing:	O.D.:	9 5/8"
	I.D.:	8.681"
	Wt/Ft:	47 ppf
	Length:	12,141'

Mud: Casing set in 9.1 ppg water base mud.
After Drilling to Total Depth (19,281): 18.3 ppg

Cement: Top of Cement Outside 9 5/8": 11,500'
Wt. Of Cement: 15.9 ppg.

Geothermal Bottom hole Temperature: 371°F @ 19,281'

Geothermal Surface Temperature: 50°F @ 0'

Hole: Final Total Depth: 19,281'
Dogleg from 3100' to 3200' w/ a severity of 5°/100'

Temperature Effect

Make a temperature profile of the mud while circulating during drilling operations. Assume that the temperatures of the 9 5/8" casing are the temperatures of the mud. The initial temperatures of the casing will be the geothermal temperatures. (See the equations below)

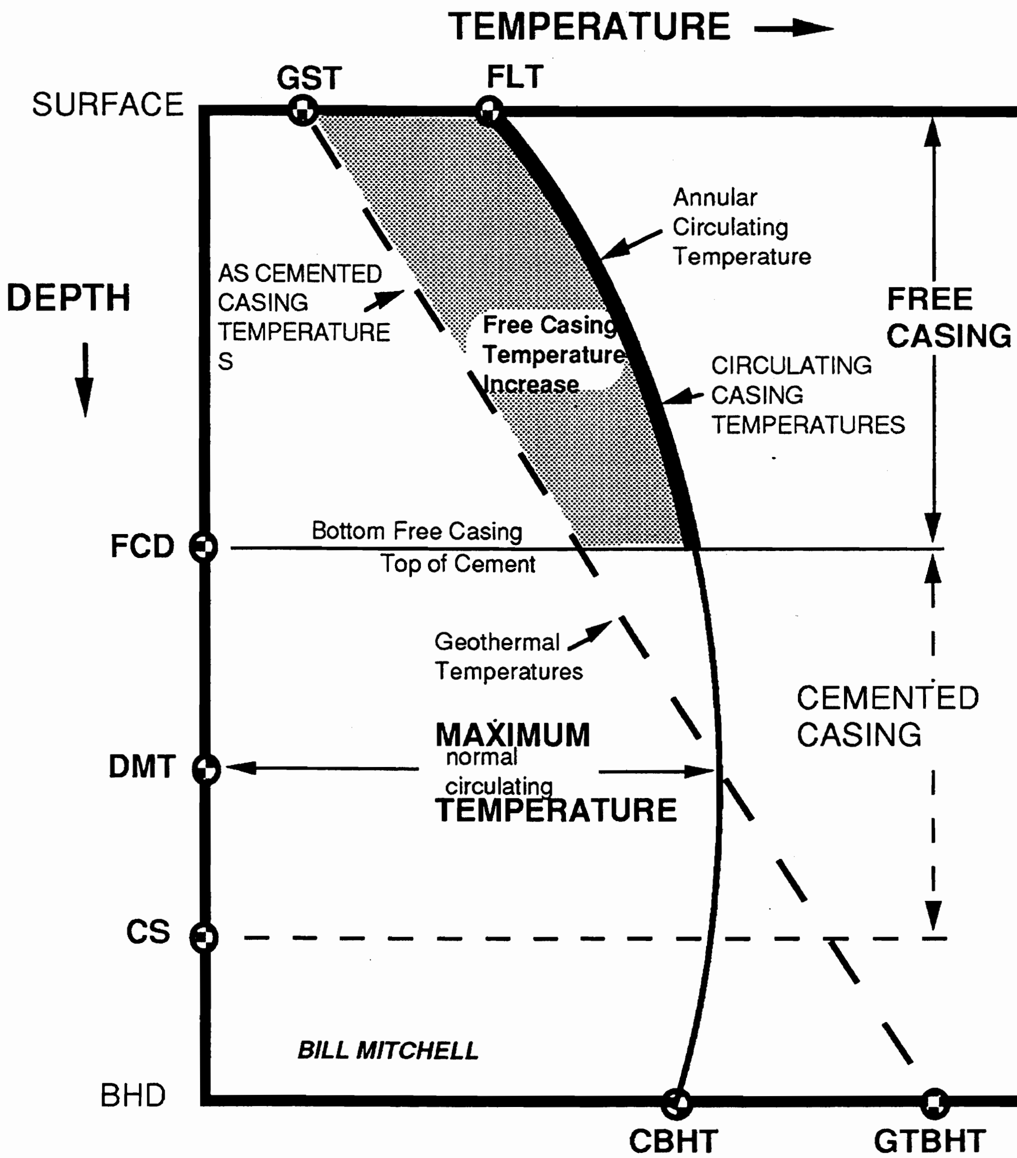
Circulating bottom hole temperature	$= .6 * (371 - 50) + 50$	**= 243 °F**
Circulating flow line temperature	$= .3 * (371 - 50) + 50$	**= 146 °F**
Circulating maximum temperature	$= .7 * (371 - 50) + 50$	**= 275 °F**
Circulating temperature at the bottom of the free casing	$x = 11{,}500 / 19{,}281$	**= 0.6**

$$\frac{T - 50}{371 - 50} = .3 + .6 - .4 * .6^2 - .3 * .6^3 \qquad = 272\ ^\circ F$$

Circulating average temperature from the surface to bottom of the free casing

$$\frac{T_a - 50}{371 - 50} = .3 + .5 * .6 - .135 * .6^2 - .074 * .6^3$$

CIRCULATING TEMPERATURE DIAGRAM



Geothermal (and cemented) average temperature from the surface to the bottom of the free casing

$$\frac{T_a - 50}{371 - 50} = .5 * .6 \qquad = 146\ ^\circ F$$

Average Casing Temperatures

Initial = **146 °F**
Final = **222°F**

Increase in the average casing temperature to final total depth:

222 - 146 = **76°F(rise)**

EQUATIONS FOR THE VALUES ON THE SKETCH

FLT = .4 (GTBHT - GST) + GST OIL MUD

FLT = .3 (GTBHT - GST) + GST WATER MUD

DMT = .7 BHD

MCT = .7 (GTBHT - GST) + GST

CBHT = .6 (GTBHT - GST) + GST

BUCKLING TENDENCY of the casing as cemented

$$B = - h * w + .0408\ h\ (D^2 * MW_c - d^2 * MW_i)$$
$$= -641 * 47 + .0408 * 641\ (9.625^2 * 15.9 - 8.681^2 * 9.1)$$
$$= -30{,}127 + 38{,}523 - 17{,}935 \qquad = \mathbf{-9{,}539\ lb}$$

Change in buckling after drilling to 19,281 feet

$\Delta MW = 18.3 - 9.1$ = **9.2 ppg**

$\Delta t = 222 - 146$ = **76°F**

$$\Delta B = +.0408 * d^2 * (.7 * L * \Delta MW_i) + 59.8 * w * \Delta t$$
$$= + .0408 * 8.681^2\ (.7 * 11{,}500 * 9.2\ ) + 59.8 * 47 * 76$$
$$= + 227{,}711 + 213{,}606 \text{ lbs} \qquad = \mathbf{+ 441{,}317\ lb}$$

B' = -9,539 + 441,317 **= + 431,778 lb** final buckling tendency

Must pull 431,778 lbs above hook load after cement sets to prevent buckling while drilling at total depth.

WELLHEAD LOAD after cementing

$$W = + w * (h + L) - .0408 * D^2 (h * MW_c + L * MW_e) + .0408 * d^2 * (h + L) * MW_i$$

$$= + 47 * 12{,}141 - .0408 * 9.625^2 (641 * 15.9 + 11{,}500 * 9.1) + .0408 * 8.681^2 (641 + 11{,}500) * 9.1$$

$$= + 570{,}627 - 434{,}073 + 339{,}700 \qquad \mathbf{= + 476{,}254\ lb}$$

WELLHEAD LOAD after drilling to TD

$$\Delta W = + .0122 * d^2 * L * \Delta MW_i - 59.8 * w * \Delta t + \Delta s$$

$$= + .0122 * 8.681^2 * 11{,}500 * 9.2 - 59.8 * 47 * 76 + 431{,}778$$

$$= + 97{,}271 - 213{,}606 + 431{,}778 \qquad \mathbf{= + 315{,}443\ lb}$$

$$W' = + 476{,}254 + 316{,}501 \qquad \mathbf{= + 791{,}697\ lb}$$

EQUIVALENT TENSION in casing at the top of the dogleg

$$W_{TOD} = W' - w * Y$$ Newton's Law of Motion

where: Y = Depth to top of dogleg

$$W_{TOD} = + 792{,}755 - 47 * 3100$$

= + 647,055 lb load in casing at top of dogleg

LUBINSKI bending load caused by the dogleg

$$T_L = \frac{3385 * 9.625 * .05 * \sqrt{647055} * \sqrt{\dfrac{9.625^2 - 8.681^2}{9.625^2 + 8.681^2}}}{\text{TANH}\left[\sqrt{\dfrac{0.2 * 647055}{9.625^4 - 8.681^4}}\right]} \qquad \mathbf{= + 422{,}418\ lb}$$

TOTAL TENSION LOAD in outer fiber of casing wall

$$\text{Load} = W_{TOD} + T_L$$

$$= 647{,}055 + 422{,}418 \qquad \mathbf{= 1{,}069{,}473\ lb}$$

Casing Selection

The body yield strength and joint strength of the casing within the dogleg must be superior to the load caused by the dogleg.

The API listed body strength of 9 5/8" * 47 ppf * P-110 casing is 1,493,000 lbs and its joint strength is 1,500,000 lbs.

Adjusted Strength = 1,493,000 * .875 * .9 **= 1,175,738 lb**

where: .875 = wall thickness
.9 = API yield point ambiguity

The tension design factors for P-110 casing and real loads are as follows

D.F. body = 1,175,738 / 1,069,473 **= 1.10**

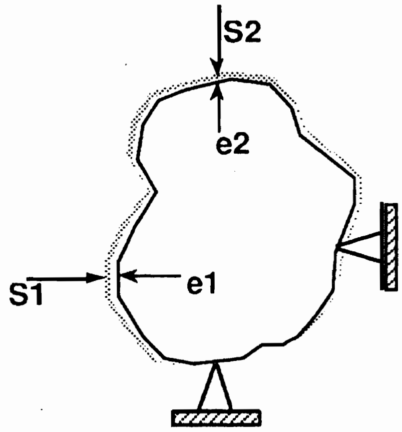
D.F. joint = 1,500,000 / 1,069,473 **= 1.40**

If the dogleg is *NOT* considered, the design factor at the depth of 3,100' is as follows

D.F. body = 1,175,738 / 647,055 **= 1.82**

TUBULAR STRENGTHS

BIAXIAL FACTOR DEFINED



API's revision of Bulletin 5C3, February 1, 1985, contains its new method for the computation of tubular collapse pressure resistance. It requires that the minimum unit yield strength of the steel in the tubular be adjusted rather than a direct adjustment of the collapse pressure resistance. API has called the adjusted minimum unit yield strength, 'axial stress equivalent grade', and has given it the symbol, 'Ypa'.

For reasons of brevity, the adjusted minimum unit yield strength is commonly called the adjusted yield strength. The following equation which is derived from the 'Maximum Strain Energy of Distortion at Yield' theory forms the basis of the biaxial factor.

$$\left[\frac{Y_{pa}}{Y_p}\right]^2 - \left[\frac{Y_{pa}}{Y_p}\right] * \left[\frac{T/A}{Y_p}\right] + \left[\frac{T/A}{Y_p}\right]^2 = 1$$

Let U $= \dfrac{T/A}{Y_p}$

A = plain end area of the tubular or casing; in^2

T = axial load in the wall of the tubular; lbs

Substitution and solving for Y_{pa}, gives the API equation for the adjusted biaxial yield strength of the steel:

$$Y_{pa} = Y_p \left[\left[1 - \frac{3}{4}\left(\frac{S_a}{YP}\right)^2 \right]^{\frac{1}{2}} - \frac{1}{2}\frac{S_a}{YP} \right]$$

Care must be observed while using this equation, because it is well known that the "axial stress equivalent grade" of steel may be increased as well as reduced. For example, tension reduces the collapse pressure of pipe while it increases the internal yield pressure.

The commensurate triaxial stress equation is inherently more accurate than the biaxial equation. Several authors have written that the percent differences in required pipe strength between the two theories will be less than 18%. This means that if 95,000 psi casing, as computed with the biaxial, is required, then in reality 112,100 psi casing may be needed.

Biaxial factors, Bx, for both tension and compression are defined by the following equations. Because of the adjustment of the sign which occurs before the last term, both tension and compression loads may be substituted in the appropriate position as positive values.

Biaxial Factor Equations

Tension-collapse $$Bx = \left[1 - \frac{3}{4}\left(\frac{S_a}{YP}\right)^2 \right]^{\frac{1}{2}} - \frac{1}{2}\frac{S_a}{YP}$$

Tension-burst $$Bx = \left[1 - \frac{3}{4}\left(\frac{S_a}{YP}\right)^2 \right]^{\frac{1}{2}} + \frac{1}{2}\frac{S_a}{YP}$$

Compression-collapse $$Bx = \left[1 - \frac{3}{4}\left(\frac{S_a}{YP}\right)^2 \right]^{\frac{1}{2}} + \frac{1}{2}\frac{S_a}{YP}$$

Compression-burst $$Bx = \left[1 - \frac{3}{4}\left(\frac{S_a}{YP}\right)^2 \right]^{\frac{1}{2}} - \frac{1}{2}\frac{S_a}{YP}$$

EXAMPLE

Calculate the adjusted yield strengths of 7" 26 ppf N-80 casing for both collapse and burst purposes for a tension load of 400,000 pounds. Its nominal plain end area is 7.549 square inches.

$$U = \frac{400000/7.549}{80000} \qquad \mathbf{= 0.6623}$$

$$Y_{pa} = 80{,}000 * [(1 - .75 * .6623^2)^{.5} - .5 * .6623]$$

Y_{pa} **= 39,037 psi** (for collapse computations)

$$Y_{pa} = 80{,}000 * [(1 - .75 * .6623^2)^{.5} + .5 * .6623]$$

Y_{pa} **= 92,024 psi** (for burst computations)

API COLLAPSE RESISTANCE

The API Bulletin 5C3, fourth edition, February 1, 1985, recommends four equations for computing the collapse pressure resistance of tubulars. The choice of equations is based only on the D/t ratio of the tubular.

Equation 1 is a theoretical equation which is based on the yielding of the inner wall of a tubular. It is Lame's equation and it is well known in the literature of mechanics of materials.

Equation 2 is an empirical equation which was derived from collapsing 2,488 tubes composed of K-55, N-80, and P-110 grades. API calls these values the 'plastic collapse pressure' of tubes. It should be noted that API expects one-half percent of the tubulars which fall into the range of equation 2 to fail with a frequency of 95 percent.

Equation 3 is an arbitrary equation which was generated by connecting the plastic collapse equation 2 with the elastic collapse equation 4. It holds no theoretical or empirical substance.

Equation 4 is a theoretical equation which is based on the bifurcation of the wall of the tubular without yielding. Bifurcation of the wall of a tube is similar to the axial buckling of a column in which the material of the column is not yielded. Bresse reported the solution to elastic buckling in 1866. The yield strength of the steel, Y_p , is not adjusted in this equation.

Bulletin 5C3: Formulas and Calculations for Pipe Properties 15

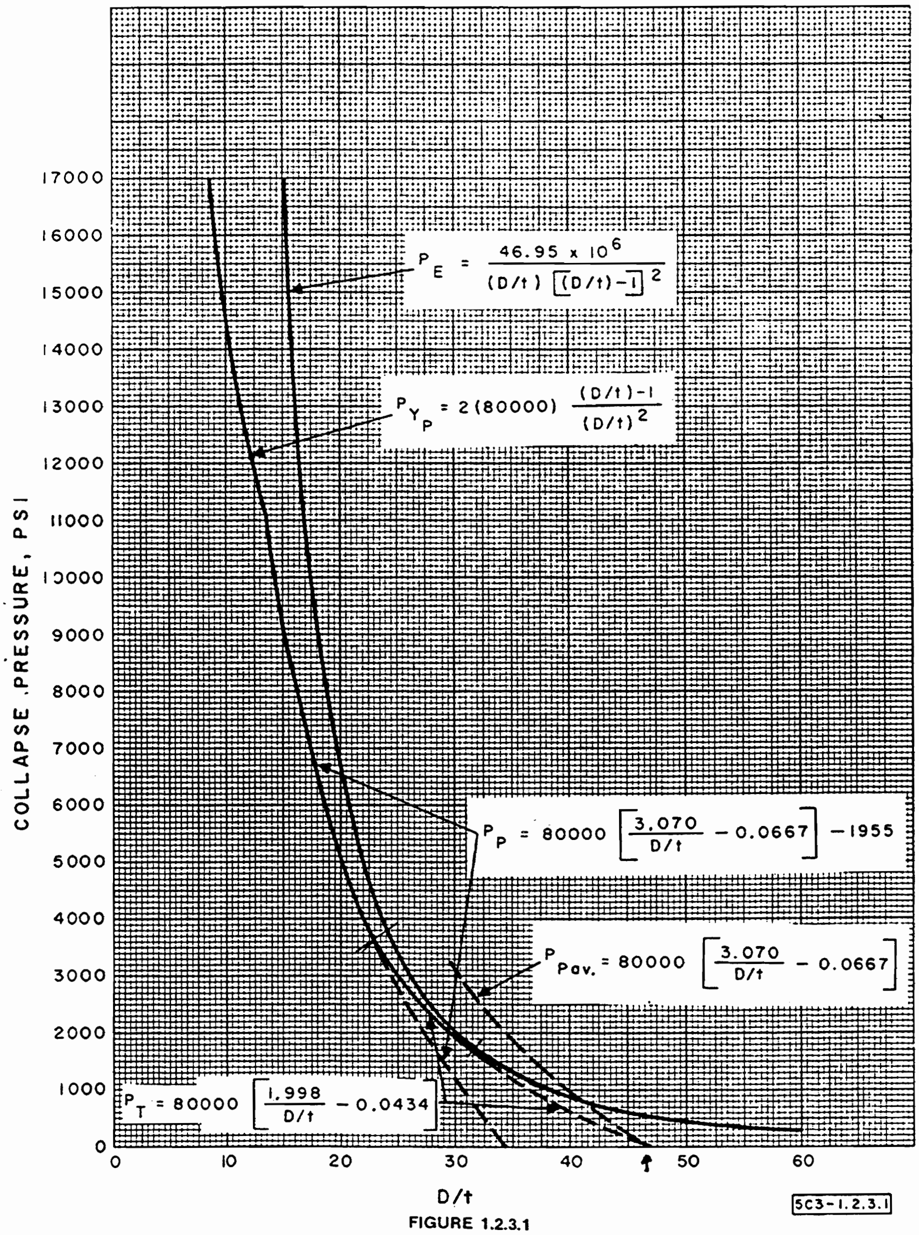


FIGURE 1.2.3.1
Grade N80 Transition Collapse
Formula Derivation

API COLLAPSE RESISTANCE OF TUBULARS TABLE

PROPERTIES		API FACTORS FOR COLLAPSE					CRITICAL D/t RATIOS		
GRADE	Yp	A	B	C	F	G	COL. 1	COL. 2	COL. 3
H40	40,000	2.950	0.0465	755	2.063	0.0325	16.40	27.01	42.64
J,K55	55,000	2.991	0.0541	1,206	1.989	0.0360	14.81	25.01	37.21
C75	75,000	3.054	0.0642	1,806	1.990	0.0418	13.60	22.91	32.05
L,N80	80,000	3.071	0.0667	1,955	1.998	0.0434	13.38	22.47	31.02
C95	95,000	3.124	0.0743	2,404	2.029	0.0482	12.85	21.33	28.36
P105	105,000	3.162	0.0794	2,702	2.052	0.0515	12.57	20.70	26.89
P110	110,000	3.181	0.0819	2,852	2.066	0.0532	12.44	20.41	26.22
P125	125,000	3.239	0.0895	3,301	2.106	0.0582	12.11	19.63	24.46
V150	150,000	3.336	0.1021	4,053	2.174	0.0666	11.67	18.57	22.11

API COLLAPSE EQUATIONS

[1] $P_{ypc} = 2 * Y_p * \frac{D/t - 1}{(D/t)^2}$ (D/t < Column 1)

[2] $P_{pc} = Y_p * [\frac{A}{D/t} - B] - C$ (Col 1 < D/t < Col 2)

[3] $P_{tc} = Y_p * [\frac{F}{D/t} - G]$ (Col 2 < D/t < Col 3)

[4] $P_{ec} = \frac{46950000}{(D/t) * (D/t - 1)^2}$ (D/t > Col 3)

$A = +2.8762 + 0.10679*10^{-5} * Y_{pa} + 0.21301*10^{-10} * Y_{pa}^{2} - 0.53132*10^{-16} * Y_{pa}^{3}$

$B = +0.026233 + 0.50609 * 10^{-6} * Y_{pa}$

$C = -465.93 + 0.030867 * Y_{pa} - 0.10483*10^{-7} * Y_{pa}^{2} + 0.36989*10^{-13} * Y_{pa}^{3}$

$$F = \frac{46.95 * 10^6 * \left[\frac{3 * X}{2 + X}\right]^3}{Y_{pa} \left[\frac{3 * X}{2 + X} - X\right] * \left[1 - \frac{3 * X}{2 + X}\right]^2} \qquad X = \frac{B}{A}$$

$$G = F * \frac{B}{A}$$

API COLLAPSE RESISTANCE (NO TENSION)

To ascertain the API collapse pressure resistance of tubulars, it is only necessary to adhere to the following procedure (see the table on the preceding page).

1. Compute the D/t ratio of the tubular.
2. Find the minimum yield strength of the steel in the column denoted by 'Yp' which is the second column from the left.
3. Compare the D/t ratio for the tubular with the critical D/t ratios in the far right hand columns.
4. Select the correct collapse equation by comparing the values of the critical D/t ratios with the D/t ratio of the tubular.
5. Substitute the appropriate API constants, A, B, C, F, and G, into the selected collapse equation.
6. Compute the collapse pressure resistance of the tubular.

EXAMPLE

Compute the API collapse pressure resistance of 7" 26ppf N-80 casing.

$D/t = 7.000/.362$ **= 19.34**

This value of D/t ratio lies between columns 1 and 2 of the critical D/t ratio columns on the right side of the table. Thus, the correct collapse equation is equation 2, the plastic collapse equation.

$$P_{pc} = Y_p * \left[\frac{A}{D/t} - B\right] - C$$

$$P_{pc} = 80{,}000 * \left[\frac{3.071}{19.34} - .0667\right] - 1.955$$ **= 5,414 psi**

The rounded off table value published by API is **5,410 psi**.

API ADJUSTED COLLAPSE RESISTANCE (WITH TENSION)

The API, within Bulletin 5C3, 1985, recommends two procedures for the computation of the effect of axial stress (tension or compression) on the collapse pressure of tubulars.

EXAMPLE

Compute the adjusted collapse pressure resistance of 7" 26 ppf P-110 with an axial tension of 83,040 pounds. Cross-sectional area is 7.549 square inches.

The procedure begins with the computation of the biaxial factor and the adjusted minimum unit yield strength of the steel in the tubular.

1. Compute the relative stress variable U.

$$U = \frac{T/A}{Y_p}$$

$$U = \frac{83040/7.549}{110000} \qquad \mathbf{= 0.1}$$

2. Compute the biaxial factor.

$$Bx = [(1 - .75 * U^2)^{.5} - .5 * U]$$
$$Bx = [(1 - .75 * 0.1^2)^{.5} - .5 * .1] \qquad \mathbf{= 0.946}$$

3. Compute the adjusted yield strength (*axial stress equivalent grade*) of the steel.

$$Y_{pa} = Y_p * Bx$$

$$Y_{pa} = 110{,}000 * 0.946 \qquad \mathbf{= 104{,}087\ psi}$$

4. Compute the constants A, B, C, F, and G.

$$A = +\,2.8762 + 0.10679*10^{-5} * Y_{pa} + 0.21301*10^{-10} * Y_{pa}^{\;2} - 0.53132*10^{-16} * Y_{pa}^{\;3}$$

$$A = +\,2.8762 + 0.10679*10^{-5} * 104{,}087 * 0.21301*10^{-10} * 104087^2 - 0.53132 * 10^{-16} * 104{,}087^3$$

$$\mathbf{= 3.158}$$

$$B = +\,0.026233 + 0.50609 * 10^{-6} * Y_{pa}$$

$B \quad = + 0.026233 + 0.50609 \; * 10^{-6} * 104{,}087$ **= 0.0789**

$C \quad = - 465.93 + 0.030867 * Y_{pa} - 0.10483 * 10^{-7} * Y_{pa}^{\;2} + 0.36989 * 10^{-13} * Y_{pa}^{\;3}$

$C \quad = - 465.93 + 0.030867 * 104{,}087 - 0.10483 * 10^{-7} * 104087^2 + 0.36989 * 10^{-13} * 104087^3$

= 2,675

Let $X = \frac{B}{A}$ in the next formula.

$X \qquad = 0.0789/3.158$ **= 0.02498**

Y_{pa} **= 104,087**

$$F \;= \frac{46.95 * 10^6 * \left[\frac{3 * X}{2 + X}\right]^3}{Y_{pa}\left[\frac{3 * X}{2 + X} - X\right] * \left[1 - \frac{3 * X}{2 + X}\right]^2}$$ **= 2.051**

$$G \;= F * \frac{B}{A} \;= 2.051 * \frac{0.0789}{3.158}$$ **= 0.0512**

5. Compute the critical (D/t) ranges in order to select the proper collapse equation.

$$(D/t)_{yp} \;= \frac{[\,(A - 2)^2 + 8 * (B + C/Y_{pa})\,]^{.5} + (A - 2)}{2 * (\,B + C/Y_{pa})}$$

$$= \frac{[\,(3.158 - 2)^2 + 8 * (.0789 + 2675/104087)\,]^{.5} + (3.158 - 2)}{2 * (\,.0789 + 2675/104087)}$$

= 12.59 and less

$$(D/t)_{pt} \;= \frac{Y_{pa} * (A - f)}{C + Y_{pa} * (B - G)}$$

$$= \frac{104087 * (3.158 - 2.051)}{2675 + 104087 * (.0789 - .0512)}$$ **= 20.75**

$$(D/t)_{te} \;= \frac{2 + B/A}{3 * B/A}$$

$$= \frac{2 + .0789/3.158}{3 * .0789/3.158} \qquad = \mathbf{27.02}$$

6. Collapse ranges for selection of collapse equation.

(D/t) for YIELD collapse	= 12.59 and less
(D/t) for PLASTIC collapse	= 12.59 to 20.75
(D/t) for TRANSITION collapse	= 20.75 to 27.02
(D/t) for ELASTIC collapse	= 27.02 and more

Note the Elastic collapse value is not altered by the adjusted yield strength of the steel. Always use Y_p in equation 4, and never use Y_{pa}.

7. Compute the nominal D/t ratio for the casing.

$$D/t = \frac{7}{.362} \qquad = \mathbf{19.34}$$

8. Select the correct collapse equation.

Because the real D/t ratio for the casing is in the D/t range of PLASTIC collapse, the plastic collapse equation is to be used.

$$P_p = Y_{pa} * [\frac{A}{D/t} - B] - C$$

$$= 104{,}087 * [\frac{3.158}{19.34} - .0789] - 2{,}675 \qquad = \mathbf{6{,}110\ psi}$$

API INTERNAL PRESSURE RESISTANCE

The API, in Bulletin 5C3, February 1, 1985, divides the internal pressure resistance of tubulars (BURST) into three areas:

1. INTERNAL YIELD PRESSURE OF THE WALL OF THE TUBE
2. INTERNAL YIELD PRESSURE OF THE COUPLINGS
3. INTERNAL PRESSURE LEAK RESISTANCE

The API lists the smallest of the values calculated within the three areas as the API internal pressure resistance of the tubular. This procedure could lead to problems. For example one may be interested in the internal yield pressure of the wall of the tube rather than the listed value which may be the leak resistance.

INTERNAL YIELD PRESSURE OF THE WALL OF THE TUBE

The internal yield pressure (burst) of the wall of a tube is given by the following API formula. The factor 0.875 accounts for the minimum wall tolerance. This is the thin wall formula of mechanics and loses accuracy as walls thicken.

$$P = 0.875 * \left[\frac{2 * Y_p}{\frac{D}{t}} \right]$$

P	= internal yield pressure of the wall; psi
Y_p	= yield strength of the steel; psi
t	= wall thickness of the tube; inches
D	= external diameter of the tube; inches

EXAMPLE

Compute the internal yield pressure for the wall of 7" 6.276" 26 ppf LT&C N-80 casing. t = 0.362

$$P = 0.875 * \left[\frac{2 * 80000 * 0.362}{7}\right] \qquad \mathbf{= 7{,}240\ psi}$$

The API lists the same value of **7,240 psi**.

INTERNAL YIELD PRESSURE OF COUPLINGS

Computation of the internal yield pressure of couplings requires two publications of API. These are the Bulletin 5C3 and Standard 5B.

EXAMPLE AND EQUATIONS

Compute the internal yield pressure for the coupling of 7" x 6.276" 26 ppf LT&C N-80 casing. Wall thickness, t, is 0.362.

$$P = Y_c * \left[\frac{W - d_1}{W}\right]$$

P	= minimum internal yield pressure of couplings; psi
Y_c	= minimum yield strength of couplings; psi
	= 80,000 psi
W	= nominal external diameter of coupling; inches
	= 7.656 inches
d_1	= diameter at the root thread of the coupling; inches

For round thread casing and tubing use

$$P = 80{,}000 * \left[\frac{7.656 - 6.77162}{7.656}\right] \qquad \mathbf{= 9{,}241 \text{ psi}}$$

Computation of variables for API's equation.

$$d_1 = E_1 - [L_1 + A] * T + H - 2 * S_{rn}$$

$$= 6.90337 - [2.921 + .375] * .0625 + 0.10825 - 2 * .017 \qquad \mathbf{= 6.77162}$$

E_1 = pitch diameter at hand tight plane; in. (Std 5B)

= 6.90337 inches (TABLE 2.3, page 7, Std 5B)

L_1 = length, end of pipe to hand tight plane; in (Std 5B)

= 2.921 inches (TABLE 2.3, page 7, Std 5B)

A = hand tight standoff; in. (Std 5B)
= column 12 / column 3 (TABLE 2.3, page 7, Std 5B)
= 3/8 **= .375 in**

T = taper **= 0.0625 in/in**

H = thread height, inches
= 0.08660 for 10 TPI
= 0.10825 for 8 TPI

S_{rn} **= 0.014** inches for 10 TPI
= 0.017 inches for 8 TPI

For Buttress thread casing use:

$$P = 80{,}000 * \left[\frac{7.656 - 6.846}{7.656}\right] \qquad \mathbf{= 8{,}464 \text{ psi}}$$

Computation of variables for API's equation.

$$d_1 = E_7 - [L_7 + I] * T + .062$$

$$= 6.954 - [2.2160 + 0.5] * .0625 + .062 \qquad \mathbf{= 6.846}$$

E_7 = pitch diameter; inches (Std 5B) **= 6.954 in**

L_7 = length of perfect threads; inches (Std 5B) **= 2.2160 in**

	PIPE SIZE inches		
	4.5	5 through 13.375	over 13.375
I =	0.400	0.500	0.375
T =	0.0625	0.0625	0.0833

INTERNAL LEAK RESISTANCE OF COUPLINGS

The appropriate equations and tables for the computation of the internal pressure leak resistance of couplings in the API Bulletin 5C3 and Standard 5B.

EXAMPLE AND EQUATIONS

For 8 round thread casing use

$$P = \frac{117188 * N * [W^2 - E_1^2]}{E_1 * W^2}$$

$$P = \frac{117188 * 4.5 * [7.656^2 - 6.90337^2]}{6.90337 * 7.656^2} \qquad \mathbf{= 14{,}281\ psi}$$

P = internal pressure leak resistance of couplings; psi

A = hand tight standoff; turns (Std 5B) **= 3 turns**

N = A + 1.5 = 3 + 1.5 **= 4.5**

W = external diameter; inch **= 7.656 inches**

E_1 = pitch dia. at hand tight plane; inches (Std 5B) **= 6.90337**

For Buttress thread casing use

$$P = \frac{3000000 * T * N * [W^2 - E_7^2]}{E_7 * W^2}$$

$$P = \frac{3000000 * .0625 * (1 + 1.5) * [7.656^2 - 6.954^2]}{6.954 * 7.656^2} = \mathbf{11{,}795\ psi}$$

PIPE SIZE			
		13.375" and smaller	16" and larger
T	=	.0625	.0833
N	=	A + 1.5	A + 1

PIPE BODY YIELD STRENGTH

The API pipe body strength is the axial load required to yield the tube. This equation does not reduce the body strength of a tubular for tolerances on dimensions or for the API yield point anomaly.

$$P_y = 0.784 * Y_p * [D^2 - d^2]$$

P_y = pipe body yield strength; lbs

Y_p = minimum yield strength of steel; psi

D = external diameter of tube; inches

d = internal diameter of tube; inches

EXAMPLE

Compute the API body strength for 7" 6.276" 26 ppf N-80 casing.

$$P_y = 0.784 * 80{,}000 * [7^2 - 6.276^2] \qquad \mathbf{= 603{,}930\ lb}$$

CASING STRENGTHS TABLE (API SPEC. 5CT)

Group	Grade	Type	Yield psi min	Yield psi max	Tensile Ultimate psi	Hardness HRC max	Specified Wall Thickness inch	Extension for yield percent
1	H40		40,000	80,000	60,000			.5
	J55		55,000	80,000	75,000			.5
	K55		55,000	80,000	95,000			.5
	N80		80,000	110,000	100,000			.5
2	C75	1,2,3	75,000	90,000	95,000			.5
	C75	9Cr	75,000	90,000	95,000	22		.5
	C75	13Cr	75,000	90,000	95,000	22		.5
	L80	1	80,000	95,000	95,000	23		.5
	L80	9Cr	80,000	95,000	95,000	23		.5
	L80	13Cr	80,000	95,000	95,000	23		.5
	C90	1,2	90,000	105,000	100,000	25.4 ±3.0	0.500 or less	.5
	C90	1,2	90,000	105,000	100,000	24.5 ±4.0	0.501 to 0.749	.5
	C90	1,2	90,000	105,000	100,000	25.4 ±5.0	0.750 to 0.999	.5
	C90	1,2	90,000	105,000	100,000	24.5 ±6.0	1.000 and above	.5
	C95		95,000	110,000	105,000			.5
	T95	1,2	95,000	110,000	105,000	25.4 ±3.0	0.500 or less	.5
	T95	1,2	95,000	110,000	105,000	25.4 ±4.0	0.501 to 0.749	.5
	T95	1,2	95,000	110,000	105,000	25.4 ±5.0	0.750 to 0.999	.5
3	P105		105,000	135,000	120,000			.6
	P110		110,000	140,000	125,000			.6
4	Q125		125,000	150,000	135,000	.... ±3.0	0.500 or less	.65
	Q125		125,000	150,000	135,000	.... ±4.0	0.501 to 0.749	.65
	Q125		125,000	150,000	135,000	.... ±5.0	0.749 and above	.65

#Additional page in book after page 69.

^Minimum elongation in 2 inches in percent rounded to nearest 1/2 percent is

$$e = 625{,}000\frac{A^{0.2}}{U^{0.9}}$$

CASING SHORT-THREAD DIMENSIONS

All dimensions in inches, except as indicated. See Fig. 2.1.
See Appendix B for metric tables.

1	2	3	4	5	6	7	8	9	10	11	12	13	14
Size: Outside Diameter D	Major Diameter	Nominal Weight: Threads and Coupling, lb. per ft.	No. of Threads Per Inch	Length: End of Pipe to Hand-Tight Plane L_1	Length: Effective Threads L_2	Total Length: End of Pipe to Vanish Point L_4	Pitch Diameter at Hand-Tight Plane E_1	End of Pipe to Center of Coupling, Power-Tight Make-Up J	Length: Face of Coupling to Hand-Tight Plane M	Diameter of Coupling Recess Q	Depth of Coupling Recess q	Hand-Tight Standoff, Thread Turns A	Minimum Length, Full Crest Threads, From End of Pipe Lc*
4½	4.500	9.50	8	0.921	1.715	2.000	4.40337	1.125	0.704	4 19/32	½	3	0.875
4½	4.500	Others	8	1.546	2.340	2.625	4.40337	0.500	0.704	4 19/32	½	3	1.500
5	5.000	11.50	8	1.421	2.215	2.500	4.90337	0.750	0.704	5 3/32	½	3	1.375
5	5.000	Others	8	1.671	2.465	2.750	4.90337	0.500	0.704	5 3/32	½	3	1.625
5½	5.500	All	8	1.796	2.590	2.875	5.40337	0.500	0.704	5 19/32	½	3	1.750
6⅝	6.625	All	8	2.046	2.840	3.125	6.52837	0.500	0.704	6 23/32	½	3	2.000
7	7.000	17.00	8	1.296	2.090	2.375	6.90337	1.250	0.704	7 3/32	½	3	1.250
7	7.000	Others	8	2.046	2.840	3.125	6.90337	0.500	0.704	7 3/32	½	3	2.000
7⅝	7.625	All	8	2.104	2.965	3.250	7.52418	0.500	0.709	7 23/32	½	3½	2.125
8⅝	8.625	24.00	8	1.854	2.715	3.000	8.52418	0.875	0.709	8 23/32	½	3½	1.875
8⅝	8.625	Others	8	2.229	3.090	3.375	8.52418	0.500	0.709	8 23/32	½	3½	2.250
9⅝	9.625	All	8	2.229	3.090	3.375	9.52418	0.500	0.709	9 23/32	½	3½	2.250(1)
9⅝	9.625	All	8	2.162	3.090	3.375	9.51999	0.500	0.713	9 23/32	½	4	2.250(2)
10¾	10.750	32.75	8	1.604	2.465	2.750	10.64918	1.250	0.709	10 27/32	½	3½	1.625(1)
10¾	10.750	Others	8	2.354	3.215	3.500	10.64918	0.500	0.709	10 27/32	½	3½	2.375(1)
10¾	10.750	Other	8	2.287	3.215	3.500	10.64499	0.500	0.713	10 27/32	½	4	2.375(2)
11¾	11.750	All	8	2.354	3.215	3.500	11.64918	0.500	0.709	11 27/32	½	3½	2.375(1)
11¾	11.750	All	8	2.287	3.215	3.500	11.64499	0.500	0.713	11 27/32	½	4	2.375(2)
13⅜	13.375	All	8	2.354	3.215	3.500	13.27418	0.500	0.709	13 15/32	½	3½	2.375(1)
13⅜	13.375	All	8	2.287	3.215	3.500	13.26999	0.500	0.713	13 15/32	½	4	2.375(2)
16	16.000	All	8	2.854	3.715	4.000	15.89918	0.500	0.709	16 3/32	½	3½	2.875
18⅝	18.625	87.50	8	2.854	3.715	4.000	18.52418	0.500	0.709	18 23/32	½	3½	2.875
20	20.000	All	8	2.854	3.715	4.000	19.89918	0.500	0.709	20 3/32	½	3½	2.875(3)
20	20.000	All	8	2.787	3.715	4.000	19.89499	0.500	0.713	20 3/32	½	4	2.875(4)

Included taper on diameter, all sizes, 0.0625 in. per in.

Coupling Grades Less than P110

P110 and Greater

Couplings Grades Less than K-55

K-55 and Greater

CASING LONG-THREAD DIMENSIONS

All dimensions in inches, except as indicated. See Fig. 2.1.
See Appendix B for metric tables.

1	2	3	4	5	6	7	8	9	10	11	12	13
Size: Outside Diameter D	Major Diameter D_4	No. of Threads Per Inch	Length: End of Pipe to Hand-Tight Plane L_1	Length: Effective Threads L_2	Total Length: End of Pipe to Vanish Point L_4	Pitch Diameter at Hand-Tight Plane E_1	End of Pipe to Center of Coupling, Power-Tight Make-Up J	Length: Face of Coupling to Hand-Tight Plane M	Diameter of Coupling Recess Q	Depth of Coupling Recess q	Hand-Tight Standoff, Thread Turns A	Minimum Length, Full Crest Threads, From End of Pipe Lc*
4½	4.500	8	1.921	2.715	3.000	4.40337	0.500	0.704	4 19/32	½	3	1.875
5	5.000	8	2.296	3.090	3.375	4.90337	0.500	0.704	5 3/32	½	3	2.250
5½	5.500	8	2.421	3.215	3.500	5.40337	0.500	0.704	5 19/32	½	3	2.375
6⅝	6.625	8	2.796	3.590	3.875	6.52837	0.500	0.704	6 23/32	½	3	2.750
7	7.000	8	2.921	3.715	4.000	6.90337	0.500	0.704	7 3/32	½	3	2.875
7⅝	7.625	8	2.979	3.840	4.125	7.52418	0.500	0.709	7 23/32	½	3½	3.000
8⅝	8.625	8	3.354	4.215	4.500	8.52418	0.500	0.709	8 23/32	½	3½	3.375
9⅝	9.625	8	3.604	4.465	4.750	9.52418	0.500	0.709	9 23/32	½	3½	3.625(1)
9⅝	9.625	8	3.537	4.465	4.750	9.51999	0.500	0.713	9 23/32	½	4	3.625(2)
20	20.00	8	4.104	4.965	5.250	19.89918	0.500	0.709	20 3/32	½	3½	4.125(3)
20	20.00	8	4.037	4.965	5.250	19.89499	0.500	0.713	20 3/32	½	4	4.125(4)

Included taper on diameter, all sizes, 0.0625 in. per in.

BUTTRESS CASING THREAD DIMENSIONS

All dimensions in inches, except as indicated. See Fig. 2.2. See Appendix B for metric tables.

1	2	3	4	5	6	7	8	9	10	11	12	13	14
Size: Outside Diameter D	Major Diameter D_4	No. of Threads Per Inch	Length: Imperfect Threads g	Length: Perfect Threads L_7	Total Length: End of Pipe to Vanish Point L_4	Pitch Diameter[1] E_7	End of Pipe to Center of Coupling, Power-Tight Make-Up J	End of Pipe to Center of Coupling, Hand-Tight Make-Up J_n	Length: Face of Coupling to Plane E_7	Length: End of Pipe to Triangle Stamp A_1	Hand-Tight Standoff, Thread Turns A	Diameter of Counterbore in Coupling Q	Minimum Length, Full Crest Threads From End of Pipe Lc*
4½	4.516	5	1.984	1.6535	3.6375	4.454	0.500	0.900	1.884	3 11/16	½	4.640	1.2535
5	5.016	5	1.984	1.7785	3.7625	4.954	0.500	1.000	1.784	4 1/16	1	5.140	1.3785
5½	5.516	5	1.984	1.8410	3.8250	5.454	0.500	1.000	1.784	4 ⅛	1	5.640	1.4410
6⅝	6.641	5	1.984	2.0285	4.0125	6.579	0.500	1.000	1.784	4 1/16	1	6.765	1.6285
7	7.016	5	1.984	2.2160	4.2000	6.954	0.500	1.000	1.784	4 ½	1	7.140	1.8160
7⅝	7.641	5	1.984	2.4035	4.3875	7.579	0.500	1.000	1.784	4 11/16	1	7.765	2.0035
8⅝	8.641	5	1.984	2.5285	4.5125	8.579	0.500	1.000	1.784	4 11/16	1	8.765	2.1285
9⅝	9.641	5	1.984	2.5285	4.5125	9.579	0.500	1.000	1.784	4 11/16	1	9.765	2.1285
10¾	10.766	5	1.984	2.5285	4.5125	10.704	0.500	1.000	1.784	4 11/16	1	10.890	2.1285
11¾	11.766	5	1.984	2.5285	4.5125	11.704	0.500	1.000	1.784	4 11/16	1	11.890	2.1285
13⅜	13.391	5	1.984	2.5285	4.5125	13.329	0.500	1.000	1.784	4 11/16	1	13.515	2.1285
16	16.000	5	1.488	3.1245	4.6125	15.938	0.500	0.875	1.313	4 11/16	⅞	16.154	2.7245
18⅝	18.625	5	1.488	3.1245	4.6125	18.563	0.500	0.875	1.313	4 11/16	⅞	18.779	2.7245
20	20.000	5	1.488	3.1245	4.6125	19.938	0.500	0.875	1.313	4 11/16	⅞	20.154	2.7245

Included taper on diameter: Sizes 13⅜ in. and smaller — 0.0625 in. per in.
Sizes 16 in. and larger — 0.0833 in. per in.

Std 5B: Specification for Inspection of Pipe Threads

API HYDROSTATIC TEST PRESSURES

The API's hydrostatic test pressures for plain-end pipe, extreme-line casing, and integral joint tubing are calculated with the following formula. (Exceptions for line pipe are noted in Bulletin 5C3).

Test Pressure for the Wall of Casing & Tubing

$$P = \frac{2 * S}{\frac{D}{t}}$$

P = hydrostatic test pressure rounded to nearest 100 psi for casing and tubing; psi

S = fiber stress as a percent of specified minimum yield strength as given in the following table.

t = nominal wall thickness; inches

D = external diameter of pipe; inches

TABLE OF FACTORS FOR TEST PRESSURES

GRADE	SIZE	'S'	MAXIMUM W/O AGREEMENT
H40, J55 & K55	9.625" & under	80	3,000
"	10.750" & larger	60	3,000
L80 & N80	all	80	10,000
C75 thru P110	all	80	10,000

EXAMPLE

Compute the API hydrostatic test pressure for 7" 6.276" 26 ppf N-80 casing (wall thickness = 0.362").

$$P = \frac{2 * (.80 * 80000) * 0.362}{7} \qquad \mathbf{= 6{,}619\ psi}$$

Note that this value is 80 percent of minimum internal yield strength.

Test Pressures for Coupling Yield and Leak Resistance

Coupling yield test pressure is 80% of yield and leak resistance test pressure is 100% of leak resistance.

TOLERANCES ON DIMENSIONS

The following table shows the tolerance on manufacturing dimensions for API tubulars. The following problem shows by example the affect of tolerances on the strength of API casing.

EXAMPLE

Compute the minimum body yield strength of 7" 6.276" 26 ppf N-80 casing.

Nominal wall thickness, t	$= \frac{7 - 6.276}{2}$	**= 0.362 in**
Wall thickness tolerance	= .125 * t	
Minimum wall thickness	= 0.362*(1 - .125)	**= 0.317 in**
Tolerance on OD	= +.01 or -.005 of nominal OD	
Minimum OD	= 7 * (1 - .005)	**= 6.965 in**
Minimum ID	= Minimum OD - Wall thickness	
	= 6.9475 - 2*0.317	**= 6.314 in**
Minimum Cross-sectional Area	$= .7854 * (OD^2 - ID^2)$	
	$= .7854 * (6.965^2 - 6.314^2)$	
		= 6.789 sq in
Adjustment of API yield strength factor	= 0.9	
Yield strength	= 80,000 * .9	**= 72,000 psi**
Minimum body yield strength	= 72,000 * 6.789	**= 488,842 lb**

The difference between API and the Adjusted **= 115,158 lb**

Percent difference below the API minimum body yield strength value is

$$\% \text{ diff } = \frac{\text{API value - Adjusted Value}}{\text{API Value}} * 100$$

$$= \frac{604000 - 488842}{604000} * 100 \qquad \mathbf{= 19\ \%}$$

TOLERANCES ON DIMENSIONS AND WEIGHTS

OUTSIDE DIAMETER, *D*

	Tolerance
4 in. and smaller	± 0.79 mm
4½ in. and larger	+1.00 percent -0.50 percent

For upset casing the following tolerances apply to the outside diameter of the pipe body immediately behind the upset for a distance of approximately 127 mm for sizes 5½ in. OD and smaller, and a distance approximately equal to the OD for sizes larger than 5½ in. Measurements shall be made with calipers or snap gages.

Pipe Size, OD, in.	Tolerances Behind M_{eu} or L_{0}, mm	
1.050-3.500	+2.38	-0.79
4.000-5.000	+2.78	-0.75% D
5.500-8.625	+3.18	-0.75% D
9.625 and larger	+3.97	-0.75% D

For 2⅜ in. and larger external-upset tubing the following tolerances shall apply to the outside diameter at distance L_a from the end of the pipe. The measurements shall be made with snap-gages or calipers. Changes in diameter between L_a and L_4, shall be smooth and gradual. Pipe body OD tolerances do not apply for a distance of L_4, from the end of the pipe.

Pipe Size, OD, in.	Tolerance, mm	
2⅜-3½	+2.38	-0.79
4	+2.78	-0.79
4½	+2.78	-0.75% D

WALL THICKNESS, *t* .... -12.5 percent

WEIGHT:

Single lengths......................	+6.5 percent -3.5 percent
Carload lots	-1.75 percent

A carload is considered to be a minimum of 40,000 lb (18144 kg).

INSIDE DIAMETER, *d*, is governed by the outside diameter and weight tolerances.

UPSET DIMENSIONS:
Tolerances on upset dimensions are given in Tables 6.4, 6.6, 6.7, and 6.8.

API DRILL PIPE LIST†

1		2	3		4	5		6
Size: Outside Diameter, *D*		Nominal Weight, lb/ft	Calculated Plain-end Weight, W_{pe}		Grade	Wall Thickness, *t*		Upset Ends, for Weld-on Tool Joints
in.	mm		lb/ft	kg/m		in.	mm	
2⅜	60.3	6.65	6.26	9.32	E X, G, S	0.280	7.11	Ext. Upset
2⅞	73.0	10.40	9.72	14.48	E X, G, S	0.362	9.19	Int. Upset or Ext. Upset
3½	88.9	9.50	8.81	13.12	E	0.254	6.45	Int. Upset or Ext. Upset
3½	88.9	13.30	12.31	18.34	E X, G, S	0.368	9.35	Int. Upset or Ext. Upset
3½	88.9	15.50	14.63	21.79	E	0.449	11.40	Int. Upset or Ext. Upset
3½	88.9	15.50	14.63	21.79	X, G, S	0.449	11.40	Ext. Upset or Int.-Ext. Upset
4	101.6	14.00	12.93	19.26	E X, G, S	0.330	8.38	Int. Upset or Ext. Upset
4½	114.3	13.75	12.24	18.23	E	0.271	6.88	Int. Upset or Ext. Upset
4½	114.3	16.60	14.98	22.31	E X, G, S	0.337	8.56	Ext. Upset or Int.-Ext. Upset
4½	114.3	20.00	18.69	27.84	E X, G, S	0.430	10.92	Ext. Upset or Int.-Ext. Upset
5	127.0	16.25	14.87	22.15	X, G, S	0.296	7.52	Int. Upset
5	127.0	19.50	17.93	26.71	E	0.362	9.19	Int.-Ext. Upset
5	127.0	19.50	17.93	26.71	X, G, S	0.362	9.19	Ext. Upset or Int.-Ext. Upset
5	127.0	25.60	24.03	35.79	E	0.500	12.70	Int.-Ext. Upset
5	127.0	25.60	24.03	35.79	X, G, S	0.500	12.70	Ext. Upset or Int.-Ext. Upset
5½	139.7	21.90	19.81	29.51	E X, G, S	0.361	9.17	Int.-Ext. Upset
5½	139.7	24.70	22.54	33.57	E X, G, S	0.415	10.54	Int.-Ext. Upset
6⅝	168.3	25.20	22.19	33.05	E	0.330	8.38	See footnote

†Drill pipe with special end finish is available under this specification. See Par. 7.1.
Upset requirements for 6⅝ in. — 25.20 lb. drill pipe not established.

BUTTRESS THREAD CASING COUPLING

Size*	Outside Diameter: Regular W	Outside Diameter: Special Clearance W_c	Minimum Length N_L
4½	5.000	4.875	8⅞
5	5.563	5.375	9⅛
5½	6.050	5.875	9¼
6⅝	7.390	7.000	9⅝
7	7.656	7.375	10
7⅝	8.500	8.125	10⅜
8⅝	9.625	9.125	10⅝
9⅝	10.625	10.125	10⅝
10¾	11.750	11.250	10⅝
11¾	12.750	.....	10⅝
13⅜	14.375	.....	10⅝
16*	17.000	.....	10⅝
18⅝*	20.000	....	10⅝
20*	21.000	....	10⅝

ROUND-THREAD CASING COUPLING

Size¹	Outside Diameter W	Minimum Length: Short* N_L	Minimum Length: Long N_L
4½	5.000	6¼	7
5	5.563	6½	7¾
5½	6.050	6¾	8
6⅝	7.390	7¼	8¾
7	7.656	7¼	9
7⅝	8.500	7½	9¼
8⅝	9.625	7¾	10
9⅝	10.625	7¾	10½
10¾	11.750	8	....
11¾	12.750	8	....
13⅜	14.375	8	....
16	17.000	9	....
18⅝	20.000	9	....
20	21.000	9	11½

API PLAIN-END CASING LINER LIST

1		2		3	4	
Size: Outside Diameter, *D*		Plain-End Weight		Grade	Wall Thickness, *t*	
in.	mm	lb/ft	kg/m		in.	mm
3½	88.9	9.91	14.76	J	0.289	7.34
4	101.6	11.34	16.89	J	0.286	7.26
4½	114.3	13.04	19.42	J	0.290	7.37
5	127.0	17.93	26.71	J	0.362	9.19
5½	139.7	19.81	29.51	J	0.361	9.17
6⅝	168.3	27.65	41.18	J	0.417	10.59

TENSILE REQUIREMENTS

1	2		3		4	
Grade	Yield Strength min.		Yield Strength max.		Tensile Strength min.	
	psi	MPa	psi	MPa	psi	MPa
E-75	75,000	517	105,000	724	100,000	689
X-95	95,000	655	125,000	862	105,000	724
G-105	105,000	724	135,000	931	115,000	793
S-135	135,000	931	165,000	1138	145,000	1000

MAKE-UP TORQUE FOR API COUPLINGS

It is written in the API Bulletin 5C3, February 1, 1985, on page 39 that the optimal make-up torque is one percent (1%) of the calculated joint pull-out strength for round thread casing and tubing. The minimum value is 75% of the optimal value and the maximum is 125% of the optimal.

ROUND THREAD WITH BENDING AND TENSION

API Bulletin 5C3, page 24, gives the following formula which was taken from the work of Clinedinst for the affect of bending and tension on the fracture strength of round thread pipe. Two bending cases exist.

No Bending

The joint strength of pipe is the lesser of its ultimate fracture strength, Pu, and its jump out strength, Pj.

$$P_u = 0.95 * A_{jp} * U_p$$

$$P_j = 0.95 * A_{jp} * L \left[\frac{0.74\, D^{-.59} * U_p}{0.5\, L + 0.14\, D} + \frac{Y_p}{L + 0.14\, D}\right]$$

Bending with Tension

The strength of a joint with combined bending and tension is found by reducing the ultimate fracture joint strength. Two cases exist and depend on the axial tensile stress.

Case #1 $\quad \dfrac{P_b}{A_{jp}} > Y_p$

$$P_b = 0.95 * A_{jp} * \left[U_p - \left(\frac{140.5 * B * D}{(U_p - Y_p)^{.8}}\right)^5 \right]$$

Case #2 $\quad \dfrac{P_b}{A_{jp}} < Y_p$

$$P_b = 0.95 * A_{jp} * \left[\frac{U_p - Y_p}{0.6444} + Y_p - 218.15 * B * D\right]$$

$$A_{jp} = .7854 * [\,(D - 0.1425)^2 - (D - 2 * t)^2\,]$$

B = bending; degrees/100 ft
D = nominal external diameter of pipe; inches
d = nominal internal diameter of pipe; inches

L	$= L_4 - M$ (API Std. 5B) p. 81 of text	
Y_p	= minimum yield strength of pipe; psi	
t	= nominal wall thickness; inches	
$\dagger U_p$	= minimum ultimate strength of pipe; psi (API Spec. 5CT) see page 80	

EXAMPLE

Compute the coupling strength of 7" 6.276" 26ppf LT&C N80 casing if it is in a 20 deg/100ft dogleg.

$U_p = 100{,}000$ psi
$B = 20.0$ deg/100ft
$D = 7.0$ inches
$L = L_4 - M = 4 - 0.704$ **= 3.296**
$t = 0.362$ inches

$$A_{jp} = .7854 * [(D - 0.1425)^2 - (D - 2 * t)^2$$

$$= .7854 * [(7 - 0.1425)^2 - (7 - 2 * .362)^2] \qquad \mathbf{= 5.998\ in^2}$$

No Bending

$$P_u = 0.95 * A_{jp} * U_p \qquad = 0.95 * 5.998 * 100{,}000 \qquad \mathbf{= 569{,}829\ lb}$$

$$P_j = 0.95 * A_{jp} * L \left[\frac{0.74\ D^{-.59} * U_p}{0.5\,L + 0.14\,D} + \frac{Y_p}{L + 0.14\,D} \right]$$

$$P_j = 0.95 * 5.998 * 3.296 \left[\frac{0.74 * 7^{-.59} * 100000}{0.5 * 3.296 + 0.14 * 7} + \frac{80000}{3.296 + 0.14 * 7} \right]$$

$$\mathbf{= 514{,}145\ lb}$$

The value of the joint strength listed by API is 519,000 lbs which is the smaller of the two values.

Bending

$$P_b = 0.95 * A_{jp} * [U_p - (\frac{140.5 * B * D}{(U_p - Y_p)^{.8}})^5]$$

$$P_b = 0.95 * 5.998 * [100000 - (\frac{140.5 * 20 * 7}{(100000 - 80000)^{.8}})^5]$$

$$P_b \qquad \mathbf{= 464{,}945\ lbs\ (case\ \#1)}$$

$$\frac{P_b}{A_{jp}} = \frac{464945}{5.998} \qquad = \mathbf{77{,}517}$$

Note that this value of 77,517 psi is less than 80,000 psi which the Yp value for N80. Thus, this value does not apply and is not the reduced joint tensile strength.

$$P_b = 0.95 * A_{jp} * [\frac{U_p - Y_p}{0.644} + Y_p - 218.15 * B * D\ ]$$

$$P_b = 0.95 * 5.998 * [\frac{100000 - 80000}{0.644} + 80000 - 218.15 * 20 * 7\ ]$$

= 458,782 lb (case#2)

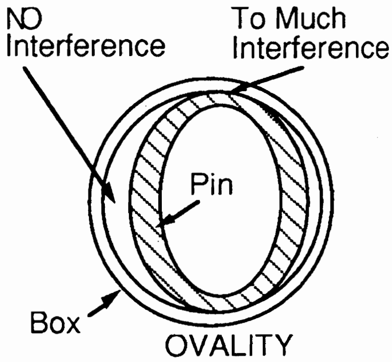
$$\frac{P_b}{A_{jp}} = \frac{527699}{6.899} \qquad = \mathbf{76{,}489\ psi}$$

Note that this value of 76,489 psi is less than 80,000 psi which is the Yp value for N80. Thus, the joint strength of this casing in a 20 degree per 100 foot dogleg is 458,782 lbs.

Thus, it has been ascertained that the maximum tensile load which this casing can bear is 458,782 lbs while it is in a 20 degree per 100 foot dogleg.

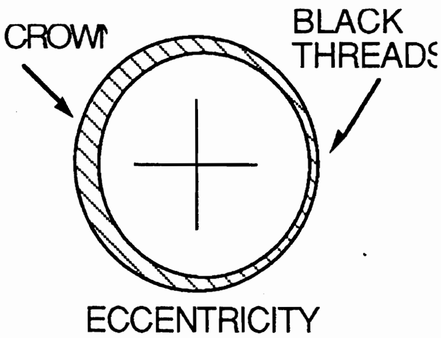
TUBULAR CONNECTIONS

Tubular connections have two functions: (1) support tension and compression loads and (2) contain fluids under pressure. Connections fall into two categories: API and proprietary. The API recognizes four connections



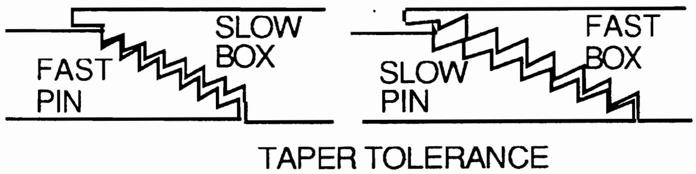
1. Round thread
   a. long
   b. short
2. Buttress thread
3. Extreme-line
4. Line pipe

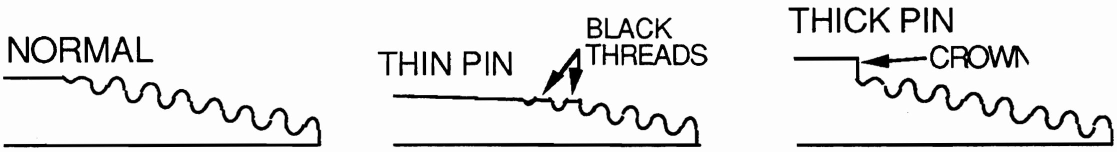
Loads are supported by thread dope seals the buttress thread which platings are tin, zinc,



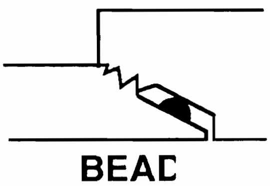
threads. API modified connections except for must be plated. Common and metallic phosphate.

The illustrations show some of the problems manufacturers have in making a connection. All of the API connections are classified as interference connections.





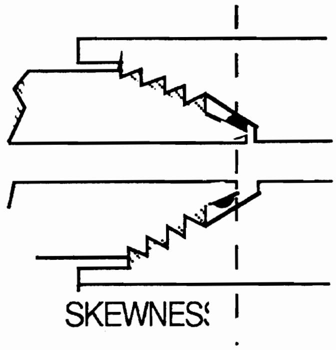
The bearing stress, which is a result of interference, is produced while torquing the connection and is the most critical aspect of making-up a connection. Bearing stress is the stress between the pin and collar. Proper bearing stress is achieved for an ideal connection by making buttress thread up to the base of a stamped triangle, and by making round thread up until the collar covers the last thread on the pin. Because there are few ideal connections, torque turn devices are popular.



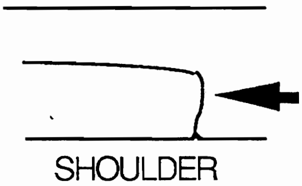
To prevent galling of the surfaces of the pin and collar, API recommends that make-up speeds do not exceed 25 rpm for all sizes of pipe. Others recommend that the surfaces not have a relative velocity of more than 50 feet per minute. The rpm's which corresponds to this relative velocity for 7 inches pipe is

$$\text{rpm} = \frac{50}{\pi\, 7} 12 \qquad = \mathbf{27.3\ rpm}$$

Proprietary connections use threads for supporting loads and one or more of the following for sealing the connection.

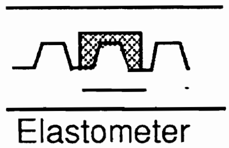


1. Elastomer which may act as O-ring or as packing
2. Bevelled shoulder and a bead
3. Interference of the threads on the pin and collar
4. Near square shoulders on the pin and collar



Except in unusual events only one of the multiple seals which are manufactured within the connection will be effective, and the others may or may not serve as back ups in cases where the effective seal begins to leak.

Galling, dirt, and seal damage are the primary causes of leaks. Gaining proper torque may also be a problem.



SLACK-OFF BENDING LOADS

Slacking off weight can cause the tubular being slacked-off to buckle. The bending stress at the location in the tubular at which the effective tension is being applied is given by Lubinski's equation

$$S_b = \frac{8}{\Pi} * \frac{OD * (H - OD) * T_{eff}}{OD^4 - ID^4}$$

Lubinski also published the following equation for the purpose of ascertaining whether the tubular is bent or buckled. He called this load which induces buckling the, CRITICAL BUCKLING LOAD.

$$F_{cbl} = 80.13 * [B_f^2 (OD^2 - ID^2)^3 * (OD^2 + ID^2)]^{1/3}$$

S_b	= bending stress in outer fiber; psi
OD	= outside diameter of tube; in
ID	= inside diameter of tube; in
H	= hole diameter containing the tube; in
T_{eff}	= effective tension; lb
B_f	= buoyancy factor
F_{cbl}	= critical buckling load; lb

EXAMPLE

20,000 pounds are slacked-off onto the bottom of the hole with 7" * 6.094" casing in a 36" drill hole which contains 14 ppg mud. Is the casing buckled and what is the stress caused by buckling?

$$B_f = 1 - \frac{14}{65.45} \qquad \text{(bouyancy factor)} \qquad = \mathbf{.786}$$

$$F_{cbl} = 80.13 * [.786^2 (7^2 - 6.094^2)^3 * (7^2 + 6.094^2)]^{1/3} \qquad = \mathbf{3{,}576\ lb}$$

$$S_b = \frac{8}{\Pi} * \frac{7 * (36 - 7) * 20000}{7^4 - 6.094^4} \qquad = \mathbf{10{,}118\ psi}$$

The tube **is buckled** and the bending stress alone **will not fail** N 80 steel.

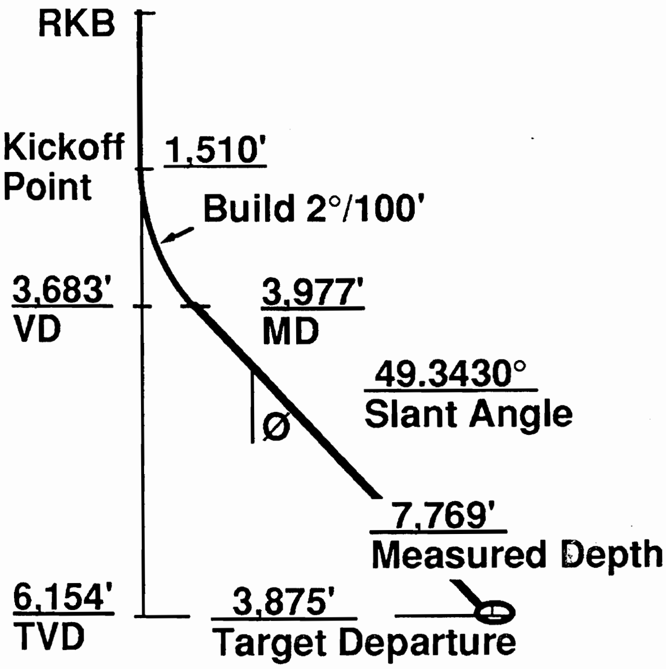
SURFACE AND DOGLEG RUNNING LOADS IN DIRECTIONAL AND VERTICAL HOLES

Tubulars have been broken and parted while being run. Axial tension loading, at any one point in the tube, may be at its highest intensity while that point is being lowered through the rotary table. Further, if a tube enters a dogleg, an additional bending load must be added to the tension load.

In directional holes, the true vertical length of a section of tube changes as the section passes from one position to another position. Thus, the tension at one particular point in the tube changes with its depth in a hole. Also, in a directional hole, the build section is a dogleg and the maximum tension load may occur while the top of a section of tube is in the top of the build. This is also true for a dogleg which has increasing inclination with depth. If the inclination angle is decreasing, most likely the maximum tension will occur while the tube is at the bottom of the dogleg.

EXAMPLE

Casing:	16"*15.124"*75ppf *7,769'
buoyed wt.:	64.2 ppf
Mud:	9.4 ppg
buoyancy factor:	0.8564
Directional:	
target departure:	3,875'
target tvd:	6,154'
build gradient:	2 °/100'
kop:	1510'



The following contains a table of values and their explanations.

1. Final resting depth (MD) of a point, 'X', in the casing after the casing has been run to total depth. Depths are selected to yield sufficient casing loadings for the purpose of selecting casing.

2. Length (MD) of a segment of casing hanging below the point of interest, 'X'. This length (MD) can not and will not change.

3. TVD of the bottom of the casing with the top of the segment at the surface. This is also the true vertical length of the casing for

computing surface running loads. Column (3) = Column (2) transposed.

4. Surface running load. It is also the buoyed weight of the casing as it is being run at the surface. Column (4) = Column (3) * wt per foot * buoyancy factor.

5. Length (MD) of casing which is hanging below the top of the dogleg. This is also the depth (MD) of the bottom of the casing with 'X' at the top of the dogleg.

6. TVD of the bottom of the casing with 'X' at the top of the dogleg. Column (6) = Column (5) transposed.

7. True vertical length of the casing segment below the top of the dogleg. Column (7) = Column (6) - dogleg depth (TVD).

8. Buoyed weight of the casing hanging below the dogleg. Column (8) = Column (7) * weight per foot * buoyancy factor.

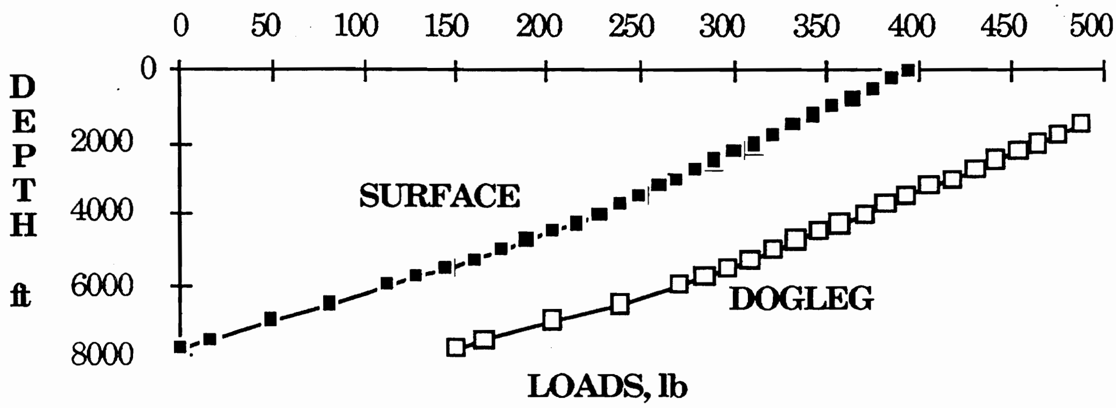
9. Lubinski's bending force is computed with the load in Column (8).

10. This is the 'continuous wall contact' bending force.

11. The dogleg running load is the sum of Column (8) and the larger of Column (9) or Column (10).

TABLE OF SURFACE & DOGLEG RUNNING LOADS IN A DIRECTIONAL WELL										
'X' IN THE CSG (1)	LENGTH BELOW POINT 'X' (2)	TVD WITH 'X'AT SURF (3)	WEIGHT WITH 'X'AT SURF (4)	MD WITH 'X'AT DGL (5)	TVD WITH 'X'AT DGL (6)	VERTICAL LENGTH OF SEGMENT (7)	buoyed WEIGHT IN DLG (8)	LUBINSKI BENDING FORCE 1000'S (9)	CIRCULAR BENDING LOAD 1000'S (10)	DOGLEG RUNNING LOAD 1000'S (11)
0	7769	6154	395261	--	--	--	--	--	--	--
500	7269	5828	374313	--	--	--	--	--	--	--
1000	6769	5502	353384	--	--	--	--	--	--	--
1510-	6259	5169	331997	7769	6154	4644	298277	--	--	--
1510+	6259	5170	332060	7769	6154	4644	298277	190	150	488
2000	5769	4851	311572	7279	5828	4318	277338	187	150	464
2500	5269	4525	290634	6779	5509	3999	256849	184	150	441
3000	4769	4199	269695	6279	5183	3673	235911	181	150	417
3500	4269	3873	248757	5779	4857	3347	214973	178	150	393
3977	3792	3542	227497	5302	4546	3036	194997	176	150	371
4500	3269	3160	202962	4779	4205	2695	173096	173	150	346
5000	2769	2729	175279	4279	3880	2370	152221	170	150	322
5500	2269	2260	145156	3779	3549	2039	130962	167	150	298
6000	1769	1769	113620	3279	3169	1659	106555	163	150	270
6500	1269	1269	81506	2779	2738	1228	78872	159	150	238
7000	769	769	49392	2279	2270	760	48814	155	150	204
7500	269	269	17277	1779	1779	269	17277	150	150	167
7515	254	254	15698	1754	1754	244	15698	150	150	166
7769	0	0	0	1510	1510	0	0	0	150	150

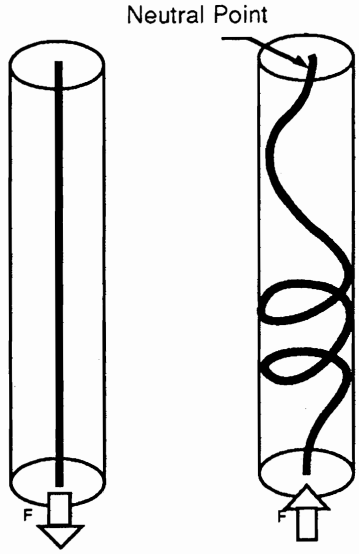
DOGLEG RUNNING & SURFACE RUNNING LOADS



The chart is a plot of the measured depth of the casing after being run in the drill hole (listed in column 1) in the above table versus the surface running load (listed in column 5 and the dogleg running load (listed in column 11)).

TUBING DESIGN

TUBING MOVEMENT AND STRESS COMPUTATIONS



Tubing designs are resolved with the paper written by **Lubinski, A., "HELICAL BUCKLING OF TUBING SEALED IN PACKERS"**, AIME TRANSACTIONS (Jornal of Petroleum Technology), JUNE, 1962.

ASSUMPTIONS

The following problem is one in which tubing is used to squeeze cement casing perforations. The following assumptions, which are standard in most tubing design methods, are made

1. the tubing fills with the fluid which is within the casing as it is run
2. no surface pressures are applied while running the tubing (no snubbing)
3. tubing is at geothermal temperatures immediately after running
4. initial length of tubing is as measured in the hole
5. movement of tubing depends only on the following
   a. slack-off
   b. pressure area forces
   c. gravitational attraction of steel
   d. ballooning
   e. fluid flow friction
   f. fictitious force (Lubinski's)
   g. temperature changes
6. stresses in tubing depend only on the following
   a. packer to tubing force
   b. internal and external tubing pressures
   c. bending
   d. gravitational attraction of steel
   e. temperature changes

Properties of Steel:

Poisson's ratio	**v = .3 in/in**
Adiabatic linear expansion coefficient	**b = 6.9E-6 in/in/F**
Modulus of Elasticity	**E = 30E+6 psi**

STEPS - WHAT TO DO?

1. Record the tubing O.D.; **D = 2.875 inch**
2. Record the tubing I.D.; **d = 2.441 inch**

3. Record the nominal weight of the tubing; W_s = **6.5 ppf**

4. Record the diameter of the hole; H = **6.094 inch**

5. Record the packer bore diameter; D_{pb} = **3.25 inch**

6. Record the depth of the packer; D_p = **10,000 ft**

7. Record the initial surface pressure within the tubing; p_{si} = **0.0 psi**

8. Record the initial surface pressure within the tubing annulus;

   p_{oi} = **0.0 psi**

9. Record the final surface pressure within the tubing;

   p_{if} = **5,000 psi** Δp_{if} = **5,000 psi**

10. Record the final surface pressure within the tubing annulus;

    p_{of} = **1,000 psi** Δp_{of} = **1,000 psi**

11. Record the initial API gravity of the fluid within the well;

    API = **30.0 °API**

12. Record the final weight of the fluid within the tubing; MW_{if} = **15.0 ppg**

13. Record the force slacked-off(+)/picked-up(-) on to packer; F_p = **20,000 lb**

14. Record the initial temperature of the tubing; T_i = **150°F**

15. Record the final temperature of the tubing; T_f = **130°F**

16. Compute the outside end area of the tubing; sq in.

    $A_o = .7854 * D^2 = .7854 * \text{STEP } 1^2 = .7854 * 2.875^2$ **= 6.49 sq in**

17. Compute the inside end area of the tubing; sq in.

    $A_i = .7854 * d^2 = .7854 * \text{STEP } 2^2 = .7854 * 2.441^2$ **= 4.68 sq in**

18. Compute the nominal cross-sectional area of the steel of the tubing; sq in.

$$A_s = .7854 * (D^2 - d^2) = .7854 * (\text{STEP } 1^2 - \text{STEP } 2^2)$$

$$= .7854 * (2.875^2 - 2.441^2) \qquad \mathbf{= 1.81\ sq\ in}$$

19. Compute the ratio of the tubing O.D. to the tubing I.D.; in./in.

$$R = \frac{O.D.}{I.D.} = \frac{\text{STEP 1}}{\text{STEP 2}} = \frac{2.875}{2.441} \qquad \mathbf{= 1.178\ in/in}$$

20. Compute the weight per inch of steel in the wall of the tubing; lb/in.

$$w_s = \frac{W_s}{12.0} = \frac{\text{STEP 3}}{12.0} = \frac{6.5}{12.0} \qquad \mathbf{= .542\ lb/in}$$

21. Compute the moment of inertia of the tubing; in^4

$$I = \frac{\pi}{64} * (D^4 - d^4) \qquad = .0491 * (\text{STEP } 1^4 - \text{STEP } 2^4)$$

$$= \frac{\pi}{64} * (2.875^4 - 2.441^4) \qquad \mathbf{= 1.61\ in^4}$$

22. Compute the radial clearance; in.

$$r = \frac{H - D}{2} = \frac{\text{STEP 4 - STEP 1}}{2} = \frac{6.094 - 2.875}{2} \qquad \mathbf{= 1.61\ in}$$

23. Compute the depth of the packer; in.

$$L_p = D_p * 12 \qquad = \text{STEP } 6 * 12 \qquad = 10{,}000 * 12 \qquad \mathbf{= 120{,}000\ in}$$

24. Compute the cross-sectional area of the packer bore; sq. in.

$$A_p = \frac{\pi}{4} * D_{pb}^{\ 2} = \frac{\pi}{4} * \text{STEP } 5^2 = \frac{\pi}{4} * 3.25^2 \qquad \mathbf{= 8.30\ sq\ in}$$

25. Compute the initial specific gravity of the fluid within the well; ppg.

$$sg_{oil} = \frac{141.5}{131.5 + \text{`API}} = \frac{141.5}{131.5 + \text{STEP 11}} = \frac{141.5}{131.5 + 30} \qquad \mathbf{= .8762}$$

26. Compute the initial fluid weight within the tubing; ppg.

$$MW_{ii} = sg_{oil} * \text{weight of water @ } 60^0\text{F (lb/gal)}$$

$$= \text{STEP } 25 * 8.337 \qquad = .8762 * 8.337 \qquad \mathbf{= 7.30\ ppg}$$

27. Compute the initial fluid weight within the annulus; ppg.

$MW_{oi} = sg_{oil}$ * weight of water @ 60`F (lb/gal)

= 7.30 ppg (same steps as in STEP 26)

28. Compute the final fluid weight within the annulus; ppg.

$MW_{of} = sg_{oil}$ * weight of water @ 60`F (lb/gal)

= 7.30 ppg (same steps as in STEP 26)

29. Compute the change in the fluid weight within the tubing; ppg.

$\Delta MW_i = MW_{if} - MW_a$ = STEP 12 - STEP 26 = 15.0 - 7.3 **= 7.7 ppg**

30. Compute the change in the fluid weight within the annulus; ppg.

$\Delta MW_o = MW_{of} - MW_{oi}$ = STEP 28 - STEP 27 = 7.30 - 7.30 **= 0.0 ppg**

31. Compute the initial pressure within the tubing at the depth of the packer; psi.

$P_{ii} = .052 * MW_{ii} * D_p + p_{si}$ = .052 * STEP 26 * STEP 6 + STEP 7

= .052 * 7.30 * 10,000 + 0.0 **= 3,796 psi**

32. Compute the initial pressure within the annulus at the depth of the packer; psi.

$P_{oi} = .052 * MW_{oi} * D_p + p_a$ = .052 * STEP 27 * STEP 6 + STEP 8

= .052 * 7.30 * 10,000 + 0.0 **= 3,796 psi**

33. Compute the final pressure within the tubing at the depth of the packer; psi.

$P_{if} = .052 * MW_{if} * D_p + p_{if}$ = .052 * STEP 12 * STEP 6 + STEP 9

= .052 * 15.0 * 10,000 + 5,000 **= 12,800 psi**

34. Compute the final pressure within the annulus at the depth of the packer; psi.

$P_{of} = .052 * MW_{of} * D_p + p_{of}$ = .052 * STEP 28 * STEP 6 + STEP 10

$= .052 * 7.30 * 10{,}000 + 1{,}000$ **= 4,796 psi**

35. Compute the change in pressure within the tubing at the depth of the packer; psi.

$\Delta P_{if} = P_{if} - P_{ii}$ = STEP 33 - STEP 31 = 12,800 - 3,796 **= 9,004 psi**

36. Compute the change in pressure within the annulus at the depth of the packer; psi.

$\Delta P_{of} = P_{of} - P_{oi}$ = STEP 34 - STEP 32 = 4,796 - 3,796 **= 1,000 psi**

37. Compute the fictitious force on the end of the tubing; lb.

$F_f = A_p * (P_{if} - P_{of})$ = STEP 24 * (STEP 33 - STEP 34)

$= 8.30 * (12{,}800 - 4{,}796)$ **= 66,433 lb**

38. Compute the change in the fictitious force; lb.

$\Delta F_f = A_p * (\Delta P_{if} - \Delta P_{of})$ = STEP 24 * (STEP 35 - STEP 36)

$= 8.30 * (9004 - 1000)$ **= 66,433 lb**

39. Compute the real pressure area force on the end of the tubing; lb.

$F_a = (A_p - A_i) * P_{if} - (A_p - Aa) * P_a$

= (STEP 24-STEP 17)*STEP 33 - (STEP 24-STEP 16)*STEP 34

$= (8.30 - 4.68) * 12{,}800 - (8.30 - 6.49) * 4{,}796$ **= 37,655 lb**

40. Compute the change in the real pressure area force on the end of the tubing; lb.

$\Delta F_a = (A_p - A_i) * \Delta P_{if} - (A_p - A_o) * \Delta P_{of}$

= (STEP 24-STEP 17)*STEP 35 - (STEP 24-STEP 16)*STEP 36

$= (8.30 - 4.68) * 9004 - (8.30 - 6.49) * 1000$ **= 30,784 lb**

41. Compute the final weight per inch of the fluid within the tubing; lb/in.

$w_i = MW_{if} * A_i / 231.0$ = STEP 12 * STEP 17 / 231.0

$= 15.0 * 4.68 / 231.0$ **= .3039 lb/in**

42. Compute the final weight per inch of the fluid displaced by the tubing; lb/in.

$$w_o = MW_{of} * A_o / 231.0 \quad = \text{STEP 26} * \text{STEP 16} / 231.0$$

$$= 7.30 * 6.49 / 231.0 \qquad \mathbf{= .2051\ lb/in}$$

43. Compute the buoyed weight of the tubing at final conditions; lb/in.

$$w_f = w_s + w_i - w_o \quad = \text{STEP 20} + \text{STEP 41} - \text{STEP 42}$$

$$= .5420 + .3039 - .2051 \qquad \mathbf{= .6408\ lb/in}$$

44. Compute the distance from the lower end of the tubing to the neutral point; in.

$$n = \frac{F_f}{w_f} = \frac{\text{STEP 37}}{\text{STEP 43}} = \frac{66433}{.6408} \qquad \mathbf{= 103{,}672\ in\ or\ 8{,}639\ ft}$$

45. Compute the change in tubing length caused by **Hooke's law (STRETCH)**; inches.

$$\Delta L_1 = -\frac{L_p * DF_a}{E * A_s} = -\frac{\text{STEP 23} * \text{STEP 40}}{E * \text{STEP 18}} = -\frac{120000 * 30784}{30E{+}6 * 1.81} \qquad \mathbf{= -68.03\ in}$$

46. Compute the change in tubing length caused by **buckling**; in.

$$\Delta L_2 = -\frac{r^2 * A_p^{\ 2} * (\Delta P_{if} - \Delta P_{of})^2}{8 * E * I * w_f}$$

$$= -\frac{\text{STEP } 22^2 * \text{STEP } 24^2 * (\text{STEP 35} - \text{STEP 36})^2}{8 * E * \text{STEP 21} * \text{STEP 43}}$$

$$= -\frac{1.61^2 * 8.30^2 * (9004{-}1000)^2}{8 * 30E{+}6 * 1.61 * .6408} \qquad \mathbf{= -46.20\ in}$$

47. Enter the Pressure Friction Loss because of fluid motion P_f; psi/in (flow direction: +down/-up).

$$P_f \qquad \mathbf{= .0083\ psi/in}$$

48. Compute the interim variable ρ_{ii}; lb/in^3.

$$\rho_{ii} = \frac{MW_{ii}}{231} = \frac{\text{STEP 26}}{231} = \frac{7.30}{231} \qquad \mathbf{= .0316\ lb/in^3}$$

49. Compute the interim variable ρ_{if}; lb/in^3.

$$\rho_{if} = \frac{MW_{if}}{231} = \frac{\text{STEP 12}}{231} = \frac{15.0}{231} \qquad \mathbf{= .0649\ lb/in^3}$$

50. Compute the interim variable $\Delta\rho_i$; lb/in^3.

$$\Delta\rho_i = \rho_{if} - \rho_{ii} = \text{STEP 49 - STEP 48} = .0649 - .0316 \qquad \mathbf{= .0333\ lb/in^3}$$

51. Compute part #1 of the **ballooning** effect; in.

$$\text{Part \#1} = -\frac{2v}{E} * \frac{\Delta p_{if} - R^2 * \Delta p_{of}}{R^2 - 1} * L$$

$$= -\frac{2 * .3}{30E+6} * \frac{\text{STEP 9 - STEP 19}^2 * \text{STEP 10}}{\text{STEP 19}^2 - 1} * \text{STEP 23}$$

$$= -\frac{2 * .3}{30E+6} * \frac{5000 - 1.178^2 * 1000}{1.178^2 - 1} * 120000 \qquad \mathbf{= -22.36\ in}$$

52. Compute part #2 of the **ballooning** effect; in.

$$\text{Part \#2} = -\frac{v}{E} * \frac{\Delta\rho_i - R^2\,\Delta\rho_o - (\frac{1+2v}{2v}) * P_f}{R^2 - 1} * L_p^{\,2}$$

$$= -(.3/30E+6) * \frac{\text{STEP 50 - 0 - }(\frac{1+.6}{.6}) * \text{STEP 47}}{\text{STEP 19}^2 - 1} * \text{STEP 23}^2$$

$$= -\frac{.3}{30E+6} * \frac{.0333 - (1+.6/.6) * .0083}{1.178^2 - 1} * 120{,}000^2 + (-22.36) \qquad \mathbf{= -4.15\ in}$$

Tubing **ballooning** effect change = STEP 51 + STEP 52 **= - 26.51 in**

ΔL_3 **= - 26.51 in**

53. Compute the change in **temperature** of the tubing; °F

$$\Delta T = T_f - T_i = \text{STEP 15 - STEP 14} = 130 - 150 \qquad \mathbf{= -20\ ^\circ F}$$

54. Compute the change in tubing length caused by the temperature expansion/contraction effect; in.

$\Delta L_4 = L_p * b * \Delta T \qquad = \text{STEP 23} * b * \text{STEP 53}$

$= 120{,}000 * 6.9E\text{-}6 * (-20)$ **= - 16.56 in**

55. Compute the required **seal length** for a free motion packer; inch

$\Delta L = \Delta L_1 + \Delta L_2 + \Delta L_3 + \Delta L_4$

$= \text{STEP 45} + \text{STEP 46} + \text{STEP 52} + \text{STEP 54}$

$= (-68.03) + (-46.20) + (-26.51) + (-16.56)$ **= -157.30 in**

56. Compute the virtual tubing length change caused by the packer force effect; in.

$$\Delta L_p = \frac{L_p F_p}{E A_s} + \frac{r^2 F_p^{\,2}}{8 E I w_f}$$

$$= \frac{\text{STEP 23} * \text{STEP 13}}{E * \text{STEP 18}} + \frac{\text{STEP } 22^2 * \text{STEP } 13^2}{8 * E * \text{STEP 21} * \text{STEP 43}}$$

$$= \frac{120000 * 20000}{30E{+}6 * 1.81} + \frac{1.61^2 * 20000^2}{8 * 30E{+}6 * 1.61 * .6408}$$ **= 48.39 in**

57. Compute the required **seal length if the tubing locator lifts off** of the packer; in.

$\Delta L_6 = \Delta L + \Delta L_p \quad = \text{STEP 55} + \text{STEP 56} \quad = -157.30 + 48.39$ **= -108.91 in**

IF THE TUBING IS FIXED TO PACKER OR TUBING DOES NOT LIFT OFF

58. Plot a force-length change curve as shown by using several values of F and computing ΔL with the following equations.

$L_p\, w_f = 120{,}000 * 0.6408$ **= 76,896 lb**

EQN 1 $\quad \Delta L_a = \dfrac{L_p F_p}{E A_s} \qquad F_p \le 0$

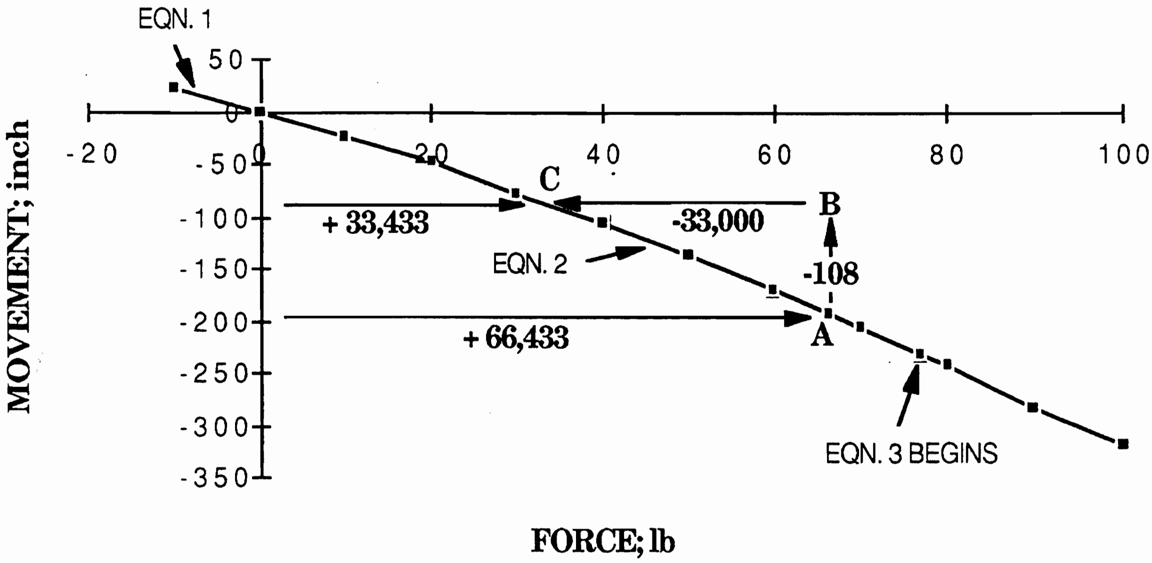
$$\text{EQN 2} \quad \Delta L_p = \frac{L_p F_p}{E A_s} + \frac{r^2 F_p^{\,2}}{8 E I w_f} \qquad 0 < F_p \leq L_p w_f$$

$$\text{EQN 3} \quad \Delta L_p = \frac{L_p F_p}{E A_s} + \frac{r^2 F_p^{\,2}}{8 E I w_f} * \left[\frac{L_p w_f}{F_p}\left(2 - \frac{L_p w_f}{F_p}\right)\right] \qquad F_p > L_p w_f$$

FORCE LENGTH CURVE

F	dL_p	F	dL_p
-20,000	+ 40.0	+60,000	-170.1
0	0	+70,000	-205.8
+10,000	- 23.1	+80,000	-243.6
+20,000	- 48.3	+90,000	-282.0
+30,000	- 75.6	+100,000	-319.9
+40,000	-104.9	+110,000	-358.0
+50,000	-136.5		

FORCE - LENGTH TUBING PLOT



59. 'A' is located by intersecting the Force-Length Change curve with the value of F_f (STEP 37), + 66,433 lb.

60. 'B' is located by moving a vertical distance equal to dL_6 (STEP 57) up or down from point 'A'. The direction to move is found from the sign of dL_6. If it is negative,then move up; if it is positive, then move down, up 108.91 in.

61. 'C' is located by drawing a horizontal line from 'B' to the curve.

62. Line 'BC', 33,000 lb, is the new packer force, F_p'. The change in packer force is caused by the changes in temperature, pressure, and fluid density within the tubing.

F_p' **= -33,000 lb.**

63. Compute the new fictitious force; lb.

$F_f' = F_f + F_p'$ = STEP 37 + STEP 62 = 66,433 + (-33,000) **= 33,433 lb**

64. Compute the new pressure area force on the end of the tubing; lb.

$F_a' = F_a + F_p'$ = STEP 39 + STEP 62 = 37,655 + (-33,000) **= 4,655 lb**

65. Compute the axial stress; psi.

$s_a = F_a' / A_s$ = STEP 64 / STEP 18 = 4,655 / 1.81 **= 2,572 psi**

66. Compute the bending stress; psi.

$$s_b = \frac{D * r * F_f'}{4 * I} = \frac{\text{STEP 1 * STEP 19 * STEP 63}}{\text{4 * STEP 21}}$$

$$= \frac{2.875 * 1.178 * 33433}{4 * 1.61}$$ **= 17,582 psi**

67. Compute the triaxial stress at the outer diameter of the tubing wall (tension and compression); lb.

$$s_{o_1} = \left[3\left(\frac{P_{if} - P_{of}}{R^2 - 1}\right)^2 + \left(\frac{P_{if} - R^2 P_{of}}{R^2 - 1} + s_a + s_b\right)^2\right]^{.5}$$

$$=[3\,(\frac{\text{STEP 33 - STEP 34}}{\text{STEP 19}^2 - 1})^2 + (\frac{\text{STEP 33 - STEP 19}^2 * \text{STEP 34}}{\text{STEP 19 - 1}} + \text{STEP 65 + STEP 66})^2]^{.5}$$

$$=[3\,(\frac{12800 - 4796}{1.178^2 - 1})^2 + (\frac{12800 - 1.178^2 * 4796}{1.178^2 - 1} + 2572 + 17582)^2]^{.5}$$

= 50,744 psi

$$S_{o_2} = \left[3\left(\frac{P_{if} - P_{of}}{R^2 - 1}\right)^2 + \left(\frac{P_{if} - R^2 P_{of}}{R^2 - 1} + s_a - s_b\right)^2\right]^{.5}$$

$$=[3\,(\frac{\text{STEP 33 - STEP 34}}{\text{STEP 19}^2 - 1})^2 + (\frac{\text{STEP 33 - STEP 19}^2 * \text{STEP 34}}{\text{STEP 19 - 1}} + \text{STEP 65 - STEP 66})^2]^{.5}$$

$$=[3\,(\frac{12800 - 4796}{1.178^2 - 1})^2 + (\frac{12800 - 1.178^2 * 4796}{1.178^2 - 1} + 2572 - 17582)^2]^{.5}$$

= 36,842 psi

68. Compute the triaxial stress at the inner diameter of the tubing wall (tension and compression); lb.

$$S_{i_1} = \left[3\left(\frac{R^2(P_{if} - P_{of})}{R^2 - 1}\right)^2 + \left(\frac{P_{if} - R^2 P_{of}}{R^2 - 1} + s_a + \frac{s_b}{R}\right)^2\right]^{.5}$$

$$=[3(\frac{\text{STEP 19}^2(\text{STEP 33-STEP 34})}{\text{STEP 19}^2\text{-1}})^2 + (\frac{\text{STEP 33-STEP 19}^2 * \text{STEP 34}}{\text{STEP 19 - 1}} + \text{STEP 65} + \frac{\text{STEP 66}}{\text{STEP 19}})^2]^{.5}$$

$$=[3\,(\frac{1.178^2(12800 - 4796)}{1.178^2 - 1})^2 + (\frac{12800 - 1.178^2 * 4796}{1.178^2 - 1} + 2572 + \frac{17582}{1.178})^2]^{.5}$$

= 59,787 psi

$$S_{i_2} = [3\,(\frac{R^2\,(P_{if} - P_{of})}{R^2 - 1})^2 + (\frac{P_{if} - R^2 P_{of}}{R^2 - 1} + s_a - \frac{s_b}{R})^2]^{.5}$$

$$=[3(\frac{\text{STEP 19}^2(\text{STEP 33-STEP 34})}{\text{STEP 19}^2\text{-1}})^2 + (\frac{\text{STEP 33-STEP 19}^2 * \text{STEP 34}}{\text{STEP 19 - 1}} + \text{STEP 65} - \frac{\text{STEP 66}}{\text{STEP 19}})^2]^{.5}$$

$$=[3\,(\frac{1.178^2(12800 - 4796)}{1.178^2 - 1})^2 + (\frac{12800 - 1.178^2 * 4796}{1.178^2 - 1} + 2572 - \frac{17582}{1.178})^2]^{.5}$$

= 49,746 psi

69. Pick the largest value computed in steps 67 and 68. This is the required minimum yield value of the steel in the tubing.

Required minimum yield value **= 59,787 psi**

70. Ascertain if the initial slack-off, F_p, yields the steel in the tubing.

$$YP > \text{abs}\left[\frac{F_p}{A_s} + \frac{D * r * F_p}{4 * I}\right]$$

$$> \text{abs}\left[\frac{\text{STEP 13}}{\text{STEP 18}} + \frac{\text{STEP 1*STEP 22*STEP 13}}{\text{4*STEP 21}}\right]$$

$$> \text{abs}\left[\frac{20000}{1.81} + \frac{2.875 * 1.61 * 20000}{4 * 1.61}\right]$$

NO Yield, if Yield of steel is **> 25,425 lb**

DRILLPIPE DESIGN

INTRODUCTION

Drillpipe may fail because of stresses induced by one or a combination of the following phenomena.

1. Tension load
   - (a) Gravity
   - (b) Pick up
   - (c) Temperature
2. External pressure (collapse)
3. Internal pressure
   - (a) Burst
   - (b) Leaking of joint
4. Bending
   - (a) Shear
   - (b) Equivalent tension
5. Fatigue
   - (a) Rotation in doglegs
6. Crushing
   - (a) Slips
   - (b) Stacking
7. Torsion (twisting)
   - (a) Twist off in body
   - (b) Over make up of joint
   - (c) Unscrewing (ratcheting)
8. Buckling (stability)
   - (a) Slack off
   - (b) Temperature rise
   - (c) Pressure rise
   - (d) Fluid density rise
9. Acceleration (hard braking)
10. Abrasion (wear)
    - (a) Wall of tube
    - (b) Tooljoints
11. Erosion (fluid wear)
12. Corrosion (H_2S, CO_2, O_2)

TAPERED DRILLPIPE STRINGS

A tapered string contains more than one OD, tooljoint, weight, grade, or levels of wear.

The problem with tapered strings is ascertaining how much weight can be placed on the weight indicator without failing any one of the sections of the tapered string.

COMBINED TENSION, TORSION, BENDING, & PRESSURE LOADS

Combined tension, torsion, bending, and pressure loads are frequently placed on drillpipe. The normal drilling operation of making hole does this. However, the damaging combined loads will most likely occur while fishing. The tension is created by the hook pulling on the drillpipe; the torsion arises while twisting the pipe with either tongs or the rotary table; the bending is derived either from doglegs in the drill hole or the misalignment of the crown block with the rotary table; and pressures are created by surface pressures or different fluid weights on the inside and the outside of the drillpipe.

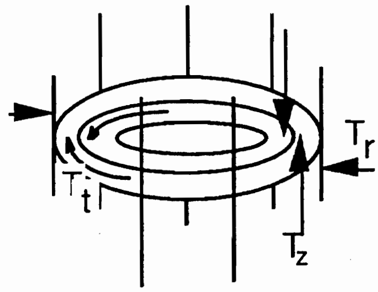
Von Mises triaxial equation depicts the combined stress. This stress is also known as general stress. If failure is forecasted by the equation it is known as general failure, because the mode of failure is not forecasted. For example if drillpipe is subjected to tension and pressure simultaneously and it fails, it would not be possible to ascertain from Von Mises equation whether it was the tension or the pressure which caused the failure.

The fundamental approach to drillpipe selection is to compute all the loads separately to determine if the strength of the drillpipe is exceeded by the individual loads and then combine the individual loads with Von Mises equation to ascertain if general failure is likely.

VON MISES STRESS

von Mises equation is

$$2\,V^2 = \left[(S_a - S_r)^2 + (S_t - S_r)^2 + (S_a - S_t)^2\right] + 6\left[T_t^{\,2} + T_r^{\,2} + T_z^{\,2})\right]$$



V = von Mises stress or general stress; psi
S_a = axial stress in the wall of the drillpipe; psi
S_r = radial stress in the wall of the drillpipe; psi
S_t = tangential stress in the wall of the drillpipe; psi
T_t = tangential shear stress normal to the longitudinal axis of the drillpipe; psi
T_r = radial shear stress normal to the longitudinal axis of the drillpipe; psi
T_z = axial shear stress parallel to the radial axis of the drillpipe; psi

The formula for computing the external diameter and cross sectional area of worn drillpipe is the following. The API specifies that class 1 drillpipe may have as much as 20% of its external wall worn away; i.e. 80% will be remaining.

It should be noted that much of the drillpipe will have all of its wall remaining.

EXAMPLE

Compute the worn diameter and the remaining minimum wall area of 5" x 4.276" 19.5 ppf API class 1 drillpipe.

$D_w = (0.8\ D_n) + (0.2\ d)$ no wear on ID

$D_w = (0.8 * 5) + (0.2 * 4.276)$ **= 4.86 inch**

The worn wall area is

$$A_w = \frac{\pi}{4}(D_w{}^2 - d^2)$$

$$A_w = \frac{\pi}{4}(4.86^2 - 4.276^2) \qquad \mathbf{= 4.190\ inch^2}$$

D_n = API nominal outside diameter; inch
D_w = worn diameter; inch
d = API nominal internal diameter; inch

AXIAL STRESS

The axial tension and tensile stress is found with free bodies and Newton's laws of motion. Thereafter, overpull and equivalent bending stress is added.

$$S_a = \frac{T_{real}}{A_w} + \frac{OP}{A_w} + \frac{F_{Lub}}{A_w}$$

REAL TENSION

T_{real}, at the rotary table, is the sum of the hook load and the internal pressure area force acting on the end of the drillpipe. At any other depth T_{real} is the value at the rotary table less the air weight of the drillpipe between the rotary table and depth where T_{real} is desired. The equations are the following.

AT THE ROTARY TABLE

$$T_{real} = HOOKLOAD + P_{surface} * A_i$$

AT ANY OTHER DEPTH

$$T_{real} = HOOKLOAD + P_{surface} * A_i - W_{air} * DEPTH$$

EXAMPLE DRILLPIPE TENSION

The weight indicator shows 230,000 lbs. The block and lines weight 30,000 lbs. A pump pressure gauge in line with the drillpipe shows 7,000 psi. The drillpipe is 5" by 4.276" by 19.5 ppf and the average weight per foot of the steel in the drillpipe is 21.4 lb/ft. What is the real tension in the drillpipe at the rotary table and at a depth of 5,000 feet?

The inside end area of 5" drillpipe = $(\pi/4)\, 4.276^2$ = **14.360 sq. in.**

$T_{real} = 230{,}000 - 30{,}000 + 7{,}000 * 14.360$ = **300,520 lb** (rotary table)

$T_{real} = 300{,}520 - 21.4 * 5{,}000$ = **193,520 lb** (at 5,000 ft)

DRAG LOADS

Drag loads may be measured with the weight indicator or estimated with equations.

MEASURED DRAG

The driller, as a standard practice, routinely measures three drillstring loads:

1. Free rotating weight. It is the weight of the drill string with the string off bottom and while the string is being rotated. The rotary speed should be above 35 rpm.

2. Pick-up weight. It is the weight of the drill string while raising the string at a normal working speed. Most drillers raise the string too slowly. Be careful not to add in the load of accelerating the drill string and breaking gel of the mud.

3. Slack-off weight. It is the weight of the drillstring while lowering the string at a normal working speed. The above potentials for error must be once again avoided.

The differences in the pick-up and slack-off weights from the free rotating weight are the pick-up drag and the slack-off drag.

EXAMPLE DRAG

The free rotating weight of a drill string is 200,000 lbs, the pick-up weight is 250,000 lbs, and the slack-off weight is 170,000 lbs. What are the pick-up and slack-off drag loads?

Pick-up drag = 250,000 - 200,000 = **50,000 lb**

Slack-off drag = 200,000 - 170,000 = **30,000 lb**

ESTIMATION OF DRAG

In a hole which is free of severe doglegs the tension drag may be estimated with the coefficient of friction method.

The method requires that the coeficient of friction be multiplied by the product of the buoyed weight and departure length of and for each section of the drill string and then summed. Drillers in the North Sea have measured coefficient of friction between 0.2 and 0.3. In the example which follows a value of the coefficient of friction of 0.28 was chosen.

The equation for tension drag is the following.

$$\text{Tension drag} = \Sigma\, (\mu * W_{buoyed} * L_{dep})$$

EXAMPLE TENSION DRAG

The drill string is composed of three sections: 1,500' of 5" drillpipe which weighs 21.4 lb/ft, 1,000 feet of 5" by 50 lb/ft heavy weight, and 100 feet of 8" by 147 lb/ft drill collars. The heavy weight and drill collars are in a 55 degree hole. The departure length of the drillpipe is 1,500 feet. The mud weight is 12.0 ppg. What is the estimated tension drag?

The buoyant factor is $1 - \frac{12}{65.45}$ = **0.817**

The departure factor for the 55 degree hole is sin(55) = **0.819**

The tension drags are

Drill collars	= 0.28 * 0.817 * 147 * 100 * 0.819	= **2,753 lb**
Heavy wt.	= 0.28 * 0.817 * 50 * 1,000 * 0.819	= **9,365 lb**
Drillpipe	= 0.28 * 0.817 * 21.4 * 1,500	= **7,343 lb**
Total tension drag for the drill string		= **19,461 lb**

PRESSURE LOADS

The radial and tangential stresses account for the pressure acting on the inner and outer walls of the drillpipe and are computed with Lames' equations.

$$S_r = \frac{d^2 P_i - D^2 P_e}{D^2 - d^2} - \frac{d^2 D^2 (P_i - P_a)}{b^2 (D^2 - d^2)}$$

$$S_t = \frac{d^2 P_i - D^2 P_e}{D^2 - d^2} + \frac{d^2 D^2 (P_i - P_e)}{b^2 (D^2 - d^2)}$$

P_i = pressure acting within the drillpipe; psi
P_e = pressure acting outside the drillpipe; psi
b = any diameter within the wall of the drillpipe; inch

Diameters and pressures are always positive in the two Lames' equations. As always compressional stresses are negative and tensile stresses are positive.

TORSIONAL LOADS

TANGENTIAL STRESS

Two formulae are available for computing the tangential torsional shear stress. The first uses the number of turns of the drillpipe while the second uses the torque applied to the drillpipe.

Number of turns formula $\quad T_t = \frac{\pi G d N}{12 L}$

Torque formula $\quad T_t = 6 d \frac{Q}{J}$

and eliminating T_t from the two formulae relates Q and N

$Q = \frac{\pi G J N}{72 L}$
G = shear modulus of steel, $12 * 10^6$; psi
N = number of turns in the free drillpipe; rev
Q = torque applied to the drillpipe; lb-ft
L = length of drillpipe; ft
J = polar moment of inertia, $\frac{\pi}{32}(D^4 - d^4)$; inch^4

AXIAL STRESS

The equation for axial shear stress which is associated with bending of the drillpipe is

$$T_z = \frac{T_{real}}{A_s} * \sin(15\ C)$$

A_s = Cross-sectional area of steel in wall of pipe; in^2

RADIAL STRESS

T_r is equal to zero for most drillpipe load conditions.

BENDING STRESS

Lubinski's bending stress formula is the preferred formula for computing bending stresses in drillpipe derived from the curvature of the drill hole.

$$F_{LUB} = \frac{3385 * d * C * \sqrt{T_{eff}} * \sqrt{\dfrac{D^2 - d^2}{D^2 + d^2}}}{A_S * \tanh\left[0.2 * \sqrt{\dfrac{T_{eff}}{D^4 - d^4}}\right]}$$

$$\tanh(x) = \frac{e^{2x} - 1}{e^{2x} + 1}$$

C = tube curvature; °/ foot

EXAMPLE

5" * 4.276" * 21.3 ppf S-135 drillpipe became stuck at a depth of 9,000 ft. In an attempt to get loose a hook load of 300,000 lbs is pulled, the pressure within the drillpipe is increased to 3,000 psi, and the drillpipe is twisted 9 revolutions. A dogleg with a severity of 5°/100ft exists at a depth of 1,000 feet. Mud weight is 11.0 ppg. Will the 5" drillpipe yield and most likely fail?

von Mises stress must be resolved. The internal and external pressures at the depth of 1000 feet are

$$P_{i\,1000} = \frac{11 * 1000}{19.25} + 3000 \qquad \mathbf{= 3571\ psi}$$

$$P_{o\,1000} = \frac{11 * 1000}{19.25} + 0 \qquad \mathbf{= 571\ psi}$$

$A_i = \frac{\pi}{4} * 4.276^2$ **$= 14.36\ in^2$**

$A_o = \frac{\pi}{4} * 5^2$ **$= 19.63\ in^2$**

$A_s = A_o - A_i$ **$= 5.275\ in^2$**

Curvature of the drillpipe **= .05 °/ft**

Buoyancy factor $= 1 - \frac{11}{65.45}$ **= .83**

Buoyed weight per foot = .83 * 21.3 **= 17.72 lb/ft**

$Tension_{1000} = 300{,}000 + 3000 * 14.36 - 21.3 * 1000$ **= 321,781 lb**

$Tensile\ stress_{1000} = \frac{321781}{5.27}$ **= 61,005 psi**

$Effective\ tension_{1000} = 321781 - 3571*14.36 + 571*19.63$ **= 281,710 lb**

$$Radial\ stress_{1000} = \frac{4.276^2 * 3571 - 5^2 * 571}{5^2 - 4.276^2} - \frac{4.276^2 * 5^2(3571 - 571)}{4.276^2\,(5^2 - 4.276^2)}$$

= -3,571 psi

$$Tang'tl\ stress_{1000} = \frac{4.276^2 * 3571 - 5^2 * 571}{5^2 - 4.276^2} + \frac{4.276^2 * 5^2(3571 - 571)}{4.276^2\,(5^2 - 4.276^2)}$$

= 19,335 psi

$Tang'tl\ shear_{1000} = \frac{\pi\ 12E6 * 4.276 * 9}{12\ 9000}$ **= 13,433 psi**

$Axial\ shear_{1000} = \frac{321781}{5.27} * \sin(15\ .05)$ **= 798 psi**

$$\text{Bending stress}_{1000} = \frac{3385 * 4.276 * .05 * \sqrt{281714} * \sqrt{\frac{5^2 - 4.276^2}{5^2 + 4.276^2}}}{5.275 * \tanh\left[0.2 * \sqrt{\frac{281714}{5^4 - 4.276^4}}\right]}$$

= 28,746 psi

von Mises stress_{1000}

$$= \sqrt{.5[(61005+28746-(-3000))^2 + (19335-(-3000))^2 + (61005+28746-19335)^2] + 3[13433^2 + 798^2 + 0]}$$

= 87,497 psi

Thus, the S-135 grade drillpipe will not fail. A design factor for the drillpipe at a depth of 1,000 feet is

$$DF = \frac{135000}{87497}$$

= 1.54

SLIP CRUSHING OF TUBULARS

Reinhold, Spiri, and Vreeland published the following equation and table which relates the tube crushing tangential stress derived from slips and axial tensile loads. The steel in the pipe will be crushed when and if S_h is equal to the unit yield strength, YP, of the steel.

$$S_h = S_t \left[1 + \frac{DK}{2L} + \left(\frac{DK}{2L}\right)^2\right]^{1/2}$$

S_h	=	tangential stress in the pipe derived from the crushing action of slips and the effective tension in the pipe; psi
S_t	=	axial effective tensile stress in the pipe at the slips; psi
	=	$\frac{T_{eff}}{A}$
D	=	outside diameter of the pipe; inch
L	=	length of the slips (usually either 12" or 16"); inch
K	=	lateral load factor of slips
	=	$\frac{1}{\tan(y + z)}$
	=	4.0 for a coefficient of friction of 0.08
y	=	taper of slip, usually 9° 27' 45"
z	=	friction angle = $\tan^{-1}u$
u	=	coefficient of friction between slips and bushing.
	=	reasonable value for lubricated slips = 0.08
A	=	cross-sectional area of the wall of the drillpipe; $inch^2$
A	=	$\frac{\pi}{4}(D^2 - d^2)$

EXAMPLE

How deep can 5" 4.276" 21 ppf D-60,000 grade drillpipe be run in 11.2 ppg if only slip crushing is considered?

$$60000 = S_t \left[1 + \frac{5 * 4}{2 * 16} + \left(\frac{5 * 4}{2 * 16}\right)^2\right]^{1/2}$$

S_t **= 42,262 psi**

The effective tension at the rotary table while running 5" drillpipe is

$$S_{eff} = \frac{\left(1 - \frac{11.2}{65.45}\right) * 21 * \text{Depth}}{\frac{\pi}{4}(5^2 - 4.276^2)}$$ **= 42,262 psi**

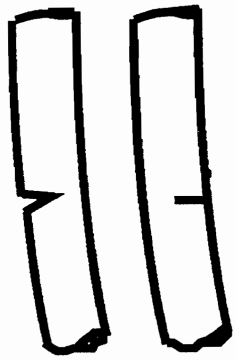
Depth **= 12,806 ft**

FATIGUE OF DRILLPIPE

In deep holes most drillpipe failures are the direct result of fatigue. Drillpipe fatigues while it is rotated within doglegs. The amount of fatigue damage depends on (1) the tensile load in the drillpipe at the dogleg, (2) the severity of the dogleg, (3) the number of turns of the drillpipe while it is within the dogleg, and (4) the mechanical dimensions and metallurgical properties of the drillpipe.

Because tension within the drillpipe is a major factor, the most critical location for a dogleg is at shallow depths. In regard to the fatigue of drillpipe, rotating off bottom is not recommended because of the increased load created by the suspended BHA.

Fatigue cracks are recognized by a characteristic polished area within the crack. The cracks are normal to the longitudinal axis of the drillpipe. Field people often call the cracks "washouts".



The maximum permissible dogleg severity with respect to the fatigue of drillpipe is given by the following equation (Lubinski's equation with modifications)

$$DLS = 137{,}500 * \frac{S_b}{E * D} * \frac{\tanh(K * L)}{K * L}$$

DLS	= maximum permissible dogleg severity; deg/100 ft
K	$= \sqrt{\frac{T_{eff}}{E\,I}}$
E	= modulus of elasticity of steel; psi
T	= buoyed weight of drill string below the dogleg; lbs
I	$= \frac{\pi}{64}(D^4 - d^4)$
D	= outside diameter of the drillpipe; inch
d	= inside diameter of the drillpipe; inch
L	= half the distance between tooljoints; inch
L	= 180" for range 2 drillpipe
Sb	= allowable bending stress; psi

The tensile stress in the drillpipe within the dogleg is

$$S_t = \frac{T}{A}$$

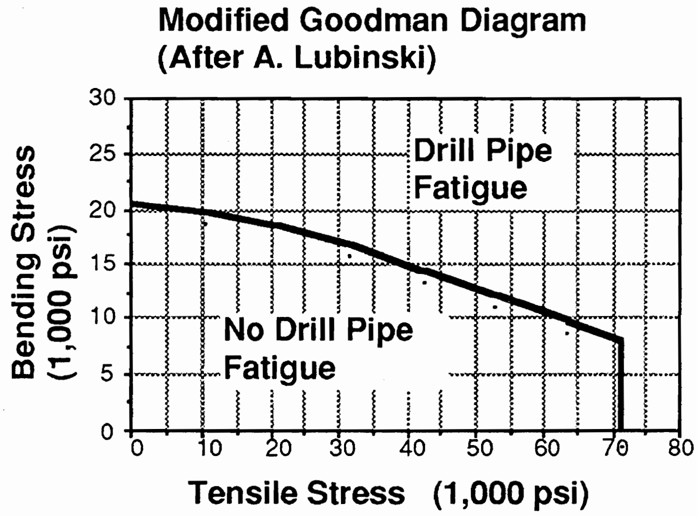
The cross-sectional area of the wall of drillpipe is

$$A = \frac{\pi}{4}(D^4 - d^4)$$

The allowable bending stress in the outer fiber of the wall of the drillpipe if the steel is grade E and the tensile stress is less than or equal to 67,000 psi is

$$S_b = 19500 - \frac{10}{67} * S_t - \frac{0.6}{670^2} * (S_t - 33{,}500)^2$$

The allowable bending stress in grade S drillpipe and if the tensile stress is less than or equal to 133,400 psi is

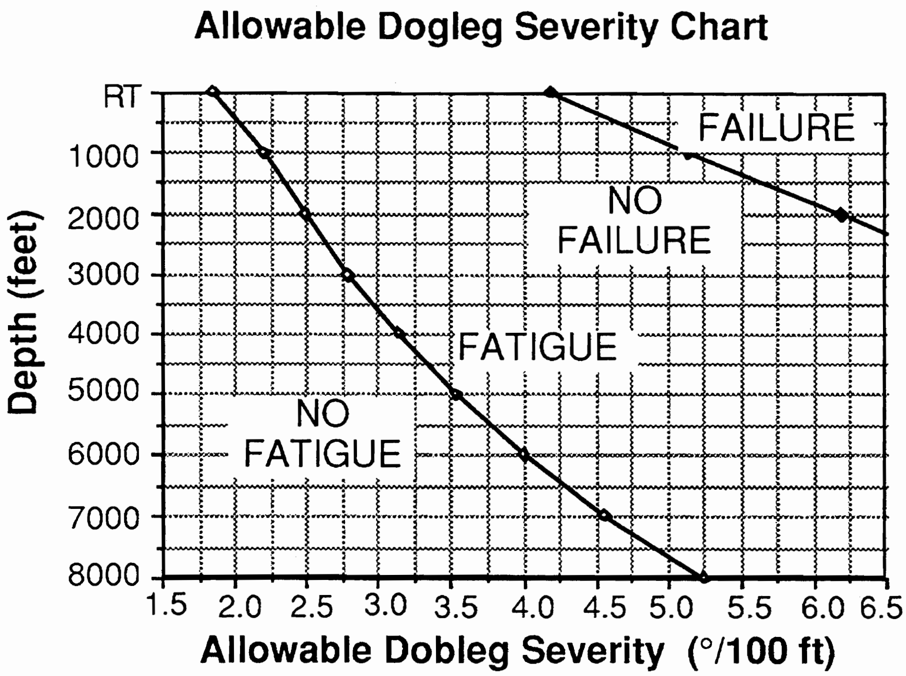


$$S_b = 20{,}000 * \left(1 - \frac{S_t}{145000} \right)$$

The maximum permissible lateral dogleg severity in regard to the excessive wear of tooljoints caused by contact between the wall of the hole and the tooljoint is

$$DLS = 34{,}400 * \frac{F}{L * T}$$

F	=	chosen maximum lateral tooljoint force on the wall of the hole; lbs
T	=	buoyed weight of drill string below the dogleg; lbs
L	=	half the distance between tooljoints; inch
L	=	180" for range 2 drillpipe



Lubinski suggests a value of 2,000 lbs for the maximum force in water base muds.

An allowable dogleg severity chart, such as the one presented, can be published in the drilling plan if doglegs are anticipated. The chart is constructed with the highest anticipated loads in the possible doglegs on the drillpipe during the drilling of the hole versus the depth of the dogleg.

EXAMPLE

A dogleg with a severity of 4 deg/100 feet is at a depth of 500 feet. The drill string is composed of 300 feet of 8" * 3" drill collars and 42 feet of tools and 14,000 feet of 5" * 4.276" * 21 ppf * grade S drillpipe. The mud weight is 11.3 ppg. Assume a zero weight on bit for a worst case scenario.

The buoyancy factor is

$$= 1 - \frac{11.3}{65.45} \qquad \mathbf{= 0.827}$$

The cross-sectional area of the drillpipe wall is

$$= \frac{\Pi}{4} * (5^2 - 4.276^2) \qquad \mathbf{= 5.275}$$

The weight of the drill string below the dogleg is the sum of the weights of the collars, the tools, and the drillpipe

$$= 147 * 300 + 147 * 42 + 13{,}500 * 21 \qquad \mathbf{= 333{,}774\ lbs}$$

The buoyed weight below the dogleg is

$$= .827 * 333{,}774 \qquad \mathbf{= 276{,}147\ lbs}$$

The buoyant tensile stress in the drillpipe within the dogleg is

$$= \frac{276147}{5.275} \qquad \mathbf{= 52{,}354\ psi}$$

The value of I, the cross-sectional moment of inertia is

$$= \frac{\Pi}{64}\,(5^4 - 4.276^4) \qquad \mathbf{= 14.269\ in^4}$$

The value of K is

$$= \left[\frac{276147}{3E7 * 14.269}\right]^{1/2} \qquad \mathbf{= 0.0254}$$

The maximum permissible bending stress is

$$= 20{,}000 * \left(1 - \frac{52354}{145000}\right) \qquad \mathbf{= 12{,}779\ psi}$$

The maximum permissible dogleg severity as not to fatigue the drillpipe is

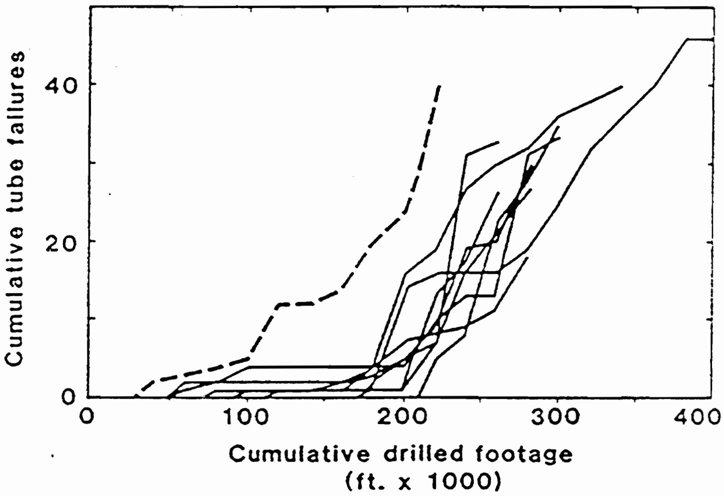
$$= 137{,}500 * \left[\frac{12779}{3E7 * 5}\right] * \left[\frac{\tanh(.0254 * 180)}{.0254 * 180}\right] \qquad \mathbf{= 2.56\ deg/100ft}$$

The maximum permissible dogleg severity as not to excessively wear the tooljoint is

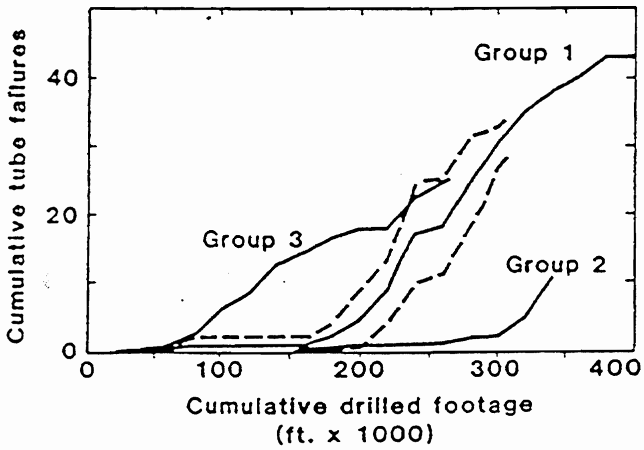
$$= 34{,}400 * \frac{2000}{180 * 276147} \qquad \mathbf{= 1.38\ deg/100ft}$$

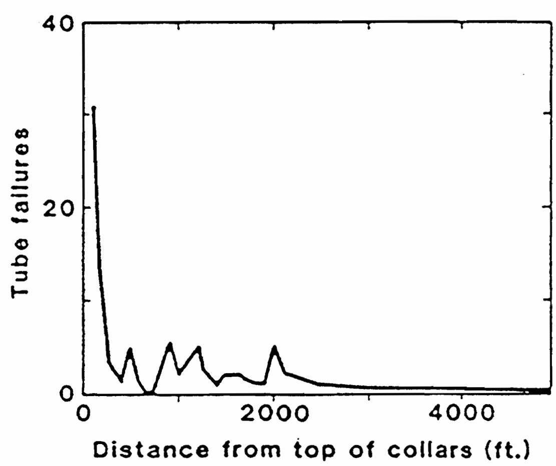
LIFE OF DRILLPIPE

Gensmer reported the life of drillpipe within the Rocky Mountain region in terms of cumulative tube failures and cumulative drilled footage. He categorized the drillpipe into 3 groups. Group 1 was internally plastic coated drillpipe, group 2 was drillpipe operated with oxygen scavengers, and group 3 was bare drillpipe in normal mud. The figures show that the life of internally plastic coated drillpipe has a life of 200,000 to 250,000 drilled footage and that failures within 200 feet of the top of the collars dominate. He also wrote that about 90% of the failures were fatigue cracks and occurred within 17 inches of the upset area below the box and few failures were near the pin end.



Group 1 string failure rates (dashed line—uncoated string).

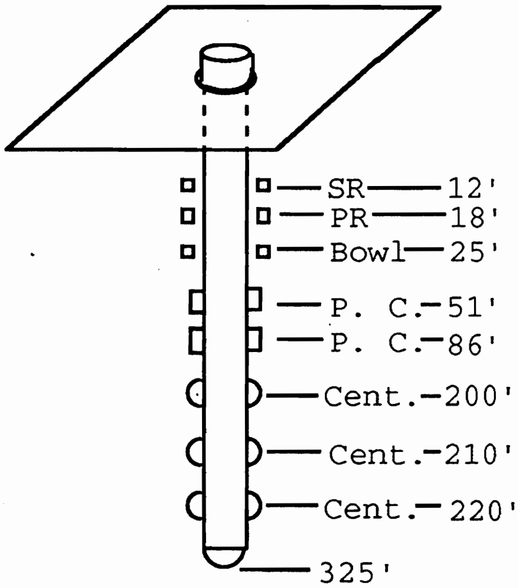




CASING TALLY

An example problem is presented to illustrate the problems encountered in making a casing tally and the subsequent location of centralizers on the casing string.

Caution must be exercised in regard to any attempt to set the casing shoe near the bottom of a drill hole. The depth of the bottom of a hole as determined by the strapping of a drill string while in tension or in the derrick may disagree by as much as ten feet with its depth as determined by a casing tally. Logging cables give similar disagreements. If the casing tally finds the bottom ten feet high or low, then the following negative events may occur:



1. A casing collar may be opposite either a shear or pipe B. O. P. E. ram which in turn would prevent its proper operation.

2. A casing collar may be in the bowl of its spool which would prevent the setting of the casing slips.

3. The casing shoe inadvertently may contact the bottom of the hole which would partially or totally eliminate the buoyant force acting on the end of the casing. Thereafter, it may be impossible to pull the casing off bottom.

4. The static head of the cement head and lines would be ten feet higher than planned which would not expedite the subsequent cementation. The best height above the rig floor for the top of the casing is 5 to 6 feet.

Joints of casing are assigned a number and this number is written on the joint while it is on the pipe racks at the rig site. While numbering a joint, its length which includes its collar but excludes its threads is recorded. The resulting list is called a casing tally.

EXAMPLE

Suppose the following casing tally has been recorded and it is desired that the casing shoe be set at a depth of 325 feet. Depths as measured from the rotary table of the various components are as follows.

1. shear rams: 12 feet
2. pipe rams : 18 feet
3. casing bowl: 25 feet
4. positive centralizers: 37 and 55 feet
5. spring centralizers: Three beginning at 200 feet and placed 10 feet apart.

Casing guide shoe is 1.12 feet long. Cementing collar is 1.96 feet long.

CASING TALLY TABLE

Joint No.	Length feet	Cumulative length
1	44.28	44.28
2	36.19	80.47
3	39.22	119.69
4	41.00	160.75
5	42.86	203.55
6	43.57	247.12
7	38.41	285.53
8	37.38	322.91
9	42.06	364.97

Plan to place the longest joint of casing (#1-44.28') at the top of the casing. This will provide a maximum space between the next casing collar and the bowl. The length of casing required (excluding the top joint) is then

L. C. = Setting depth - top joint + floor clearance - shoe length - cementation collar length

L. C. = 325 - 44.28 + 5 - 1.12 - 1.96 **= 282.64**

By trial and error it is found that joint #7 should not be used. The length of the casing with the shoe and collar is

364.97 - 38.41 + 1.12 + 1.96 **= 329.64** and the floor clearance is the setting depth subtracted from the length of the made up casing.

= 329.64 - 325 **= 4.64 feet**

The depth of first collar = length of top joint - floor clearance

= 44.28 - 4.64 **= 39.64 feet**

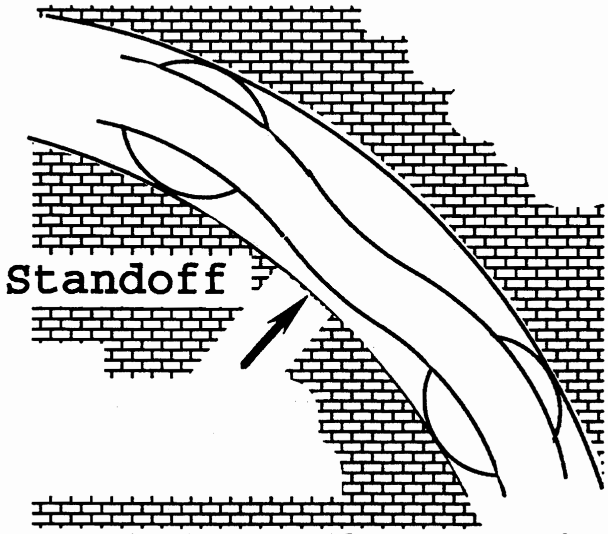
Note that this collar is 14.64 feet (39.64 - 25) below the casing bowl.

The depth of the second collar is 81.70 feet (39.64 + 42.06)

CASING COLLAR DEPTH TABLE

Joint No.	Length	Collar Depth	Notes
1	44.28	-4.64	Place positive centralizer down on joint #1
9	42.06	39.64	Place positive centralizer down on joint #9 (55 - 39.64 = 15.36)
8	37.38	81.70	
6	43.57	119.08	
5	42.86	162.65	Place centralizer down 37.35' on joint #5
4	41.00	205.51	Place centralizers down 4.49' and 14.49' on joint #4
3	39.22	246.51	
Collar	1.96	285.73	
2	36.19	287.69	
Shoe	1.12	323.88	
Bottom		325.00	

CASING CENTRALIZER SPACING



The purpose of centralizers on casing is to provide separation for the casing from the wall of the hole. This separation is called standoff. The strength of a centralizer is the amount of force required to compress its bows a given distance. Thus, stronger bows provide more standoff for casing.

Wall force is the sum of two components: (1) the product of the tension in the pipe and the dogleg severity of the hole and (2) buoyed weight of the pipe. The moments transmitted through the string of pipe in not considered and therefore, closer spacings of centralizers in holes which have "S" shaped doglegs may require additional thought. The following equations apply to all pipe (casing, tubing, drill pipe, etc.) which is in normal oil field service.

CENTRALIZER SPACING EQUATION

$$F = 2\,T \sin\left(DLS * \frac{CS}{2}\right) + (WF)_b * CS * \sin ø \qquad \text{(exact transcendental function)}$$

$$CS = \frac{F}{.0175 * T * DLS + (WF)_b * \sin ø} \qquad \text{(with an 0.02\% precision)}$$

F	= force on each centralizer if spaced CS feet apart; lbf
CS	= centralizer spacing; feet
DLS	= dogleg severity; degrees/100 feet; example DLS = .03
w	= weight per foot of pipe (steel only); lbf/foot
F_b	= buoyancy factor; no units
ø	= average inclination angle near the centralizer; degrees
T	= tension in the wall of the pipe (for drill pipe see "tension in drill pipe in directional wells"); lbf

T = $.0408 * TVD * (\rho_i d^2 - \rho_e D^2) + \cos\phi * w * S$ (for casing only in the slant portion of the hole); lbf

$(WF)_b$ = $w + .0408 (\rho_i d^2 - \rho_e D^2)$; lbf/foot

ρ = mud weights inside and outside casing; ppg
d,D = ID and OD of casing; inches
TVD = true vertical depth to the shoe of the casing; feet
S = distance from the casing shoe to the centralizer in question; ft
M = measured depth to the casing shoe; feet

EXAMPLE CENTRALIZER SPACING

A centralizer will provide a standoff of 1 inch while bearing a wall force of 1200 lbf. The centralizer is in a gradual dogleg where the average survey data and other data are as listed.

Inclination angles 36° at 6122 ft and 44° at 6302 ft

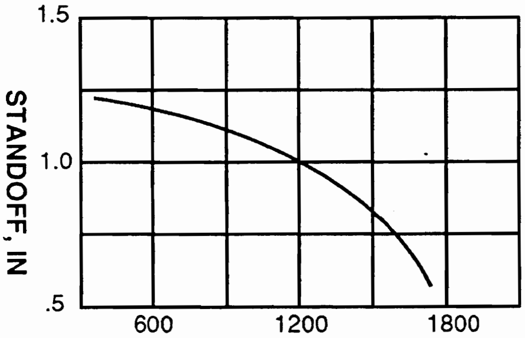
Direction angles N 2°E at 6122 ft and N 7°E at 6302 ft

N = 9989 ft TVD = 4111 ft

ρ_i = 15 ppg ρ_e = 12 ppg (cement)

w = 53 lbf/ft d = 8.535" D = 9.625"

LOAD DEFLECTION CURVE



What centralizer spacing is required in the dogleg?

$$S = 9989 - \frac{(6122 + 6302)}{2}$$ **= 3777 ft**

$$T = .0408 * 4111 * (15 * 8.535^2 - 12 * 9.625^2) + \cos\left(\frac{36+44}{2}\right) * 53 * 3777$$

T **= 150,162 lbf**

$$\phi = \frac{36+44}{2}$$ **= 40°**

Dogleg length = 6302 - 6122 **= 180 ft**

$$DLS = \frac{100}{180}\left[(44-36)^2 + ((7\text{-}2) * \sin 40)^2\right]^{1/2} \qquad \mathbf{= 4.79\ deg/100\ ft}$$

$$(WF)_b = 53 + .0408\,(15 * 8.535^2 - 12 * 9.625^2) \qquad \mathbf{= 52.23\ lb/ft}$$

Exact Equation

$$1200 = 2 * 150{,}162 * \sin(.0479 * CS) + 52.23 * CS\,(\sin 40)$$

$$CS \qquad \mathbf{= 7.53\ ft\ (trial\ and\ error\ solution)}$$

Thus, the centralizers would be placed 7.53 feet apart throughout the 180 feet of the dogleg within the hole.

If the dogleg were not present, then the centralizer spacing would have been

$$CS = \frac{1200}{(52.23 * \sin 40)} \qquad \mathbf{= 35.7\ ft}$$

CASING SAG BETWEEN CENTRALIZERS

The approximate sag (maximum downward deflection of the casing between the centralizers) MD of the pipe is given by the following equation (Timoshenko, "Strength of Materials", Part 2, page 45, equation 52)

$$MD = \left(\frac{WF_b * \sin ø * CS^4}{EI}\right) * \left(\frac{1296}{U^4}\right)\left(\frac{U^2}{2} - \frac{U \cosh U - U}{\sinh U}\right)$$

$$U = \left(\frac{36 * T * CS^2}{EI}\right)^{1/2} \quad ; \text{no units}$$

$$E = 30 * 10^6\ ;\ lbf/in^2$$

$$I = \frac{\pi}{64} * (D^4 - d^4)\ ;\ in^4$$

EXAMPLE

The sag in the casing in the case where there is not a dogleg but the hole is inclined at 40 degrees as in the example problem is

$$MD = \frac{52.23 * \sin 40 * 35.7^4}{30 * 10^6 * 160.8} * \frac{1296}{1.195^4}\left[\frac{1.195^2}{2} - \frac{1.195 \cosh 1.195 - 1.195}{\sinh 1.195}\right]$$

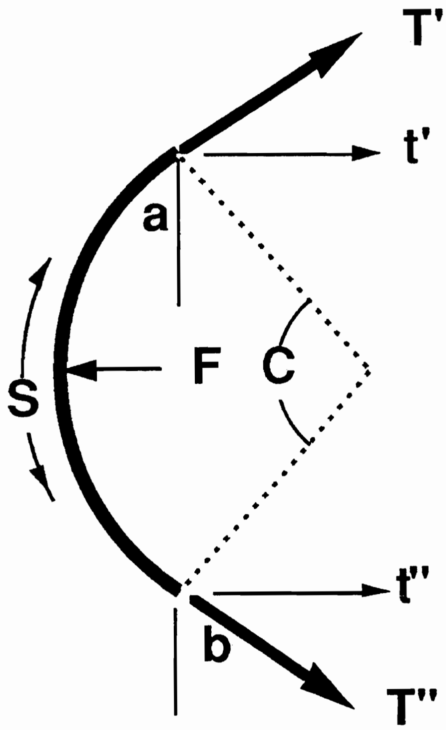
$$U = \frac{36 * 150162 * 35.7^2}{30 * 10^6 * 160.8}$$ **= 1.195 no units**

$I = .04909 * (9.625^4 - 8.535^4)$ **= 160.8 in^4**

MD **= 0.534 inch**

WALL FORCE EQUATION

The equation relating wall force with pipe tension and dogleg severity of the hole is established in the following



$$c = a + b$$

$$F = t' + t''$$

If t' = t", then $F + 2\,t'$

$$DLS = \frac{c}{S}$$

$$t' = T \sin\left(\frac{c}{2}\right)$$

thus

$$F = 2\,T \sin\left(\frac{DLS * S}{2}\right)$$

Note: $\sin\left(\frac{DLS * S}{2}\right) = \frac{DLS * S}{2}$

If DLS (rad/foot), C = rad

$$F = T\,(DLS * S)$$

$$\frac{\text{Wall Force}}{\text{Foot}} = \frac{F}{S} = \frac{T * DLS * S}{S} = T * DLS$$

(rad/ft)

EXAMPLE

Compute the wall force if pipe is in 100,000 of effective tension and in a dogleg of 5°/100feet which is 100 feet long.

DLS = 5°/100' S = 100' T = 100,000

$F = 2 * T * \sin\left(\frac{5°/100 * 100}{2}\right)$ **= 8723.88 lb**

$F = 2 * 100{,}000 \sin(2.5°)$ **= 8723.88 lb**

$F = 2 * 100{,}000 \sin\left(2.5° * \frac{\pi}{180}\right)$ **= 8723.88 lb**

$F = T * DLS * S$

$F = 100{,}000 * \left(\frac{5}{100} * \frac{\pi}{180}\right) * 100$ **= 8726.65 lb**

NOTES: $DLS\,(\frac{rad}{ft}) = DLS\,(\frac{o}{100'}) * (\frac{\pi}{180})$

$.0008726 = (\frac{5}{100}) * (\frac{\pi}{180})$

STRETCH OF PIPE WITH HELICAL BUCKLING

If weight is slacked-off at the surface and placed on the bottom of the hole or on tools at the bottom of the hole, the pipe will shorten because of resulting compression of the steel in the walls of the pipe and because the pipe will settle into a helical coil within the wellbore. For practical problems both occur simultaneously.

Lubinski gives the following equation for the change of pipe length with helical buckling

$$\Delta L = \frac{r^2 \; F_s^{\;2}}{8 \; EI \; W} L \quad \text{(Shortening because of helical buckling)}$$

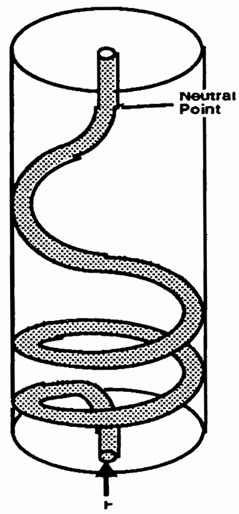
F_s = slack-off weight after helical buckling has commenced

$W = W_s + W_i - W_e$ (Buoyed weight of the tube)

W_s = (pipe weight per foot/12) $= \frac{W_p}{12}$

$W_i = \rho f_i A_i = \rho f_i \frac{\pi}{4} d^2$

$W_e = \rho f_e A_e = \rho f_e \frac{\pi}{4} D^2$

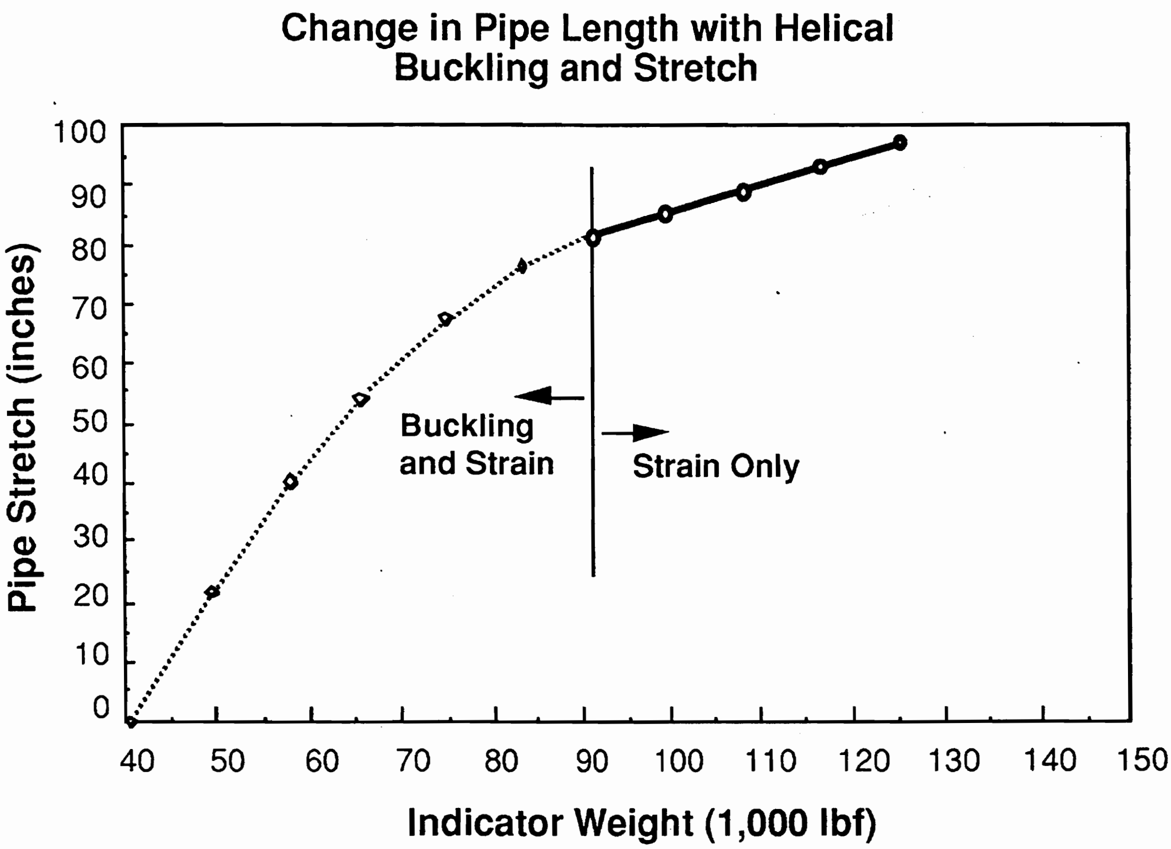


$$W = \frac{W_p}{12} + \frac{\rho_i \frac{\pi}{4} d^2}{231} - \frac{\rho_o \frac{\pi}{4} D^2}{231}$$

NOTE: ON UNITS: $\rho \frac{\text{lbf}}{\text{in}^3} = \rho' \frac{\text{lbf}}{\text{gal}} / 231$

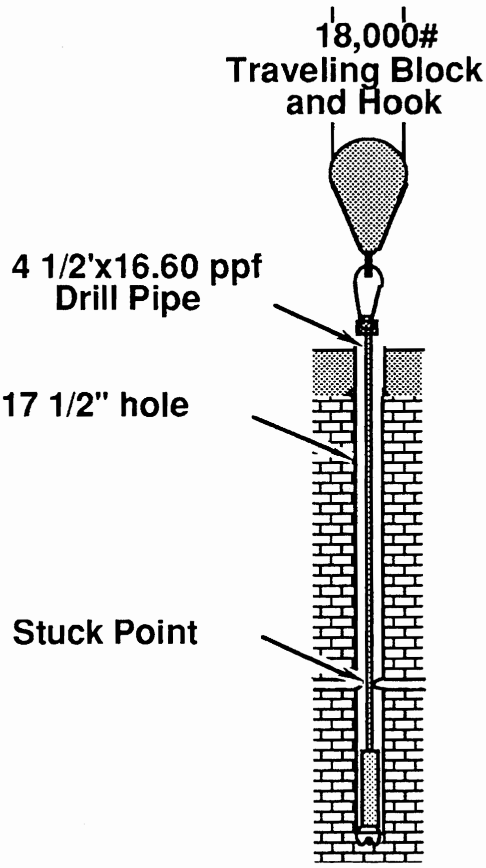
and Hooke's law with appropriate units gives the axial strain elongation equation

$$\Delta L = \frac{.283\, \Delta F\, L}{E\, W_s} \quad \text{(can cause shortening or lengthening)}$$



EXAMPLE

The drill pipe has become stuck while drilling with $4\frac{1}{2}$" * 16.60 ppf drill pipe in a $17\frac{1}{2}$" hole. Mud weight is 12.6 ppg. Find the depth to the stuck point and indicator weight such that the neutral point with regard to axial stress is at the stuck point. The block and tackle weighs 18,000 lbf.



The following data, stretch (inches) of the drill pipe versus weight indicator reading (1,000 lbf), has been recorded.

Analysis of the plot constructed with the data shows that the last 5 points are linear which indicates that the buckles have been removed and that only axial strain is occurring. Hooke's law applies:

$$L = \frac{E\, W_s\, \Delta L}{.283 * \Delta F}$$

In oilfield units

$$L = \frac{(8.82E6)\ W_s(ppf)\ \Delta L(inches)}{\Delta F(lbf)}$$

$$L = \frac{30 * 10^6 * \frac{16.6}{12} * (98.5 - 90.3)}{.283 * (141{,}000 - 121{,}000)}$$

= 60,124 inch or 5,010 feet

Δe	Reading
0 .0	41
21.7	51
40.2	61
55.5	71
67.6	81
76.5	91
82.1	101
86.2	111
90.3	121
94.4	131
98.5	141

A free body of the drill pipe shows that the hook load required to place the neutral point of axial stress at the stuck point is

Hook load	= 16.6 * 5010	**= 83,166 lbf**
Weight Indicator	= 83,166 + 18000	**= 101,166 lbf**

Slacking off more weight than 101,166 lbf will cause buckling of the drill pipe. The change in the length of the pipe will be the sum of the helical buckles and Hooke's Law effects.

Suppose 30,000 lbf are slacked off from the neutral position of 101,166 lbf; then

$$\Delta L_{total} = \frac{r^2 F_s^{\,2}}{8\,EIW} + \frac{.283 * L\,\Delta F}{E\,W_s}$$

$$r^2 = [\frac{17.5 - 4.5}{2}] \qquad = \mathbf{42.25}$$

$$F_s^{\,2} = 30{,}000^2 \qquad = \mathbf{9 * 10^8}$$

$$I = \frac{\pi}{64}(4.5^4 - 3.826^4) \qquad = \mathbf{9.61}$$

$$W = \frac{16.6}{12} + \frac{12.6}{231}\frac{\pi}{4}\,3.826^2 - \frac{12.6}{231}\frac{\pi}{4}\,4.5^2 \qquad = \mathbf{1.143}$$

$$L = 5{,}010 * 12 \qquad = \mathbf{60{,}120}$$

$$\Delta F = 30{,}000$$

$$W_s = \frac{16.6}{12} \qquad = \mathbf{1.38}$$

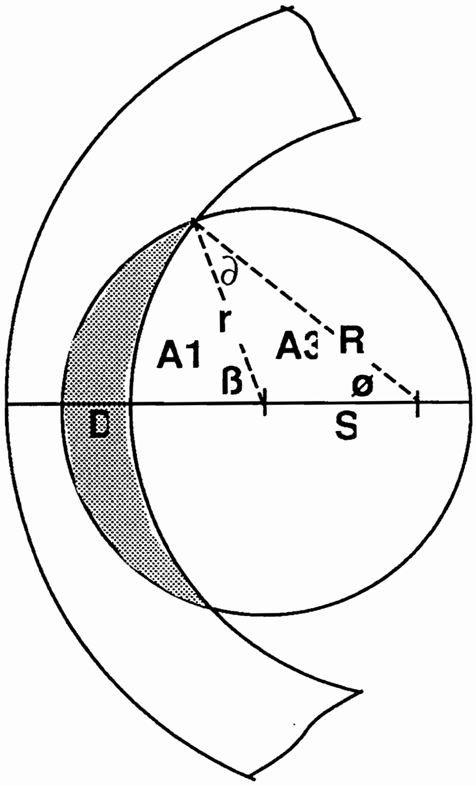
$$\Delta L_{total} = \frac{42.24 * 9 * 10^8}{8 * 3 * 10^7 * 9.61 * 1.143} + \frac{.283 * 60120 * 30000}{3 * 10^7 * 1.38}$$

$$\Delta L_{total} = 14.42 + 12.33 \qquad = \mathbf{26.75\ inch}$$

Because weight is being slacked off, the pipe will shorten 26.75 inches.

CASING WEAR

Laboratory physical tests show that tool joints of drillpipe are most responsible for casing wear under all but the most extreme circumstances. Casing wear is reported either as the depth of wear into the wall of the casing or the volume of steel worn from the wall. An ideal crescent is shown in the figure.



AREA AND VOLUME OF CRESENT

1. $A_c = A_1 - (A_2 - A_3)$
2. A_1 = Segment of circle with r and ß
3. A_2 = Segment of circle with R and ø
4. $S = R - r - D$
5. $\partial = \mathrm{acos}[(R^2 + r^2 - S^2)/2Rr]$
6. $ß = \mathrm{asin}[\frac{r \sin \partial}{S}]$
7. $ø = ß + \partial$
8. $p = \frac{R + r + S}{2}$
9. $A_1 = \frac{ß\, r^2}{2}$
10. $A_2 = \frac{ø\, R^2}{2}$
11. $A_3 = [p(p-S)\,(p-r)\,(p-R)]^{.5}$
12. $V_c = L * A_c$

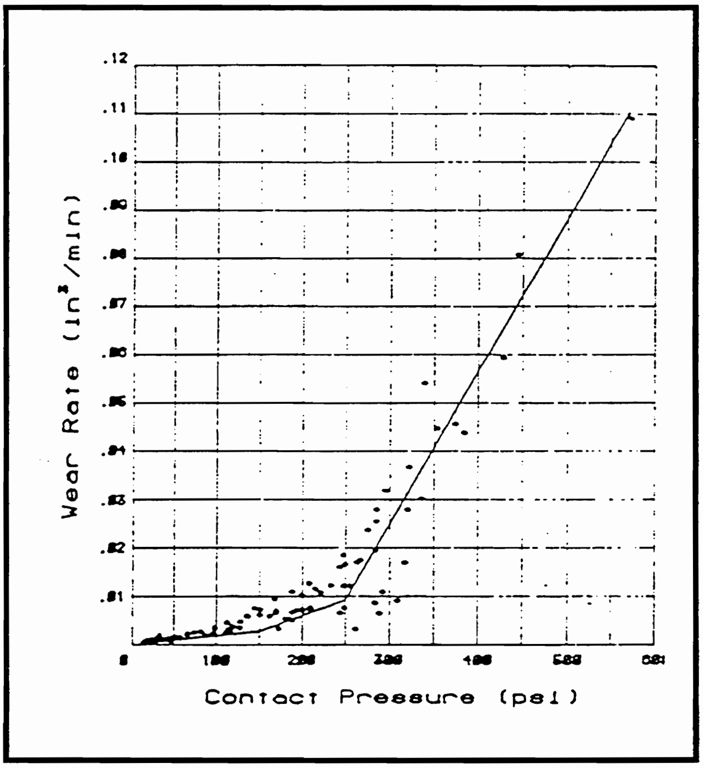
Note: 231 cubic inches	= 1 gallon
16 cups	= 1 gallon
16 tablespoons	= 1 cup

EXAMPLE WEAR PROBLEM

7" 23ppf 6.366" ID 40 foot long casing is worn in a cresent shape to a depth of 0.0234 inches by 3.5" drillpipe which has 5" tooljoints. How many cups of steel is contained in the cresent?

Radius of casing = 3.183 in.
Radius of tooljoint = 2.5 in.
S = .7064 in.
∂ = 0.0639 radian
π = 3.1947 in.
ß = 0.2281 radian
ø = 0.2920 radian
A_1 = 0.9125 sq. in.
A_2 = 1.1554 sq. in.
A_3 = 0.2542 sq. in.
A_c = 0.0227 sq. in.

$$V_c = A_c * L$$
$$= 0.0227 * 40 * 12$$
$$= 10.896 \text{ cubic inches}$$
$$V_c = 10.896 * \frac{16}{231}$$
$$= \mathbf{0.755\ cups}$$



The literature identifies three wear phenomena by which steel is removed from the wall of casing:

1. lubricated friction wear at low contact pressures between smooth tool joints and ~~the~~ casing (no meaningful wear occurs)
2. abrasive and grinding wear at medium contact pressures (sand intensifies wear)
3. galling wear at highest of contact pressures.

These are depicted by the slopes of the lines in the figure.

Lubinski writes that contact loads of 2,000 or more pounds per tool joint will cause excessive wear.

Beware of corrosion under drillpipe rubbers.

All grades (K-55, N-80, P-110, etc.) of casing wear at about the same rate. Guard against casing wear with extra wall thickness.

Run a casing inspection log to ascertain wear. Statements like, "I can drill for 30 days before wear is serious", is foolish.

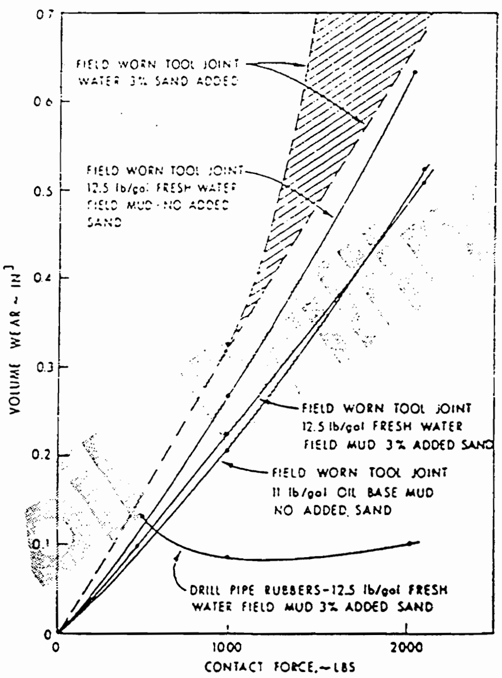
WEAR DEPTH ESTIMATION

Bradley published an equation for estimating casing wear; however, it should be pointed out that he himself claims errors of 800%.

$$D = .002 \frac{F_{tj}}{1000} * RD * RS$$

D	= depth of penetration (cresent shape); inch
F_{tj}	= lateral tooljoint force; lb
RD	= days of rotation inside casing; day
RS	= rotary speed; rpm

EXAMPLE WEAR DEPTH



A 3,810 lb lateral tooljoint force has existed for 4.375 days. The rotary speed has been 88 rpm. Estimate the wear penetration depth?

$$D = .002 * \frac{3810}{1000} * 4.375 * \frac{88}{120}$$

D = **0.0244 in.**

The figure compares wear of various drilling components with contact force.

SCRAPING FORCED IN A DOGLEG

CRITICAL LENGTH OF A TOOL

A tool will be bent in a dogleg if it exceeds a critical length. The figure depicts the points of contact. The equations give the value of the scraping force and the friction force of pushing the tool through the dogleg. The equations only relate to the rigidity of the tools and not to axial tension which may add to the forces. The industry recognizes two type of doglegs: circular and abrupt.

BENT CONFIGURATIONS OF A TOOL

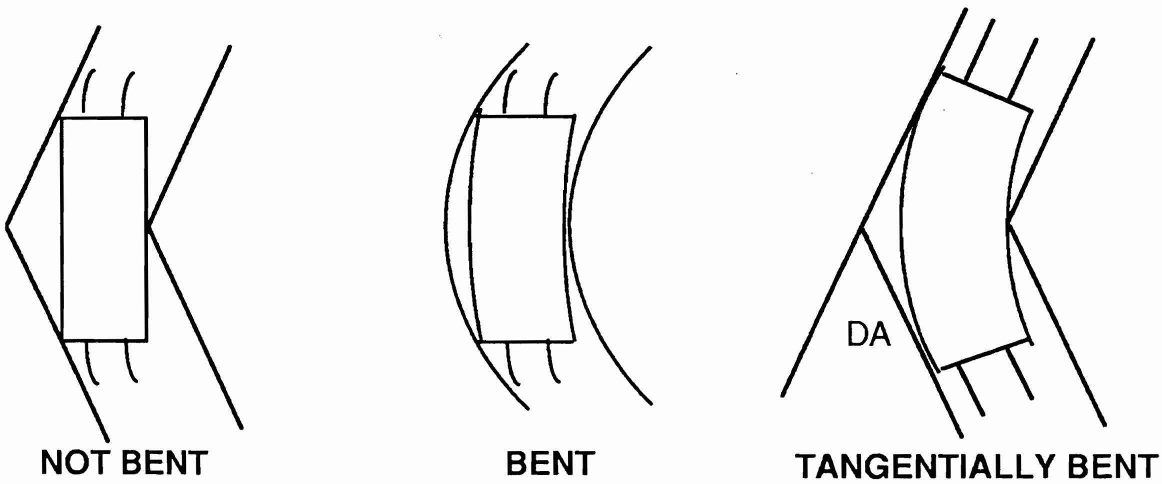
Three bending modes are considered and depicted in the following sketches. These are NOT BENT, BENT, AND TANGENTIALLY BENT. The equations which give the radius of curvatures of tool for each of the configurations are as listed.

Not Bent: R_t = infinity

Bent: $$R_t = \frac{L^2 + 4\,U^2}{8\,U}$$ Circular dogleg

U	$= R - C - 0.5\,[4\,R^2 - L^2]^{1/2}$
R_t	= radius of curvature of the tool; in
C	= Clearance between hole ID and tool OD; in
R	= radius of curvature of the hole; in
L	= length of tool; in
DA	= dogleg angle; deg

Tangentially Bent: $$R_t = \frac{C}{1 - \cos(\frac{DA}{2})}$$ Abrupt dogleg



SCRAPING FORCE OF A TOOL

The standard mechanics of material equation gives the scraping force.

$$F_s = \frac{4\,E\,I}{R_t\,L}$$

E	= 3E7;psi
I	$= \frac{\Pi}{64}\,[OD^4 - ID^4]$;for cylinder tool; in^4
L	= length of tool; ft, for circular dogleg
L	$= R_t\,DA\,\frac{\Pi}{180}$; for abrupt dogleg and tangentially bent

CIRCULAR DOGLEGS

The length of a tool which may be run through a circular dogleg and not be bent is given by the following equation.

$$L_c = 2\,[C\,(2\,R - C)]^{1/2}$$

$$R = \frac{687.55}{DLS}$$

L_c	= critical length; ft
C	= clearance between hole ID and tool OD; in.
R	= radius of curvature of hole; in
DLS	= dogleg severity of hole; deg/ft

EXAMPLE CRITICAL LENGTH, RADIUS OF CURVATURE, AND SCRAPING FORCE

An 8" OD by 3" ID by 30' (360") tool is run in 9 5/8" by 8.535" casing where a 5 deg/100ft dogleg exists. What is the critical length of the tool?

$$C = 8.535 - 8 = 0.535 \text{ in}$$

$$R = \frac{687.55}{.05} \qquad \mathbf{= 13,751\ in}$$

$$L_c = 2\,[0.535\,(1 * 13,7541 - 0.535)]^{.5} \qquad \mathbf{= 242.6\ in}$$

Note that the tool is bent.

$$U = 13,751 - 0.535 - 0.5\,[413,751^2 - 360^2]^{1/2} \qquad \mathbf{= 0.64315}$$

$$R_t = \frac{360^2 + 4 * 0.64315^2}{8 * 0.64315} \qquad = \mathbf{25{,}189\ in}$$

$$I = \frac{\Pi}{64}[\,8^4 - 3^4\,] \qquad = \mathbf{197.1\ in^4}$$

$$F_s = \frac{4 * 3E7 * 197.1}{25189 * 360} \qquad = \mathbf{2{,}608\ lb}$$

ABRUPT DOGLEG

The maximum length of a tool which may be run through an abrupt dogleg without be bent is given by the following equation.

$$L_c = \frac{2\left[H - D\cos\left(\frac{DA}{2}\right)\right]}{\sin\left(\frac{DA}{2}\right)}$$

L_c	= critical length of tool; in
DA	= abrupt dogleg angle; deg
H	= hole diameter; in
D	= tool diameter; in

EXAMPLE CRITICAL LENGTH, RADIUS OF CURVATURE, AND SCRAPING FORCE

An 8"OD by 3"ID by 30' (360") tool is run in 9 5/8" by 8.535" casing where a 1 degree abrupt dogleg exists. What is the critical length of the tool?

$$L_c = \frac{2\,[8.535 - 8\cos(\frac{1}{2})]}{\sin(\frac{1}{2})} \qquad = \mathbf{122.7\ in}$$

Note that the tool is bent.

$$R_t = \frac{0.535}{1 - \cos(\frac{1}{2})} \qquad = \mathbf{14{,}050\ in}$$

$$I = \frac{\Pi}{64}[8^4 - 3^4] \qquad = \mathbf{197.1\ in^4}$$

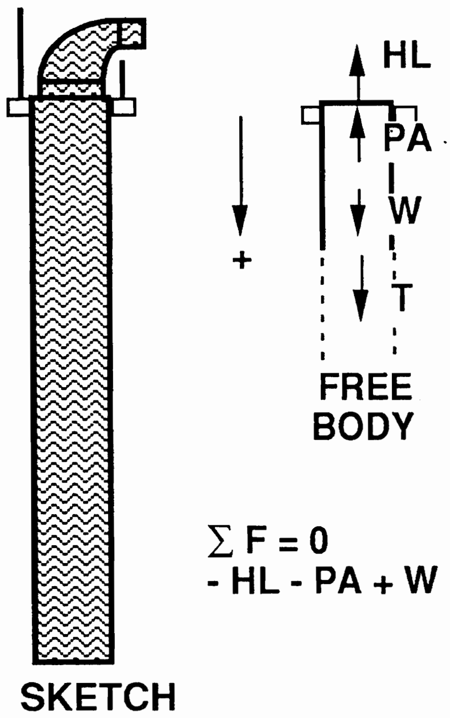
$$L = 14{,}050 * 1 * \frac{\Pi}{180} \qquad = \mathbf{245.2\ in}$$

$$F_s = \frac{4 * 3E7 * 197.1}{14050 * 245.2} \qquad = \mathbf{6{,}865.5\ lb}$$

FREE BODIES

The steps in constructing, analyzing and using a free body are embodied in the following steps.

a. Draw sketch
b. Select free body
c. Direction of forces
d. Select unknown force on free body
e. Place known forces on free body
f. Writ force equation (Newton's Law)
g. Solve for unknown force
h. Select direction of unknown force



$$\sum F = 0$$

$$- HL - PA + W + T = 0$$

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CHAPTER II

DRILLING OPTIMIZATION METHODS

Drilling optimization is the application of technology which yields a reduction of drilling costs associated with making hole. The following optimization techniques are popular in drilling.

1. Cost per foot equation
2. Time value of money
3. Expected value method
4. Lagrangian multiplier
5. Multiple regression
6. Confidence intervals
7. Lagrange's interpolation formula

COST PER FOOT EQUATION

The cost per foot equation is used for the comparison of alternative equipment, chemicals, and procedures for the drilling of a formation or an interval. The comparisons are often called break-even calculations and are usually between drill bit types or manufacturers; however, any of the variables may be compared.

$$C = \frac{\text{Bit} + \text{Tools} + \text{Mud} + [\text{Drill Time} + \text{Trip} + \text{Lost}]\,[\text{Rig} + \text{Support} + \text{Tool Rental}]}{\text{Drill Rate} * \text{Drill Time}}$$

C	= Cost per foot for the interval of concern; \$/ft
Bit	= Cost of delivered bit at the drill site; \$
Tools	= Cost of tools or repairs to tools; \$
Mud	= Cost of mud to drill the interval; \$
Drill Time	= Time required to drill the interval or bit life; hr
Trip	= Time to pull and run a bit; hr
Lost	= Time chargeable to non-drilling task; hr
Rig	= Contract rental rate of a rig; \$/hr
Support	= Third party contractors rates; \$/hr
Tool Rental	= Rental of tools; \$/hr
Drill Rate	= Average drilling rate over the interval; ft/hr

In a comparison of drill bits, the drilling rate and life of the proposed bit will always be in question. The usual procedure is to compute the cost per foot with the data from a standard bit with the proposed bit; and, then construct a chart of required drilling rate and bit life for the proposed bit. The following example illustrates the method.

EXAMPLE

A study of bit data from a data base predicts that the expected values from a bit run with a tooth bit are as follows.

Bit	= $1,491
Tool	= Stabilizer $250
Mud	= $2,000
Drill Time	= Bit life; 33 hr
Trip	= 7 hr
Lost	= Surveying after every trip; 0.3 hr
Rig	= 550 $/hr
Support	= Contractor and Supervisors; $250/hr
Tool Rental	= Large drill collars; 9 $/hr additional
Drill rate	= 13 ft/hr

$$C = \frac{1491 + 250 + 2000 + [33 + 7 + .3][550 + 250 + 9]}{13 * 33}$$

C = **84.72 $/ft** Cost per foot for the data base tooth bit.

Cost of PDC bit

The costs for a proposed PDC bit which is to be run on a bottom hole motor are the following.

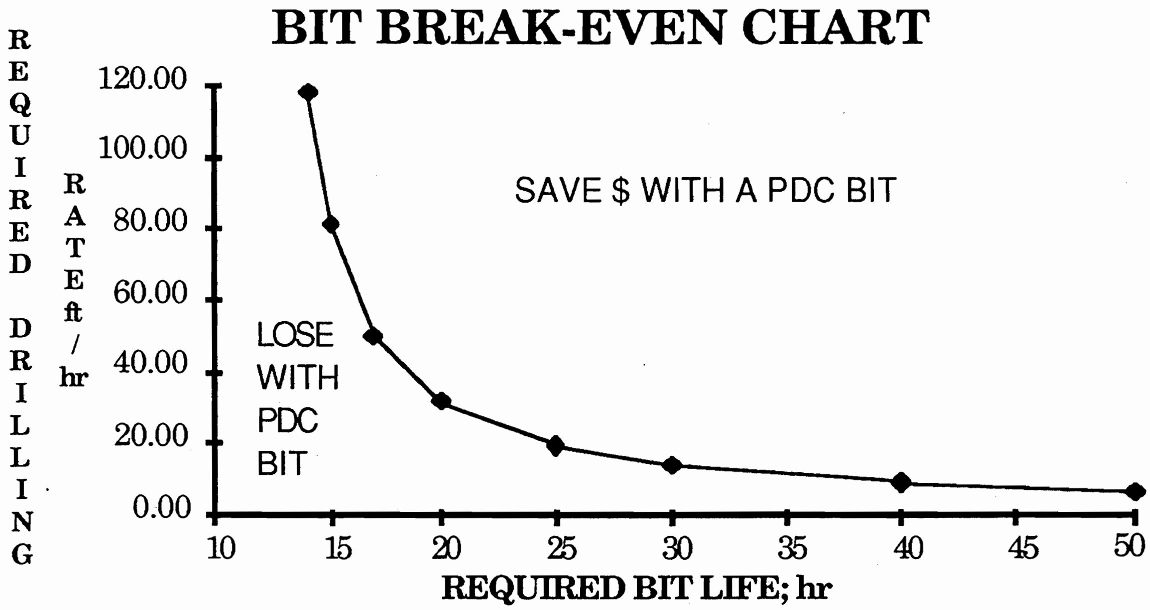
Bit	= $9,283
Tool	= (1) Bottom hole motor repair charge; $1,894
	(2) Two stabilizers; 2 * $250 = $500
Mud	= $2,500 ($500 additional required by the BHM)
Drill Time	= Bit life; Unknown; hr
Trip	= 7.5 hr (.5 additional hour)
Lost	= Surveying after every trip; 0.3 hr
Rig	= 550 $/hr
Support	= Contractor and Supervisors; $250/hr
Tool Rental	= Bottom hole motor; 200 $/hr
Drill rate	= Unknown; ft/hr

$$84.72 = \frac{9283 + (1894 + 500) + 2500 + [\text{Drill Time} + 7.5 + .3][550 + 250 + 200]}{\text{Drill Rate} * \text{Drill Time}}$$

Reduction of the equation yields,

[Drill rate - 11.80] * Drill Time **= 259.41**

A break-even chart of required drill rates versus required drill time (bit life) may be prepared with the equation.



It is seen from the chart and the equation, that if the proposed bit is break-even with the data bit, it must drill at a rate of 20 ft/hr for 25 hours. The proposed bit can not break-even at any bit life if its drilling rate is less than 11.80 ft/hr.

EXAMPLE OF EXPECTED VALUE

A mud company claims that a spending $3,000 more dollars on the mud will cause bits to drill 10% faster and 20% longer. Thus, the values for the data bit adjusted for the new mud are the following.

Bit	= $1,491
Tool	= Stabilizer $250
Mud	= $2,000 + $3,000 = $5,000
Drill Time	= Bit life; 33 hr * 1.2 = 39.6 hr
Trip	= 7 hr
Lost	= Surveying after every trip; 0.3 hr
Rig	= 550 $/hr
Support	= Contractor and Supervisors; $250
Tool Rental	= Large drill collars; 9 $/hr additional
Drilling rate	= 13 ft/hr * 1.1 = 14.3 ft/hr

$$C = \frac{1491 + 250 + 5000 + [39.6 + 7 + .3][550 + 250 + 9]}{14.3 * 39.6} \qquad \mathbf{= 78.91\ \$/ft}$$

Because the cost is 78.91 $/ft versus 84.72 $/ft, the cost increase in the mud is justified by this analysis.

TIME VALUE OF MONEY

Because interest is paid on savings, the value of money varies with time. Suppose that a company has had an average rate of return of 21% over the past five years. Further, if an employee had bought a stabilizer five years ago for $1,000 and did not use it, what would its value need to be today to break-even?

The equation for computing the time value of money is the following.

$$V = P * \left(1 + \frac{r}{q}\right)^{n*q}$$

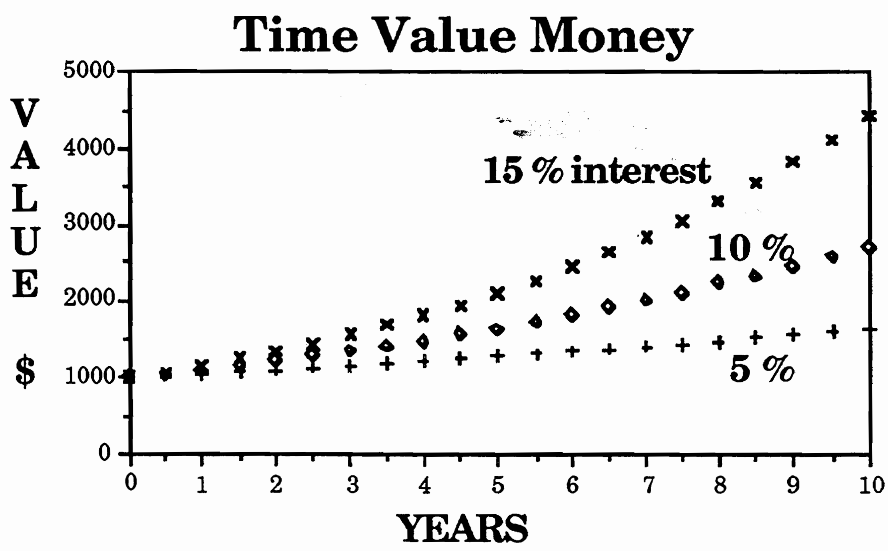
V	= Value of principal five years later and is unknown; $
P	= Amount of dollars spent five years ago; $1,000 $
r	= Yearly rate of return or interest rate; 0.21 fraction
n	= Number of years which have transpired; 5 yr
q	= Number of payments per year; 4 quarterly

EXAMPLE

With the numbers given above, the future value of the $1,000 is the following.

$$V = 1{,}000 * \left(1 + \frac{0.21}{4}\right)^{5*4} \qquad = \mathbf{\$2{,}782.50}$$

Current required value to break-even.



EXPECTED VALUE METHOD

The expected value method provides a means of reaching a decision based on expected costs and the probabilities of their occurrences. The fundamental form of the expected value equations are the following.

$$EV = C_1 * P_1 + C_2 * P_2 \qquad \text{and} \qquad P_1 + P_2 = 1$$

EV	= Expected Value; \$
C_1	= Cost of first event; \$
P_1	= Probability of first event; fraction
C_2	= Cost of second event; \$
P_2	= Probability of second event; fraction.

EXAMPLE

If a coin is tossed 100 times, and one dollar is won on heads and one dollar is lost on tails, what is the expected value of our 100 bets? The answer is zero, because no money should be won or lost. A step by step solution with the expected value method gives a value of zero as expected.

Choose a basis for cost. The cost of 100 tosses and bets is chosen.

Choose the first event to be the winning of one dollar and assign it a positive value of one.

Choose the second event to be the losing of one dollar and assign it a negative value of one.

On the toss of a coin, only two possibilities exist; either a head or a tail.

The probability of a head is 1/2 because a head is one of two possibilities.

The probability of a tail is 1/2 because a tail is one of two possibilities.

Check to ascertain if the individual probabilities add to the value of one.

$$\frac{1}{2} + \frac{1}{2} = 1$$

Now substitute the values reasoned into the expected value equation.

$$EV = (1 * 100) * \frac{1}{2} + (-1 * 100) * \frac{1}{2}$$

$$EV \qquad\qquad = \mathbf{0}$$

The expected value is zero as was reasoned.

EXAMPLE

Should 4" or 5" drillpipe be rented if the following data are factual? In this problem the expected values of renting the 4" and 5" drillpipes will be computed and compared for the drilling of an interval of hole.

TABLE OF COST AND EXPECTED PERFORMANCE

DRILLPIPE SIZE inch	DELIVERY CHARGE $	RENTAL COST $/day	DRILL RATE ft/hr	Prob
4	1,000	800	290	0.20
			315	0.80
5	1,100	900	300	0.40
			350	0.60

HOLE: 10,000 ft of hole are to be drilled.

Cost basis: The drillpipe cost while drilling 10,000 ft of hole.

C_1 = cost of drilling 10,000 ft of hole at 290 ft/hr

$$C_1 = \frac{10000}{290} * \frac{800}{24} \qquad \text{note: } C = \frac{\text{ft}}{\text{ft/hr}} * \frac{\$/\text{day}}{24\text{hr/day}} = \$$$

P_1 = probability of drilling at 290 ft/hr is 0.2

C_2 = cost of drilling 10,000 ft of hole at 315 ft/hr

$$C_2 = \frac{10000}{315} * \frac{800}{24}$$

P_2 = probability of drilling at 315 ft/hr is 0.8

The $1,000 entry is the delivery charge for the drillpipe.

$$EV_{4"} = [\frac{10000}{290} * \frac{800}{24}] * .2 + [\frac{10000}{315} * \frac{800}{24}] * .8 + 1,000 \qquad = \mathbf{\$2,076}$$

The costs and probabilities for the 5" selection are reasoned in a manner similar to the 4" drillpipe.

$$EV_{5"} = [\frac{10000}{300} * \frac{900}{24}] * .4 + [\frac{10000}{350} * \frac{900}{24}] * .6 + 1,100$$

$EV_{5"}$ = **$2,242**

The expected saving if 4" drillpipe is selected is **$166.**

EXAMPLE OF EXPECTED VALUE

A decision is required as to whether 8", 9", or 10" outside diameter drill collars are to be used to drill a tough 1,000 ft section of hole. A data base and an analysis have yielded the following expected information.

C	= OD of the collars
Drill rate;ft/hr	= 3 * C
Bit Cost	= $2,500
Mud Cost; $	= 1,500 + 500 * C
Bit consumption	= 16/C
Trip time; hr	= 8 + C/5
Rig rate; $/hr	= 150
Cost of fishing	= 3,000 * C

Probability of fishing: In the past 72 sections drilled, the number of fishing jobs was 6 plus the collar size divided by 2.

The problem will be resolved by comparing the expected computed costs of the three collar sizes.

Cost basis: cost to drill the 1,000 ft section.

C_1 = cost with fishing cost

$$C_1 = \underbrace{(2500 * \frac{16}{C})}_{\text{bits}} + \underbrace{(1500 + 500 * C)}_{\text{mud}} + [\underbrace{\frac{1000}{3 * C}}_{\text{drill}} + \underbrace{(8 + \frac{C}{5})}_{\text{trip}}] * 150 + \underbrace{3000 * C}_{\text{fish}}$$

$$C_1 = \frac{90000}{C} + 2{,}700 + 3{,}530 * C$$

P_1 = probability of fishing is given by the equation

$$P_1 = \frac{(6 + \frac{C}{2})}{72}$$

C_2 = cost per foot and not fishing

$$C_2 = \underbrace{(2{,}500 * \frac{16}{C})}_{\text{bits}} + \underbrace{(1{,}500 + 500 * C)}_{\text{mud}} + [\underbrace{\frac{1000}{3 * C}}_{\text{drill}} + \underbrace{(8 + \frac{C}{5})}_{\text{trip}}] * 150$$

$$C_2 = \frac{90000}{C} + 2{,}700 + 530 * C$$

$$P_2 = \text{probability of not fishing} = \quad 1 - P_1 = \frac{(66 - \frac{C}{2})}{72}$$

$$EV = (\frac{90000}{C} + 2{,}700 + 3{,}530\ C) * \frac{(6 + \frac{C}{2})}{72} + (\frac{90000}{C} + 2{,}700 + 530\ C) * \frac{(66 - \frac{C}{2})}{72}$$

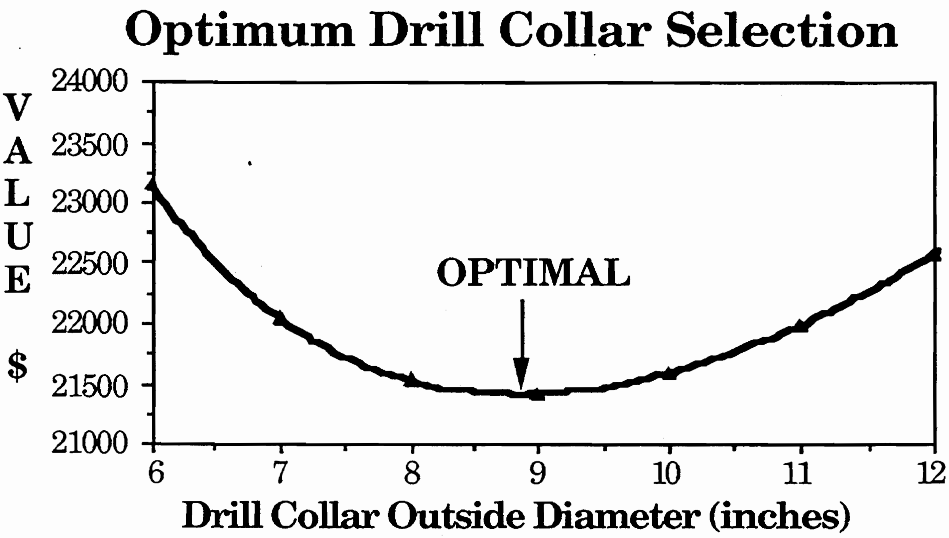
Substituting the values of 8", 9", and 10" yields

$EV_{8"}$ = **\$21,523**

$EV_{9"}$ = **\$21,407**

$EV_{10"}$ = **\$21,583**

This analysis predicts that the 9" drill collars should be selected.



EXAMPLE OF EXPECTED VALUE

A study with a database shows that for 100 trips, if the hole were circulated for two hours no fishing jobs occurred; however, if the holes were not circulated, ten fishing jobs occurred. Also, the study showed that for every hour of circulation, fishing costs were reduced from \$20,000 by \$7,000. How many hours of circulation should there be between cessation of drilling and tripping?

Cost basis: cost to drill 500 ft of the section. 500 ft is an average bit run. Values of the variables are as shown in the equations.

C_1 = cost with fishing cost

$$C_1 = \underset{\text{bit}}{2{,}000} + \underset{\text{mud}}{5{,}000} + [\underset{\text{drill}}{16} + \underset{\text{trip}}{8}] * \underset{\text{rig}}{400} + \underset{\text{fish}}{(20{,}000 - 7{,}000\ T)} + \underset{\text{circulation}}{400\ T}$$

$$C_1 = 36{,}600 - 6{,}600\ T$$

P_1 = probability of fishing is given by the equation

$$P_1 = \frac{(10 - 5\,T)}{100} = 0.1 - 0.05\,T$$

C_2 = cost per foot and not fishing

$$\begin{array}{ccccccccc} & \text{bit} & & \text{mud} & & \text{drill trip} & & \text{rig} & & \text{circulation} \\ C_2 = & 2{,}000 & + & 5{,}000 & + & [16 + 8] & * & 400 & + & 400\,T \end{array}$$

$$C_2 = 16{,}600 + 400\,T$$

P_2 = probability of not fishing is $1 - P_1 = 1 - \dfrac{(10 + 5\,T)}{100}$

$$P_2 = 0.9 + 0.05\,T$$

$$EV = (36{,}600 - 6{,}600\,T) * (0.1 - .05\,T) + (16{,}600 + 400\,T) * (0.9 + .05\,T)$$

$$EV = 18{,}600 - 1{,}300\,T + 350\,T^2$$

From calculus, by taking the derivative of the equation with respect to T and setting the resulting equation equal to zero yields,

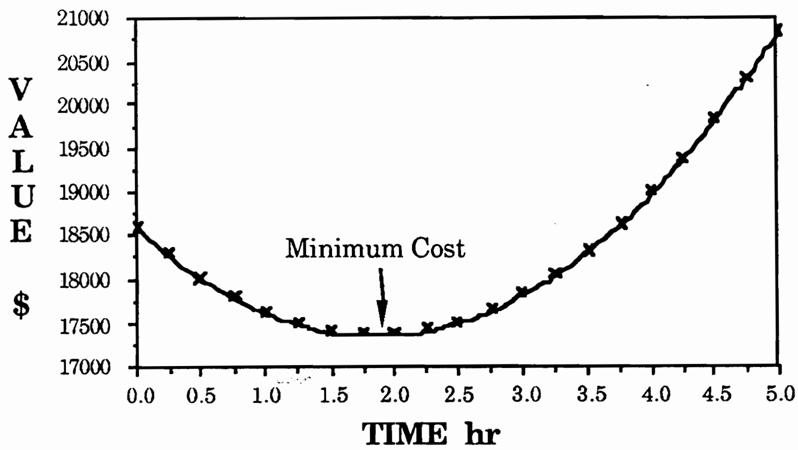
$$700\,T - 1{,}300 = 0$$

From which

T **= 1.86 hr**

The analysis predicts that the optimal circulation time is **1.86 hours**.

OPTIMUM CIRCULATING TIME



LAGRANGIAN MULTIPLIER

The Lagrangian multiplier technique may find maximums and minimums where an objective and a constraining equation exist. The technique is shown with an example.

EXAMPLE OF OPTIMUM PIT DIMENSIONS

The perimeter of a rectangular mud pit is to be 144 ft. Its depth is 8 ft. What should its length, L, and its width, w, be to maximize its volume?

Identify the objective equation; it is, **f = 8 w L**. The objective equation gives the parameter to be maximized.

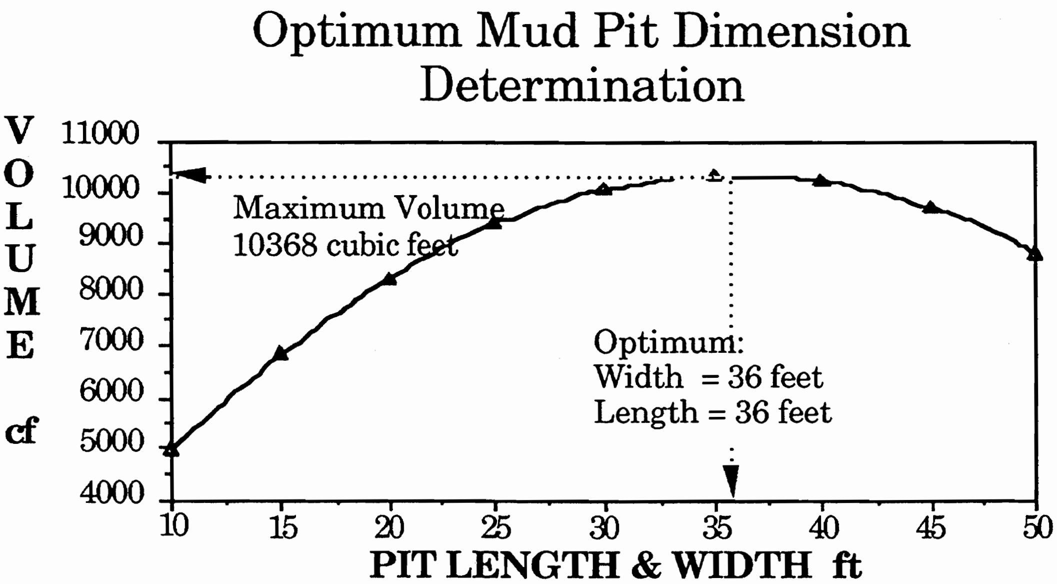
Identify the constraint equation; it is **g = 2 w + 2 L - 144**. The constraint equation limits the values of the independent variables which are w and L.

Lagrange's equation is the following.

$$\frac{\partial f}{\partial L}\frac{\partial g}{\partial w} - \frac{\partial f}{\partial w}\frac{\partial g}{\partial L} = 0$$

Take partials of f and g, and substitute into Lagrange's equation.

$$\frac{\partial f}{\partial L} = 8w \qquad \frac{\partial f}{\partial w} = 8L \qquad \frac{\partial g}{\partial L} = 2 \qquad \frac{\partial g}{\partial w} = 2$$



Substitution yields,

$$8\,w * 2 - 8\,L * 2 = 0$$

and then w = L

Substitute L for W in the g equation and find the L.

$$2L + 2L - 144 = 0$$

Yields

$L = 36$ ft and then w **= 36 ft.**

These are the required length and width to maximize the volume of the mud pit.

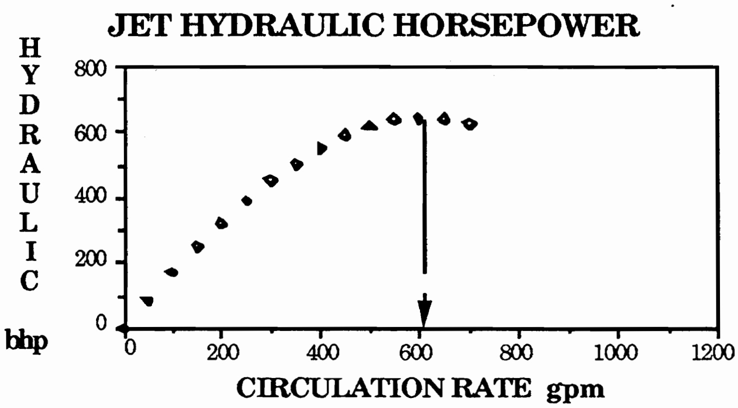
EXAMPLE

Maximize the hydraulic horsepower through the jets of a bit. The following relationships are assumed.

$$J_{hhp} = P_j Q \qquad P_j = MW\, Q^2 S^{-4} \qquad P_f = jQ^m \qquad P_p = P_f + P_j$$

J_{hhp}	= bit jet hydraulic horsepower
P_j	= pressure drop through jets
Q	= circulation rate
S	= jet number (size in 32nds of an inch)
P_f	= friction loss through circulation system except jets
P_p	= operating pump pressure
m w	= mud weight

Lagrange's objective equation and the variable to maximize is



$f = P_j\, Q$

Subsitution for P_j gives

$f = mw\, Q^3\, S^{-4}$

The constraint equation is

$g = P_f + P_j - P_p$

Substitution for P_f and P_j gives

$g = jQ^m + mw\, Q^2\, S^{-4} - P_p$

Lagrange's partial functions are

$$\partial f/\partial s = -4\, mw\, Q^3\, S^{-5}$$

$\partial f/\partial Q = 3 \text{ mw } Q^2 S^{-4}$

$\partial g/\partial s = -4 \text{ mw } Q^2 S^{-5}$

$\partial g/\partial Q = m\, jQ^{m-1} + 2 \text{ mw } Q\, S^{-4}$

Lagrangian equation

$$(-4 \text{ mw } Q^3 S^{-5})\,(m\, jQ^{m-1} + 2 \text{ mw } Q\, S^{-4}) - (3 \text{ mw } Q^3 S^{-4})\,(-4 \text{ mw } Q^2 S^{-5}) = 0$$

which reduces to the following

$$MW\, Q^2 S^{-4} - m\, jQ^m = 0$$

after replacing parameters becomes

$$P_f = \frac{1}{m} P_j$$

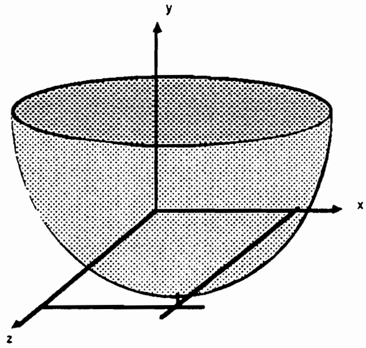
substituting Pf in g yields

$$P_j = \frac{m}{m+1} P_p$$

$$P_f = \frac{1}{m+1} P_p$$

Of course these are the equations for maximizing Jet Hydraulic Horsepower.

MULTIPLE REGRESSION WITH LEAST SQUARES



Multiple regression is used to find the parameters of an equation which causes the equation to best represent the data. If a computer is available an alternative which is not associated at all with multiple regression and is very powerful is the table look-up method.

The multiple regression method is illustrated with two independent variables, X, Z, and three parameters, k, b, and c.

Compute k, b, and c with the least squares method for this popular engineering equation.

$$Y = k\, X^b\, Z^c$$

Begin by putting the equation into linear form by taking logarithms

$$\mathrm{Ln}\, Y = \mathrm{Ln}\, k + b\, \mathrm{Ln}\, X + c\, \mathrm{Ln}\, Z$$

For ease of presentation

let $a = \mathrm{Ln}\, k$ $\quad X_1 = \mathrm{Ln}\, Y$ $\quad X_2 = \mathrm{Ln}\, X$ $\quad X_3 = \mathrm{Ln}\, Z$

The transformed equation is

$$X_1 = a + b\, X_2 + c\, X_3$$

Next, the three least squares equations are solved simultaneously for a, b, and c.

$$\Sigma X_1 = a\, N + b\, \Sigma X_2 + c\, \Sigma X_3$$

$$\Sigma X_1X_2 = a\, \Sigma X_2 + b\, \Sigma X_2X_2 + c\, \Sigma X_3X_2$$

$$\Sigma X_1X_3 = a\, \Sigma X_3 + b\, \Sigma X_2X_3 + c\, \Sigma X_3X_3$$

Cramer's rule gives the solutions

$$D = \begin{vmatrix} N & \Sigma X_2 & \Sigma X_3 \\ \Sigma X_2 & \Sigma X_2X_2 & \Sigma X_3X_2 \\ \Sigma X_3 & \Sigma X_2X_3 & \Sigma X_3X_3 \end{vmatrix}$$

$$a = \frac{\begin{vmatrix} \Sigma X_1 & \Sigma X_2 & \Sigma X_3 \\ \Sigma X_1X_2 & \Sigma X_2X_2 & \Sigma X_3X_2 \\ \Sigma X_1X_3 & \Sigma X_2X_3 & \Sigma X_3X_3 \end{vmatrix}}{D}$$

$$b = \frac{\begin{vmatrix} N & \Sigma X_1 & \Sigma X_3 \\ \Sigma X_2 & \Sigma X_1X_2 & \Sigma X_3X_2 \\ \Sigma X_3 & \Sigma X_1X_3 & \Sigma X_3X_3 \end{vmatrix}}{D}$$

$$c = \frac{\begin{vmatrix} N & \Sigma X_2 & \Sigma X_1 \\ \Sigma X_2 & \Sigma X_2X_2 & \Sigma X_3X_1 \\ \Sigma X_3 & \Sigma X_2X_3 & \Sigma X_3X_1 \end{vmatrix}}{D}$$

k is computed with the equation: $\mathbf{k = e^a}$

EXAMPLE OF LEAST SQUARES

Find: k, b, and c, of the equation,

$$Y = k\ X^b Z^c$$

The linear form of the equation is

$$\text{Ln } Y = \text{Ln } k + b \text{ Ln } X + c \text{ Ln } Z$$

For ease of presentation let

$X_1 = \text{Ln } Y \quad X_2 = \text{Ln } X \quad X_3 = \text{Ln } Z \quad a = \text{Ln } k$

==DATA==			----------------COMPUTATIONS----------------						
X	Z	Y	X2	X3	X1	X1X2	X1X3	X2X2	X2X3
3	5	34	1.10	1.61	3.53	3.87	5.68	1.21	1.77
8	10	164	2.08	2.30	5.10	10.60	11.74	4.32	4.79
17	15	514	2.83	2.71	6.24	17.69	16.90	8.03	7.67
26	24	1252	3.26	3.18	7.13	23.24	22.67	10.62	10.35
37	35	2594	3.61	3.56	7.86	28.39	27.95	13.04	12.84
50	48	4804	3.91	3.87	8.48	33.16	32.82	15.30	15.14

$$\begin{aligned} n &= 6.00 \\ \Sigma X_1 &= 38.34 \\ \Sigma X_2 &= 16.79 \\ \Sigma X_3 &= 17.22 \\ \Sigma X_1X_2 &= 116.95 \\ \Sigma X_1X_3 &= 117.76 \\ \Sigma X_2X_2 &= 52.52 \\ \Sigma X_2X_3 &= 52.57 \end{aligned}$$

D	=	2.06
a	=	0.74
b	=	0.85
c	=	1.14
k	=	2.10

The sought after equation becomes

$$Y = 2.10\ X^{0.85}\ Z^{1.14}$$

EXAMPLE

A popular friction loss hydraulic equation is $P_f = j\ Q^m$

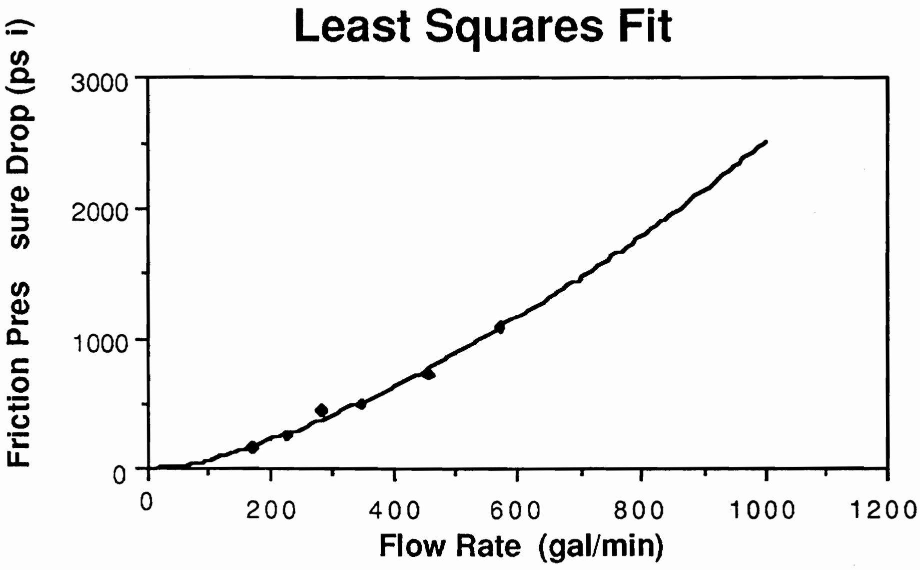
Find: j and m with the hydraulic rig test given in the table below.

Linear form: $\text{Ln}\ P_f = \text{Ln}\ j + m\ \text{Ln}\ Q$

Let: $X_1 = \text{Ln}\ P_f$ $\quad X_2 = \text{Ln}\ Q$ $\quad a = \text{Ln}\ j$

--DATA--		---------COMPUTED--------				---CHECK---	
Q	P_f	X_2	X_1	X_1X_2	X_2X_2	Y '	%error
569	1085	6.34	6.99	44.34	40.24	1084	0.12
455	738	6.12	6.60	40.42	37.46	776	-5.18
347	505	5.85	6.22	36.41	34.21	518	-2.59
285	444	5.65	6.10	34.46	31.95	386	13.01
228	272	5.43	5.61	30.44	29.48	277	-1.79
171	172	5.14	5.15	26.47	26.44	180	-4.79

n	=	6.00
ΣX_1	=	36.67
ΣX_2	=	34.54
ΣX_1X_2	=	212.53
ΣX_2X_2	=	199.78
D	=	5.89
a	=	-2.48



The values of j and m are computed to be

$j = 0.084$
$m = 1.492$

and the equation is

$$P_f = 0.084\ Q^{1.492}$$

CONFIDENCE LINES

Confidence lines are computed and drawn to provide the following type of answer: 'if 100 holes are drilled, 95 of the holes will have an expected maximum cost of $125,000 per hole and a minimum expected cost of $75,000 per hole and an expected cost of $100,000 per hole'. These are often called the maximum, minimum, and best values.

Draw two confidence lines and the least squares line for the linear equation

$$Y = a + b\,X$$

Begin by computing a, b, and c

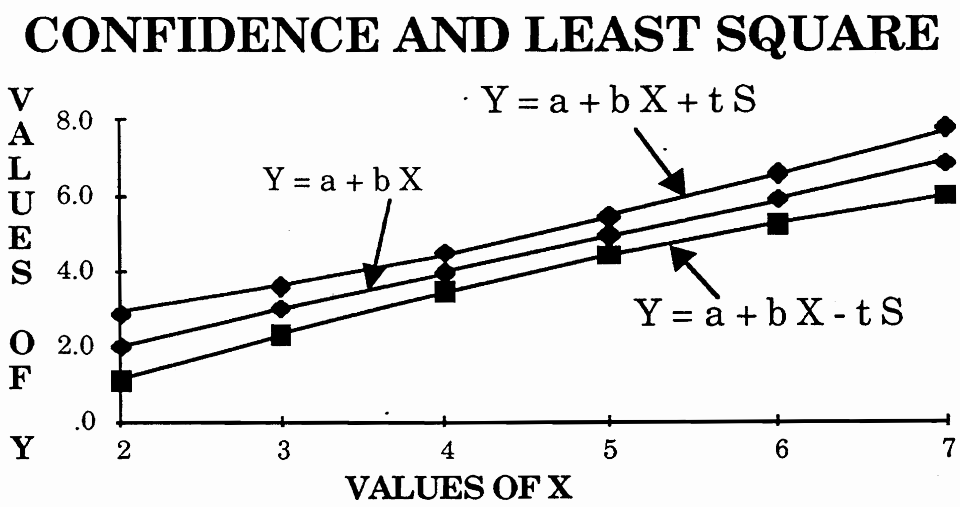
$$b = \frac{N\,\Sigma XY - \Sigma X\,\Sigma Y}{N\,\Sigma X^2 - (\Sigma X)^2} \qquad a = \frac{\Sigma Y}{N} - b\frac{\Sigma X}{N} \qquad c = \frac{\Sigma X}{N}$$

X, Y are data point pairs
N is number of data point pairs

The least squares line is given by the equation

$$Y = a + b\,X$$

Ascertain the value of the student's 't' distribution



For example:

if 95% confidence is desired and N = 4

then $\Gamma = 1 - 95$ and

then $t_{n\text{-}2,\,1\text{-}\Gamma/2} = t_{2,\,.975} = 4.30$ (value depends on number of data points)

The two confidence lines as shown in the sketch are

$Y = a + b\,X + t\,S$ upper line

$Y = a + b\,X - t\,S$ lower line

Where S is given by the equation below

$$S = \left[\frac{1}{N} + \frac{(X - c)^2}{\Sigma(X - c)^2}\right]^{1/2} * \left[\frac{\Sigma(Y - a - bX)^2}{N - 2}\right]^{1/2}$$

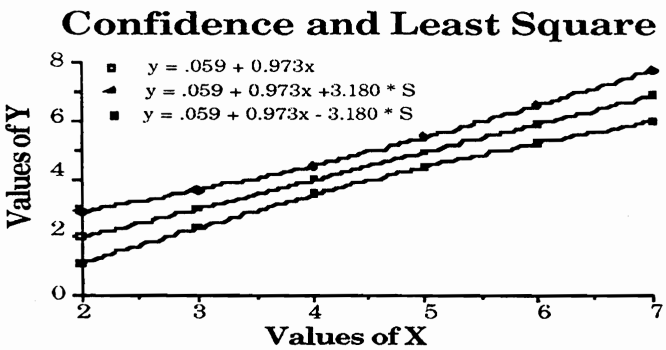
EXAMPLE

Find the confidence lines for the following data.

X DATA	Y DATA	X*Y	X*X	$(X-c)^2$	$(Y-a-bX)^2$
2.106	2.321	4.888	4.435	5.609	.046
3.406	3.159	10.760	11.601	1.141	.045
4.692	4.264	20.007	22.015	.047	.129
5.432	5.692	30.919	29.507	.917	.122
6.736	6.621	44.599	45.374	5.115	.000

n	= 5.000	Σ X	=	22.372
Xbar	= 4.474	Σ Y	=	22.057
Ybar	= 4.411	$\Sigma(X-Xbar)^2$	=	12.830
Σ XY	= 111.172	$\Sigma(Y-a-bX)^2$	=	.342

Σ X2	**= 112.931**
a	**= .059**
b	**= .973**
n-2	**= 3.000**
1-Gam/2	**= .975**
student t	**= 3.180**



Selected pairs for plotting chart

Xo	Yo	Yo+	Yo-	S
2	2.004	2.888	1.121	.278
3	2.977	3.630	2.325	.205
4	3.950	4.451	3.449	.157
5	4.923	5.428	4.417	.159
6	5.895	6.558	5.232	.208
7	6.868	7.765	5.972	.282

LAGRANGE'S NON-LINEAR INTERPOLATION FORMULA

The following abbreviated form of Lagrange's interpolation formula fits a second order polynomial ($y = a + b\,x + c\,x^2$) through three data points and then computes a value of y corresponding with a value of a selected x.

$$y = y_1 \frac{(x-x_2)(x-x_3)}{(x_1-x_2)(x_1-x_3)} + y_2 \frac{(x-x_1)(x-x_3)}{(x_2-x_1)(x_2-x_3)} + y_3 \frac{(x-x_1)(x-x_2)}{(x_3-x_1)(x_3-x_2)}$$

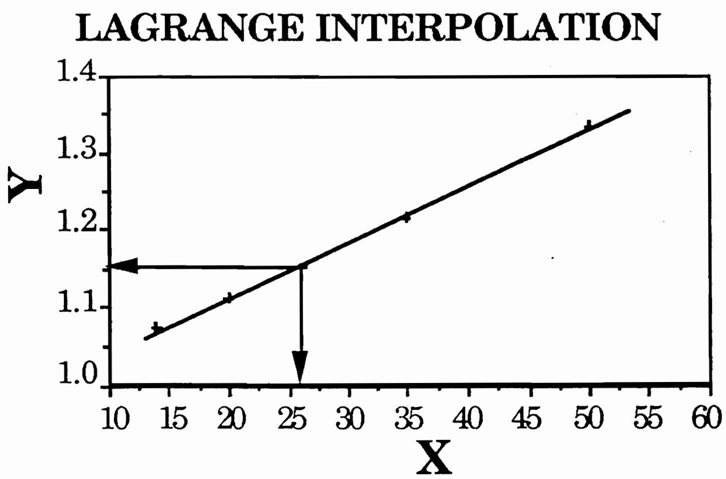
EXAMPLE

Given the following table of x and y values find the value of y which corresponds with the value of x of 26.

x	14	20	26	35	50
y	1.0764	1.1134	?	1.2160	1.3350

Choose the points 20,1.1134; 35,1.2160; 50,1.3350 to substitute into Lagrange's interpolation formula. Note that the points 14, 20, and 35, could have also been chosen.

$$y = 1.1134 \frac{(26-35)(26-50)}{(20-35)(20-50)} + 1.2160 \frac{(26-20)(26-50)}{(35-20)(35-50)} + 1.3350 \frac{(26-20)(26-35)}{(50-20)(50-35)}$$



Reduction gives the value of y corresponding to the value of **26 for x,**

y = **1.1525**

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CHAPTER III

DRILL HOLE MECHANICS

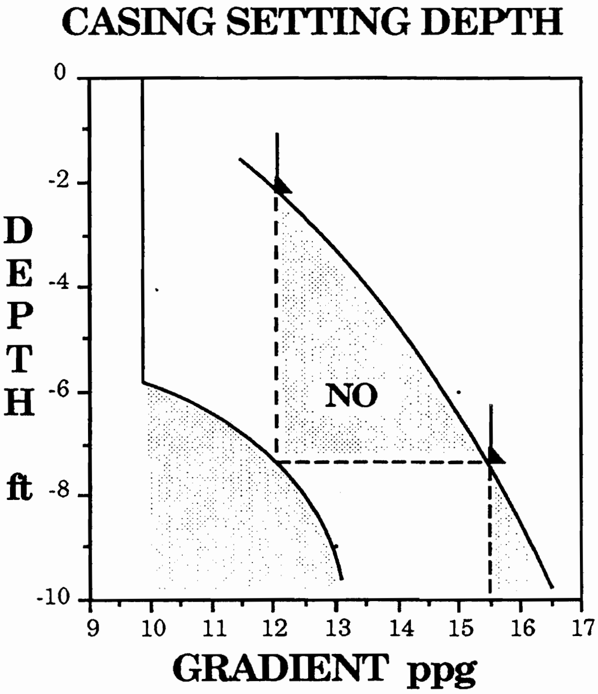
INTRODUCTION

Drill hole mechanics is the topic which aids most of all in the choice of a mud weight for drilling a section of hole. The choice of mud weights is one of the most analytically complex, political taxing, and critical task. The following list are those factors which may have an effect.

1. Fracture gradients (there are two)
2. Pore pressure
3. Kick tolerance
4. Casing shoe depths
5. Borehole stability (sloughing formations)
6. Surface pressure control equipment
7. Annular circulating pressures
8. Pressure surges (swabbing and running pipe)
9. Differential sticking of pipe
10. Filtration of mud
11. Filling of the hole
12. Gas cutting of the mud
13. Bit hydraulics
14. Mud cost
15. Drilling rate
16. Removal of drill solids
17. Formations porosity, permeability, and fluids
18. Safety margin over the pore pressure
19. Safety margin under the fracture gradients
20. Electric log analyst
21. Geologist (cuttings analysis)
22. Reservoir engineering (formation damage)

SELECTING CASING SETTING DEPTHS

Of the topics in the above list, casing shoe depth is most critical. Casings must be set at correct depths in order to drill a well into the earth with the rotary drilling system. The sketch considers only a pore pressure gradient and well depth.



The least feet of casing will be run and set in a practical well if the casings are planned from the bottom to the top of the well.

The crucial points are that the mud must be greater than the pore pressure gradient but less than the weakest fracture gradient in the open hole which is usually at the shoe of the last casing string.

Thus, as shown in the sketch the deepest that the hole can be drilled below the casing which is set at a depth of 2,000' is 7,200'. An attempt to drill deeper will result in a kick if permeable beds are present at 7,200' or lost circulation at the casing shoe if the mud weight is increased above the pore pressure at 7,200'.

As a consequence, the casing string must be set below 2,000' to drill deeper than 7,200'.

A casing setting depth equation is

$$P_{ff} = \frac{C * LOT}{19.25} \geq `P_{m@shoe} = \frac{{}_eP_f * D - \rho_g * L - MW\,(D - C - L)}{19.25}$$

$$C = \frac{({}_eP_f - MW)\,D + (MW - \rho_g)\,L}{LOT - MW}$$

C	= required shoe depth; feet
${}_eP_f$	= equivalent formation pressure; ppg
D	= drilling depth; feet

MW	= drilling mud weight; ppg
L	= gas kick length; feet
ρ_g	= down hole density of gas; ppg
LOT	= leak off test value at depth C; ppg
P_{ff}	= resistance of the formation to fracturing; psi

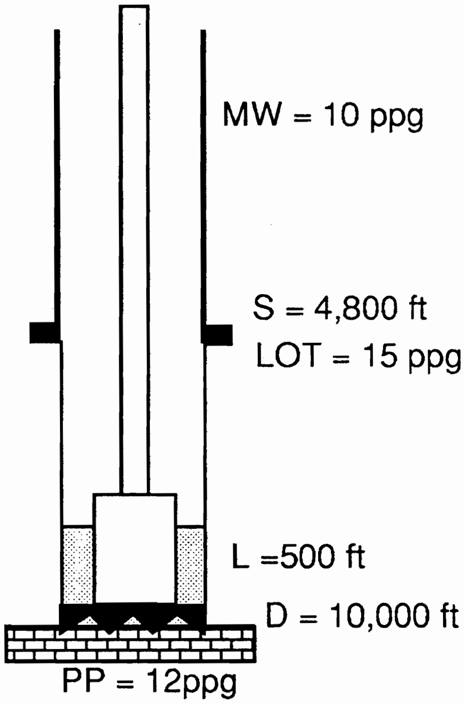
EXAMPLE

What is the required casing setting depth?

LOT = 15.0 ppg ${}_eP_f$ = 12 ppg

D = 10,000 feet   MW = 10 ppg

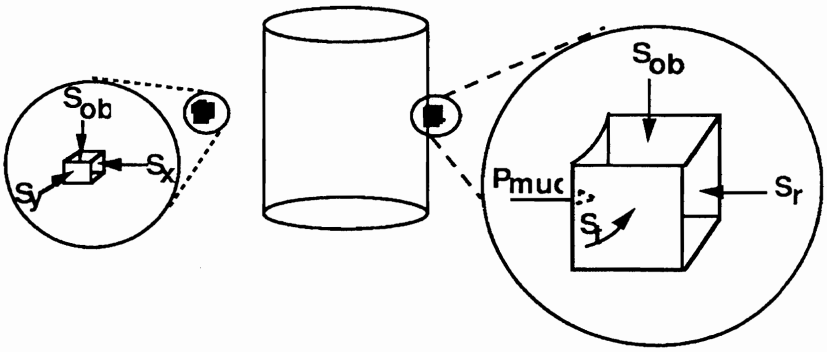
L = 500 feet ρ_g = 2 ppg



$$C = \frac{(12 - 10) * 10000 + (10 - 2) * 500}{15 - 10}$$ = **4,800 feet** (required shoe depth)

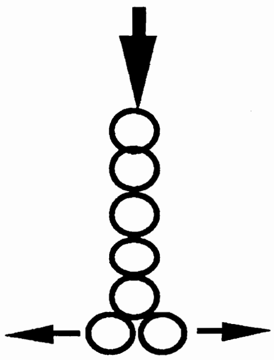
ENLARGEMENT & FRACTURE OF DRILL HOLES

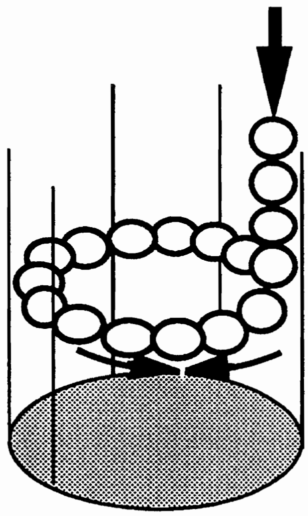
Drill holes are not 'washed out' by the annular mud stream. Stresses within the wall of a drill hole fails the rock (sand, shale, limestone, etc.) which is the wall of the hole causing this rock to fall into the hole where it is carried away by the annular mud stream.



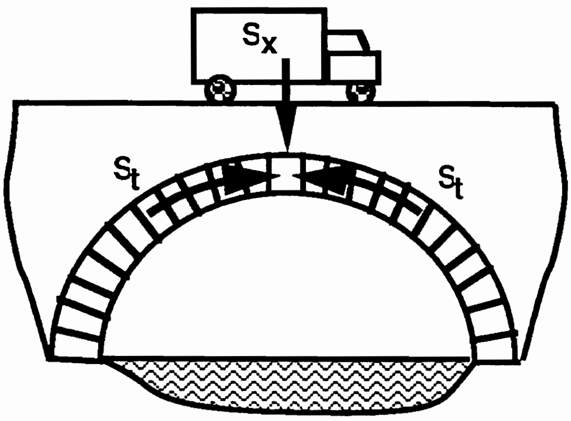
The stresses associated with a drill hole are the overburden, radial, tangential, and mud pressure.

A perception of the stresses may be gained by considering the sketch. Sand grains are resting on top of other creating an overburden stress. The wedging of the two grains not aligned creates the x direction and y direction stresses.



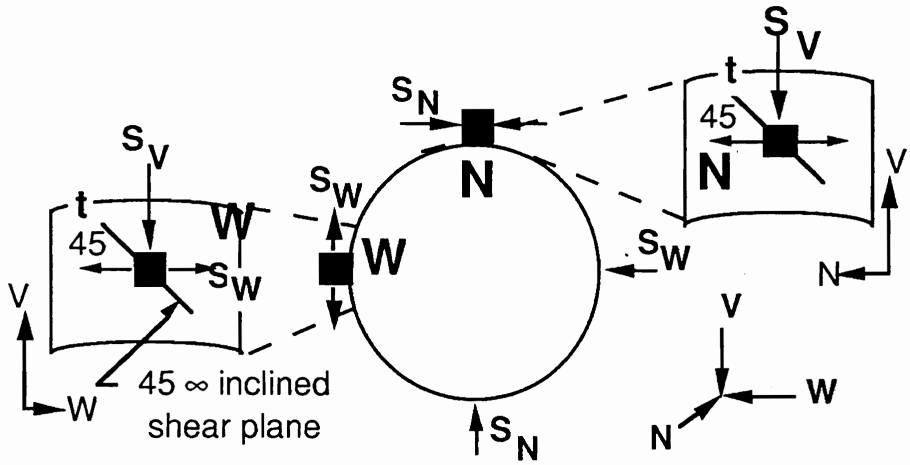


The two x and y stresses are used to discuss stresses removed from the hole while radial and tangential stresses are used to discuss stress near the drill hole. All stresses are orthogonal.



The radial and tangential stress may be perceived by imagining a masonry bridge which spans a stream and is supporting a truck. This is one-half of a drill hole viewed horizontally rather than vertically. The bricks will have a tangential stress which will be compressive in this case; however, in a drill hole the mud pressure could put the bricks into tension. If the resulting tension were larger than the tensile strength of the bricks a fracture would occur.

Mills and Topping developed equations for the compressive, tensile, and shear stresses around a drill hole for the case in which the two horizontal forces are not equal in value.



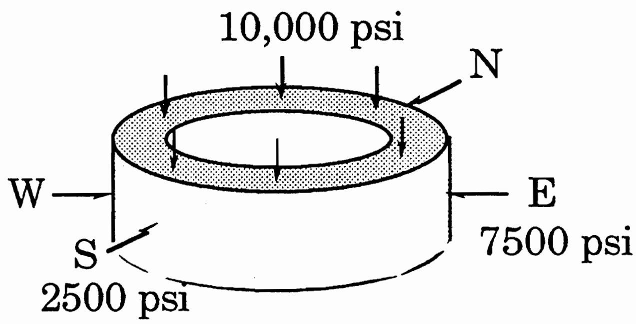
S_V	= overburden stress; psi
S_W	= tangential stress on west and east sides of hole; psi
S_N	= tangential stress on north and south sides of hole; psi
τ	= shear stress; psi
V	= vertical direction
W	= west or east side of hole
N	= north or south side of hole

Their equations for the stresses at the north and south sides and west and east sides are

$S_{N,tangential}$	$= -3 * (S_N - S_W)$	(compression at north side)
$S_{W,tangential}$	$= +1 * (S_N - S_W)$	(tension at west side)
$S_{V,N}$	$= -S_V - 2\,m\,(S_N - S_W)$	(compression at north side)
$S_{V,W}$	$= -S_V + 2\,m\,(S_N - S_W)$	(compression at west side)
τ_N	$= \dfrac{S_{V,N} - S_N}{2}$	(shear at north side)
τ_W	$= \dfrac{S_{V,W} - S_W}{2}$	(shear at west side)

EXAMPLE

The drill hole has penetrated a zone in which the overburden stress is 10,000 psi, the horizontal stress in the west east direction is 7,500 psi, and the horizontal stress in the north south direction is 2,500 psi. What are the stresses at the edge of the drill hole at the west east sides and north south sides?

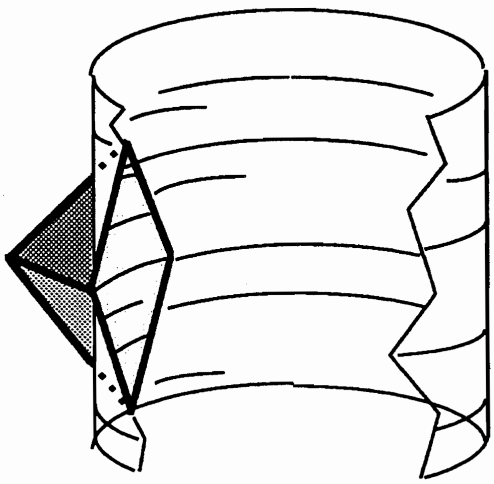


$S_N - S_W = -2500 - (-7500)$	**= + 5,000 psi**	(net stress at west side)
$S_{N,tangential} = -\ 3 * 5000$	**= -15,000 psi**	(compression at north side)
$S_{W,tangential} = +\ 1 * 5000$	**= +5,000 psi**	(tension at west side)
$S_{V,N} = -\ 10000 - 2\,(.25)\,5000$	**= -12,500 psi**	(compression at north side)
$S_{V,W} = -\ 10000 + 2\,(.25)\,5000$	**= -7,500 psi**	(compression at west side)
$\tau_N = \dfrac{-12500 - (-15000)}{2}$	**= +1,125 psi**	(shear at north side)
$\tau_W = \dfrac{-7500 - 5000}{2}$	**= -6,250 psi**	(shear at west side)

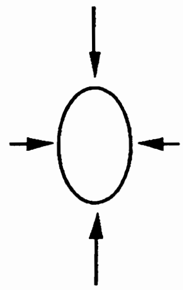
POINTS DEDUCED FROM THE ABOVE EXAMPLE

1. Because rocks are weaker in shear than in compression and from this example, it is expected that the hole is to enlarge in the west east direction preferentially over the north south direction because of the large shear stress on the west east sides.

2. The drill hole is expected to enlarge into an oval shape with the major axis aligned in the direction of the major stress, west east stress.



3. It may also be surmised that directional drill holes are expected to enlarge vertically preferentially over horizontally because the overburden stress is the major stress.



4. Directional drill holes are expected to require higher mud weights because the difference between the overburden stress which is acting on the upper side of a directional hole and the applicable horizontal stress is more than the differences in the horizontal stresses which would be acting in a vertical hole.

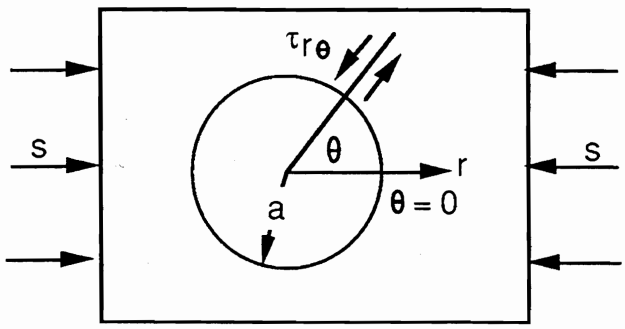
5. Directional drill holes are expected to have lower leakoff values because the sides of the drill hole which are aligned with the major stress is in tension. In the above example, the west and east sides are in 5,000 psi of tension.

Kirsch's equations are the following.

$$\sigma_r = \frac{s}{2}\left[1 - \left(\frac{a}{r}\right)^2 + \cos 2\theta \left(1 - 4\left(\frac{a}{r}\right)^2 + 3\left(\frac{a}{r}\right)^4\right)\right]$$

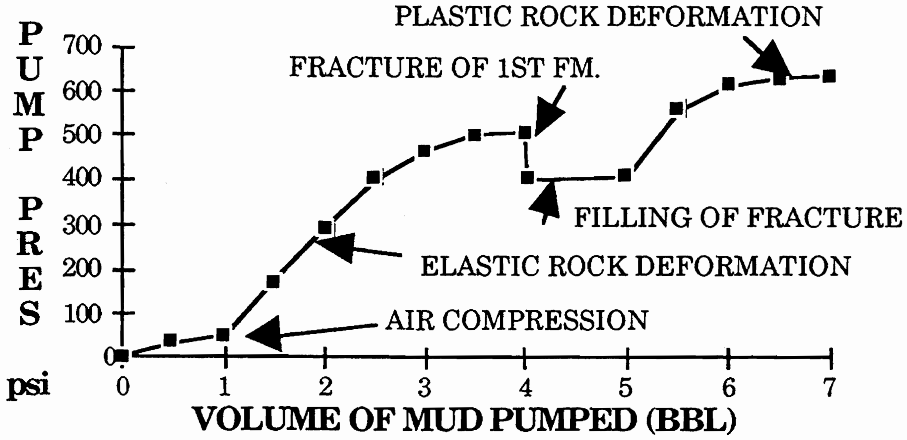
$$\sigma_\theta = \frac{s}{2}\left[1 + \left(\frac{a}{r}\right)^2 - \cos 2\theta \left(1 + 3\left(\frac{a}{r}\right)^4\right)\right]$$

$$\tau_{r\theta} = -\frac{s}{2}\left[1 + 2\left(\frac{a}{r}\right)^2 - 3\left(\frac{a}{r}\right)^4\right]\sin 2\theta$$



LEAKOFF TEST

LEAK OFF TEST AT 4,398'



Pressure integrity tests are conducted to ascertain the resistance of a formation penetrated by a wellbore to the initiation of a fracture within the formation. There are two phases of fracturing a formation: the fracture is initiated and then extended. The pressure required to cause initiation of a fracture is usually greater than the pressure causing extension.

A pressure integrity test is a test in which the drill hole is pressured at the surface by pumping mud into the drill hole. Usually a graph of surface casing pressure versus mud volume pumped is made.

A critical problem occurs during an integrity test because the magnitude of the strain at the initiation of the test is not known; i.e., the strain is not zero in value and one does not know at what point on the curve that the test commenced. This could lead to the fracturing of the drill hole.

Air residing in the surface equipment may also change the shape of the plot if not purged.

In practice the pump rate must be continuous and sufficiently reduced to permit the recording and the plotting of the data during the tests. Thus static equilibrium is not met but it is approached and bleed-off is minimized. Do not expect all of the mud pumped into the annulus to return after removing the surface pressure because a portion of a test will occur in the plastic deformation region of the open formations.

There are four categories of muds with which a L.O.T. may be run: penetrating versus non-penetrating mud (oil base base versus water base and high filtrate versus low filtrate mud) and clean versus contained solids.

Cementations and stimulations (with sand) show that plugging of fractures which were extending causes most drill holes to regain their initial strength and in some cases to surpass it. Thus, muds containing a high concentration of solids should, and in most cases do, have higher L.O.T. values than clean muds.

It is argued that penetrating muds are more likely to intersect a plane of weakness within the rock surrounding the drill hole than a non-penetrating mud. A second reason is that a mud with a low filtrate will act as a non-penetrating mud. Thus, it is frequently found that oil base muds will have a lower L.O.T. than a high filtrate water base mud.

Many formations show a significant amount of creep. Two factors are important: one is that all the mud pumped may not be returned from the drill hole and the second is that the slope of the plot depends on the pumping rate. Multiple open formations may give an 'S' plot.

EXAMPLE

Estimate the wellbore pressure required to initiate a fracture in a formation if

mw = 10.2 ppg   D = 5,200 feet

INTEGRITY TEST DATA

Casing Pressure, psig	Volume Mud Pumped, Bbls
240	1
460	2
680	3
800	4
940	5
1020	6
1090	7
1140	8

The data is plotted on the adjoining chart. Extrapolation of the data (dashed line) indicates a maximum pressure of 1220 psig. The equivalent density of this pressure at 5200 feet is

$$ED = \frac{1220}{.052 * 5200}$$

$$\mathbf{= 4.5\ ppg}$$

Thus the pressure required to initiate a fracture is

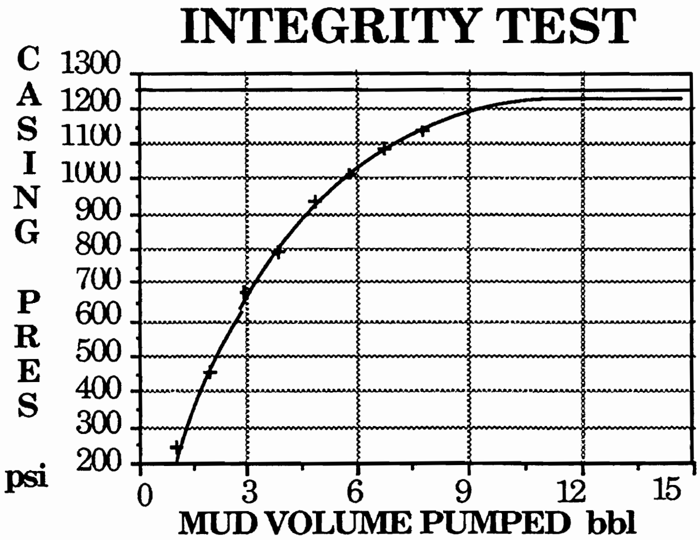
$$P_{init} = .052 * 10.2 * 5200 + 1220$$

$$\mathbf{= 3978\ psig}$$

and the equivalent mud density is

$$mwe = \frac{3978}{.052 * 5200}$$

$$\mathbf{= 14.7\ ppg}$$

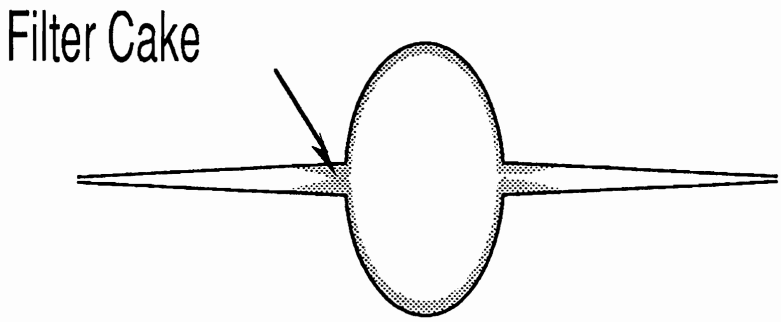


FRACTURES IN A DRILL HOLE

In order for a drill hole to fracture, the wall of the drill hole must be placed in tension in the tangential direction. Further, the magnitude of the tension must be greater than the tensile strength of the rock comprising the drill hole.

In drilling the resistance of a rock to fracturing is known as the 'leakoff test' value (LOT).

The leakoff test depends on the type of fluid within the drill hole which is adjacent to the rock to be tested. Clean water and oil have the lowest leakoff tests of all muds. Clean gas may have the lowest values of all fluids. Water base and oil base muds with high solids concentration have the highest leakoff tests.



Cementing has shown that zones may fracture and take all the mud or cement being circulated until the fracture is packed with solids and then circulation continues. In this case it seems that the pressure of the cement acting on the face of the zone in addition to other stresses place the zone in sufficient tension to cause it to fracture and diminish the tension. Thereafter, the packing of the fracture and building a filter cake within the hole raises the leakoff test.

It has been observed that leakoff tests for a particular zone increases with time.

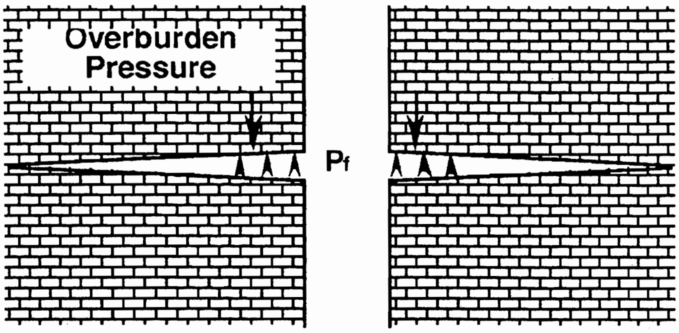
Clean fluids have a lower leakoff test because they penetrate and pressure zones further from the drill hole. As a consequence, a plane of weakness is more likely to be intersected and fractured.

Every zone appears to have two values for leakoff tests. One is the value required to fracture a drill hole for the first time. It is called the initiation fracture pressure. The other is the pressure required to extend a fracture and it is called the extension fracture pressure.

TYPES OF FRACTURES

Fractures within a wellbore may be aligned horizontally or vertically. Vertical fractures usually occur at depth. In regard to the drilling of a wellbore, fracturing is synonymous with loss of circulation.

HORIZONTAL FRACTURES



The pressure of the fluid in the wellbore must surpass the unit weight of the overburden (includes the rock and the fluid within its pores) and the tensile strength of the rock in order to lift the overburden and create a fracture.

$P_{hor} = s_{ob} + s_t$

P_{hor} = wellbore pressure to create a horizontal fracture; psi

s_{ob} = overburden stress; psi

s_t = unit tensile of rock being fractured; psi

VERTICAL FRACTURES

Two vertical fracture gradients exist for all points within a wellbore only because of the size and geometry of common drill holes. These two fracture gradients are called the fracture extension and initiation gradients.

Equal stress model

If a rock is equally stressed in all horizontal directions and the rock is totally contained and can not strain, the equation which relates vertical and horizontal stress may be derived with Poisson's ratio and Hooke's law.

$$e_y = \frac{S_y}{E} - \frac{v}{E}(S_x + S_z) \qquad \text{Hooke's law and Poisson's ratio}$$

s = stress; psi

v = Poisson's ratio; in/in

e_y = horizontal strain after stresses S_x and S_z; in/in

E = Young's modulus; psi

Because there is no strain, $e_y = 0$

$$\frac{S_y}{E} - \frac{v}{E}(S_x + S_z) = 0$$

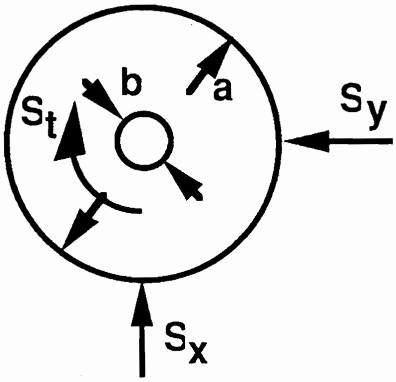
Because the horizontal stresses are equal ($S_x = S_y$), then

$$S_y = \frac{v}{1-v} S_z$$

Note that Poisson's ratio v has never been observed to be less than 0 or greater than 1/2.

Thus, the horizontal stresses may have values near zero if compared with that of the overburden stress.

If a cylindrical hole is drilled into a rock layer supporting overburden, the horizontal stresses (radial) will be relaxed at the wall of the hole. However, tangential stress will develop in the rock.



This tangential stress has been quantified by Lame and his equation is

$$S_t = 2 \frac{a^2}{a^2 - b^2} S_y$$

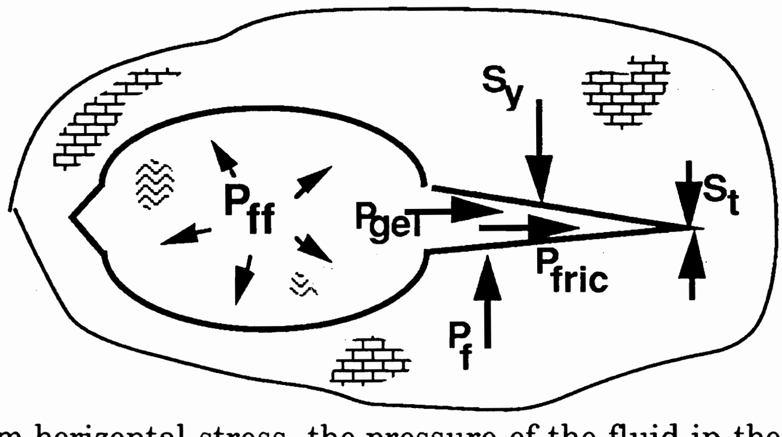
S_t = tangential stress; psi

If a is large relative to b as is the case of drill holes, then

$$S_t = 2 S_y$$

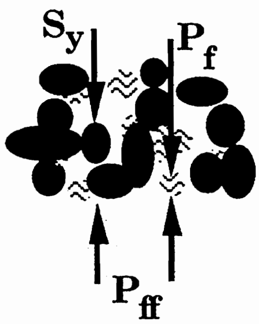
FRACTURE EXTENSION

If a vertical crack exists in a rock which is supporting an overburden, then the magnitude of the stress which is required to be placed on the face of the fracture in order to prevent its closure is that of the horizontal stress which is perpendicular to the fracture. If the fracture is to be extended, then the pressure required in the drill hole is the sum of, the minimum horizontal stress, the pressure of the fluid in the formation, tensile strength of the rock, and the friction (and gel if applicable) pressure loss in the fracture.



$$P_{ff} = S_y + P_f + S_t + P_{fric} + P_{gel}$$

EFFECT OF FLUID



If the rock contains a fluid and is both porous and permeable, then the pressure on the wall of the fracture will need to counter both the pressure of the fluid within the pores of the rock and the horizontal stress transmitted by the grains of the rock.

$$P_{fracture} = P_f + S_y$$

P_f = Pressure of fluid in the pores of the rock; psi

OVERBURDEN CONTAINING FLUID

Terzaghi suggested that an overburden stress is the sum of the fluid pressure and the stress transmitted by the grains of the rock.

$$S_{ob} = P_f + S_z$$

$$S_z = S_{ob} - P_f$$

and for a vertical fracture

$$S_y = \frac{v}{1\text{-}v}(s_{ob} - P_{fluid})$$

and for a cylinder (a drill hole)

$$S_y = \frac{2\,v}{1\text{-}v}(s_{ob} - P_{fluid})$$

INITIATION AND EXTENSION FRACTURE GRADIENTS

The equation for fracture extension is

$$P_{ext} = \frac{v}{1\text{-}v}(s_{ob} - P_{fluid}) + P_{fluid} + s_t$$

P_{ext} = pressure in the wellbore to cause fracture extension; psi

The equation for fracture initiation is

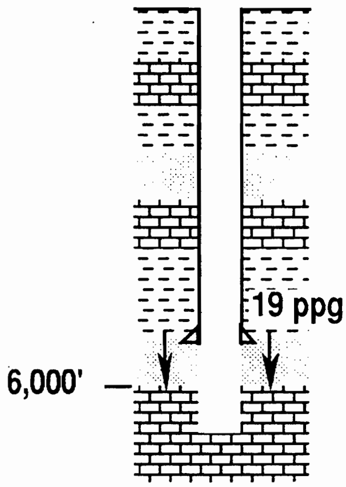
$$P_{init} = \frac{2\,v}{1 - v}(s_{ob} - P_{fluid}) + P_{fluid} + s_t$$

P_{init} = pressure in the wellbore to create a vertical fracture; psi

The two equations may be converted to equivalent densities by dividing both by the depth D.

EXAMPLE

The overburden gradient of a rock layer at 6,000' is 19 ppg. Poisson's ratio for the rock layer is 0.3 and its unit tensile strength is 50 psi. The effect of fluid pressure is to be illustrated.



Under pressured. Fluid pressure is 4 ppg.

$$MWE = \frac{.3}{1-.3}(19-4) + 4 + \frac{50}{.052 * 6000}$$
= **10.4 ppg** (extension)

$$MWE = \frac{.3 * 2}{1-.3}(19-4) + 4 + \frac{50}{.052 * 6000}$$
= **16.9 ppg** (initiation)

Normal pressured. Fluid pressure is 9 ppg.

$$MWE = \frac{.3}{1-.3}(19-9) + 9 + \frac{50}{.052 * 6000}$$ = **13.3 ppg** (extension)

$$MWE = \frac{.3 * 2}{1-.3}(19-9) + 9 + \frac{50}{.052 * 6000}$$ = **17.6 ppg** (initiation)

Overpressured. Fluid pressure is 17 ppg.

$$MWE = \frac{.3}{1-.3}(19-17) + 17 + \frac{50}{.052 * 6000}$$ = **17.9 ppg** (extension)

$$MWE = \frac{.3 * 2}{1-.3}(19-17) + 17 + \frac{50}{.052 * 6000}$$ = **18.7 ppg** (initiation)

It may be noted that fluid pressure gradients have a marked affect on the fracture extension gradients and a lesser affect on the initiation gradients.

EXAMPLE

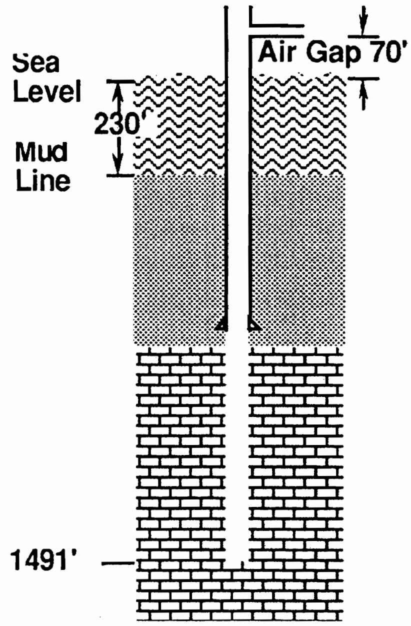
Calculate the fracture gradients of a formation at 1491 feet if the air gap (highest rise of the mud in the conductor pipe above sea level) is 70 feet, the sea depth is 230 feet, and the formations have an average specific gravity of 1.9, a Poisson's ratio of .2, and a tensile strength of 100 psi.

If the wellbore has not been previously fractured then the smaller value of either the horizontal or initiation fracture pressures must be chosen.

$$s_{ob} = 0*70 + .052 * 8.5 * 230 + .052 * 1.9 * 8.33 * (1491-300)$$ = **1082 psi**

The horizontal fracture pressure is

$$P_{hor} = 1082 + 100 \qquad = \mathbf{1182\ psi}$$



The expected pore pressure is

$$P_p = .052 * 8.5 * (1491\text{-}70) \qquad = \mathbf{628\ psi}$$

The vertical initiation fracture pressure is

$$P_{init} = (1082 - 628)\ 2\ \frac{.2}{1 - .2} + 628 + 100 \quad = \mathbf{955\ psig}$$

The maximum allowable mud weight is

$$0.052 * MW * 1491 = 955$$

$$MW \quad = \mathbf{12.3\ ppg}$$

Thus the wellbore at a depth of 1491 feet is expected to fracture vertically at a mud weight or equivalent mud weight of **12.3 ppg.**

EQUIVALENT MUD DENSITY

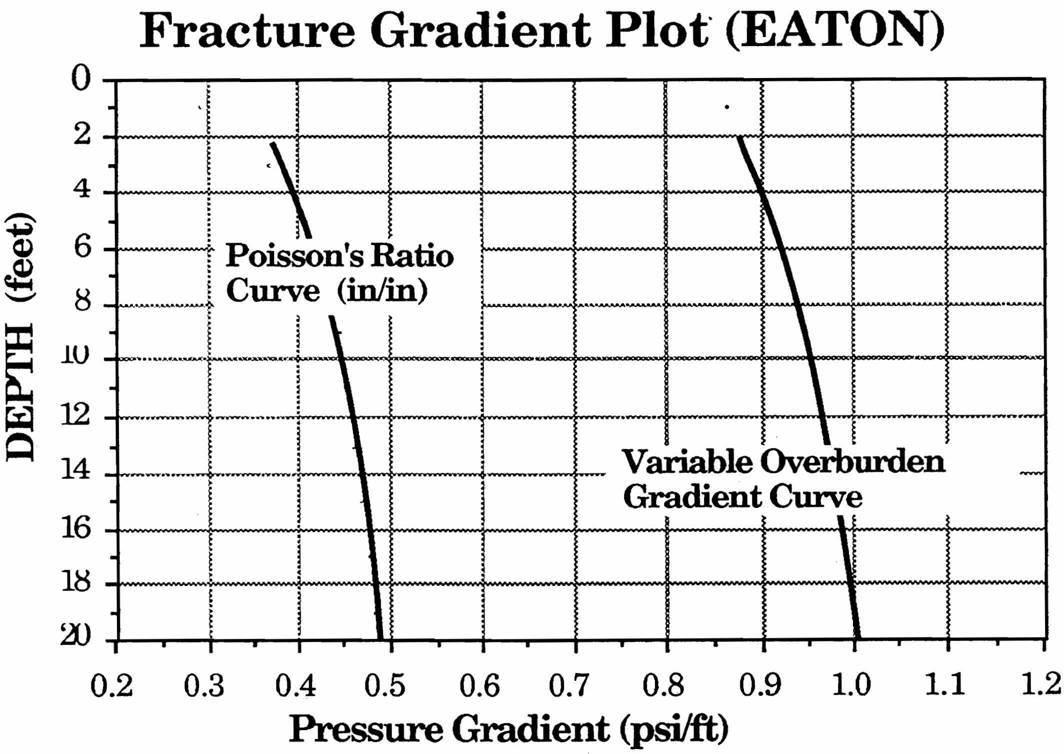
A pressure with the units of psi acting within a wellbore at a depth D may be converted to equivalent mud density or equivalent gradient in the units of ppg with either of the equations

$$MWE = \frac{P}{.052 * D} \text{ (not exact)} \qquad\qquad MWE = 19.25\,\frac{P}{D} \text{ (exact)}$$

MWE = equivalent mud weight; ppg
P = pressure; psi
D = depth; feet

FRACTURE GRADIENT PLOT (EATON)

In practical designs of fracture gradient plots problems may be solved by generating fracture gradient lines depicting feasible minimum mud weights. The mud weight selection sketch shows pressure gradients versus depth. The pore pressure gradients were chosen to illustrate several points. The two fracture gradients were calculated with the fracture initiation and extension equations; with Eaton's gulf coast variable overburden gradient and Poisson's ratio charts; and with a minimum formation tensile strength of 100 psi.



Equations which very closely approximate **Eaton's** published curves are:

Variable Overburden Gradient (psi/ft)

$$S_{ob} = \text{VOBG} = 0.84753 + 0.01494\, D - 0.0006\, D^2 + 1.199\text{E-}5\, D^3$$

r $> 0.99\%$
D = depth in 1,000's of feet

Poisson's Ratio (no units)

$$v = 0.23743 + 0.05945\, D - 0.00668\, D^2 + 0.00035\, D^3 - 6.71\text{E-}6\, D^4$$

r $> 0.99\%$
D = depth in 1,000's of feet

EXAMPLE OF FRACTURE GRADIENT

Calculate the fracture initiation and extension gradients at a depth of 8,000 feet. Use the approximations to Ben Eaton's data.

S_{ob} = VOBG $= 0.84753 + 0.01494 * 8 - 0.0006 * 8^2 + 1.199E\text{-}5 * 8^3$

= .935 psi/ft (also known as s_{ob})

v $= 0.23743 + 0.05945 * 8 - 0.00668 * 8^2 + 0.00035 * 8^3 - 6.71E\text{-}6 * 8^4$

= .437 in/in

MWE = 14 ppg (taken from the sketch at a depth of 8,000')

$$MWE_{init} = (.935 - .052 * 14)\ \frac{2 * .437}{1 - .437} + .052 * 14 + \frac{100}{8000}$$

= 1.062 psi/ft or 20.44 ppg

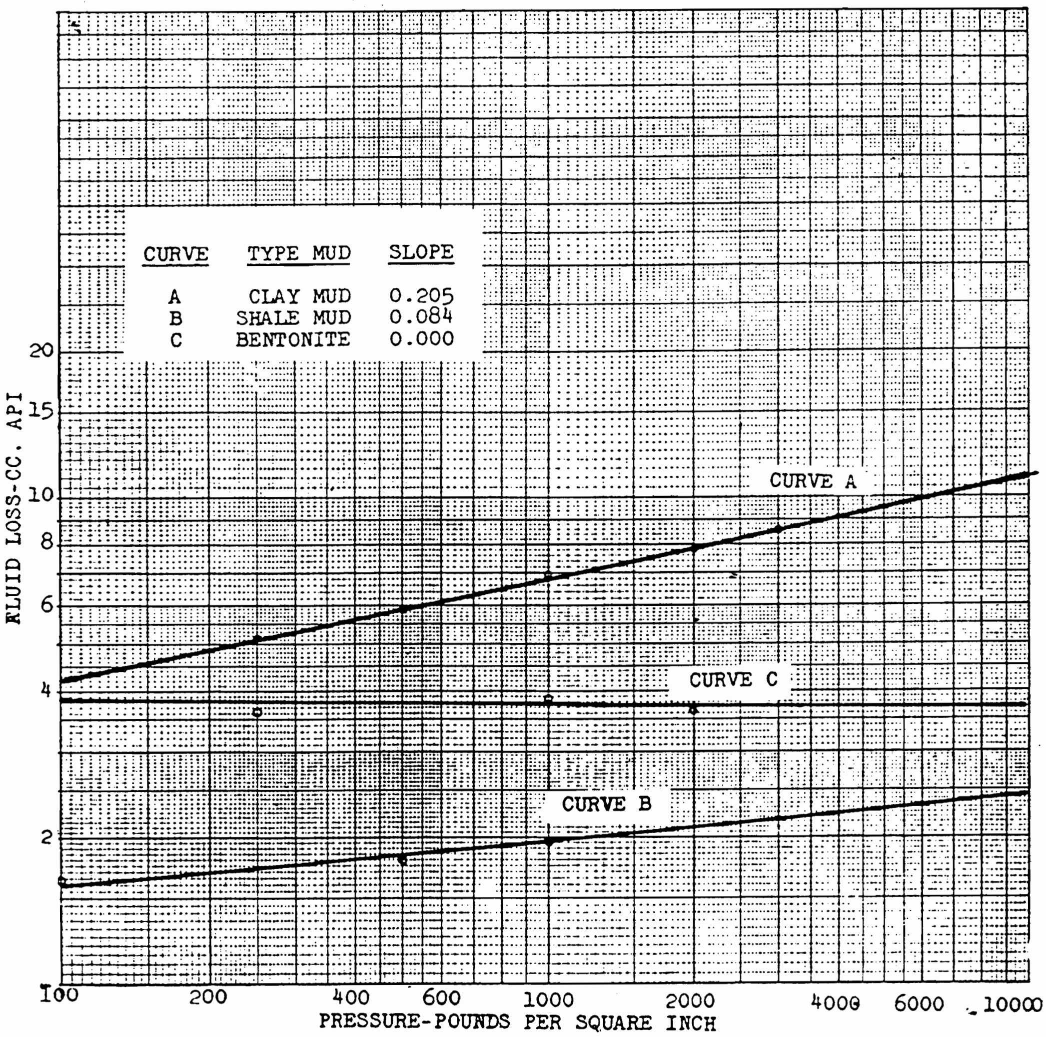
$$MWE_{ext} = (.935 - .052 * 14)\ \frac{.437}{1 - .437} + .052 * 14 + \frac{100}{8000}$$

= .901 psi/ft or 17.34 ppg

FILTRATION OF MUD INTO THE FORMATION

Two types of filtration exists within the borehole: static and dynamic. Dynamic filtration occurs when mud is being circulated and occurs at both the wall and the bottom of the hole. Static filtration tests have shown that bentonite muds have no additional filter loss with increases in pressure. This is depicted in the graph. Outman's has shown that dynamic filtration is also only slightly dependent on pressure (or pressure differentials) for incompressible filter cakes.

EFFECT OF PRESSURE



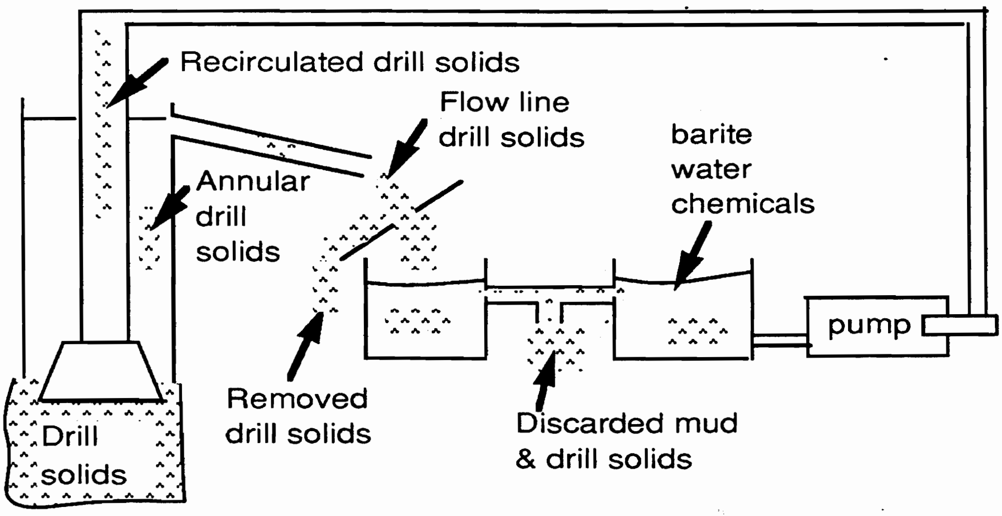
DRILL SOLIDS OPTIMIZATION

Shown below is a sketch of a typical flow loop for rotary drilling rigs. The drill solids are generated by drilling new hole and hole enlargement. Because drill solids accumulate in the annulus, the drill solids are at their highest concentration there. A portion of the solids are removed by the solids removal equipment and a portion are retained. This is called 'separation of solids'.

Because of the difficulties in removing solids, few old solids, i.e., those which have been recirculated and reduced in size, can not be mechanically separated; and only about 60% of the new drill solids can be separated. A fraction of the drill solids are removed by the discarding of mud. This is called 'discarding solids'.

Chemicals, barite, and water are added to reduce the concentration of drill solids; but, this does not remove the drill solids. This is called 'dilution of drill solids'. The solids which are retained in the mud are recirculated. Drill solids concentration is the ratio of the volume of the drill solids to the volume of the mud including the drill solids, and is reported as percent drill solids.

$$\%\text{Drill Solids} = \frac{\text{Volume Drill Solids}}{\text{Volume Mud}}$$



One major problem exists in the optimization of mud costs. It is the selection of the concentration of recirculated drill solids to be allowed in the mud. As drill solids are allowed to increase, drill rate decreases and mud consumption decreases. A decrease in drill rate means more bit to drill a section of hole, more rotation time, more trip time, and more support time. Thus, cost per foot for the rig goes up and mud costs goes down. A balance is required.

For a weighted mud the recirculated solids concentration can be estimated by comparing the barite and water added to the mud while drilling a section of hole with the computed amounts required. For an un-weighted mud, the comparison must be based on only the water added.

BARITE & WATER REQUIRED TO DRILL A SECTION OF HOLE

A definite quantity of barite and water are required to drill a section of hole. The quantity of each depends on the length and size of the hole drilled, efficiency of the solids removal equipment, percentage of retained solids, volume of the active mud system, and the mud weight. Thus, following formula may be reasoned.

$$\boxed{\begin{array}{c}\text{INITIAL MUD}\\ V_i\\ \rho_i\\ S_{rds\ i}\end{array}} + \boxed{\begin{array}{c}\text{New}\\ \text{Drill}\\ \text{Solids}\end{array}} - \boxed{\begin{array}{c}\text{Separated}\\ \text{Drill}\\ \text{Solids}\end{array}} - \boxed{\begin{array}{c}\text{Discarded}\\ \text{Drill}\\ \text{Solids}\\ \&\\ \text{Liquid}\\ V_d \ \ \rho_d\\ S_{rds\ d}\end{array}} + \boxed{\begin{array}{c}\text{Barite}\\ \&\\ \text{Water}\end{array}} = \boxed{\begin{array}{c}\text{FINAL MUD}\\ V_f\\ \rho_f\\ S_{rds\ f}\end{array}}$$

The volume and weight of the new drill solids which are not separated and retained in the mud are

$$V_{nds} = \frac{H^2}{1029} * L * [1 - EFF_{sre}] \qquad \text{\{Expected cavings may be reasoned into H\}}$$

$$W_{nds} = SG * \rho_w * V_{rds} * 42$$

The objective is to ascertain the volume of water and weight of barite to be added, and the volume of mud to be discarded, for the purpose of controlling the solids concentration in the recirculated mud while drilling a section of hole. The following equation is based on the conservation of mass of the drill solids. If the pits are to be maintained at a constant volume, it is only necessary to set the following equation equal to zero and solve for the quantity of mud to be discarded, V_d.

$$\Delta V_{mud} = V_f - V_i$$

$$\Delta V_{mud} = \frac{S_{rds\ i} * V_i - S_{rds\ d} * V_d + V_{rds}}{S_{rds\ f}} - V_i$$

The volumes and weights of the barite and water to be added are resolved in the following.

From the sketch and the conservation of mass of the mud, the following equation is written.

$$\rho_i V_i + \rho_{rds} V_{rds} - \rho_d V_d + W_b + \rho_w V_w = \rho_f V_f$$

The conservation of volume which is adequate for barite and water, but is not satisfactory for salts which can dissolve into water gives

$$V_i + V_{rds} - V_d + \frac{W_b}{\rho_b} + V_w = V_f$$

The simultaneous solution of the last two equations gives the equations for the computation of the weight of barite and the volume of water required to drill a chosen interval of formation.

$$W_b = \frac{V_f\left(\frac{\rho_f}{\rho_w} - 1\right) + V_d\left(\frac{\rho_d}{\rho_w} - 1\right) - V_i\left(\frac{\rho_i}{\rho_w} - 1\right) - V_{rds}\left(\frac{\rho_{rds}}{\rho_w} - 1\right)}{\left(\frac{1}{\rho_w} - \frac{1}{\rho_b}\right)}$$

$$V_w = \frac{V_f\left(\frac{\rho_f}{\rho_b} - 1\right) + V_d\left(\frac{\rho_d}{\rho_b} - 1\right) - V_i\left(\frac{\rho_i}{\rho_b} - 1\right) - V_{rds}\left(\frac{\rho_{rds}}{\rho_b} - 1\right)}{\left(\frac{\rho_w}{\rho_b} - 1\right)}$$

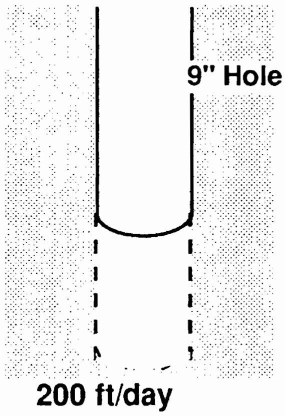
V_{rds} = Volume of new drill solids which are not separated from the mud (retained); bbl
W_{rds} = Weight of retained drill solids; lbm
H = Hole diameter; inch
L = Length of section drilled; ft
Eff_{sre} = Efficiency of solids removal equipment; fraction
SG = Specific gravity of drill solids; ppg/ppg
ρ_w = density of water; (350 ppb) (8.33 ppg) (62.3 ppcf)
ρ_b = Density of barite; (1488 ppb) (35.43 ppg) (265 ppcf)
ΔV_{mud} = Increase in mud volume during the drilling of the section; bbl
$S_{rds\ i}$ = Recirculated drill solids concentration at the beginning of drilling the section; fraction
$S_{rds\ f}$ = Recirculated drill solids concentration at the end of drilling the section; fraction
$S_{rds\ d}$ = Recirculated drill solids concentration discarded during the drilling of the section; fraction
V_i = Volume of mud at the beginning of the drilling of the section; bbl
V_f = Volume of mud at the end of the drilling of the section; bbl
V_d = Volume of mud discarded during the during of the section; bbl
ρ_f = Mud density at the end of drilling the section; ppg
ρ_i = Mud density at the beginning of drilling the section; ppg
ρ_d = Density of the discarded mud; ppg
W_b = Quantity of barite to be added to the mud; lb
V_w = Quantity of water to be added to the mud; bbl
ρ_{rds} = Density of retained drill solids; ppg

EXAMPLE

The minimum volume of a mud system is 800 bbls. The native solids (S. G. = 2.65) are to be kept at 5% volume with the addition of barite and water while drilling a 9" hole at 200 feet per day. In all cases the mud weight is 10 ppg initially and the shale shaker and other equipment remove 1/3 of the solids. Calculate the sacks of barite required to increase and maintain the system to and at higher mud weights.

W_{bar} **= 11,406 lb**

V_{water} **= 191.3 bbl**



SOLIDS CONCENTRATION SELECTION

The selection of the concentration of recirculated drill solids to be maintained in the mud can be based on minimum drilling cost per foot. The basic equation combines the cost of the rig, tools, services, and mud for the drilling of a section of hole. The optimal solids concentration equation is

cost = rig cost + mud cost

$$CPF = \frac{\$RT\&S}{ROP} + \$MUD * \frac{V_{nm}}{L}$$

CPF = cost per foot to drill a section of hole; $\frac{\$}{ft}$

R T & S = hourly cost rates of the rig, tools & services; $\frac{\$}{hr}$

ROP = rate of penetration of the bit; $\frac{ft}{hr}$

Mud = cost of mud per barrel during the drilling of the section; $\frac{\$}{bbl}$

V_{nm} = mud which must be mixed and added to the mud system during the drilling of the section of hole; bbl

L = length of a section of hole to drill; ft

ROP versus SOLIDS CONCENTRATION

A popular relationship for ROP versus S_{rds} is

$$ROP = k\, e^{-(I * S_{rds})}$$

I = exponent for drill rate impedence caused by recirculated drill solids

k = correlation coefficient

A suggestion for finding the values of k and I are to measure the ROP during normal drilling and then measure the ROP after pumping a slug of new mud without solids to the bit. The S_{rds} will be that which is being maintained during normal drilling and will be zero while drilling with the new mud.

VOLUME OF MUD TO BE MIXED during the drilling of an interval of hole
If the final and beginning mud densities, retained drill solids, and mud volumes are to be equal, and mud is to be discarded and mixed on a continuous basis, then the equation for the volume of mud to be mixed and added is developed and given in the following equations.

$$\boxed{V_i} + \boxed{\begin{array}{c}\text{New}\\\text{Drill}\\\text{solids}\\\text{retained}\end{array}} + \boxed{\begin{array}{c}\text{New}\\\text{Mud}\\\text{no}\\\text{solids}\end{array}} - \boxed{\begin{array}{c}\text{Discarded}\\\text{Mud}\\\text{with solids}\end{array}} = \boxed{V_f}$$

$V_i = V_f$ {no change in mud volume in the system}

which becomes

$$\boxed{\begin{array}{c}\text{New}\\\text{Mud}\end{array}} = \boxed{\begin{array}{c}\text{Discarded}\\\text{Mud}\end{array}} - \boxed{\begin{array}{c}\text{New}\\\text{Drill}\\\text{solids}\end{array}}$$

The algebraic equation is

$$V_{nm} = V_{dm} - V_{rds}$$

Conservation of solids volume requires

$$0 = S_{rds} * V_{dm} - V_{rds}$$

$$V_{rds} = S_{rds} * V_{dm}$$

$$V_{dm} = \frac{V_{rds}}{S_{rds}}$$

Conservation of mud excluding solids requires

$$V_{nm} = V_{dm} - S_{rds} * V_{dm} \qquad V_{nm} = V_{dm}(1 - S_{rds})$$

substituting for V_{dm} gives

$$V_{nm} = \frac{V_{rds}}{S_{rds}} (1 - S_{rds})$$

The volume of new mud to be mixed and consumed for the section is

$$V_{nm} = \frac{H^2}{1029} * L * \frac{[1 - EFF_{sre}]\ [1 - S_{rds}]}{S_{rds}}$$

Substitution of the above gives the equation for selecting optimal solids concentration. In order to ascertain the variance of cost per foot with retained solids concentration a plot is made with the two variables.

$$CPF = \frac{\$R\,T\,\&\,S}{k e^{-[I * S_{rds}]}} + \$MUD \left[\frac{H^2}{1029} * \frac{[1 - EFF_{sre}]\ [1 - S_{rds}]}{S_{rds}}\right]$$

EXAMPLE

Rig, tools, and supplies cost $500/hr. Other variables are as listed.

$S_{rds} = 0.05$
Mud = $20/bbl
$Eff_{sre} = 0.4$
Hole dia = 9.0 in

TABLE OF ROP V. SOLIDS

ROP	S_{rds}
24	.05
15	.08

The appropriate equations and values from the table of ROP versus Solids give the following values for k and I.

$$I = \frac{\mathrm{Ln}\left(\dfrac{ROP_2}{ROP_1}\right)}{S_{rds1} - S_{rds2}} \qquad k = ROP_1\, e^{I * S_1}$$

$$I = 15.67 \qquad k = 52.67$$

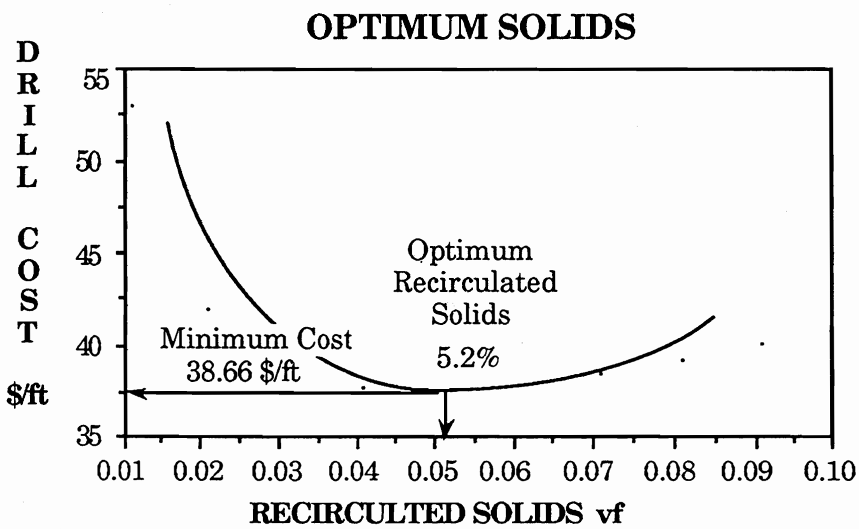
Now, the cost per foot equation can be completed.

$$CPF = \frac{500}{52.67\, e^{-[15.67 * S_{rds}]}} + 20\left[\frac{9^2}{1029.5} * \frac{[1 - .4]\,[1 - S_{rds}]}{S_{rds}}\right]$$

TABLE OF COST VS. SOLIDS

S_{rds}	CPF
.03	45.72
.05	38.72
.052	38.66
.07	40.98
.09	48.44

It is seen from the plot that 5.2 volume % of solids is the optimal recirculated drill solids concentration for this example.



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CHAPTER IV

KICK REMOVAL

The driller's and engineer's removal methods are two reliable methods of circulating a drilling kick from a hole. The other methods presented below are satisfactory in special circumstances. The depicting attributes of each method are the following:

Driller's

1. kill mud is pumped after the kick is removed from the hole
2. two circulations of the hole are required
3. annular and surface pressures will be higher while removing the kick than those of the engineer's method

Engineer's

1. the drilling mud is weighted to kill mud weight prior to pumping
2. kill mud is pumped while removing the kick
3. one circulation is required to kill the hole

Concurrent

1. drilling mud is weighted as it is pumped into the hole but not necessarily to the weight of kill mud
2. the hole will contain a variable weight mud
3. annular and surface pressures will be higher than the engineer's and less than the driller's

Gas Migration

1. the gas bubble is allowed to rise in the annulus without circulating
2. the casing pressure is allowed to rise to a selected value without bleeding mud
3. mud is bled from the annulus while keeping the pressure at the selected value
4. after the kick rises to the surface, heavy mud is lubricated into the annulus to kill the annulus and well.

Dynamic

1. kill weight mud is pumped at a rate sufficient to raise the pressure at the bottom of the hole above or equal to that of the kicking formation. The increase in bottom hole pressure occurs because mud is occupying more and more of the volume of the

annulus, and friction pressure losses in the annulus. (pump mud faster than gas entry rate)
2. a choke pressure may or may not be applied

Low choke pressure

1. a pressure is held at the choke which will prevent the fracturing of a formation in the hole
2. additional gas will enter the hole during the removal of the kick
3. similar to the dynamic method

Partition

1. the kick is pumped out of the hole in partitions
2. pressures will be the lowest of the methods
3. mud may or may not be weighted prior to the circulation of a partition

There are three primary causes of kicks, 1) drilling into a permeable formation which has a higher pressure gradient than the drill mud, 2) pulling of the drilling string without filling the hole, and 3) lowering the mud weight after drilling a section of hole. The following development discusses the type of kick which arises from drilling into higher pressured formations.

In order to successfully remove a kick with any methods other than the dynamic or the low choke methods, two basic rules must be observed. The first is that additional gas must be prohibited from entering the drill hole. The second is that the pressure of the mud within the entire open drill hole must be less than the fracture strength of the formation or the last casing. The first rule requires that the pressure at the bottom of the hole be equal to or greater than the formation pressure from which the gas came. The second requires that the choke be continually adjusted. In equation form these two rules are

$P_{m\,bh} \geq P_f$ Rule number 1

$P_{m\,x} \leq P_{ff\,x}$ Rule number 2

After the kick is detected and the blowout preventers are closed, the standpipe and casing pressures as well as the gain in the pits are recorded.

In the following discussion it is assumed that

1. the gas kick enters the drill hole at the bottom of the hole as a contiguous bubble and remains as one through its removal from the drill hole
2. the mass of the gas in the contiguous bubble does not change (absorption into the mud, adsorption onto solids, loss into formations, etc.)

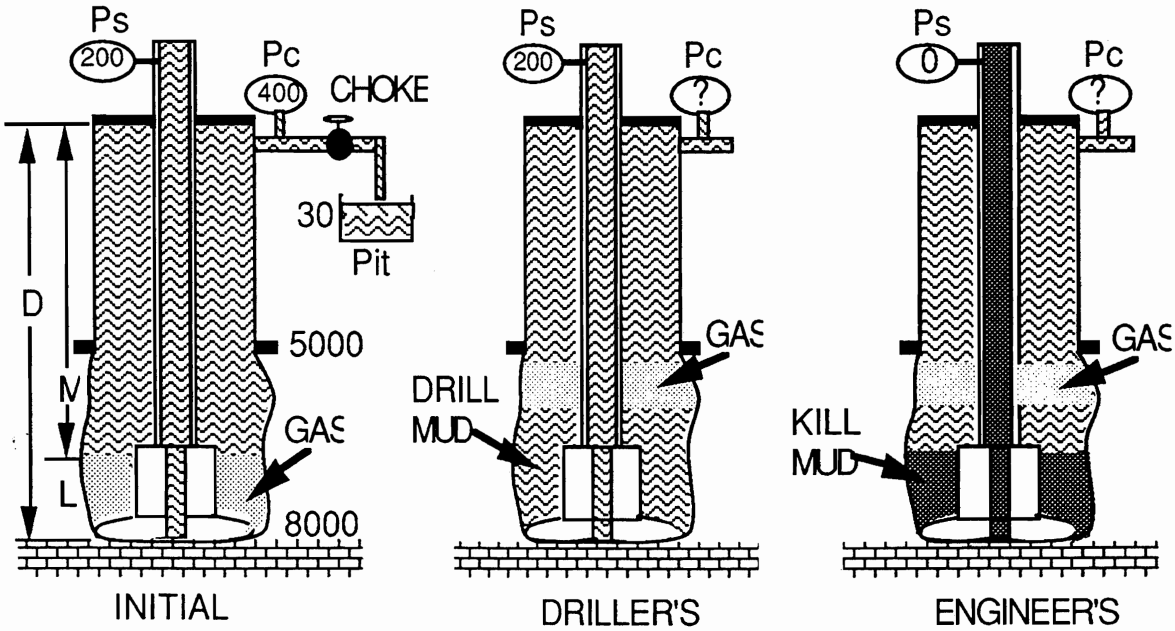
3. the intrinsic gas properties do not change (molecular weight, density, etc.)
4. the temperature of the kick does not change during its removal
5. gel and friction pressure losses in the annulus are neglected
6. the kick will not rise in the hole without circulation.

The removal of the kick with the driller's method is segmented into five mud pumping intervals

1. ascertaining initial conditions before pumping begins
2. pumping the top of the kick to the surface
3. venting the kick at the surface
4. pumping the kill mud to the drill bit
5. pumping the kill mud to surface.

EXAMPLE

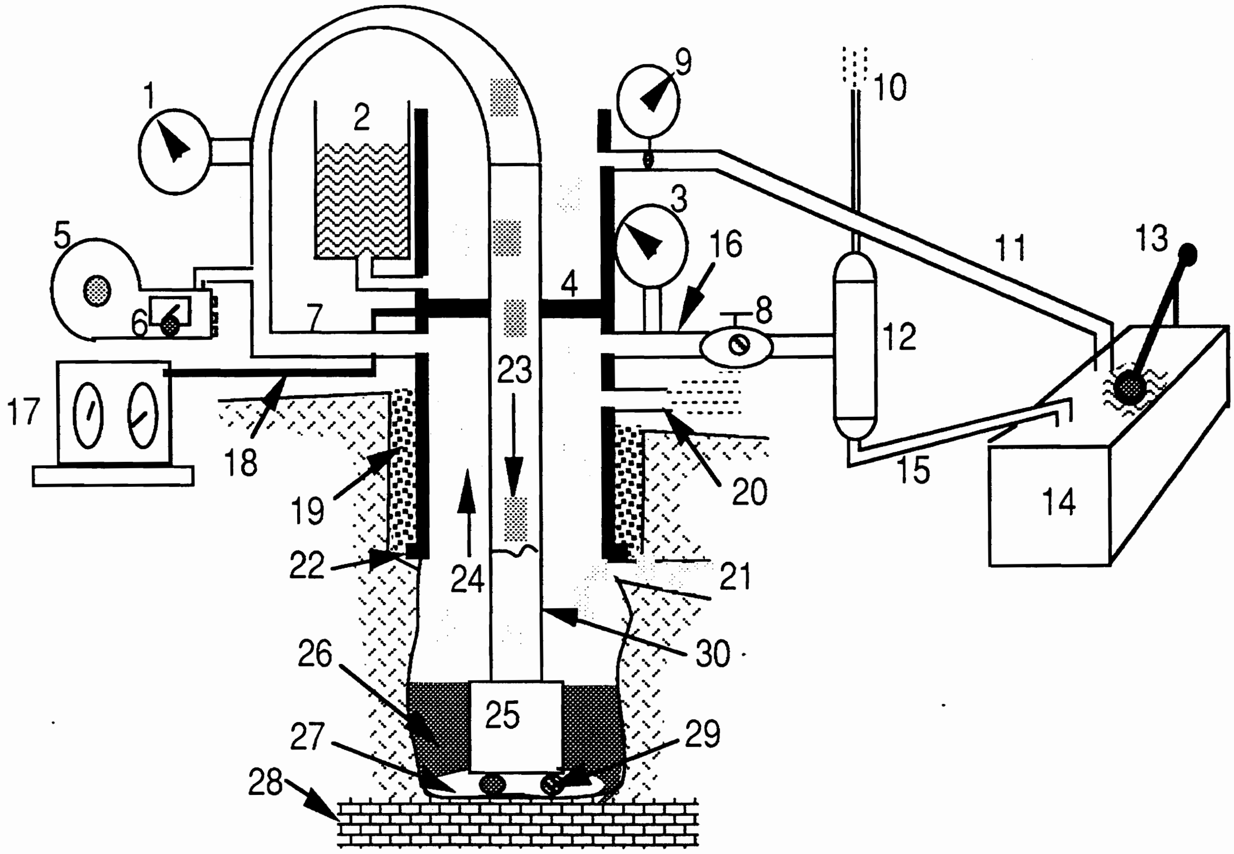
The following example is carried throughout the discussions. Suppose a 30 barrels kick has flowed into the hole and the standpipe and casing pressures are recorded to be 200 psig and 400 psig, respectively. The mud weight is 14.3 ppg. The drill string is 600 feet of 8" by 3" bha and 7,400 feet of 5" by 19.5 ppf drill pipe. The last casing was set at 5,000 feet and the formation pressure integrity test at the shoe gave a value of 16.0 ppg. The low circulating rate pressure is 1,500 psig. A sketch of the drill hole follows.



SKETCH OF HOLE WITH GAS KICK

RIG KICK REMOVAL EQUIPMENT

Larger drilling rigs have the equipment shown in the sketch.



NUMBER EQUIPMENT

1	standpipe pressure gauge	2	hole fill tank
3	casing pressure gauge	4	blowout preventer rams (or bag)
5	mud pump	6	mud pump stroke sensor
7	kill line	8	choke in choke line
9	flow line mud flow sensor	10	gas flare and flare line
11	mud flow line	12	gas mud separator
13	mud pit level sensor	14	active mud pit
15	separator mud flow line	16	choke line
17	accumulator	18	bope lines
19	cement for last casing	20	vent line
21	fracture in formation and loss of mud		
22	shoe of last casing		
23	kill mud and inside drillpipe		
24	drilling mud and drillpipe annulus		
25	drill collars	26	kick fluid and drill collar annulus
27	drill bit	28	kicking formation
29	jets in the drill bit	30	drillpipe

Notes

KILL PARAMETERS

Volumetric capacities of the annuli around the bha and drillpipe

$$C = \frac{H^2 - D^2}{1029.5} \qquad C_{ca} = \frac{12.25^2 - 8^2}{1029.5} \qquad \mathbf{C_{ca} = 0.0836\ bbl/ft}$$

$$C_{pa} = \frac{12.25^2 - 5^2}{1029.5} \qquad \mathbf{C_{pa} = 0.121\ bbl/ft}$$

Volumetric capacities within the bha and drillpipe

$$C = \frac{ID^2}{1029.5} \qquad C_c = \frac{3^2}{1029.5} \qquad \mathbf{C_c = .00874\ bbl/ft}$$

$$C_p = \frac{4.276^2}{1029.5} \qquad \mathbf{C_p = .0178\ bbl/ft}$$

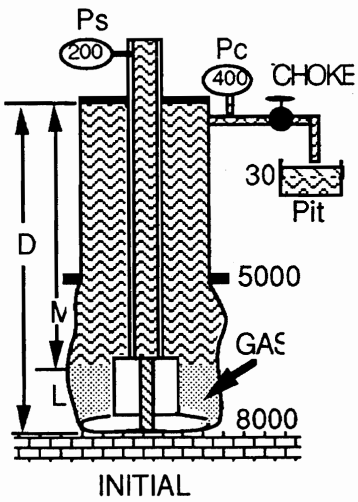
INITIAL CONDITIONS

The initial length of the gas Kick is

$$L_i = \frac{V_g}{C_{ca}} \quad \text{(if kick is not above the drill collars)}$$

$$L_i = \frac{30}{.0836}$$

L_i **= 359 ft** (initial length of the gas kick)



The formation Pressure is

$$P_f = P_{si} + .052\ \rho_m\ D$$

D	=	Drilling depth
ρ_m	=	Drilling mud weight
P_{si}	=	Initial shut-in standpipe pressure

$P_f = 200 + .052 * 14.3 * 8000$ **= 6,149 psig**

The pressure gradient of the formation is

$$G_f = \frac{P_f}{.052\ D} = \frac{6149}{.052 * 8000} \qquad \mathbf{= 14.8\ ppg}$$

The initial density of the kick fluid is found with the equation

$$P_f = P_{ci} + .052 * \rho_m * (D - L) + .052\, \rho_k\, L_i$$

P_{ci} = Initial shut-in casing pressure

ρ_{ki} = Initial kick density

$$6149 = 400 + .052 * 14.3 * 7641 + .052 * \rho_k * 359$$

ρ_{ki} **= 3.6 ppg** (Gas and a little oil)

The initial pressure drop across the length of the kick and at the top of the kick is

$\Delta P_i = .052 * \rho_{ki} * L_i$ = .052 * 3.6 * 359 **= 67.2 psig**

$P_{ti} = Pf - \Delta P_i$ = 6149 - 67.2 **= 6082 psig**

If the gas passes from one annulus to another, the pressure drop across the kick only because of the annular capacity change is

$$\Delta P_{\text{new annulus}} = \Delta P \frac{C_{ca}}{C_{pa}}$$

The gas column factor for the kick, $\varphi = \frac{mw}{ZRT}$, is computed with the gas column formula

$P_f = P_{ti}\, e^{\varphi L_i}$ $\quad 6149 = 6082\, e^{\varphi * 359}$

φ $= \frac{1}{32790}$

GAS COLUMN FORMULA

$$\frac{P_b}{P_t} = e^{\frac{mw}{ZRT} L}$$

P_b = pressure at bottom of gas column; psia

P_t = pressure at top of gas column; psia

L = length of gas column; ft

R = gas constant; R = 1544

T = average gas temperature; Rankine

Z = average gas deviation factor; no units

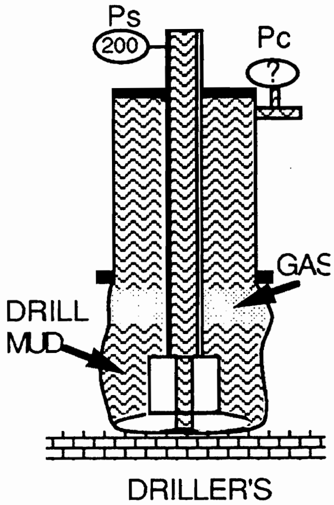
mw = molecular weight of the gas

DRILLER'S METHOD

DEFINITION OF THE DRILLER'S METHOD

The driller's method is a method of removing a kick from the hole which requires the kick to be circulated from the hole prior to circulating a kill mud into the hole. Two circulations are required and the method is sometimes called the two circulation method.

PUMPING THE KICK TO THE SURFACE with driller's method



The volume of mud to be pumped to move the top of the kick to the top of the hole is

$$V_m = L_{bha}\, C_{ca} + (D - L_{bha} - L)\, C_{pa}$$

$$V_m = 600 * .0836 + (8000 - 600 - 1311) * .121$$

$$\mathbf{V_m = 790\ bbl}$$

V_m = Total volume of mud pumped after shut-in

The static length of the gas kick while the top of the gas kick is at a depth, X, is (Note: the kick must be completely contained in the annulus of the BHA or the annulus of the drillpipe. If the kick is within the annulus of the BHA set C_p equal to C_c, if not use as is.)

$$L = \frac{1}{\varphi} \operatorname{Ln}\left\{ 1 + \frac{\Delta P_i \dfrac{C_{ca}}{C_{pa}}}{P_f - \Delta P_i \dfrac{C_{ca}}{C_{pa}} - .052\, \rho_m\, (D\text{-}X\text{-}L)} \right\}$$

The length of the kick while the kick is at the shoe of the last casing, i.e., X = 5,000 ft.

$$L = 32790 \operatorname{Ln}\left\{ 1 + \frac{67.2 \dfrac{.0836}{.121}}{6149 - 67.2 \dfrac{.0836}{.121} - .052 * 14.3(8000\text{-}5000\text{-}L)} \right\}$$

$$L \qquad\qquad \mathbf{= 363\ ft}$$

The length and the volume of the kick when the top of the kick is pumped to the surface is

$$L = 32790\ \mathrm{Ln}\left\{1 + \frac{67.2\frac{.0836}{.121}}{6149 - 67.2\frac{.0836}{.121} - .052 * 14.3(8000-0-L)}\right\}$$

L **= 1,311 ft**

$V_k = L * C_{pa}$ **= 159 bbls**

V_k = Pit gain

The static pressure acting on the drill hole at a depth of X (X = 5,000 feet) with the top of the kick at a depth of X is

$$P_X = P_f - .052 * \rho_m (D - L - X) - \Delta P\frac{C_{ca}}{C_{pa}}$$

$$P_{5000} = 6149 - .052 * 14.3 (8000 - 363 - 5000) - 67.2\frac{.0836}{.121}$$

P_{5000} **= 4,142 psi**

The casing pressure with the top of the kick at 5,000 feet is

$$P_c = P_f - .052 * \rho_m (D - L) - \Delta P\frac{C_{ca}}{C_{pa}}$$

$P_c = 6149 - .052 * 14.3 (8000 - 363) - 67.2\frac{.0836}{.121}$ **= 424 psi**

The casing pressure with the top of the kick at the surface is

$P_c = 6149 - .052 * 14.3 (8000 - 1311) - 67.2\frac{.0836}{.121}$ **= 1,129 psi**

The static standpipe pressure is

$P_s = P_{si}$ **= 200 psig**

The circulating standpipe pressure is

$$P_s = LCRP + P_{si}$$

LCRP = Low circulating rate pressure

$P_s = 1500 + 200$ **= 1700 psig**

The volume of mud to be pumped to bring the top of the kick to a depth of 5,000 is

$$V_m = L_{bha}\, C_{ca} + (D - X - L_{bha} - L)\, C_{pa}$$

L_{bha} = Length of the bottom hole assembly
L = length of kick with its top at 5,000'

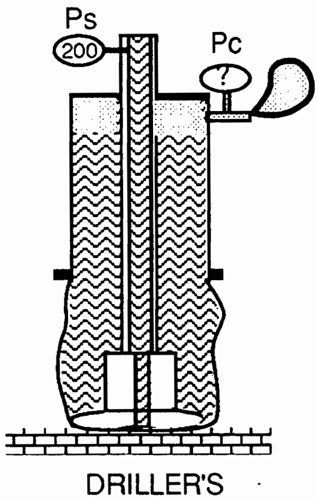
V_m = 600 * .0836 + (8000 - 5000 - 600 - 363) * .121 **= 298 bbl**

The pit volume gain after pumping 296 barrels of mud is

$V_g = L * C_{pa}$ = 363 * .121 **= 44 bbl**

VENTING THE KICK with driller's method

During the venting of the kick, the casing pressure will drop, from its highest value to the initial standpipe shut-in value, in a linear manner as the mud fills the space occupied by the kick . In our example the highest casing pressure value is 1,129 psig and the initial standpipe pressure value was 200 psig. The volume of mud to pump to remove the kick from the hole (volume of the annulus) is



$$V_m = L_{bha}\, C_{ca} + (D - L_{bha})\, C_{pa}$$

$$V_m = 600 * .0836 + (8000 - 600) * .121$$

$V_m = V_{annulus}$ **= 949 bbl**

The length of the kick is

$$L = \frac{V_{ann} - V_m}{C_{pa}}$$

L_h = length of kick with the top of the kick is at the surface (1311 ft)
V_k = volume of the kick with the top of the kick at the surface (159 bbl)

After pumping 870 barrels of mud the length of the kick is

$L = \dfrac{949 - 870}{.121}$ **= 653 ft**

The casing pressure during the venting of the kick at the surface is

$$P_c = P_{si} + \frac{V_{ann} - V_m}{V_k}(P_h - P_{si})$$

P_h = casing pressure with the top of the kick at the surface (1129 psig)

P_{si} = initial shut-in standpipe pressure (200 psig)

Let the volume of mud pumped be 870 barrels, the casing pressure is

$$P_c = 200 + \frac{949 - 870}{159}(1129 - 200) \qquad \mathbf{= 661\ psig}$$

The static standpipe pressure is

$$P_s = P_{si} \qquad \mathbf{= 200\ psig}$$

The circulating standpipe pressure is

$$P_s = LCRP + P_{si}$$

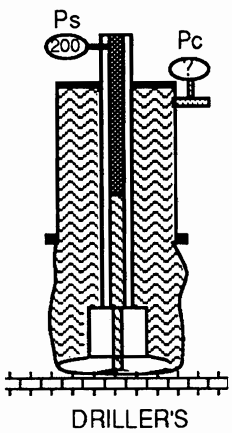
$$P_s = 1500 + 200 \qquad \mathbf{= 1700\ psig}$$

The pit volume gain after pumping 870 barrels of mud is

$$V_g = L * C_{pa} \qquad = 653 * .121 \qquad \mathbf{= 79\ bbl}$$

PUMPING THE KILL MUD TO THE BIT with driller's method

The volume of mud required to pump the kill weight mud to the bit is the volume contained within the drill string



$$V_m = L_{bha}\, c_c + (D - L_{bha})\, c_p$$

$$V_m = 600 * .00874 + (8000 - 600) * .0178 \qquad \mathbf{= 137\ bbl}$$

$$\text{total } V_m = 949 + 137 \qquad \mathbf{= 1{,}086\ bbl}$$

The static casing pressure with the kill mud at the bit is equal to the initial standpipe pressure

$$P_c = P_{si} \qquad \mathbf{= 200\ psig}$$

The circulating standpipe pressure while the kill mud is being pumped to the bit is

$$P_s = P_f + \left[\frac{(V_m - V_a)/c_p}{D}\right] \left[LCRP \;*\; \left(\frac{\rho_{km}}{\rho_m}\right)^{.75} - .052 * \rho_{km} * D\right]$$

$$+ \left[1 - \frac{(V_m - V_a)/c_p}{D}\right] \left[LCRP - .052 \;*\; \rho_m * D\right]$$

ρ_{km} = weight of kill mud (**14.8 ppg**)

After pumping a total volume of mud of 1,000 barrels which consists of 949 barrels of light mud and 51 barrels of kill mud, the circulating standpipe pressure is

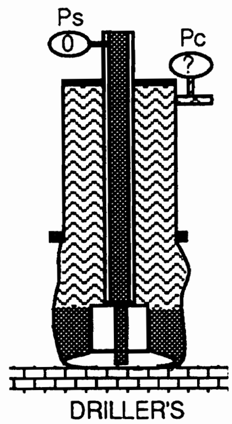
$$P_s = 6149 + \left(\frac{(1000 - 949)/.0178}{8000}\right)(1500 * \left(\frac{14.8}{14.3}\right)^{.75} - .052 * 14.8 * 8000)$$

$$+ \left(1 - \frac{(1000 - 949)/.0178}{8000}\right)(1500 - .052 * 14.3 * 8000) \qquad \mathbf{= 1{,}640\ psi}$$

The additional pit volume gain is equal only to the volumes of barite and other additives mixed into the mud to increase the density of the mud.

PUMPING THE KILL MUD TO THE SURFACE with driller's method

The additional volume of mud required to pump the kill weight mud to the surface is the volume of the annulus



$$V_m = L_{bha}\, C_{ca} + (D - L_{bha})\, C_{pa}$$

$$V_m = 600 * .0836 + (8000 - 600) * .121 \qquad \mathbf{= 949\ bbl}$$

$$V_{mt} = 790 + 159 + 137 + 949 \qquad \mathbf{= 2{,}035\ bbl}$$

V_{mt} = total mud volume to kill drill hole

The casing pressure is

$$P_c = \left[\frac{V_{mt} - V_m}{V_{mt} - V_b}\right] P_{si}$$

V_b = volume of mud pumped to put kill mud at the bit (1,086 bbl)

Let 1,600 barrels of mud be pumped, the casing pressure is

$$P_c = \frac{2035 - 1600}{2035 - 1086}(200) \qquad \mathbf{= 91\ psig}$$

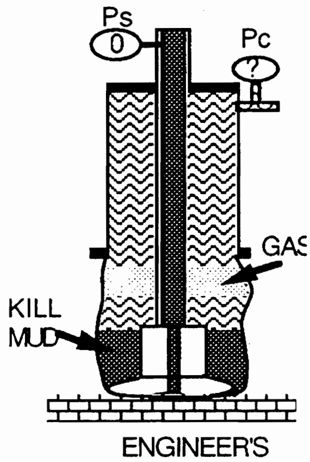
The circulating standpipe pressure is

$$P_s = LCRP * (\frac{\rho_{km}}{\rho_m})^{.75} \qquad = 1500 * (\frac{14.8}{14.3})^{.75} \qquad \mathbf{= 1{,}539\ psig}$$

ENGINEER'S METHOD

DEFINITION OF THE ENGINEER'S METHOD

The engineeer's method is a method of removing a kick from a hole which requires that the mud in the pits be weighted to a kill value prior to circulating the kick from the hole. One circulation is required. The method is often called the wait and weight method.



PUMPING THE KICK TO THE SURFACE with engineer's method

The volume of mud to be pumped to move the top of the kick to the top of the hole is

$$V_m = L_{bha}\, C_{ca} + (D - L_{bha} - L)\, C_{pa}$$

$$V_m = 600 * .0836 + (8000 - 600 - 1399) * .121 \qquad \mathbf{= 779\ bbl}$$

The pressure drop across the drill mud below the kick, and while the kill mud is in the annulus of the drillpipe is

$$\Delta P_{dm} = .052 * \rho_m * \frac{V_{ds}}{C_{pa}}$$

$$= .052 * 14.3 * \frac{137}{.121} \qquad \mathbf{= 837\ psi}$$

The static length of the gas kick while the top of the gas kick is at a of depth, X, is (Note: the kick and the kill mud must be completely contained in the annulus of the BHA or the annulus of the drillpipe.

$$L = \frac{1}{\varphi} Ln \left\{ 1 + \frac{\Delta P \frac{C_{ca}}{C_{pa}}}{Pf - \Delta P_i \frac{C_{ca}}{C_{pa}} - \Delta P_{dm} - .052\, \rho_{km}\, (D - \frac{V_{ds}}{C_{pa}} - X - L)} \right\}$$

The length of the kick while the kick is at the shoe of the last casing, i.e., X = 5,000 ft.

$$L = 32790\ \mathrm{Ln}\left\{1 + \frac{67.2\,\frac{.0836}{.121}}{6149 - 67.2\,\frac{.0836}{.121} - 837 - .052 * 14.8 * \left(8000 - \frac{137}{.121} - 5000 - L\right)}\right\}$$

$$L \qquad = \mathbf{366\ ft}$$

The length and the volume of the kick when the top of the kick is pumped to the surface is

$$L = 32790\ \mathrm{Ln}\left\{1 + \frac{67.2\,\frac{.0836}{.121}}{6149 - 67.2\,\frac{.0836}{.121} - 837 - .052 * 14.8 * \left(8000 - \frac{137}{.121} - 0 - L\right)}\right\}$$

$$L \qquad = \mathbf{1,399\ ft}$$

$$V_k = L * C_{pa} \qquad = 1,399 * .121 \qquad = \mathbf{169\ bbl}$$

The static pressure acting on the drill hole at a depth of X (X = 5,000 feet) with the top of the kick at a depth of X is

$$P_X = P_f - \Delta P_i \frac{C_{ca}}{C_{pa}} - \Delta P_{dm} - .052\,\rho_{km}\left(D - \frac{V_{ds}}{C_{pa}} - X - L\right))$$

$$P_{5000} = 6149 - 67.2\frac{.0836}{.121} - 837 - .052 * 14.8 * \left(8000 - \frac{137}{.121} - 5000 - 366\right)$$

$$P_{5000} \qquad = \mathbf{4,110\ psi}$$

The static casing pressure with the top of the kick at 5,000 feet is

$$P_c = P_f - \Delta P_i \frac{C_{ca}}{C_{pa}} - \Delta P_{dm} - .052\,\rho_{km}\left(D - \frac{V_{ds}}{C_{pa}} - X - L\right))$$

$$P_c = 6149 - 67.2\frac{.0836}{.121} - 837 - .052 * 14.8\left(8000 - \frac{137}{.121} - 0 - 366\right)$$

$$P_c \qquad = \mathbf{262\ psi}$$

The static casing pressure with the top of the kick at the surface is

$$P_c = 6149 - 67.2\frac{.0836}{.121} - 837 - .052 * 14.8\left(8000 - \frac{137}{.121} - 1399\right)$$

$$P_c \qquad = \mathbf{1,057\ psi}$$

The static standpipe pressure is

P_s **= 0 psig**

The circulating standpipe pressure after filling the drillstring with kill mud is

$$P_s = LCRP * \left(\frac{\rho_{km}}{\rho_m}\right)^{.75} = 1500 * \left(\frac{14.8}{14.3}\right)^{.75}$$ **= 1,539 psig**

The volume of mud to be pumped to bring the top of the kick to a depth of 5,000 is

$$V_m = L_{bha}\, C_{ca} + (D - X - L_{bha} - L)\, C_{pa}$$

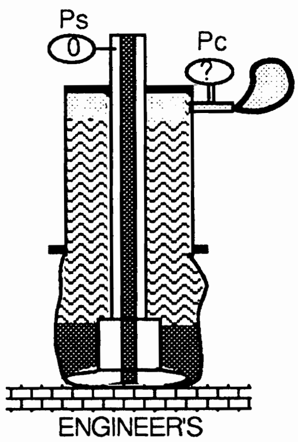
$$V_m = 600 * .0836 + (8000 - 5000 - 600 - 366) * .121$$ **= 297 bbl**

The pit volume gain after pumping 296 barrels of mud is

$$V_g = L * C_{pa} = 366 * .121$$ **= 44 bbl**

VENTING THE KICK with engineer's method

During the venting of the kick, the casing pressure will drop, from its highest value to a pressure equal to the difference in the pressure across the length of the kill mud and the pressure across a similar length of kill mud, in a linear manner as the mud fills the space occupied by the kick . In our example the highest casing pressure value is 1,057 psig and the pressure difference is 29 psig. The volume of mud to pump to remove the kick from the hole (volume of the annulus) is



$$V_m = L_{bha}\, C_{ca} + (D - L_{bha})\, C_{pa}$$

$$V_m = 600 * .0836 + (8000 - 600) * .121$$

$$V_m = V_{annulus}$$ **= 949 bbl**

The length of the kick while the kick is being vented is

$$L = \frac{V_{ann} - V_m}{C_{pa}}$$

L_h = length of kick with the top of the kick at the surface (1311 ft)

V_k = volume of the kick with the top of the kick at the surface (159 bbl)

After pumping 870 barrels of mud the length of the kick is

$$L = \frac{949 - 870}{.121} \qquad \mathbf{= 653\ ft}$$

The casing pressure during the venting of the kick at the surface is

$$P_c = P_d + \frac{V_{ann} - V_m}{V_k} (P_h - P_d)$$

P_h = casing pressure with the top of the kick at the surface (1057 psig)
P_s = pressure difference (29 psig)
P_d = casing pressure with the top of the kill mud at the surface (29 psig)

Let the volume of mud pumped be 870 barrels, the casing pressure is

$$P_c = 29 + \frac{949 - 870}{169} (1057 - 29) \qquad \mathbf{= 507\ psig}$$

The static standpipe pressure is

$$P_s \qquad \mathbf{= 0}$$

The circulating standpipe pressure is

$$P_s = LCRP * \left(\frac{\rho_{km}}{\rho_m}\right)^{.75}$$

$$P_s = 1500 * \left(\frac{14.8}{14.3}\right)^{.75} \qquad \mathbf{= 1{,}539\ psig}$$

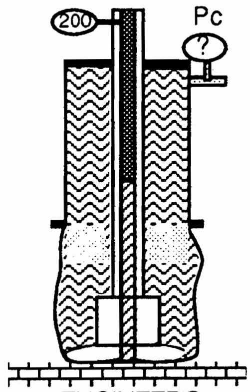
The pit volume gain after pumping 870 barrels of mud is

$$V_g = L * C_{pa}$$

$$V_g = 653 * .121 \qquad \mathbf{= 79\ bbl}$$

PUMPING THE KILL MUD TO THE BIT with engineer's method

The volume of mud required to pump the kill weight mud to the bit is the volume contained within the drill string



ENGINEER'S

$$V_m = L_{bha}\, c_c + (D - L_{bha})\, c_p$$

$$V_m = 600 * .00874 + (8000 - 600) * .0178 \qquad \mathbf{= 137\ bbl}$$

The circulating standpipe pressure is

$$P_s = P_f + \left[\frac{V_m/c_p}{D}\right]\left[LCRP * \left(\frac{\rho_{km}}{\rho_m}\right)^{.75} - .052 * \rho_{km} * D\right]$$
$$+\left[1 - \frac{V_m/c_p}{D}\right]\left[LCRP - .052 * \rho_m * D\right]$$

ρ_{km} = weight of kill mud (14.8 ppg)

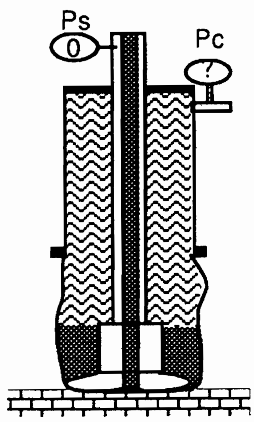
After pumping a total volume of mud of 100 barrels of kill mud, the standpipe pressure is

$$P_s = 6149 + \left(\frac{100/.0178}{8000}\right)\left(1500 * \left(\frac{14.8}{14.3}\right)^{.75} - .052 * 14.8 * 8000\right)$$
$$+ \left(1 - \frac{100/.0178}{8000}\right)(1500 - .052 * 14.3 * 8000)$$

$$P_s \qquad \mathbf{= 1{,}582\ psi}$$

PUMPING THE KILL MUD TO THE SURFACE with engineer's method

The volume of mud required to pump the kill mud to the surface after venting the gas is the volume of the drillstring



ENGINEER'S

$$V_m = V_{ds} = \mathbf{137\ bbl}$$

The static casing pressure during the venting of the kick at the surface is

$$P_c = \frac{V_m}{V_{ds}} P_d$$

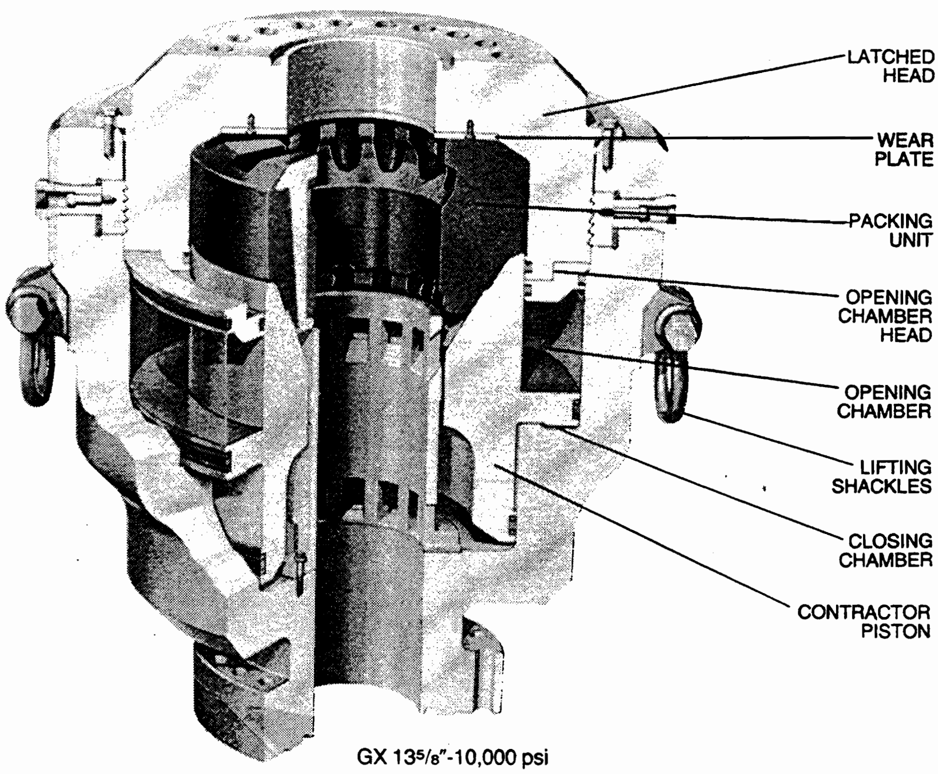
Let the volume of mud pumped after venting the gas be 100 barrels, the casing pressure is

$$P_c = \frac{100}{137} 29 \qquad \mathbf{= 21\ psig}$$

The circulating standpipe pressure is

$$P_s = LCRP * \left(\frac{\rho_{km}}{\rho_m}\right)^{.75}$$

$$P_s = 1500 * \left(\frac{14.8}{14.3}\right)^{.75} \qquad \mathbf{= 1{,}539\ psig}$$



GX 13⅝"-10,000 psi

KICK CONTROL WORK SHEET

DATE COMPLETED:________________ TIME:______________

Low Circulating Rates and Pressures

Pump #1 Circ. rate#1_______ Press#1_______ Circ. rate#2_______ Press#2_______

Pump #2 Circ. rate#1_______ Press#1_______ Circ. rate#2_______ Press#2_______

PUMP DATA

	TYPE NAME	LINER inch	STROKE inch	MAX PRESS psi	DISPLACEMENT bbl/stk
Pump No. 1	_________	________	_______	__________	_____________
Pump No. 2	_________	________	_______	__________	_____________

DRILLSTRING AND HOLE VOLUME DATA

BHA $\dfrac{ID\_\_^2}{1029.5}$ = ____CAP * LENGTH_______ = ________VOLUME

HOLE x BHA ANN $\dfrac{ID\_\_^2 - OD\_\_^2}{1029.5}$ = ____CAP * LENGTH_______ = ________VOLUME

DP#1 $\dfrac{ID\_\_^2}{1029.5}$ = ____CAP * LENGTH_______ = ________VOLUME

HOLE x DP1 ANN $\dfrac{ID\_\_^2 - OD\_\_^2}{1029.5}$ = ____CAP * LENGTH_______ = ________VOLUME

DP#2 $\dfrac{ID\_\_^2}{1029.5}$ = ____CAP * LENGTH_______ = ________VOLUME

HOLE x DP2 ANN $\dfrac{ID\_\_^2 - OD\_\_^2}{1029.5}$ = ____CAP * LENGTH_______ = ______VOLUME

TOTAL DEPTH_______ VOL_______

PIT VOLUMES

PIT #1 SIZE_____L'x_____W'x_____H' $\dfrac{L * W}{67.8}$ =_____CAP (bbl/in) $\dfrac{L*W*H}{5.65}$
=______VOL (bbl)

PIT #2 SIZE_____L'x_____W'x_____H' $\dfrac{L * W}{67.8}$ =_____CAP (bbl/in) $\dfrac{L*W*H}{5.65}$
=______VOL (bbl)

PIT #3 SIZE_____L'x_____W'x_____H' $\dfrac{L * W}{67.8}$ =_____CAP (bbl/in) $\dfrac{L*W*H}{5.65}$
=______VOL (bbl)

CASING, BOPE, AND SURFACE EQUIPMENT

CASING SIZE______" WEIGHT_______ GRADE____ SHOE DEPTH_________ BURST PRESS.________

WORKING PRESSURE OF BOPE AND SURFACE EQUIPMENT_____________PSI

(LEAKOFF_______ - DRILL MUD WT________) * SHOE DEPTH / 19.25 = FRAC CASING SIP_______PSI*

*IF THE CASING PRESSURE EXCEEDS THE FRACTURE CASING SIP BEFORE THE KICK FLUID ENTERS THE CASING, AN UNDERGROUND BLOWOUT COULD OCCUR.

WELL KILLING VOLUMES

CIRC TO BIT $= \dfrac{\text{drill string bbl}}{\text{pump vol/stk}}$ = ________ = ___________ DS stk

CIRC ANN (KICK OUT) = $\dfrac{\text{annulus bbl}}{\text{pump vol/stk}}$) = ________ = ___________ ANN stk

CIRC to KILL HOLE = 2 *ANN stk +DS stk = ________ = ___________ KILL stk

KICK DATA

SHUTIN DRILLPIPE PRESS_______SIDP SHUTIN CASING PRESS_________SICP

PIT GAIN_____________PV

COMPUTATIONS

INIT. CIRC. PRESS. = LOW CIRC. RATE PRESS + SIDP (DRILLER'S & W&W)

KILL MUD WT. $= \dfrac{\text{SIDP} * 19.25}{\text{DEPTH}}$ + DRILL MUD WT. (DRILLER'S & W&W)

FINAL CIRC. PRESS $= \dfrac{\text{KILL MUD WT.}}{\text{DRILL MUD WT.}}$ * LOW CIRC. RATE PRESS (W&W ONLY)

GRAPHICAL PUMPING SCHEDULE

WAIT AND WEIGHT

Plot the mud pump strokes on the abscissa. Plot the initial circulating rate pressure at zero strokes and the final circulating rate pressure at the strokes to circulate the mud to the bit.

Draw a straight line between the two points and then a horizontal line to the final pump strokes.

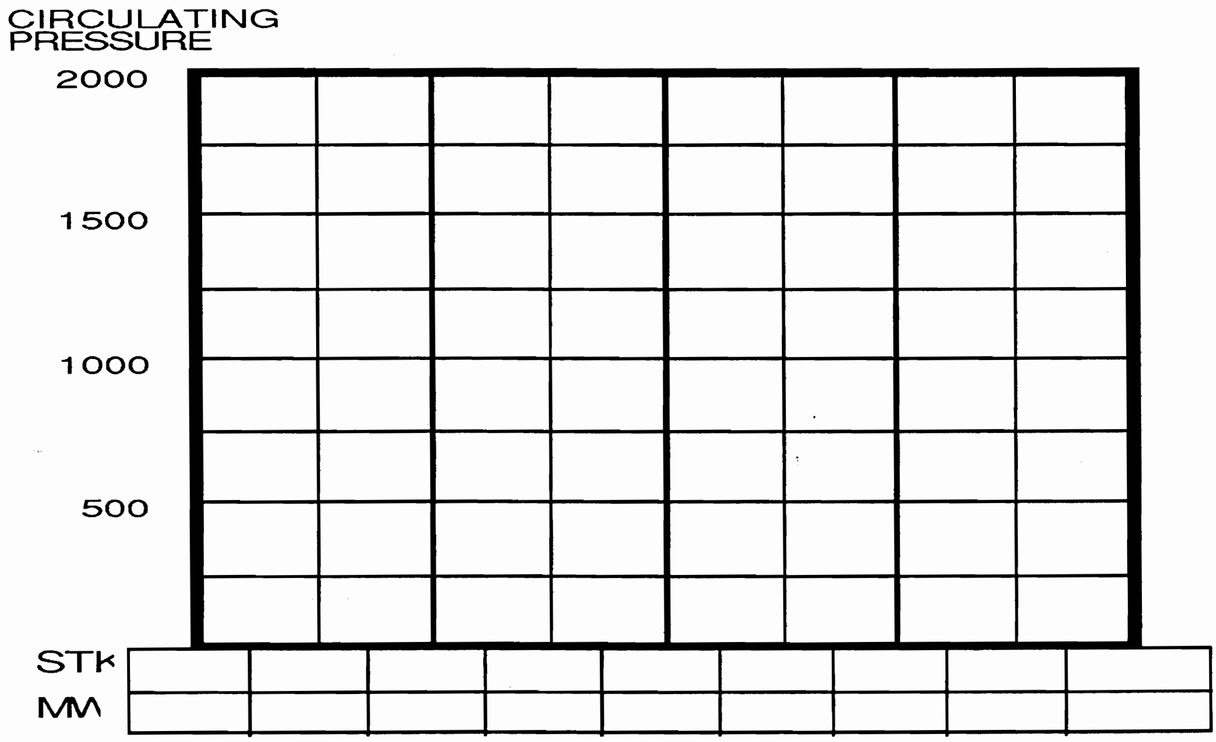
Maintain the circulating pressures as shown by the line by adjusting the choke.

Plot the initial circulating rate pressure, the drill mud weight, and zero strokes on the abscissa at the left side of the chart.

Plot on the abscissa even increments of strokes and mud weight increase (their correspondence depends on the barite mixing rate).

Plot the final circulating rate pressure on the abscissa at the number of strokes where the kill mud weight completely fills the drillstring. Draw a straight line between the two points. Maintain the circulation pressures as shown by the line.

PUMPING SCHEDULE CHART



EXAMPLE

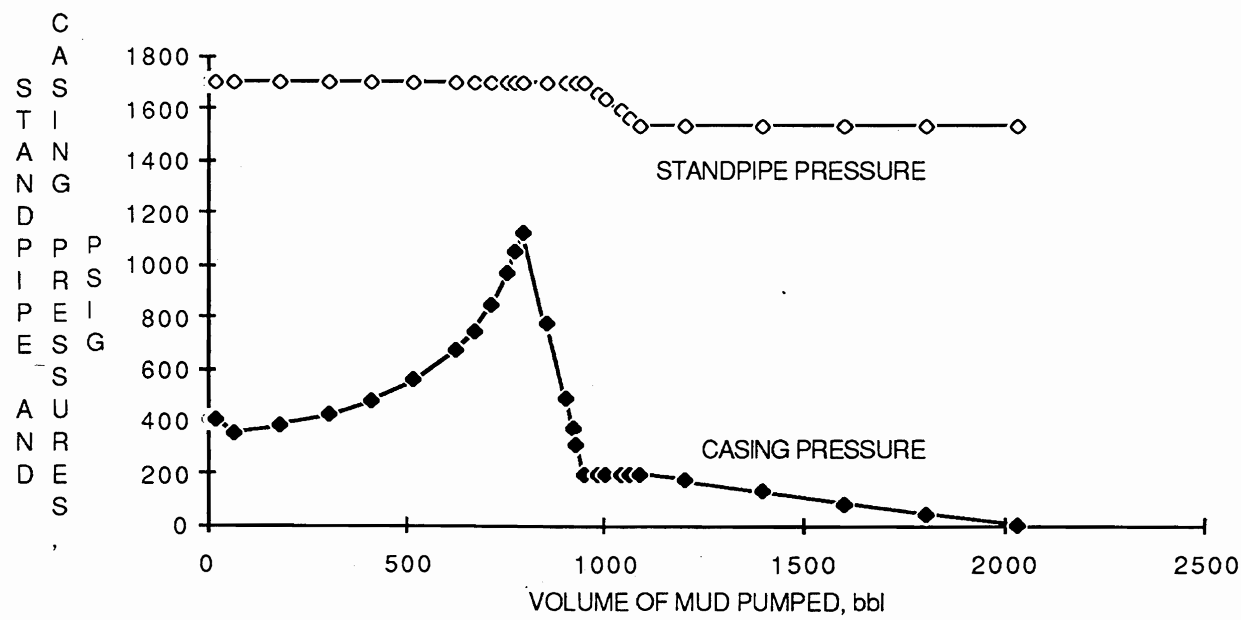
KICK REMOVAL WITH DRILLER'S AND ENGINEER'S METHODS
KICK REMOVAL PARAMETERS

HOLE DATA			KICK DATA		
Drillpipe OD	INCH	5	Pit gain	BBL	30
Drillpipe ID	INCH	4.276	Standpipe Press init.	PSIG	200
BHA OD	INCH	8	Casing Pressure init.	PSIG	400
BHA ID	INCH	3	Formation Pressure	PSIG	6149
BHA length	FEET	600	Equivalent Formation Press	PPG	14.8
Drillpipe length	FEET	7400	Initial kick length	FEET	359
Bit diameter	INCH	12.250	Initial density of kick	PPG	3.6
Drilling Depth	FEET	8000	Press head of kick in bha ann	PSI	67
Drilling mud wt.	PPG	14.3	Press head of kick in pipe ann	PSI	46
Capacity ann BHA	BBL/FT	0.0836	Gas factor - 1/phi		32827
Capacity ann dp	BBL/FT	0.12148	Mud pumped to remove kick	BBL	949
Capacity in BHA	BBL/FT	0.00874	Pres drop ac kill mud-Engr	PSI	837
Capacity in dp	BBL/FT	0.01776	Length of Kill Mud-Engr	FEET	1125
Volume of ds	BBL	137	Cas Pres w/kill mud at surf-Engr	PSIG	28
Volume of Annulus	BBL	949	Length of gas at surface-Engr	FEET	1399
Low Circ rate pres	PSI	1500	Volume of gas at surface-Engr	BBL	170

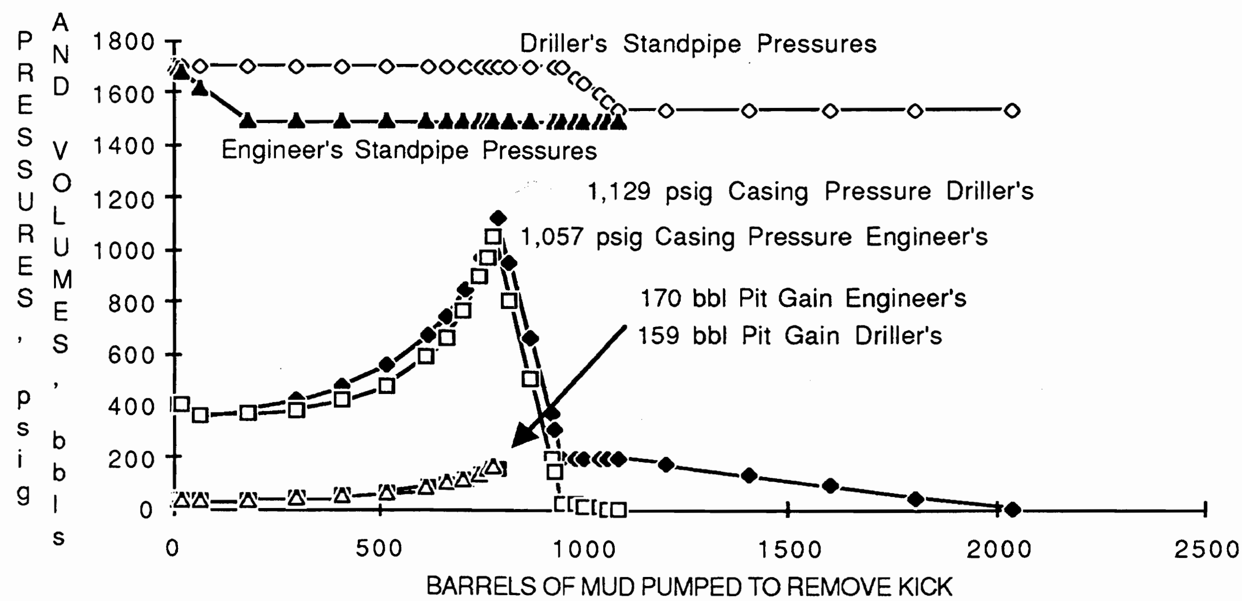
TABLE OF COMPUTED KICK PUMP OUT PARAMETERS

DRILLER								ENGINEER						
TOP OF Kick Depth feet	MUD PUMPED bbl	Trial Length feet	Kick Length feet	CASING Press psig	STAND pipe Press psig	TOP OF Kick Press psig	PIT GAIN bbl	MUD PUMPED bbl	Trial Length feet	Kick Length feet	CASING Press psig	STAND Pipe Press psig	TOP OF Kick Press psig	PIT GAIN bbl
7641	0	359	359	400	1700	6092	30	0	359	359	400	1700	6092	30
7600	3	361	361	401	1700	6059	30	3	361	361	401	1696	6059	30
7500	11	365	365	404	1700	5991	31	11	365	365	404	1687	5991	31
7400	19	369	369	409	1700	5910	31	19	369	369	409	1677	5910	31
19	VOLUME TO FILL BHA ANNULUS													
7000	66	270	270	355	1700	5550	39	65	270	270	365	1622	5570	39
137	VOLUME TO FILL DRILLSTRING											1491		
6000	189	310	310	365	1700	4046	39	182	310	311	371	1491	4633	39
5000	299	369	369	424	1700	4142	44	297	366	366	399	1491	4105	44
4000	410	435	435	477	1700	3462	53	410	442	442	422	1491	3395	54
3000	519	539	539	554	1700	2765	65	519	552	552	492	1491	2712	67
2000	622	699	699	669	1700	2155	84	619	720	721	596	1491	2079	87
1500	670	800	800	749	1700	1854	97	665	839	839	664	1491	1779	102
1000	714	935	935	849	1700	1599	114	709	995	995	765	1491	1509	120
500	754	1109	1109	974	1700	1346	134	746	1171	1171	895	1491	1267	142
250	779	1202	1202	1049	1700	1234	146	769	1279	1280	972	1491	1159	155
0	790	1311	1311	1129	1700	1129	159	779	1399	1399	1057	1491	1057	170
VOLUME TO PUMP TOP OF GAS TO SURFACE														
790	venting gas				1129	1700			779	venting gas			1059	1491
820	venting gas				959	1700			820	venting gas			810	1491
870	venting gas				651	1700			870	venting gas			507	1491
920	venting gas				370	1700			920	venting gas			204	1491
930	venting gas				311	1700			930	venting gas			144	1491
949	venting gas				201	1700			949	venting gas			29	1491
VOLUME TO PUMP GAS OUT OF ANNULUS														
949	kill mud to bit				200	1700			949	kill mud to surf			29	1491
950	kill mud to bit				200	1699			950	kill mud to surf			29	1491
980	kill mud to bit				200	1669			980	kill mud to surf			22	1491
1000	kill mud to bit				200	1640			1000	kill mud to surf			19	1491
1040	kill mud to bit				200	1592			1040	kill mud to surf			9	1491
1060	kill mud to bit				200	1569			1060	kill mud to surf			5	1491
1095	kill mud to bit				200	1539			1095	kill mud to surf			0	1491
VOLUME TO KILL WELL WITH ENGINEER'S METHOD														
1095	kill mud to surface					200	1539							
1200	kill mud to surface					175	1539							
1400	kill mud to surface					134	1539							
1600	kill mud to surface					92	1539							
1800	kill mud to surface					50	1539							
2035	kill mud to surface					0	1539							
VOLUME TO KILL WELL WITH DRILLER'S METHOD														

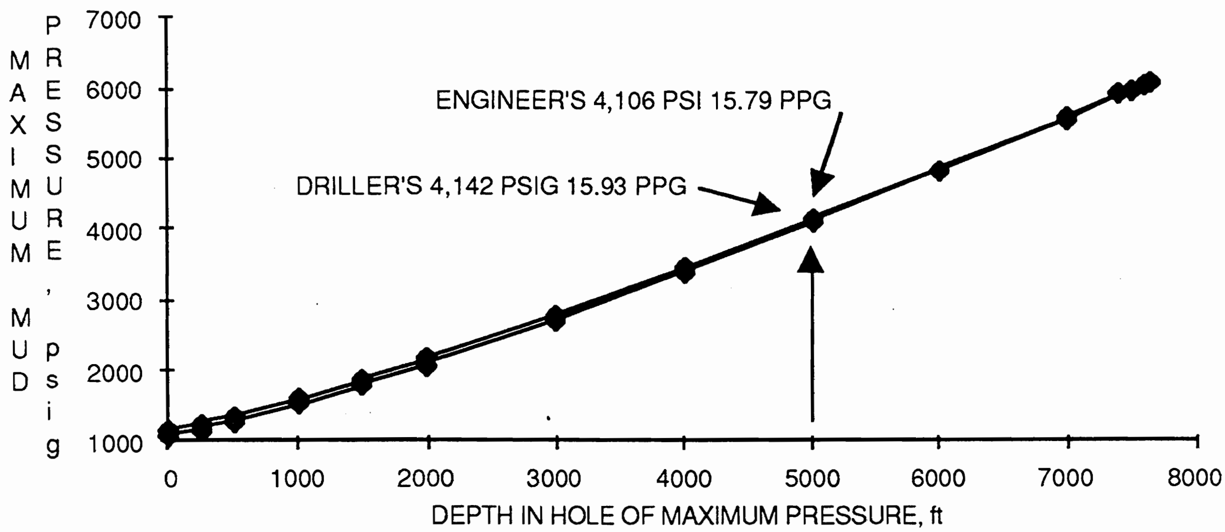
KICK REMOVAL WITH THE DRILLER'S METHOD



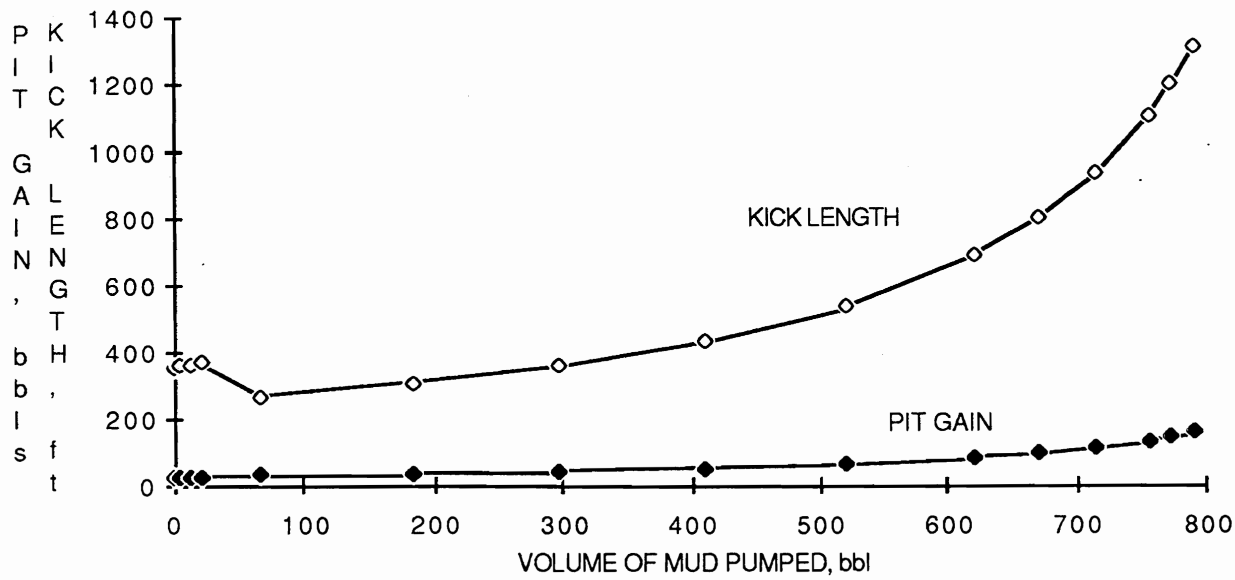
COMPARISON OF ENGINEER'S AND DRILLER'S METHODS



MAXIMUM PRESSURES - DRILLER'S V. ENGINEER'S METHOD



KICK REMOVAL WITH THE DRILLER'S METHOD



GAS MIGRATION

DEFINITION OF THE GAS MIGRATION METHOD

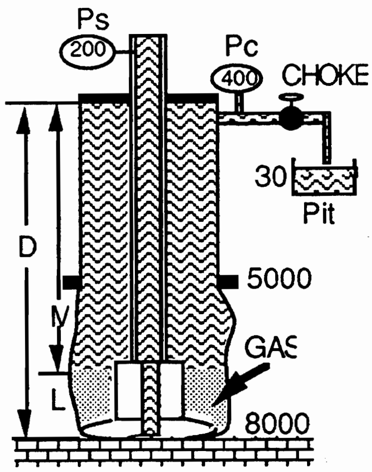
Gary Nance's gas migration method of kick removal as modified by B. J. Mitchell is a method of removing a kick from the hole which requires that the kick be allowed to rise within the hole without circulation. This method may be required if rig power is lost.

The objective is to maintain a bottom hole pressure in the mud which is never more than a selected value above the formation pressure. In the following example this value is selected to be 200 psi. The 200 psi is composed of a safety value of 100 psi and a value of 100 psi which is the upper value of a 100 psi range in which the bottom hole pressure will be purposefully varied.

The events after the kick is shut-in are (1) the pressure on the casing gauge is allowed to build by 200 psi, (2) while maintaining the casing pressure at the new pressure, mud is bled from the annulus until a mud column equivalent to 100 psi (the selected pressure range) is bled, (3) the pressure on the casing is allowed to build by another 100 psi, and (4) and the process is repeated.

The explanation of the method will be aided with the example which has been carried throughout the kick chapter.

INITIAL CONDITIONS



INITIAL

Initial conditions are the same as in Driller's and Engineer's method and are partially shown in the sketch. Also, recall the following.

Drillpipe OD	inch	5
Drillpipe Length	feet	7,400
BHA OD	inch	8
BHA Length	feet	600
Capacity DP ann	bbl/ft	0.121
Capacity BHA ann	bbl/ft	0.0836
Formation Pressure	psi	6,149
Press head of kick in BHA ann	psi	67
Press head of kick in DP ann	psi	46
Initial Kick length	feet	359
Initial kick density	ppg	3.6

SELECTION OF SAFETY PRESSURE AND THE OPERATION PRESSURE RANGE

A safety pressure is arbitrarily chosen as 100 psi to assure that additional gas will not enter the drill hole during the implementation of the method.

An operating pressure range over which the bottom hole mud pressure will be varied during the removal of the kick is arbitrarily selected as 100 psi. Thus the overbalance on the kicking formation will never be less than 100 psi or more than 200 psi. In this example the maximum BHP is 6,349 psi.

The following table contains the values expected while allowing the gas to migrate to the surface. Each row in the table corresponds with an event in the drill hole and numbers on the two sketches.

The equations following the table gives the values in the table.

ROW	BHP psi	ΔBHP psi	MUD† LENGTH	CP psi	BLEED bbl	NOTES
===	====	=====	========	===	====	====================
1	6149	0	0	400	0	initial shut-in data
2	6328	179	241	585	0	top of kick at top of BHA
3	6349	200	600	550	1.25	bottom of kick at top BHA
4	6249	100	1226	1271	118	kick rises and bleed

† NOTES:

BHP = mud pressure at the bottom of the hole; psi

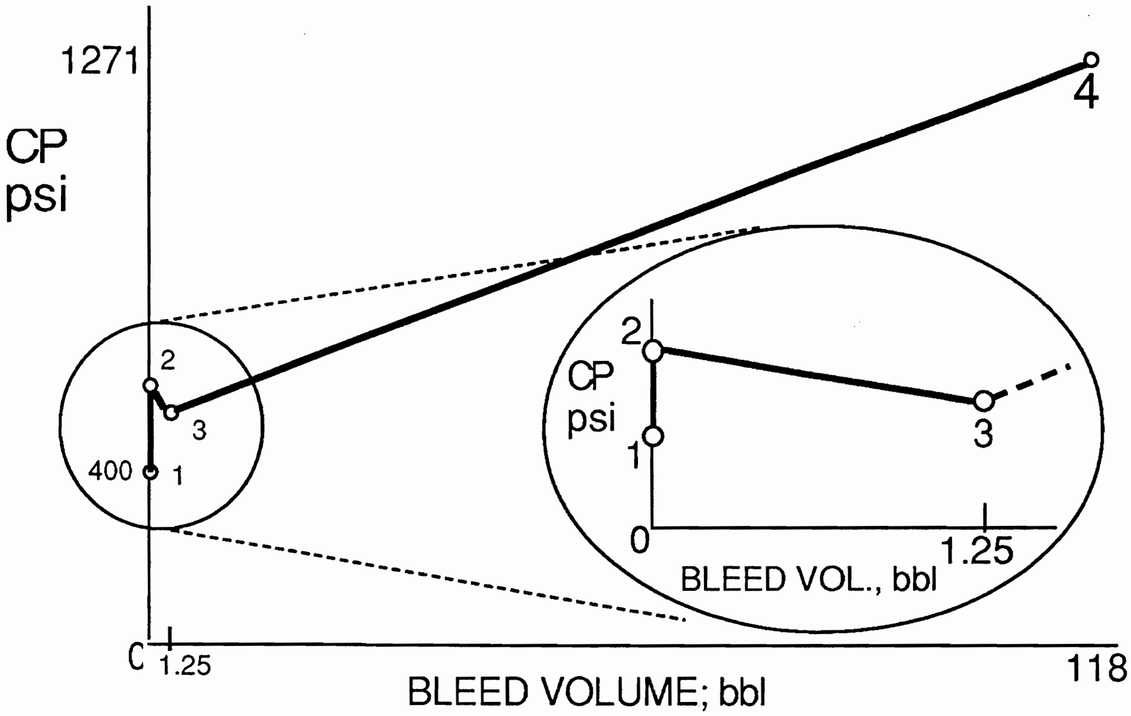
ΔBHP = difference between the BHP and the formation pressure; psi

MUD

LENGTH = length of mud below the kick; ft

CP = casing pressure (annulus); psi

Bleed = volume of mud required to be removed from the annulus; bbl



Discussion of the rows in the table.

ROW 1 AND POINT 1

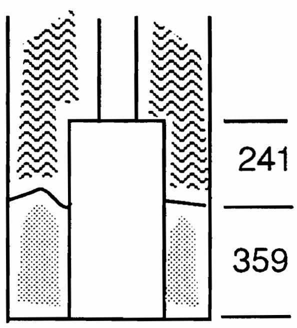
Row one is the initial shut-in conditions for this example.

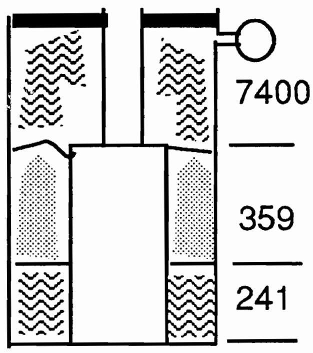
ROW 2 AND POINT 2

The bottom hole pressure (BHP) is the sum of the formation and operating pressures.

$$6328 = 6149 + 179$$

When the top of the kick rises to the top of the BHA without bleeding mud from the annulus, the kick volume, length, and pressure will remain constant. The gas will have moved up the hole (600 - 359) a distance of 241 feet. The BHP will be the sum of the 241 feet of head of mud and the pressure at the bottom of the gas which will be equal to the formation pressure (6,149 psi).





$$BHP = \frac{14.3 * 241}{19.25} + 6149 \qquad = 179 + 6149 \qquad \mathbf{= 6{,}328\ psi}$$

and the differential bottom hole pressure is

$$\Delta BHP = 6328 - 6149 \qquad \mathbf{= 179\ psi}$$

The casing pressure is the sum of the heads of fluids subtracted from the BHP.

$$CP = 6328 - \left(\frac{14.3 * 241}{19.25} + \frac{3.6 * 359}{19.25} + \frac{14.3 * 7400}{19.25} \right)$$

$$CP = 6328 - 179 - 67 - 5497 \qquad \mathbf{= 585\ psi}$$

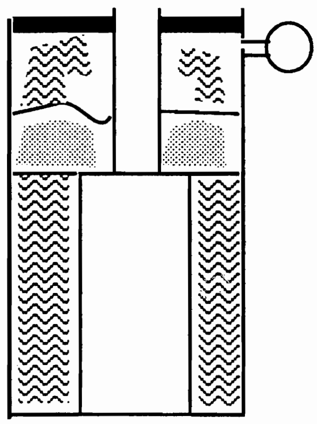
ROW 3 AND POINT 3

It may be that a casing bleed is required while the kick is rising within the annulus of the BHA. A casing bleed is required if the BHP during the rise of the gas would exceed the value of 6,328 psi. The following explains the event.

If the bottom of the kick rises to the top of the BHA without bleeding mud from the annulus, the kick volume and pressure will remain constant. The length of the kick will shorten by the ratio of the capacities of the annuli of the BHA and the drillpipe. The bottom of the gas will have moved up the hole a distance of 600 feet. The BHP will be the sum of the 600 feet of head of mud and the pressure at the bottom of the gas which will be the formation pressure (6,149 psi).

$$BHP = \frac{14.3 * 600}{19.25} + 6149 \qquad = 446 + 6149 \qquad \mathbf{= 6{,}595\ psi}$$

Because a BHP of 6,595 psi gives a ΔBHP of 246 psi (6595 - 6349) which is too large. Thus a bleed is required.
If the BHP is to be 6,349 psi and ΔBHP is to be 200 psi when the bottom of the kick rises to the top of the BHA, the pressure at the bottom of the kick (P_{bk}) will be



$$P_{bk} = 6349 - \frac{14.3 * 600}{19.25} \qquad \mathbf{= 5{,}903\ psi}$$

The volume and length of the kick will become

$$V_k = 30\ \frac{6149}{5903} \qquad \mathbf{= 31.25\ bbl}$$

$$L_k = \frac{31.25}{.121} \qquad \mathbf{= 258\ ft}$$

The required bleed volume is

$$Bleed = 31.25 - 30.00 \qquad \mathbf{= 1.25\ bbl}$$

The casing pressure will become

$$CP = 6349 - \frac{14.3 * 600}{19.25} - 46 - \frac{14.3\,(8000 - 600 - 256)}{19.25}$$

$$CP = 6349 - 446 - 46 - 5307 \qquad \mathbf{= 550\ psi}$$

ROW 4 AND POINT 4

Point 4 represents the drill hole with the top of the kick at the top of the surface. When the kick rises to the surface and the BHP is 6,349, the casing pressure and the length of the kick is

$$CP = 6349 - \frac{14.3\,(8000 - X)}{19.25} - 46$$

$$CP = 360 + 0.7429 * X$$

$$X = 264 * \frac{5903}{CP}$$

Substitution of X into the CP equation gives

$$CP = 360 + 0.7429 * \left(264 * \frac{5903}{CP}\right)$$

$$\text{then } CP^2 - 360\,CP - 1{,}157{,}663 = 0 \qquad CP \qquad \mathbf{= 1{,}271\ psi}$$

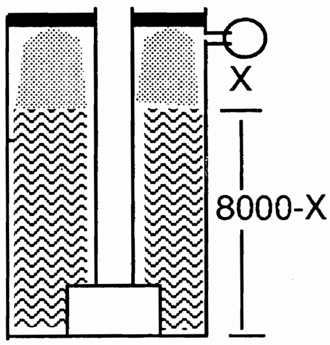
$$\text{and } X = 264 * \frac{5903}{1271} \qquad \mathbf{= 1{,}226\ ft}$$

and the volume of the kick is

$$V_k = 1226 * .122 \qquad \mathbf{= 148\ bbl}$$

and the total bleed volume will be

$$Bleed = 150 - 31.25 \qquad \mathbf{= 118\ bbl}$$



PUMPING SCHEDULE

The points 1, 2, 3, and 4 are connected with straight lines on the CP versus Bleed Volume chart. The lines are the schedules for the bleeds of the casing and the casing pressures to be held at any bleed volume.

LUBRICATING MUD INTO THE DRILL HOLE TO REPLACE THE GAS

When the gas rises to the surface it must be replaced by mud. The ideal condition would be to have the annulus of the drill hole free of gas, no pressure on the casing, and a ΔBHP equal to the selected safety pressure. The volume of the mud to be lubricated is the volume of the gas when it rises to the surface and this value is 150 bbl. The required mud weight to be lubricated is

$$MW = \left[6249 - \frac{14.3\,(8000 - 1226)}{19.25}\right] * \left[\frac{19.25}{1226}\right] \qquad \mathbf{= 19.1\ ppg}$$

BLOWOUT CONTROL

If a hole is shut-in because of a gas kick entering at the bottom of the hole while drilling, the gas will continue to enter the hole until the pressure at the bottom of the hole is equal to the pressure of the kicking formation.

The gas which entered the hole during the early period must be repressured to the formation pressure. This is the primary reason that the pressure gauges rise slowly to equilibrium pressures.

If a formation fractures, the pressure in the annulus is relieved, mud and possibly gas are lost from the hole, gas within the hole expands, and the U-tube effect between the drillpipe and the annulus is broken. Thus, it is expected that the standpipe gauge will initially rise and then fall to zero from which it may or may not rise once again.

The casing gauge is expected to initially rise, then fall, and thereafter fluctuate. The fluctuations could arise from pumping mud into the drillpipe or annulus, bridging of the annulus, temporarily repressuring the hole after plugging the fracture within the fractured formation, plugging of the drillpipe, flow of mud back into the hole from the fracture, gravity segregation of the mud and gas, and adjustments of the surface choke.

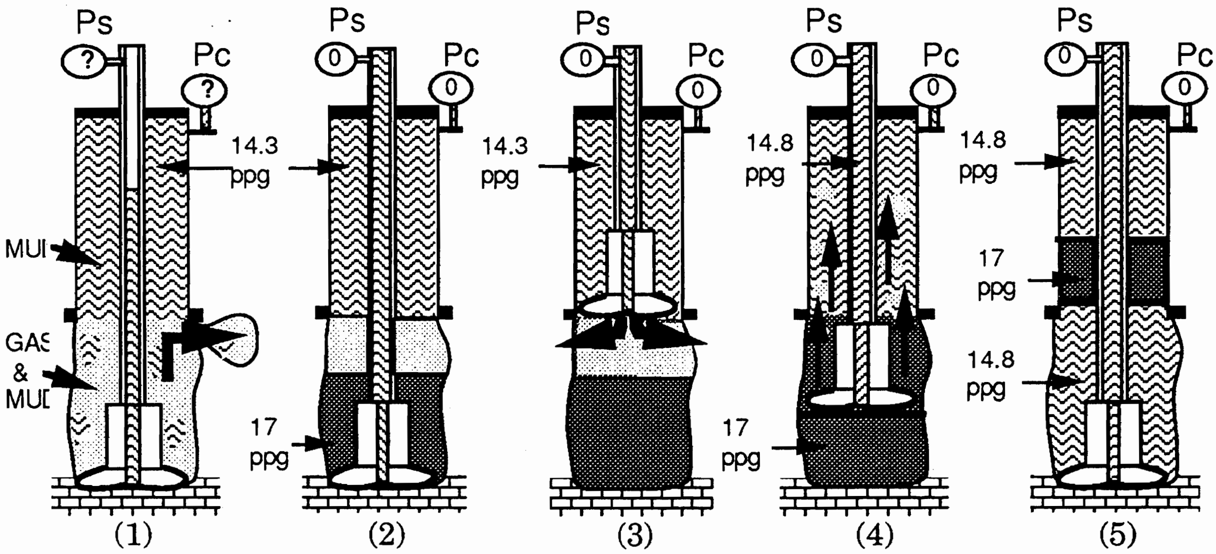
The period of these fluctuations may be as short as ten minutes to as long as a day. Most likely adjustments of the surface choke will have no affect on the drillpipe gauge.

At no time should it be assumed that the pressure of the fluids (gas or mud) within the hole at the bottom of hole is equal to the pressure of the kicking formation.

HIGH WEIGHT PILL

Neal Adams recommended the five steps as depicted in the following sketch for gaining control of an underground blowout. The method is not definitive and the normal kill information will be obscure and un-examinable in most cases.

The objective of the method is to use the existing mud in the hole and a high weight pill to terminate gas flow from the kicking formation until the hole is repaired and a higher weight drilling mud is circulated.



An underground blowout will occur if the fracture resistance of any formation in the open hole is less than the pressure in the mud opposite the formation just after the kick is taken or while the top of the kick is passing by the formation during removal. Underground blowouts can be avoided by setting casing strings deeper in most cases.

The control procedure is presented in the following steps which correspond with the sketch.

1. A gas kick has occurred and an underground blowout is in progress.
2. A pill of heavy mud is circulated to the bottom of the hole.

The density of the pill and its volume is such that the pill in combination with the estimated density of the gas and mud in the hole above the pill, prevents additional gas entry from the kicking formation. Because the density of the gas cut mud and formation pressures, as well as the volume of the hole, are unknown, the actual circulated pill should be three times the computed required volume.

3. Pull (strip) the bit to the fractured formation which in most cases will be the one just below the shoe and squeeze cement the formation to regain or increase its fracture strength. A braden head squeeze (down the annulus) will be required and any gas above the bit must be purged before the squeeze is made. A squeeze of one hundred sacks of cement could be sufficient.

4. Displace any gas cut mud below the squeeze out of the hole in stages with the intended drilling mud.

5. Displace the high density pill out in stages. Because the requisite density of the displacing and subsequent drilling mud is not known, be prepared for another kick while displacing the high density pill.

EXAMPLE

Gain control of a hole which has an underground blowout. The following are known.

$13\frac{3}{8}$" casing depth: 5,000'
leak-off test value at 5,000': 14.9 ppg
bit: $12\frac{1}{4}$"
kicking formation depth: 8,000'
BHA: 8" 3" 600'
drillpipe: 5" 4.276" 7,400'
drilling mud weight: 14.3 ppg
last observed drillpipe pressure before it fell to zero: 150 psig

The first step is to mix and circulate the high density pill into the hole.

Two estimates of the kicking formation's minimum pressure and its minimum equivalent gradient are available. One is based on the observed drillpipe pressure gauge, and the values are

$$P_f = 150 + .052 * 14.3 * 8000 \qquad \mathbf{= 6{,}099\ psig}$$

$$G_f = \frac{6099}{.052 * 8000} \qquad \mathbf{= 14.66\ ppg}$$

The second is based on the leak-off test value, and the values are

$$P_f = .052 * 14.9 * 5000 + .052 * 14.3 * 3000 \qquad \mathbf{= 6{,}105\ psig}$$

$$G_f = \frac{6105}{.052 * 8000} \qquad \mathbf{= 14.68\ ppg}$$

A reasonable estimate of the formation pressure of the kicking formation is on-half pound per gallon more than the above minimums, 15.2 ppg.

An estimate of the average weight of the gas cut mud is based on the open hole containing equal volumes of gas and mud, then the weighted average of the mud in the entire hole is `

$$\text{Est. density gas cut mud} = \frac{14.3 * 5000 + \frac{2.0 + 14.3}{2} * 3000}{8000} \qquad \mathbf{= 12.0\ ppg}$$

The combinations of the pill's length and weight are computed with the formula.

$$15.2 = \frac{12.0 * (8000 - X) + \text{weight pill} * X}{8000}$$

The possible combinations high weight pill length and weight are

Height	Weight
6,400	16.0
5,120	17.0
4,267	18.0
3,657	19.0

The combination of 17.0 ppg and 5,120 feet of high weight pill is selected.

The computed pill volume is

$$\text{Hole capacity} = \frac{12.25^2}{1029.5} \qquad \mathbf{= 0.146\ bbl/ft}$$

$$V_{\text{computed pill}} = 5120 * .146 \qquad \mathbf{= 746\ bbl}$$

Because the recommendation is to circulate three time the computed pill volume, the actual pill to mix and pump is

$$V_{\text{pill}} = 3 * 746 \qquad \mathbf{= 2239\ bbl}$$

The second step is to pull the bit to the fractured formation which will most likely be at the shoe, circulate the annulus, and squeeze cement the formation. Let the strengthen leak-off test be 15.6 ppg.

The third step is to run the bit to the top of the high weight pill and circulate the rest of the gas cut mud out of the hole.

The fourth step is to stage back to the bottom of the hole. The maximum column length of 17.0 ppg mud which can be circulated in one stage is X.

$$.052 * 15.6 * 5000 = .052 * 15.2 * (5000 - X) + .052 * 17.0 * X$$

$$X \qquad \mathbf{= 1{,}111\ ft}$$

BARITE PLUG

The objective of the barite plug method of gaining control of an underground blowout is to form a bridge of Barite solids in the hole which will terminate gas flow above the bridge.

The procedure is not concerned with column pressures and mud weights.

The procedure is to circulate about 500 feet of a 22 ppg Barite pill into the hole above the kicking formation at a rate such that little gas will penetrate the pill. The recommended mixture of materials, per barrel, for a 22 ppg pill is

Barite	**750 lbs**
Fresh water	**21 gal**
SAPP	**$\frac{1}{2}$ lbs**
Caustic Soda	**$\frac{1}{4}$ lbs**

EXAMPLE

If a Barite pill is to be set in a $12\frac{1}{4}$ inches hole, the volume of the pill is

$$\text{Hole capacity} = \frac{12.25^2}{1029.5}$$

Hole capacity = 0.146 bbl/ft

$V_{pill} = 0.146 * 500$ **= 73 bbl**

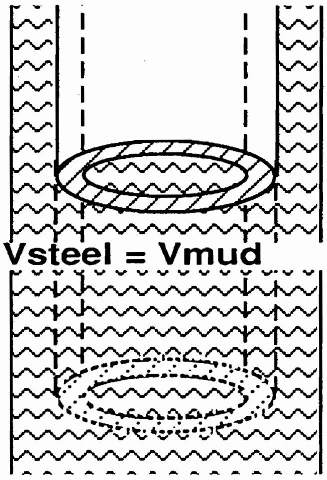
Also, **546 sacks of Barite** and **37 barrels of water** are required.

Cement mixing equipment is required in order to maintain adequate mixing rates.

Dangers are that during any cessation of mixing or pumping, the drillpipe or the annulus before the drillpipe can be pulled may become plugged or bridged.

FILLING THE HOLE ON TRIPS

Pulling of the pipe from the hole on trips causes the mud level in the hole to drop unless it is being continuously filled. The best procedure utilizes a trip tank because the trip tank allows precise accounting of the mud entering or leaving the wellbore. Regardless of the procedure or equipment used, when a stand of drill pipe is pulled from the hole, the volume of the steel of the drill pipe must be replaced by mud. Density of steel is 2,749 pounds/barrel.



$$V_{dp} = \frac{wL}{w_s}$$

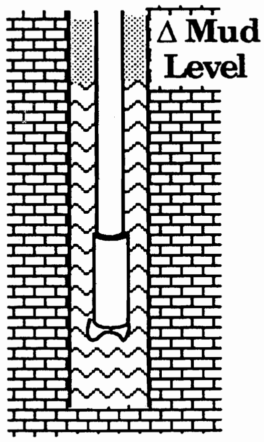
$$V_{dp} = V_h - V_{p+an}$$

$$V_{p+an} = \left[\frac{D^2}{1029} - \frac{w}{w_s}\right] * H$$

w	= weight per foot of pipe (lb/ft)
L	= length of pipe (feet)
w_s	= specific weight of steel (2750 lb/bbl)
V_{dp}	= volume steel in pipe (bbls)
v_{p+an}	= volume mud contained by the annulus and pipe (bbls)
H	= drop in mud level (feet)
D	= hole diameter (inches)

EXAMPLE

93'
5"x19.5 ppf
Drill Pipe



When one stand (93') of 5" by 19.5 ppf drill pipe is pulled, how many feet does the mud level drop in a 9" hole?

$$H = \frac{\frac{19.5 * 93}{2749}}{\frac{9^2}{1029} - \frac{19.5}{2749}} \quad \mathbf{= 9.2\ feet}$$

The mud weight equivalent at the bottom of a 16,000' hole changes from 11.8 ppg to

$$MWE = \frac{.052 * 11.8 * (16{,}000 - 9.2)}{.052 * 16000} \quad \mathbf{= 11.79\ ppg}$$

which is insignificant, but does point out that filling the hole on each stand prevents loss of bottom hole pressure.

However, the pulling of 10 stands if drilling in the same hole at 2000 feet without filling the hole reduces the mud weight equivalent from 11.8 ppg to 11.3 ppg which may be significant.

Swabbing and lax hole filling procedures combine in reducing equivalent density and could cause problems.

$$MWE = \frac{.052 * 11.8 * (2000\text{-}92)}{.052 * 2000} \qquad \mathbf{= 11.3\ ppg}$$

NOVEL TECHNIQUES

It may be possible to squeeze a viscous polymer or diesel oil gunk into a kicking formation to terminate gas entry until the hole can be repaired and the mud weight increased.

It may be possible to pull the drillstring and run a packer to terminate gas entry until corrective action is taken.

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CHAPTER V

RIG HYDRAULICS

OBJECTIVES

Objective of Rig hydraulics has eleven facets and one myth:

1. cleaning of the bottom of the hole while drilling
2. cleaning of the drill bit
3. transportation of solids to the surface at a reasonable rate
4. removal of drill solids and cavings with mud cleaning equipment
5. equivalent circulating density for the prevention of lost circulation
6. running of close clearance tools
7. circulating by close clearance tools
8. friction pressure losses through and around the drillstring and rig piping
9. hydraulic energy consumption and pressure drop through down hole tools within the drillstring
10. circulation of cement and completion fluids
11. surge and swab
12. holes do not wash out (This is the myth.)

BASIC RIG HYDRAULIC EQUATION

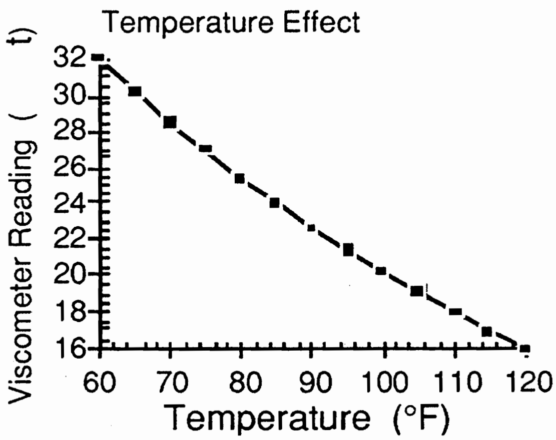
$$\text{Pump Pressure} = \underset{\substack{\uparrow \\ \text{lifts} \\ \text{cuttings}}}{\Delta P_{pipe\&ann}} + \underset{\substack{\uparrow \\ \text{cleans} \\ \text{bottom of} \\ \text{hole and} \\ \text{bit teeth}}}{\Delta P_{jets}} + \underset{\substack{\uparrow \\ \text{turns} \\ \text{bit}}}{\Delta P_{motor}} + \underset{\substack{\uparrow \\ \text{diamond} \\ \text{bits}}}{\Delta P_{liftoff}} + \underset{\substack{\uparrow \\ \text{power} \\ \text{for} \\ \text{tools}}}{\Delta P_{tools}}$$

MUD RHEOLOGY

Recall that the University of Tulsa data showed that a powerlaw Reynold's Number of 1,800 cleaned cuttings beds and prevented their accumulation in high angle holes.

TEMPERATURE EFFECT

As the temperature of a mud increases, the shear stress at a shear rate decreases. The equation for temperature adjustments of viscometer readings (shear stress) is



$$\tau_T = \frac{\tau_{T2}^{U}}{\tau_{T1}^{U-1}}$$

$$U = \frac{T - T_1}{T_2 - T_1}$$

τ_T = Shear stress or viscometer reading at temp. T

τ_{T1} = Shear stress or viscometer reading at temperature T_1

τ_{T2} = Shear stress or viscometer reading at temperature T_2

T = Elevated downhole temperature; deg F

T_1 = Low temperature at which a shear stress was measured; deg F

T_2 = High temperature at which a shear stress was measured;deg F

PRESSURE EFFECT

As the pressure in the mud increases, the shear stress at a shear rate increases. The equation for pressure adjustment of viscometer readings (shear stress) is

$$\tau_{P2} = \tau_{P1}\, e^{c(P_2 - P_1)}$$

c = Pressure constant which must be computed for each mud

τ_{P2} = Shear stress or viscometer reading at a higher pressure

τ_{P1} = Shear stress or viscometer reading at a lower pressure

P_2 = Higher pressure

P_1 = Lower pressure

EXAMPLE REYNOLD'S NUMBER AND PRESSURE LOSS

Compute the expected viscometer readings for the temperature of 92 degrees F. The temperature for the first set of readings was 62 degrees F and 147 degrees F for the second set. These values were taken with a FANN by heating the mud in the FANN cup to two temperatures. The values at 92 degrees F are to be added to the table.

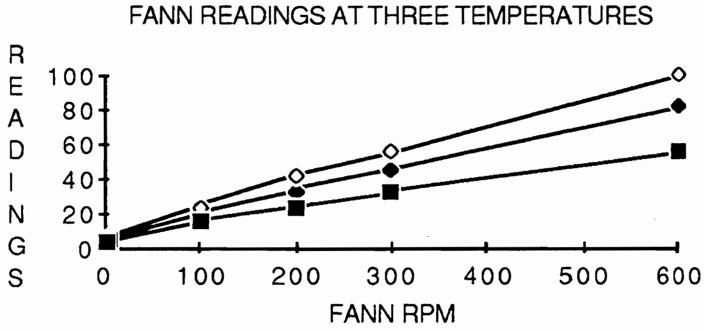


TABLE OF VISCOMETER READINGS

RPM	τ_{62}	τ_{92}	τ_{147}
===	===	===	====
600	100	82	55
300	55	45	32
200	42	33	24
100	24	20	15
6	6	5	4
3	5	4	3

≠ Low temperature data

≠ Generated column of data

≠ High temperature data

$$U = \frac{92 - 62}{147 - 62} = \mathbf{0.3529}$$

An example calculation for ascertaining the expected viscometer reading at 92 degrees F and 600 rpm is the following

$$\tau_{92\ 600} = \frac{\tau_{147\ 600}^{\ 0.3529}}{\tau_{62\ 600}^{\ 0.3529-1}} = \mathbf{82\ lb/100sq.ft.}$$

This is the value recorded in the above table and used in the following example powerlaw friction pressure loss example.

FRICTION LOSS, MODELS, & FLOW REGIMES

There are four recognized types of flow regimes and each of the four has its application. The powerlaw model predicts all of the these flow types.

1. Plug
2. Laminar
3. Transitional
4. Turbulent

If a fluid has a yield strength and its yield strength has a value greater than the shear stress at the wall of a pipe or annulus, then the fluid will be in plug flow.

A suitable criterion is if the value of the Reynold's number is less than ten (10) and the fluid has gel strength, then the flow regime is plug.

The other flow regimes are predicted with the Moody diagram and the Reynold's number.

POWERLAW FLOW MODEL

The powerlaw flow model is the superior model of those which are popular for the computation of

1. Flow regimes
2. Friction pressure losses within pipe and annuli
3. Velocity profiles

The working powerlaw equation is

$$.01066\,\tau = K'\left(1.703\,R\right)^{n'}$$

τ = Fann rotary viscometer reading
R = Fann rotary viscometer speed

For reference the laminar Newtonian equations are

$$\frac{\Delta P}{L} = \frac{\mu\ v}{1500\ id} \qquad \frac{\Delta P}{L} = \frac{\mu\ v}{1000(H - OD)}$$

ΔP = pressure loss for the length of pipe or annulus, L; psi
L = length of pipe or annulus; ft
μ = fluid viscosity; cps
v = fluid velocity; ft/sec
id = inside diameter of pipe; inch

H = hole diameter; inch
OD = outside diameter of pipe; inch

The following steps illustrate the **full, complete, no short cut** powerlaw model procedure of Rabinowitsch, Metzner, Reed, Dodge, and Savins.

STEPS

1. Make a logarithm-logarithm chart of the rotary viscometer data

   τ = rotary viscometer reading; lbf/100sq ft

   R = rotary speed; rpm

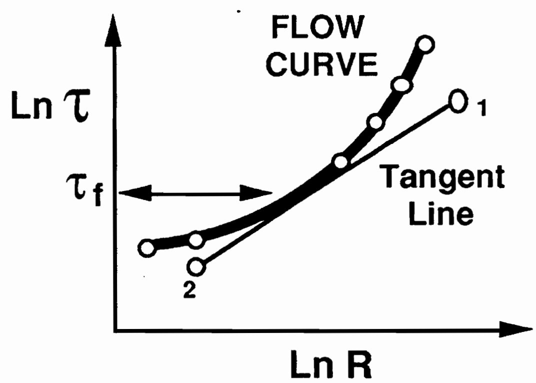
2. Draw a tangent line at a selected point and record τ_f

   f = fluid

3. Select any two points on the tangent line and record their rpm's (R_1 & R_2) and rotary viscometer readings (τ_1 & τ_2)

4. Compute the flow index (n)

$$n = \frac{\mathrm{Ln}\left(\frac{\tau_1}{\tau_2}\right)}{\mathrm{Ln}\left(\frac{R_1}{R_2}\right)}$$



5. Compute the viscometer consistency index (k_v)

$$k_v = \frac{0.01066\,\tau_1}{(1.703\,R_1)^n}$$

6. Compute the flow adjustment factors (F_p)and (F_a)

$$X = \frac{0.123/n}{1 - 1.0678^{-2/n}}$$

$$F_p = \left[\frac{3n+1}{4n\,X}\right]^n \qquad F_a = \left[\frac{2n+1}{3n\,X}\right]^n$$

p = pipe
a = annulus

7. Compute the consistency indices for pipe (k_p) and annulus (k_a)

$$k_p = F_p * k_v \qquad\qquad k_a = F_a * k_v$$

8. Compute the Reynold's numbers for pipe (N_p) and annulus (N_a) and the average velocity (V_p) or (V_a) of the fluid

$$V_p = \frac{Q}{2.45\, d^2} \qquad\qquad V_a = \frac{Q}{2.45\,(H^2\text{-}D^2)}$$

$$N_p = \frac{1.86}{k_p} * \left[\frac{d}{96}\right]^n * V_p^{(2\text{-}n)} * MW$$

$$N_a = \frac{2.79}{k_a} * \left[\frac{H\text{-}D}{144}\right]^n * V_a^{(2\text{-}n)} * MW$$

Q	=	Circulation rate; gpm
H	=	Hole diameter; inch
D	=	OD of pipe; inch
d	=	ID of pipe; inch
MW	=	mud weight; ppg
V_a	=	average fluid velocity; fps

9. Ascertain the friction factor and the flow regime from the chart or use the API friction factor equations (f)

Laminar If N_p or $N_a < 3470 - 1370\, n$

$$f_p = 16/N_p \qquad\qquad f_a = 24/N_a$$

Turbulent If N_p or $N_a > 4270 - 1370\, n$

$$f_p = a/N_p^{\,b} \qquad\qquad f_a = a/N_a^{\,b}$$

Transitional If $3470 - 1370\, n < N_p$ or $N_a < 4270 - 1370\, n$

$$f_p = \left[\frac{N_p - X}{800}\right] * \left[\frac{a}{Y^b} - \frac{16}{X}\right] + \frac{16}{X}$$

$$f_a = \left[\frac{N_a - X}{800}\right] * \left[\frac{a}{Y^b} - \frac{24}{X}\right] + \frac{24}{X}$$

$$a = \frac{\log_{10}(n) + 3.93}{50} \qquad b = \frac{1.75 - \log_{10}(n)}{7}$$

$$X = 3470 - 1370\,n \qquad Y = 4270 - 1370\,n$$

10. Compute the friction pressure loss per foot of pipe or annulus $(\frac{\Delta P_p}{L})$

$$\frac{\Delta P_p}{L} = \frac{f_p\, MW\, V_p^2}{25.8\, d} \qquad \frac{\Delta P_a}{L} = \frac{f_a\, MW\, V_a^2}{25.8\,(H\text{-}D)}$$

L = length of pipe or annulus

$\frac{\Delta P_p}{L}$ = friction pressure loss, psi/ft

11. Compute the expected rotary viscometer for the friction pressure loss per foot (τ_c)

$$\tau_{cp} = 281.4\ d\frac{\Delta P_p}{L} \qquad \tau_{ca} = 281.4\,(H\text{-}D)\frac{\Delta P_a}{L}$$

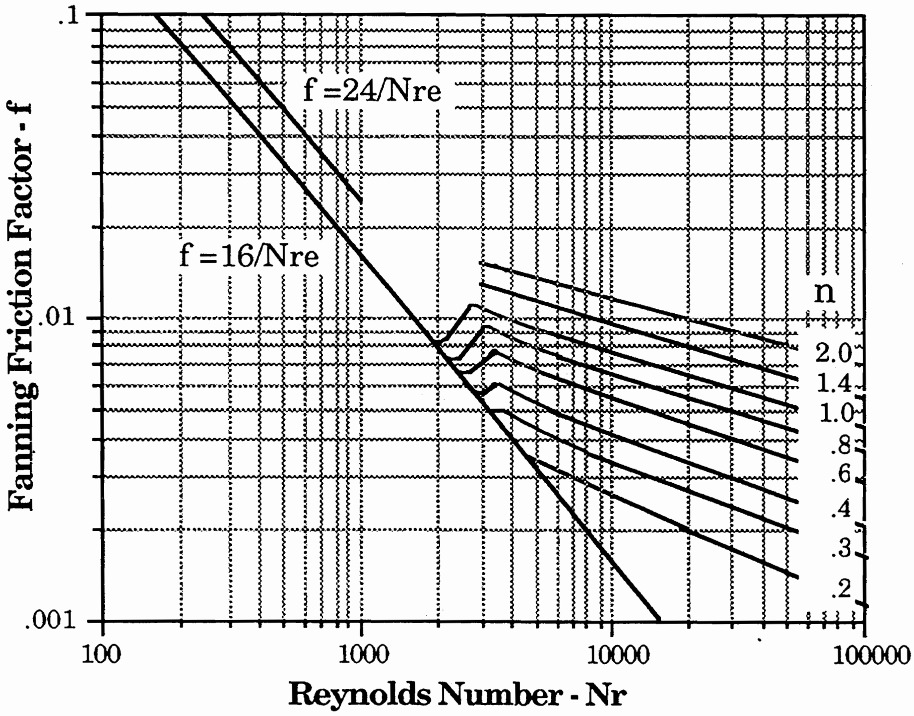
c = computed

12. If the computed rotary viscometer reading is in satisfactory agreement with the value recorded in step number two, then the friction pressure loss is correct.

$$\Delta P_p = \frac{\Delta P_p}{L} * L$$

If the computed values are not satisfactorily precise, then go to step number 2 and repeat the steps.

Friction Factor Chart



EXAMPLE PRESSURE LOSS

Compute the friction pressure losses within the pipe and annulus for the following data.

Pipe:
- OD = 5"
- ID = 4.276"

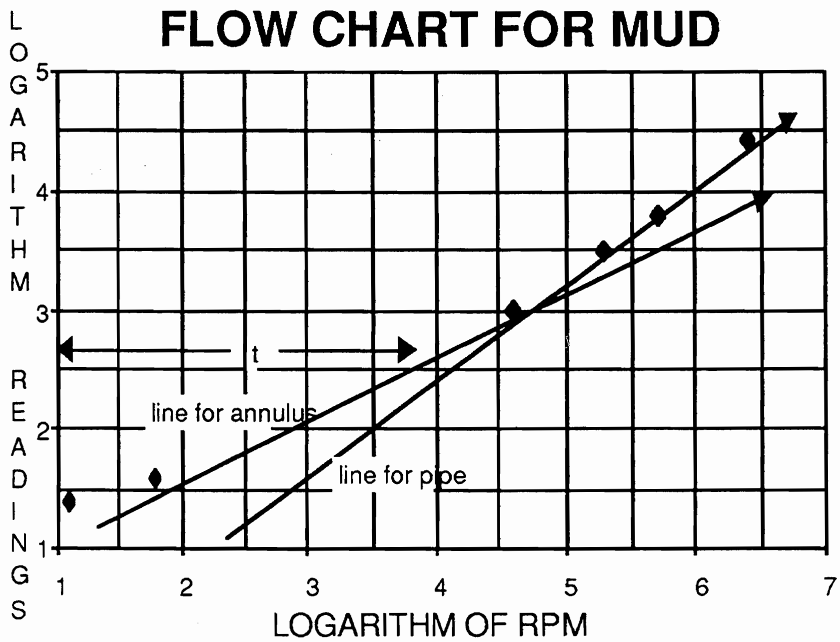
Annulus:
- Hole = 9"
- OD = 5"

Length = 1,000 ft

Mud:
- Circ. = 400 gpm
- MW = 11.4 ppg

FANN READINGS

R rpm	τ lb/100sqft
600	82
300	45
200	33
100	20
6	5
3	4



STEPS

1. The flow chart is drawn

2. Two tangent lines are drawn - one for the pipe and one for annulus - at selected values of Fann readings, τ_{fp} and τ_{fa}. The chosen values of the tangent points are

   $\tau_{fp} = 60$

   $\tau_{fa} = 15$

3. Two values on each of the tangent lines for the pipe and the annulus are selected

   PIPE

$\tau_1 = 55$	$R_1 = 403$
$\tau_2 = 3$	$R_2 = 12$

   ANNULUS

$\tau_1 = 40$	$R_1 = 403$
$\tau_2 = 3$	$R_2 = 3$

4. The values of n for the pipe and annulus are

$$n_p = \frac{\mathrm{Ln}\,\frac{55}{3}}{\mathrm{Ln}\,\frac{403}{12}} \qquad \mathbf{= 0.828}$$

$$n_a = \frac{\mathrm{Ln}\,\frac{40}{3}}{\mathrm{Ln}\,\frac{403}{3}} \qquad \mathbf{= 0.529}$$

5. The viscometer consistency indices (k_v) are

$$k_{vp} = \frac{0.01066 * 55}{(1.703 * 403)^{0.828}} \qquad \mathbf{= .0026}$$

$$k_{va} = \frac{0.01066 * 40}{(1.703 * 403)^{0.529}} \qquad \mathbf{= .0135}$$

6. The viscometer to pipe flow adjustment factors (F_p) and (F_a) are

$$X_p = \frac{0.123/0.828}{1 - 1.0678^{-2/0.828}} \qquad = \mathbf{1.014}$$

$$F_p = \left(\frac{3*0.828+1}{4*0.828*1.014}\right)^{0.828} \qquad = \mathbf{1.031}$$

$$X_a = \frac{0.123/0.529}{1 - 1.0678^{-2/0.529}} \qquad = \mathbf{1.0583}$$

$$F_a = \left[\frac{2*0.529+1}{3*0.529*1.0583}\right]^{0.529} \qquad = \mathbf{1.1135}$$

p = pipe
a = annulus

7. The consistency indices for pipe (k_p) and annulus (k_a) are

$$k_p = 1.031 * 0.0026 \qquad = \mathbf{0.0027}$$

$$k_a = 1.1135 * 0.0135 \qquad = \mathbf{0.015}$$

8. The Reynold's numbers for pipe (N_p) and annulus (N_a) and the average velocity (V_p) or (V_a) of the fluid are

$$V_p = \frac{400}{2.45 * 4.276^2} \qquad = \mathbf{8.929\ fps}$$

$$V_a = \frac{400}{2.45\,(9^2 - 5^2)} \qquad = \mathbf{2.916\ fps}$$

$$N_p = \frac{1.86}{.0027}\left[\frac{4.276}{96}\right]^{0.828} * 8.929^{(2-0.828)} * 11.4 \qquad = \mathbf{7{,}772}$$

$$N_a = \frac{2.79}{.015}\left[\frac{9-5}{144}\right]^{0.529} * 2.916^{(2-0.529)} * 11.4 \qquad = \mathbf{1{,}538}$$

9. The friction factors and the flow regimes from the chart or use API equations (f) are

PIPE

Flow regime in the pipe Y = 4270 - 1370 * 0.828 **= 3,136**

Because N_p = 7772 > 3136 = Y the flow regime is turbulent within the pipe.

$$a = \frac{\log_{10}(0.828) + 3.93}{50} \qquad b = \frac{1.75 - \log_{10}(0.828)}{7}$$

a = **0.0770** b = **0.2615**

$$f_p = a/N_p{}^b = \frac{0.0770}{7772^{0.2617}} \quad \mathbf{= .0074}$$

ANNULUS

Flow regime in the annulus: X = 3470 - 1370 * 0.529 **= 2,745**

Because N_a = 1538 < 2745 = Y the flow regime is laminar within the annulus.

$f_a = 24/1538$ **= 0.0156**

10. The friction pressure losses per foot of pipe or annulus ($\frac{DP_p}{L}$) are

$$\frac{\Delta P_p}{L} = \frac{0.0074 * 11.4 * 8.929^2}{25.8 * 4.276} \quad \mathbf{= 0.0610}$$

$$\frac{\Delta P_a}{L} = \frac{0.0156 * 11.4 * 2.916^2}{25.8 * (9 - 5)} \quad \mathbf{= 0.0147}$$

11. The expected rotary viscometer for the friction pressure losses per foot (τ_c) are

τ_{cp} = 281.4 * 4.276 * 0.0610 **= 73.4**

τ_{ca} = 281.4 * (9-5) * 0.0147 **= 16.5**

12. The computed rotary viscometer readings are in satisfactory agreement with the values recorded in step number two (15 versus 16.5 and 60 versus 73.4); therefore, the friction pressure losses per foot of the pipe (.0610 psi/ft) and the annulus (0.0147 psi/ft) are correct.

ΔP_p = 0.0610 * 1000 **= 61.0 psi**

ΔP_a = 0.0147 * 1000 **= 14.7 psi**

EFFECT OF MUD WEIGHT ON BIT HYDRAULICS

Bit hydraulics are lost because of increased friction within the rig hydraulic system as mud weight is increased. The bit hydraulic loss is proportional to the ratio of mud weights to the 0.8 power.

EXAMPLE MUD WEIGHT ON BIT HORSEPOWER

A pump is capable of a pressure of 2000 psig at a circulation rate of 400 gpm. Friction loss is 1000 psi and the mud weight is 12 ppg. Calculate the available bit hydraulic horsepower if the mud weight is increased to 15 ppg and the circulation rate remains at 400 gpm.

$$\text{Current Bit HHP} = \frac{1000 * 400}{1714} \qquad \mathbf{= 233.4\ hp}$$

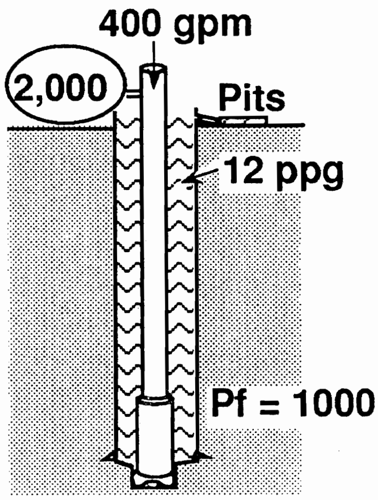
$$\text{New friction loss} = 1000 * \left(\frac{15}{12}\right)^{.8} \qquad \mathbf{= 1195\ psig}$$

$$\text{New pressure for bit} = 2000 - 1195 \qquad \mathbf{= 805\ psid}$$

If the jets on the bit are enlarged, then

$$\text{New bit HHP} = \frac{805 * 400}{1714} \qquad \mathbf{= 188\ hp}$$

and **45 hp** are lost to the additional friction of the heavier mud.

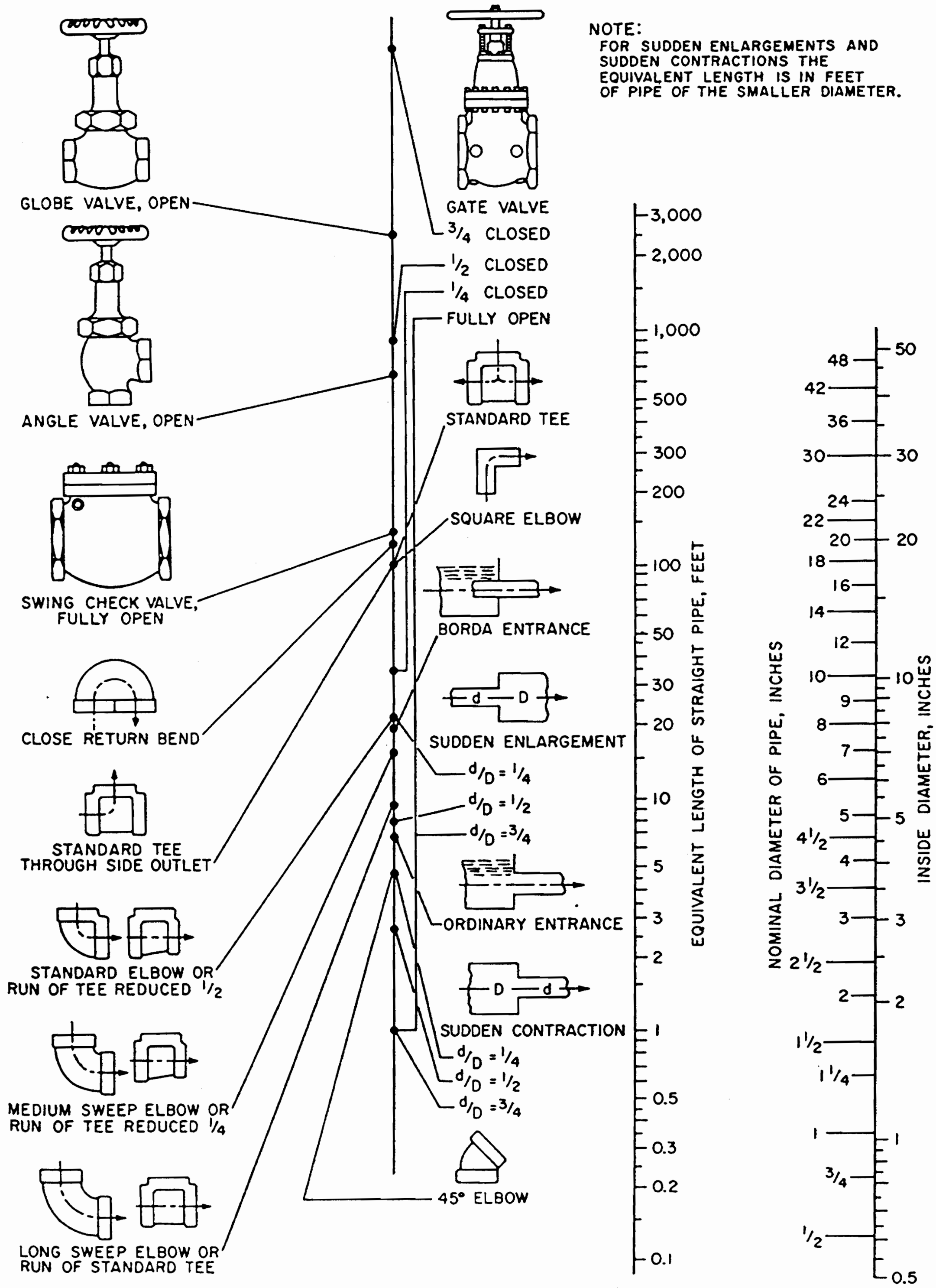


NOTE: $P_f = \dfrac{f\,w\,v^2\,L}{25.8\,d} \qquad f = .061\,N_R^{-.22}$

$$N_R = 928\,\frac{w\,d\,v}{vis}$$

(Common fluid mechanics equations)

$$\text{Therefore:} \quad P_f = \frac{vis^{.22}\,W^{.78}\,Q^{1.78}\,L}{9359\,d^{4.78}}$$



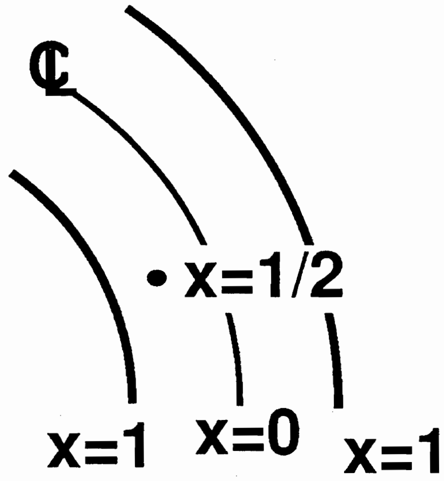
Turbulent flow resistance of valves and various fittings. (Courtesy of Crane Company.[4])

ANNULAR MUD VELOCITY PROFILES

Muds and flow regimes which have flatter velocity profiles carry solid particles and clean drill holes better.

Muds with low values of the power law parameter, n', have the flatter velocity profiles (n' < .4 are good).

The annular laminar flow velocity profile may be computed with the following equations.



$$V_{max} = 5\left[\frac{1.28}{k_a}\frac{\Delta P}{L}\right]^{1/n}\left[\frac{n}{n+1}\right]$$

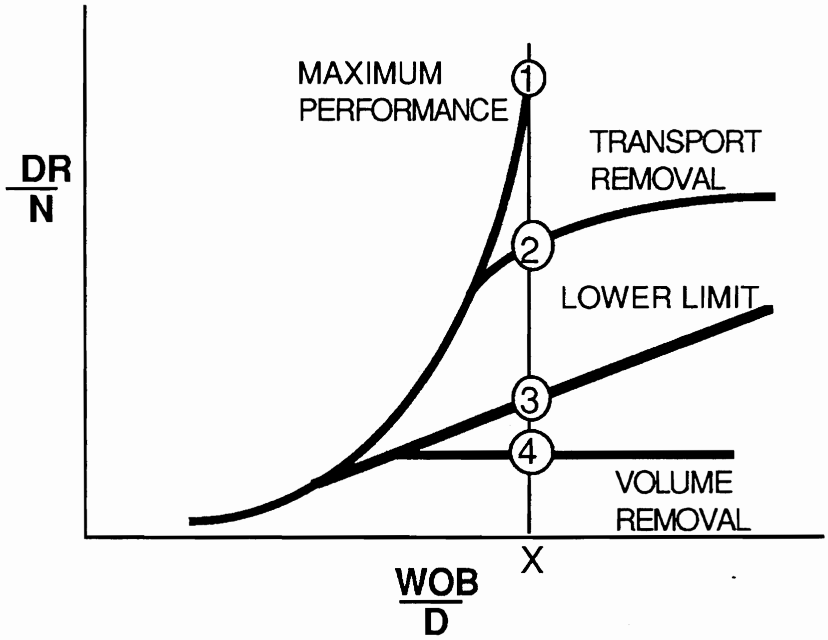
$$V_x = V_{max}\left[1 - X^{\frac{n+1}{n}}\right]$$

V_{max}	=	maximum velocity in the center of the annulus; ft/sec
V_x	=	velocity at x distance from the center of the annulus; ft/sec
X	=	distance from the center of the annulus on either side of the center; ft
DP	=	pressure loss over the length of annulus; psi
L	=	length of annulus in question; ft
n '	=	power law parameter
ka	=	power law parameter

BINGHAM'S DRILLING EFFICIENCY DIAGRAM

Bingham's drilling efficiency diagram depicts four fundamental ways in which a drill bit responds to weight on bit and rotary speed during the drilling of a formation. His diagram is based on hundreds of field and laboratory tests.

The axes of his diagram are WEIGHT ON BIT divided by DRILL BIT DIAMETER, $(\frac{WOB}{D})$ and DRILLING RATE divided by ROTARY SPEED, $(\frac{DR}{N})$. The WOB is divided by drill bit diameter so that the performances of bits of various diameters may be placed on one diagram. The ordinate is the volume of formation penetrated per revolution of the drill bit and is therefore the efficiency of the bit's teeth in cutting of the formation. (Note that if in 2 minutes 1 feet of hole is drilled by a 6 inches bit which is rotating at 100 rpm, then 339 cubic inches of rock (1 ft* 12 in./ft * .7854 * 6^2) will be drilled in 200 revolutions (2 min * 100 rev/min) or 1.7 cubic inches are drilled per revolution of the drill bit.) Based on this piece of reasoning, higher points are best. Drill bits wear faster if higher weights on bit are used.



Thus, the best points are to the left in his diagram, in regard to bit wear. Combining the two pieces of reasoning, it is seen that points which are in the upper left region of the diagram are most efficient where the bit will drill the most rock and last the longest.

Bill Maurer argued that the maximum value of the exponent, a, in the equation, $\frac{DR}{N} = k\,(\frac{WOB}{D})^a$, has a value of two. This equation represents the maximum performance line drawn on the diagram.

The transport removal region is thought to occur if the cleaning of the bottom of the hole is not thorough. For example, chips not removed from the surface of the bottom of the hole could be being reground by the teeth of the bit and preventing teeth from grinding new rock. Because cleaning is dependent on hydraulics, it is believed that a line such as that drawn in the transport removal region on the diagram results from insufficient bit hydraulics. In the above equation the exponent, a, will have a value of less than one.

The lower drilling region presents the prevalent drilling efficiencies which occur in the field. The value of the exponent, a, is one. It is thought that characteristics of the drilling mud relative to those of the formation being drilled are responsible.

The volume removal region is thought to be synonymous with the lack of cleaning of the teeth of the drill bit. This is also called bit 'balling.'

A test to be described later can be conducted which will distinguish the region in which a drilling operation is occurring. If a test shows, that while drilling with a $\frac{WOB}{D}$ equal to X, corresponding to an efficiency indicated by point number 1, then nothing can be done to raise the efficiency with the current drill bit. A different drill bit may have a pronounced affect.

If a test shows an efficiency at point number 2, then additional bit hydraulics or extended nozzles should help.

If a test shows an efficiency at point number 3, then changes in the drilling mud should help. It has been shown in laboratory tests that differential pressure will produce lower limit drilling efficiencies. Water drills as fast as air if equal differential pressures are maintained. Quar gum causes lower limit drilling.

If a test shows an efficiency at point number 4, then a higher circulation rate or a center jet to better clean the teeth of the bit should help.

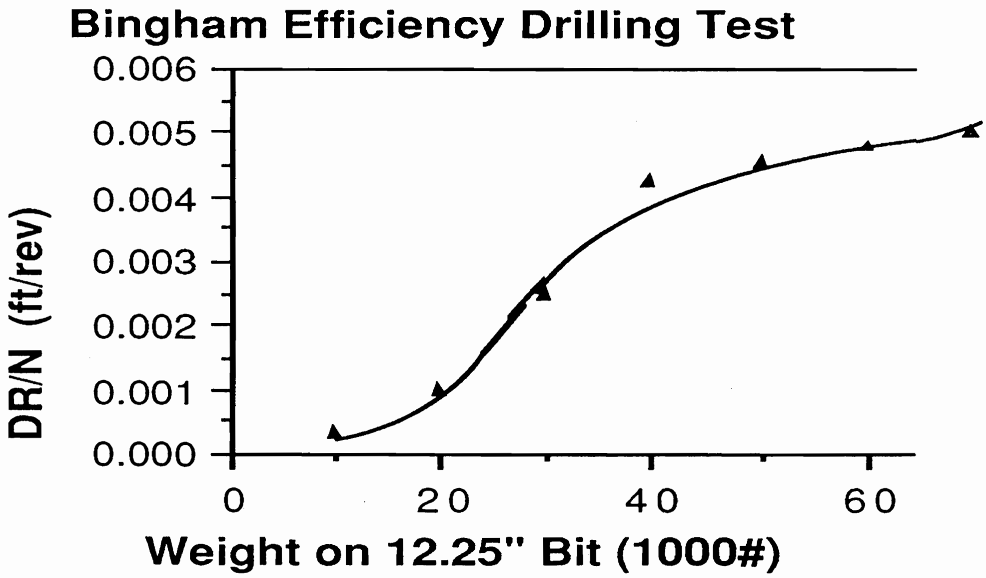
EXAMPLE

A drilling test with a 12.25" drill bit which is rotated at 100 rpm gave the following data and computed results.

DRILLING TEST DATA		COMPUTED RESULTS
WOB	DR	DR/N = DR/60/100
1,000#	ft/hr	ft/rev
10	1.8	.0003
20	6.0	.001
30	15.0	.0025
40	25.0	.0042
50	27.0	.0045
60	29.0	.0048
70	30.0	.005

The data and the computed results gave the chart on the right.

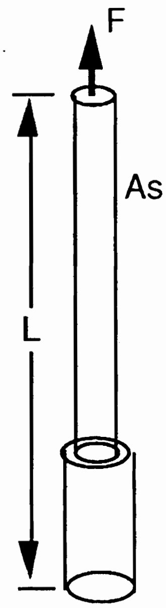
At the WOB of 50,000 lbs the drilling will be in the transport and removal region. Additional bit hydraulics is required to give better bottom hole cleaning and a higher drilling efficiency.



DRILL OFF TEST EQUATIONS

This drill off test method requires an initial weight be put on the drill bit, the drum brake be set, and the change in the weight indicator values be recorded versus time. The length of the string is also required.

$$\frac{\Delta F}{As} = E\frac{\Delta L}{L} \quad \text{Hooke's law}$$



$$\text{Drill rate} = \frac{\Delta L}{\Delta t} = \frac{\Delta F * L}{As\ E\ \Delta t}$$

$$\frac{ft}{hr} = \frac{ft}{hr} = \frac{lb * ft}{in^2 * \frac{lb}{in^2} * sec * \frac{1\ hr}{3600\ sec}}$$

$$\text{Drill rate} = \frac{\Delta F * L * 3600}{\frac{w}{3.4} * 3E7 * \Delta t}$$

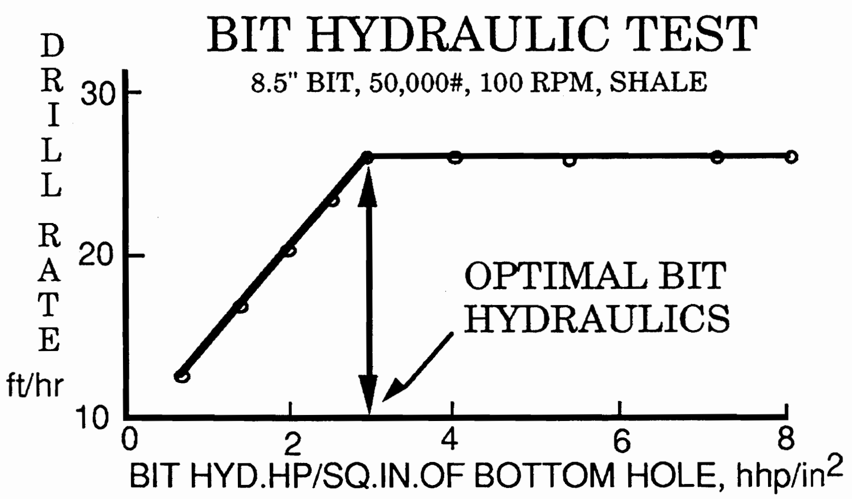
Drill rate	= ft/hr
ΔF	= change in force on the pipe; lbf
L	= length of free pipe; ft
w	= linear weight of pipe; ppf
Δt	= time for change in pipe force; sec
ΔL	= pipe stretch; ft

OPTIMAL BOTTOM HOLE CLEANING

The first and the only arbitrary step in optimal bottom hole cleaning is the selection of the desired bit hydraulic horsepower (BHHP) per square inch of hole bottom or impact force(IF) per square inch of hole bottom. After one of these values are selected the next step is compute the pump operation pressure, circulation rate, and jet sizes which maximizes that portion of the BHHP or IF produced by the pump and minimize the amount lost in circulating friction. The following information will assist in the selection of BHHP/sq.in. of hole bottom.

Required Bit Hydraulic Horsepower

Laboratory and field drilling tests show that drilling rate rises with increased bit hydraulics to a maximum value and thereafter fails to cause a further rise. This phenomenon is interpreted to mean that once the bottom of the hole is cleaned that further efforts at cleaning is a waste of bit hydraulics. The authors field data show that three hydraulic horsepower per square inch of bottom hole was satisfactory for all drilling with rolling cone drill bits. A chart of typical data is the following.



To conduct this test on the rig it is only necessary to measure the drilling rate at various circulation rates while maintaining the rotary speed and weight on bit constant.

NUMBER OF JETS AND THE JET PRESSURE DROP

Mobil Oil suggested that a cavitation-like process as the probable cause for the effect that as jets are operated at deeper depths the pressure drop across the jets decrease. The reduction can be as much as 15%. The theoretical equation with no back pressure in consistent units is

$$P_j = \frac{MW\,Q^2}{2g_c C_d^2 A^2} \qquad \text{or} \qquad P_j = \frac{MW\,V_j^2}{1120} \text{ in field units}$$

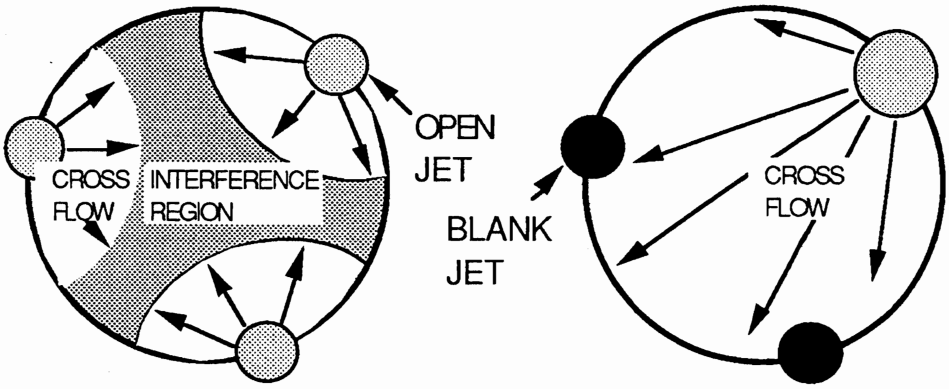
They published the following two equations. Both equations assume a 95% jet discharge coefficient, C_d, and are in practical drilling units (n is number of open jets)

$$P_j = [1 - 0.035\ F\ n]\frac{MW * V_j^2}{1120} \qquad F = \frac{Depth}{1000} \text{ if Depth} < 1000 \text{ ft}$$

$$F = 1.0 \qquad \text{if Depth} > 1000 \text{ ft}$$

$$P_j = [1 - 0.035\ F\ n]\ MW * \left\{\frac{12.51 * Q}{J1^2 + J2^2 + J3^2}\right\}^2$$

During one revolution of the drill bit, only 15% to 20% of the bottom of the hole is impacted by the jet stream.



The jet streams can only wash away the bottom of the hole on which they impact and then only if the pressure drop across the jets are about equal to the compressive strength of the formation. Thus, soft clay can normally be cut and hard rock cannot.

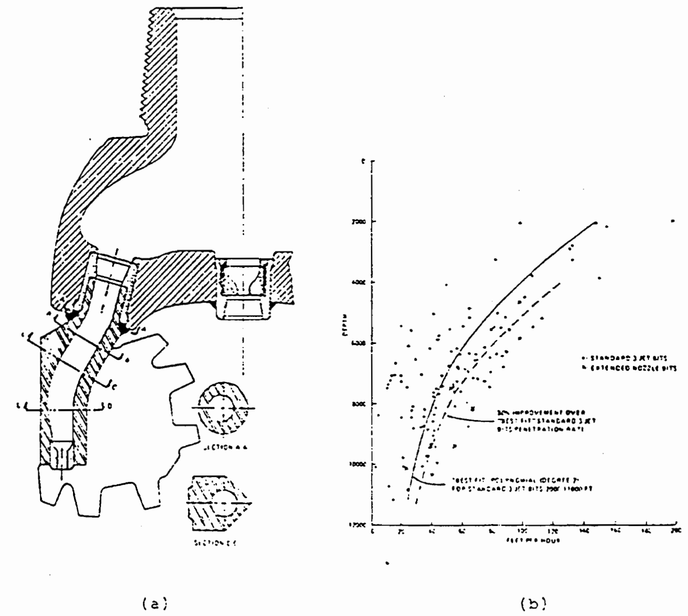
Center Jet and Extended Jet

Shell published the adjacent chart which shows that center and extended jets drilled 30% faster than conventional jets in Louisiana.

The center jet only cleans the teeth of the bit. The center jet stream impacts on the spear point of the drill bit and sends a cross-flow stream up and out across the shell of the cone. Drill solids within the rows of teeth are washed away. Of course, if drill solids are removed with normal washing of the teeth, then the center jet will not raise drilling efficiency.

Two field practices are common in regard to sizing the center jet. One is to use a center jet of a size equal to the other jets. The other is to use a minimum size center jet; a number 12 jet for example.

The extended jet has raised drilling rates in those cases where bit hydraulics are limited by a small mud pump.

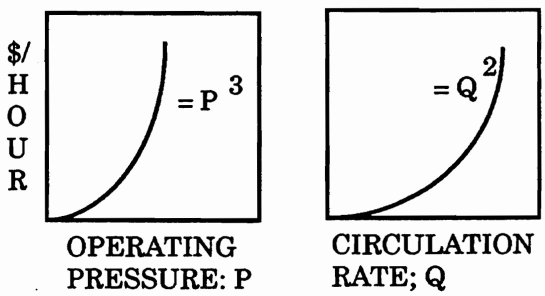


(a) Extended Center Jet Bit
(b) Extended Nozzle Bits Drill 30% Faster in Louisiana

Mud Pump Maintenance Costs

Pump maintenance and lost rig time are closely related. These two factors as well as bit hydraulic requirements should be carefully considered while selecting a mud pump operating pressure and circulation rate. The two charts show pump maintenance costs versus operation pressures and circulation rates.

PUMP MAINTENANCE COSTS



The two charts show that raising the pump pressure is the most costly means of raising bit hydraulics.

Fuel costs for pumping are proportional to either pump operating pressure, P, or circulation rate, Q.

Optimizing Rig Hydraulics

There are three popular flow variables on which bit hydraulics may be optimized. These are the bit hydraulic horsepower per square inch of bottom hole area, the pump operating pressure, and the circulation rate.

The following three problems illustrates these optimizations.

EXAMPLE

Optimization with a Chosen Bit Hydraulic Horsepower per Square Inch of Bottom Hole Area

Optimize the hydraulics while drilling with a 8.5" drill bit with 10.0 ppg mud. Jets in the bit are three #14. Choose a BHHP/sq. in. value of 3.0.

Steps

1. Ascertain the j and m in the equation $P_f = j\ Q^m$. This is done by running a circulating pressure test at the rig site. Conduct the test by varying the pump speed and recording the pump pressure and circulating rate at each speed. Let the data gathered be the following.

TEST DATA		COMPUTED	
Q	P_p	P_j	P_f
===	===	===	===
569	2550	1465	1085
455	1650	912	738
347	1035	530	505
285	800	356	444
228	500	228	272
171	300	128	172

A. A sample computation for the pressure drop across the jets is as follows.

$$P_j = MW * \left\{\frac{12.51 * Q}{J1^2 + J2^2 + J3^2}\right\}^2$$

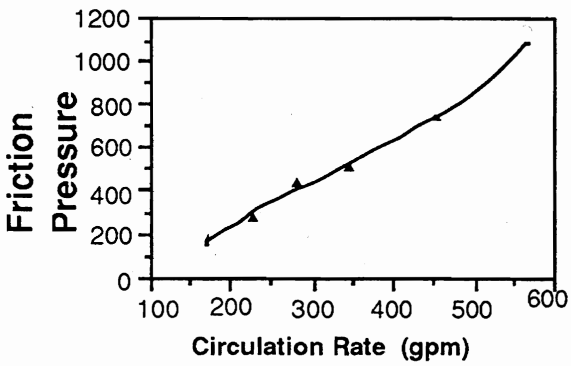
$$P_j = 10.0 * \left\{\frac{12.51 * 569}{14^2 + 14^2 + 14^2}\right\}^2 \qquad \mathbf{= 1{,}465\ psi}$$

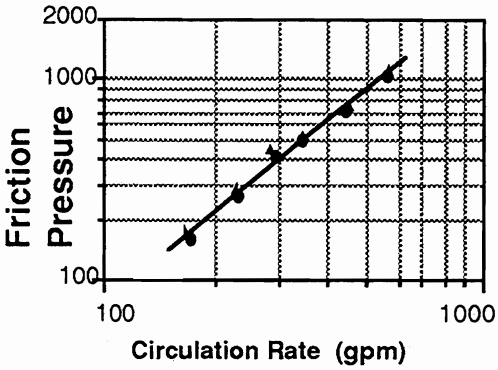
B. A sample computation for the friction loss is as follows.

$$P_f = P_p - P_j \qquad = 2{,}550 - 1{,}465 \qquad \mathbf{= 1{,}085\ psi}$$

2. Make a chart of pressure friction loss v. circulation rate.

RIG CIRCULATION TEST





3. If the chart appears reasonable compute the parameters j and m of the equation, $P_f = jQ^m$, with the multiple regression procedure demonstrated in chapter #2. The points in the left chart should have a slight upward curvature and should be smooth. The points in the right chart should approximate a straight line. Both sets of points are satisfactory.

Find: j and m

Linear form: $\text{Ln } P_f = \text{Ln } j + m \text{ Ln } Q$

Let: $X_1 = \text{Ln } P_f$ $\quad X_2 = \text{Ln } Q$ $\quad a = \text{Ln } j$

STATISTICAL COMPUTATIONS

---DATA----		--------COMPUTED---------				--CHECK---	
Q	P_f	X2	X1	X1X2	X2X2	Y'	%error
569	1085	6.34	6.99	44.34	40.24	1084	0.12
455	738	6.12	6.60	40.42	37.46	776	-5.18
347	505	5.85	6.22	36.41	34.21	518	-2.59
285	444	5.65	6.10	34.46	31.95	386	13.01
228	272	5.43	5.61	30.44	29.48	277	-1.79
171	172	5.14	5.15	26.47	26.44	180	-4.79

n	6.00
sum X1	36.67
sum X2	34.54
sum X1X2	212.53

sum X2X2	199.78
D	5.89
a	-2.48

The values of j and m are computed to be

j	**= 0.084**
m	**= 1.492**

and the equation is

$$P_f = \mathbf{0.084\ Q^{1.492}}$$

4. The optimal circulation rate, Q*, is computed with the equation

$$Q^* = \left\{\frac{1714 * BHHP}{j * m}\right\}^{\frac{1}{m+1}}$$

$$Q^* = \left\{\frac{1714 * 170}{.084 * 1.492}\right\}^{\frac{1}{1.492+1}} \qquad \mathbf{= 359\ gpm}$$

A. If the optimal circulation rate is greater than a pump limitation or other parameter, then

Q* = Maximum chosen circulation rate

B. If the optimal circulation rate is less than a minimum chosen value, then

Q* = Minimum chosen circulation rate

5. The optimal pump pressure, P_p*, is computed with the equation

$$P_p^* = j\,(Q^*)^m + \frac{1714 * BHHP}{Q^*}$$

$$P_p^* = .084 * 359^{1.492} + \frac{1714 * 170}{359} \qquad \mathbf{= 1{,}358\ psi}$$

6. The optimal jet pressure drop is found with the equation

$$P_j^* = \frac{m}{m+1} * P_p^*$$

$$P_j^* = \frac{1.492}{1.492+1} * 1{,}358 \qquad = \mathbf{813\ psi}$$

7. The optimal jet #'s are found by trial & error with the following equation. The computed jet pressure drop must approximate that of the optimal jet pressure drop.

$$P_j = MW * \left\{\frac{12.51 * Q}{J1^2 + J2^2 + J3^2}\right\}^2$$

$$P_j = 10.0 * \left\{\frac{12.51 * 359}{16^2 + 16^2 + 0^2}\right\}^2 \qquad = \mathbf{769\ psi}$$

Because the jet pressure drop of 769 psi compares well with the optimal pressure drop of 813 psi, the two #16 jets are chosen. Two jets were chosen rather than three for the better bottom hole cleaning.

Thus, for optimal drilling, use **two #16 jets** and circulate at **359 gpm**. The pump pressure should be about **1,358 psi**.

Theory for Optimizing of Jet Hydraulic Horsepower with a Selected Bit Hydraulic Horsepower per Square Inch of Hole Bottom

1. The equation, $P_f = j\ Q^m$, satisfactorily models the friction pressure loss in a rig's circulation system.

2. Bit Hydraulic Horsepower is given by the equation

$$BHHP = \frac{P_j\ Q}{1714}$$

3. At the condition of maximum hydraulic horsepower distribution of the mud pumps hydraulic horsepower through the jets of the bit, the two following equations were derived in chapter #2.

$$P_f = \frac{1}{m+1} * P_p$$

$$P_j = \frac{m}{m+1} * P_p$$

$$P_f = \frac{P_j}{m}$$

Substitutions for the variables P_f and P_j into the last equation gives

$$j\,Q^m = \frac{1714\ BHHP}{m\ Q}$$

Solving for Q gives the Q which maximizes bit hydraulic horsepower with the constraint of a selected BHHP. This Q is called optimal Q*.

$$Q^* = \left\{\frac{1714\ BHHP}{j\ m}\right\}^{\frac{1}{m+1}}$$

The optimal pump pressure is the sum of the optimal friction pressure losses and the optimal jet pressure drop.

$$P_p^* = P_f^* + P_j^*$$

The optimal friction pressure losses and optimal jet pressure drop is

$$P_f^* = j\,(Q^*)^m$$

$$P_j^* = \frac{1714\ BHHP}{m\ Q^*}$$

Substitution for P_f^* and P_j^* gives the optimal pump pressure

$$P_p^* = j\,(Q^*)^m + \frac{1714\ BHHP}{m\ Q^*}$$

The optimal jet numbers are ascertained by any trial and error procedure with the equation

$$P_j^* = MW * \left\{\frac{12.51 * Q^*}{J1^2 + J2^2 + J3^2}\right\}^2$$

EXAMPLE

Optimization with a Chosen Impact Force per Square Inch of Bottom Hole Area

Optimize the hydraulics while drilling with a 8.5" drill bit with 11.0 ppg mud. Jets in the bit are three #14. Choose an IF/sq. in. value of 9.0. IF is then

$$IF = \frac{Q\,(MW\ P_j)^{12}}{57.66} \qquad IF = 9 * .7854 * 8.5^2 \qquad \mathbf{= 511\ lbs}$$

1. Complete steps one through three as in the previous example.

4. The optimal circulation rate, Q*, is computed with the equation

$$Q^* = \left\{\frac{2 * 57.66^2\ \mathrm{IF}^2}{j * m * \mathrm{MW}}\right\}^{\frac{1}{m+2}}$$

$$Q^* = \left\{\frac{2 * 57.66^2 * 511^2}{.084 * 1.492 * 10}\right\}^{\frac{1}{1.492+2}} \qquad \mathbf{= 415\ gpm}$$

A. If the optimal circulation rate is greater than a pump limitation or other parameter, then

Q* = Maximum chosen circulation rate

B. If the optimal circulation rate is less than a minimum chosen value, then

Q* = Minimum chosen circulation rate

5. The optimal pump pressure, P_p*, is computed with the equation

$$P_f^* = j\,(Q^*)^m$$

$$P_j^* = \frac{1}{\mathrm{MW}} \left\{\frac{57.66\ \mathrm{IF}}{Q^*}\right\}^2$$

$$P_p^* = j\,(Q^*)^m + \frac{1}{\mathrm{MW}} \left\{\frac{57.66\ \mathrm{IF}}{Q^*}\right\}^2$$

$$P_p^* = .084 * 415^{1.492} + \frac{1}{10.0} \left\{\frac{57.66 * 511}{415}\right\}^2 \qquad \mathbf{= 1{,}181\ psi}$$

6. The optimal jet pressure drop is found with the equation

$$P_j^* = \frac{m}{m+2} * P_p^*$$

$$P_j^* = \frac{1.492}{1.492+2} * 1{,}181 \qquad \mathbf{= 505\ psi}$$

7. The optimal jet numbers are found by trial & error with the following equation. The computed jet pressure drop must approximate that of the optimal jet pressure drop.

$$P_j^* = MW * \left\{\frac{12.51 * Q}{J1^2 + J2^2 + J3^2}\right\}^2$$

$$P_j^* = 10.0 * \left\{\frac{12.51 * 415}{18^2 + 20^2 + 0^2}\right\}^2 \qquad \mathbf{= 514\ psi}$$

Because the jet pressure drop of 514 psi compares well with the optimal pressure drop of 505 psi, one #18 and one #20 jets are chosen. Two jets were chosen rather than three for better bottom hole cleaning.

Thus, for optimal drilling, use **one #18** and **one #20 jets** and circulate at **415 gpm**. The pump pressure should be about **1,181 psi**.

EXAMPLE

Optimization with a Chosen Pump Operating Pressure

Optimize the hydraulics while drilling with a 8.5" drill bit with 11.0 ppg mud. Jets in the bit are three #14. Choose a pump operating pressure of 3,000 psi.

Steps

1. Complete steps one through three as in the previous example.
2. Compute the optimal friction pressure loss with the equation

$$P_f^* = \frac{1}{m+1} * P_p^*$$

$$P_f^* = \frac{1}{1.492+1} * 3000 \qquad \mathbf{= 1{,}204\ psi}$$

3. Compute the optimal Q* with the equation.

$$Q^* = \left\{\frac{P_f^*}{j}\right\}^{\frac{1}{m}} \qquad Q^* = \left\{\frac{1204}{.084}\right\}^{\frac{1}{1.492}} \qquad \mathbf{= 611\ gpm}$$

A. If the optimal circulation rate is greater than a pump limitation or other parameter, then

Q* = Maximum chosen circulation rate

B. If the optimal circulation rate is less than a minimum chosen value, then

Q* = Minimum chosen circulation rate

4. Compute the optimal jet pressure drop with the equation

$$P_j^* = \frac{m}{m+1} * P_p^* \qquad P_j^* = \frac{1.492}{1.492+1} * 3000 \qquad = \mathbf{1,796\ psi}$$

5. Compute the optimal jet numbers with the following equation. A trial and error procedure is required.

$$P_j^* = MW * \left\{\frac{12.51 * Q^*}{J1^2 + J2^2 + J3^2}\right\}^2$$

$$1,796 = 10.0 * \left\{\frac{12.51 * 611}{16^2 + 18^2 + 0^2}\right\}^2$$

1,796 ≈ 1,737

One #16 and one #18 jets are the optimal jets.

Thus, for optimal drilling, use **one #16** and **one #18 jets** and circulate at **611 gpm**. The pump pressure should be about **3,000 psi**.

Theory for Maximizing Impact Force with a Selected Pump Pressure

1. The equation, $P_f = j\,Q^m$, satisfactorily models the friction pressure loss in a rig's circulation system.

2. Bit hydraulic impact force is given by the equation

$$IF = \frac{Q\,(MW\,P_j)^{1/2}}{57.66}$$

3. At the condition of maximum impact force distribution of the mud pumps hydraulic impact force through the jets of the bit, the two following equations were derived in chapter #2.

$$P_f = \frac{2}{m+2} * P_p$$

$$P_j = \frac{m}{m+2} * P_p$$

Substitutions for the variables P_f and P_j into the last equation gives

$$j\,Q^m = \frac{2}{m+2} * P_p$$

$$Q^m = \frac{2}{j(m+2)} P_p$$

Solving for Q gives the Q which maximizes the bit impact force with the constraint of a selected pump pressure. This Q is called optimal Q*.

$$Q^* = \left[\frac{2}{j(m+2)} P_p\right]^{1/m}$$

The optimal jet numbers are ascertained by any trial and error procedure with the equation

$$P_j^* = MW * \left\{\frac{12.51 * Q^*}{J1^2 + J2^2 + J3^2}\right\}^2$$

Notes

HOLE CLEANING

A solid particle in the hole is acted on by four factors 1) gravity, 2) viscous drag, 3) impact and 4) buoyancy. Stoke's law and Newton's law governs particles falling in Newtonian fluids with annular-to-particle ratios 100 or more and particle concentrations below 0.1 % by volume. None of these hold for drilling.

VERTICAL HOLES

Sifferman published transport ratios collected with an annular flow model made with a twelve inches outer steel tube and various diameter inner tubes. The model was about 100 feet long. He defined transport ratio with the equation. His transport ratio is the seen to be the solid's velocity expressed as a fraction of the annular velocity.

$$R_t = \frac{V_f - V_s}{V_f}$$

R_t	=	Transport ratio
V_f	=	Mud annular velocity; fpm
V_s	=	Solids free settling (slip) velocity; fpm

SIFFERMAN MUD TYPES

TYPE	F_{600}	F_{300}	F_{200}	F_{100}	F_6	F_3	G_i	G_{10}	TOUCH
THICK	69	53	45	36	23	20	13	29	gooey
INTER-MEDIATE	49	35	30	25	15	13	13	22	slick
THIN	24	16	13	10	3	3	2	3	slick
WATER	2	1					0	0	not slick

His table depicting the degree to which various factors affect transport ratios is the following.

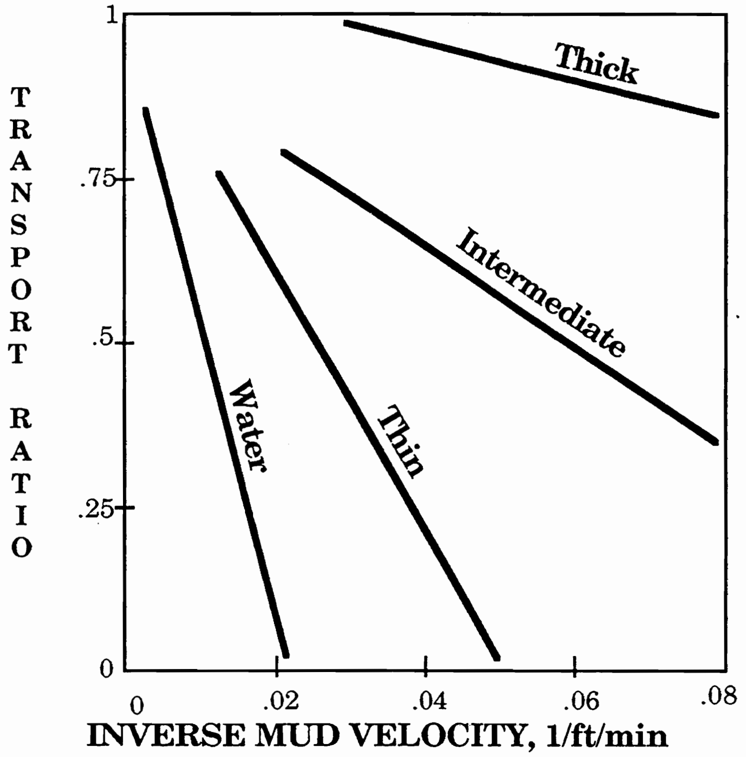
CUTTINGS TRANSPORT VARIABLES			
VARIABLE	MAJOR	MODERATE	MINOR
Annular Velocity	X		
Rheological Mud Properties	X (vertical)	X (inclined)	
Cutting Size		X	
Mud Weight		X	
Rotary Speed		X (orbiting)	X (spinning)
Annulus Size			X
Eccentricity	X (horizontal)		X (vertical)
Drilling Rate			X

Sifferman identified three thicknesses of mud for the publication of his transport ratios: thick, intermediate, and thin; and water has been added.

The Fann dial readings and gels of his muds are given in the table on the previous page.

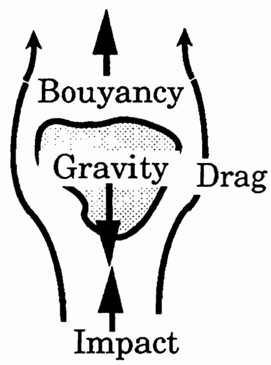
His chart as modified by Sample gives transport ratios versus inverse mud annular velocity for his mud types.

SIFFERMAN'S TRANSPORT RATIOS



DRILL CUTTINGS CONCENTRATION IN THE ANNULUS

API



API used the following steady state equation for volumetric drill cuttings concentration in the annulus.

$$V_{fc} = \frac{DR * B^2}{1471\, Q\, R_t}$$

V_{fc}	= volumetric drill cuttings concentration in the annulus
DR	= steady-state drilling rate; fph
B	= drill bit diameter; inch
Q	= circulation (pumping) rate; gpm
R_t	= drill cuttings transport ratio

EXAMPLE

Compute V_{fc} if a 8.5" drill bit is drilling at a rate of 50 fph. The mud circulation rate is 108 gpm and the mud consistency is intermediate. The drillpipe is 4".

The annular velocity is

$$AV = \frac{24.5 * Q}{H^2 - OD^2} \qquad = \frac{24.5 * 108}{8.5^2 - 4^2} \qquad \mathbf{= 47\ fpm}$$

The reciprocal of the AV is

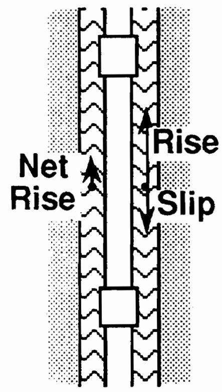
$$\frac{1}{AV} \qquad = \frac{1}{47} \qquad \mathbf{= .0213}$$

From Sifferman's transport ratio chart for intermediate mud, the transport ratio is

$$R_t \qquad \mathbf{= .81}$$

The volumetric drill cuttings concentration in the annulus is

$$V_{fc} = \frac{DR * B^2}{1471\, Q\, R_t} \qquad = \frac{50 * 8.5^2}{1471 * 108 * .81} \qquad \mathbf{= 0.028\ or\ 2.8\%}$$



NEWITT'S EQUATION

Newitt published a more precise annular concentration equation for steady state lifting of solids in a vertical tube.

$$C_a = -\frac{1}{2}\left\{\frac{V_a}{V_s} - 1\right\} + \left[\frac{1}{4}\left\{\frac{V_a}{V_s} - 1\right\}^2 + \frac{V_a}{V_s} C\right]^{\frac{1}{2}}$$

and

$$V_f = V_a \left\{1 + C_a \frac{V_s}{V_a}\right\} \qquad V_a = \frac{24.5\, Q}{H^2 - D^2} \qquad C = \frac{V_c}{V_m} \qquad V_c = \frac{DR}{60} * \frac{0.7854\, B^2}{144}$$

C_a	=	drill cuttings volumetric concentration in the annulus
V_a	=	average bulk velocity of fluid entering without drill cuttings; fpm
V_s	=	slip velocity of drill cuttings; fpm
V_f	=	adjusted annular velocity for drill cuttings effect; fpm
V_c	=	volume rate of cuttings entering the annulus; cfpm
V_m	=	mud entry rate into annulus; cfpm

EXAMPLE

Compute V_{fc} with Newitt's equation if a 8.5" drill bit is drilling at a rate of 50 fph. The mud circulation rate is 108 gpm and the mud consistency is intermediate. The drillpipe is 4".

The drill cuttings entry rate into the annulus is

$$V_a = \frac{24.5 * 108}{8.5^2 - 4^2} \qquad \mathbf{= 47\ fpm}$$

$$V_c = \frac{DR}{60}\frac{.7854\,B^2}{144} \qquad = \frac{50}{60}\frac{.7854 * 8.5^2}{144} \qquad \mathbf{= .328\ cfpm}$$

The mud entry rate into the annulus is

$$V_m = \frac{Q}{7.48} \qquad = \frac{108}{7.48} \qquad \mathbf{= 14.44\ cfpm}$$

The value of C in Newitt's equation is

$$C = \frac{V_c}{V_m} \qquad = \frac{.328}{14.44} \qquad \mathbf{= .0227}$$

The slip velocity of the drill cuttings from Sifferman's chart and the equation

$$V_t = V_f(1 - R_t) \qquad = 47\,(1 - .81) \qquad \mathbf{= 8.9\ fpm}$$

The drill cuttings concentration in the annulus ($C_a = V_{fc}$) is

$$C_a = -\frac{1}{2}\left\{\frac{V_a}{V_t} - 1\right\} + \left[\frac{1}{4}\left\{\frac{V_a}{V_t} - 1\right\}^2 + \frac{V_a}{V_t}C\right]^{\frac{1}{2}}$$

$$C_a = -\frac{1}{2}\left\{\frac{47}{8.9} - 1\right\} + \left[\frac{1}{4}\left\{\frac{47}{8.9} - 1\right\}^2 + \frac{47}{8.9}\,.0227\right]^{\frac{1}{2}} \qquad \mathbf{= .0278}$$

The adjusted annular velocity for the drill cuttings effect is

$$V_f = V_a\left\{1 + C_a\frac{V_t}{V_f}\right\} \qquad = 47\left\{1 + .0278\frac{8.9}{47}\right\} \qquad \mathbf{= 47.25\ fpm}$$

MITCHELL'S EQUATION

B. J. Mitchell published an annular concentration equation which accounts for the cessation of circulation during connections and the circulation which occurs prior to a connection but after drilling has ceased. This latter circulation is called pre-connection circulation time. The volume fraction of cuttings computed with his equation is the average in the annulus.

$$\frac{1}{V_{fc}} = 1 + \left[1 - \frac{D^2}{H^2}\right] * \left[\frac{AV - V_s}{L_j}\right] * \left[\frac{60 * L_j}{DR} + T_{pc}\right]$$

V_{fc} = average new drill cuttings concentration; vol%
D = OD of drillpipe; inch
H = bit diameter; inch
A V = average annular mud velocity; fpm
V_s = average drill cuttings slip velocity; fpm
L_j = length of one drillpipe (31'); ft
DR = average drilling rate; fph
T_{pc} = pre-connection circulation time; minute)

The required pre-connection circulation time for one connection is the time necessary to circulate the cuttings to a height which will prevent their settling to the bottom of the hole during that connection.

$$T_{pc} = \frac{V_s}{V_f - V_s} * T_c$$

T_c = time for one connection; minute

The increase in the average mud weight is

$$\Delta MW = V_{fc} * [\, 8.33 * W_s - MW]$$

W_s = specific gravity of the drill cuttings; sp. gr.

HOPKIN'S PARTICLE SLIP VELOCITY CHART

An example problem follows Hopkin's slip velocity chart.

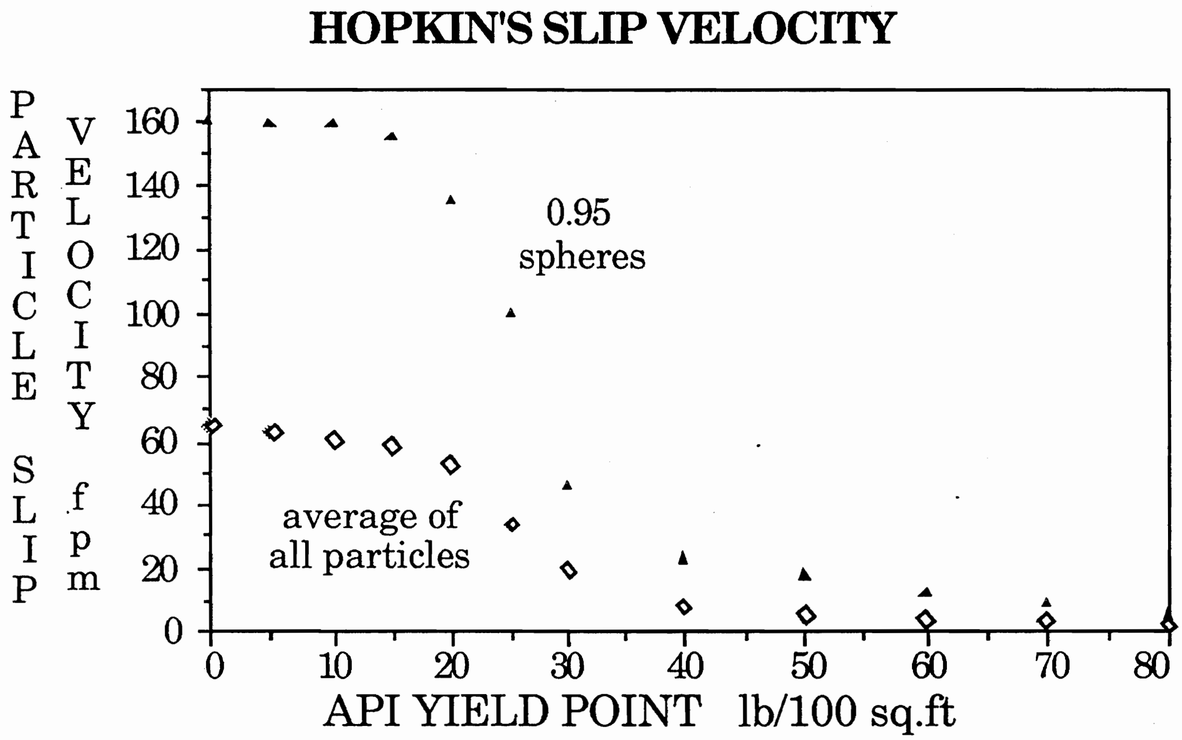
Hopkin ran over 2,000 dynamic particle lifting tests with a 8 foot long 4.5" diameter vertical tube with 13 types of mud and 52 particle shapes and sizes. The particles were circulated from the bottom of the tube to the top with the fluid in laminar, transitional, and turbulent flow. Mud Marsh funnel viscosities ranged from 26 to 1,000 seconds/qt. A minimum of three data points were collected for each test.

A curve fit of his graph (not shown) which presents the effect of mud weight on slip velocity gave

$$F_{mw} = 2.117 - 0.1648 * MW + 0.003681 * MW^2$$

Thus, the adjusted slip velocity is (V_{sc} is taken from Hopkin's figure)

$$V_s = F_{mw} * V_{sc}$$



EXAMPLE

Compute V_{fc} with Mitchell's equation if a 17.5" drill bit is drilling at a rate of 100 fph. The mud circulation rate is 1,148 gpm and the API yield point of the mud is 30 #/100sq.ft. The drillpipe is 5" and the mud weight is 9.3 ppg. Connection time is 3 minutes. The bottom hole assembly is 9.5" in diameter. Spheres of a diameter of 0.95" are to lifted.

The slip velocity read from Hopkin's chart is

V_{sc} **= 46 fpm**

The correction factor for the mud weight is

$F_{mw} = 2.117 - 0.1648 * 9.3 + 0.003681 * 9.3^2$ **= 0.903**

The adjusted slip velocity is

$$V_s = 0.903 * 46 \qquad \mathbf{= 41.5\ fpm}$$

The required pre-connection circulation time is

$$T_{pc} = \frac{41.5}{100 - 41.5} * 3 \qquad \mathbf{= 2.1\ minute}$$

The average volume fraction of new drill cuttings in the annulus is

$$\frac{1}{V_{fc}} = 1 + [1 - \frac{5^2}{17.5^2}][\frac{100 - 41.5}{31}][\frac{60 * 31}{100} + 2.1]$$

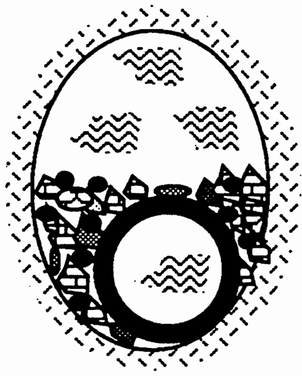
$$V_{fc} \qquad \mathbf{= 0.027\ vol\%}$$

The increase in the mud weight in the annulus from the addition of the drill cuttings and the final average mud weight in the annuluss are

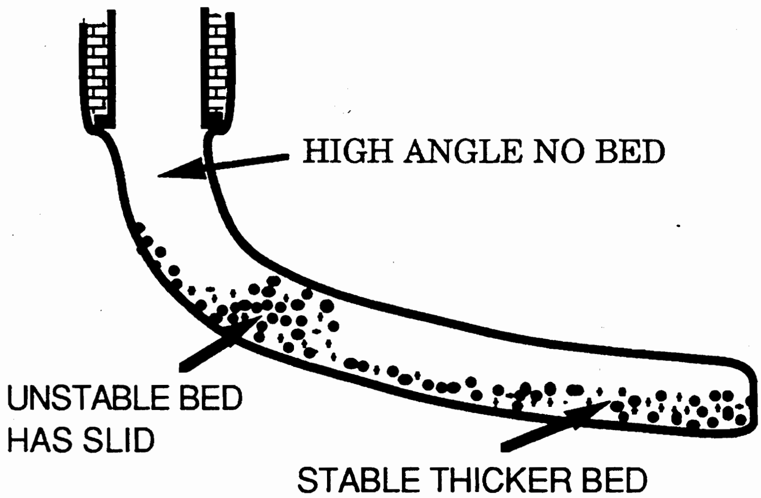
$$\Delta MW = 0.028 * [8.33 * 2.75 - 9.3] \qquad \mathbf{= 0.38\ ppg}$$

$$MW_{new} = 9.3 + 0.38 \qquad \mathbf{= 9.7\ ppg}$$

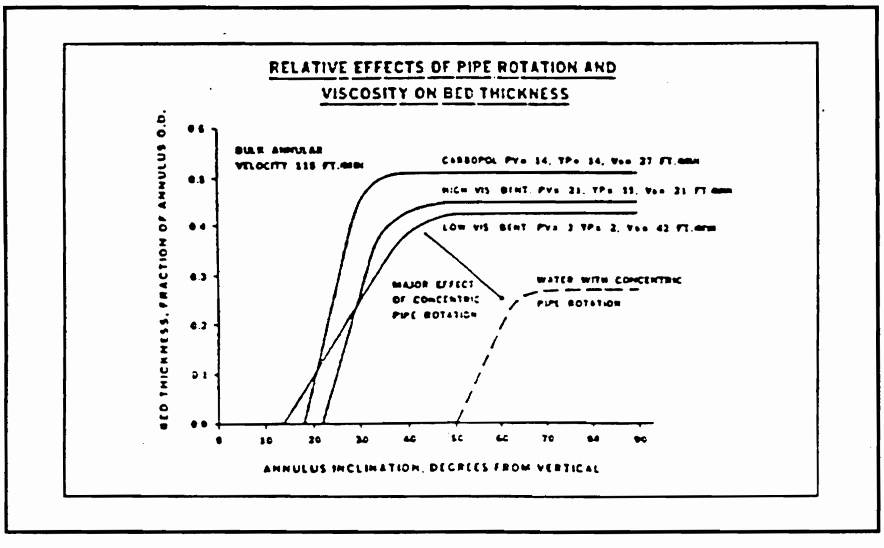
INCLINED HOLE CLEANING (DIRECTIONAL)



Cleaning of inclined and horizontal holes presents two major concerns which are not present in vertical hole cleaning. One concern is the existence and thickness of a bed of cuttings on the low side of the hole. The other is the sliding of a bed of cuttings down hole. In regard to sliding, a bed is said to be stable if it does not have a tendency to slide and unstable if it does. Having a cuttings bed in the hole while drilling places the tripping of the pipe in jeopardy. Having an unstable bed places the drill string in jeopardy and especially so any time mud circulation is halted. The thicker beds are more likely to form in more deviated holes and unstable beds are more likely in the less deviated sections of holes. The sketch shows a cross-section of a cuttings bed and stable and unstable hole sections.



Most cuttings bed problems occur with an oil base mud and normally pressured zones. It is thought that water base muds aid in the disintegration of the cuttings and there subsequent removal. Further, cuttings from over-pressured zones are more buoyant (cuttings less dense and the mud is more dense) and have less cohesiveness (internal strength) than cuttings from normally pressured zones and therefore are less likely to form as thick a bed if any at all.



University of Tulsa published the chart which shows that the viscosity of the circulating fluid has little influence on bed thickness; however, circulation rate appears important.

If an oil base mud is being circulated cuttings beds cannot be removed by mud circulation alone and this could be a major problem in cementations and sticking pipe and drillstrings.

REMOVAL OF CUTTINGS BED

It has been found that thinner cuttings beds are formed and maintained by

1. mud in turbulent flow. This statement is based on laboratory data which show that the mud flow within the area of the hole above the cuttings bed will have a powerlaw Reynold's number value of 1,900 or more. In other words the mud will erode a bed until the flow turns to laminar flow above the bed.
2. circulating slugs of 50 to 100 barrels in size of lower viscosity than the base mud. This finding is from field observations.
3. circulating slugs of 50 to 100 barrels in size of higher density than the base mud. This finding is from field observations.
4. circulating slugs of water base mud equal in viscosity and density to the oil mud. This finding is from field observations.
5. reciprocating the drill string. This finding is from field observations.
6. rotating the drill string and especially so if it is eccentric. This finding is supported by both laboratory and field testing.
7. making short trips. This finding is from field observations.
8. circulating muds which have a flat velocity profile at the circulating rate. This finding is supported by both laboratory and field testing.

TRANSPORT AND STABILITY CHART FOR INCLINED HOLE CLEANING

B. Tarr proposed that a solids transport and bed stability chart can be constructed by combining slurry pipeline and vertical hole particle transport technologies. His equation for full transport is

$$V_{ft} = (V_1 * \cos f) + (V_2 * \sin f)$$

V_{ft}	=	full transport annular velocity; fpm
f	=	hole inclination angle; degree
V_1	=	particle slip velocity in a vertical hole (drilling data); fpm
V_2	=	critical transport velocity for large solids in a horizontal annulus (slurry pipe line data); fpm

The long standing critical velocity equation (often referred at Newton's transport law) for the transport of large solids by a Newtonian liquid within a horizontal annulus is

$$V_2 = C_1 * \left[\left(\frac{SW - MW}{MW} \right) * g^3 * \left(\frac{H - D}{12}\right)^3 \right]^{1/6}$$

C_1	=	44 The value of 44 fits University of Tulsa data best; however, UT data never had full transport
C_1	=	60 Full transport
SW	=	specific weight of the solids (22 ppg); ppg
MW	=	mud weight; ppg
g	=	acceleration of gravity (32.2 ft/sec^2); ft/sec^2
H	=	hole diameter; inch
D	=	OD of inner tube; inch

EXAMPLE

Compute the annular velocity necessary to have full transport in a hole which has an inclination of 60 degrees and a 9.3 ppg which has an API yield point of 30 lb/100sq.ft. The diameter of the drill bit is 8.5 inches and the drillpipe is 4.5 inches.

From a previous problem which used Hopkin's chart and equation the slip velocity of a 0.95 sphere was 41.5 fpm.

V_1 **= 41.5 fpm**

The U of Tulsa data and the horizontal transport equation gives

$$V_2 = 44 * \left[\left(\frac{22 - 9.3}{9.3} \right) * 32.2^3 * \left(\frac{8.5 - 4.5}{12} \right)^3 \right]^{1/6}$$ **= 131 fpm**

The full transport annular velocity and circulation rate for this 60 degree inclined hole is

$$V_{ft} = (41.5 * \cos(60)) + (131 * \sin(60))$$ **= 152 fpm**

$$Q = \frac{(H^2 - OD^2) * V}{24.5}$$ **= 323 gpm**

Assuming that the transitional flow regime is required for full transport; ie, Nr = 2000

$$2000 = 928 \frac{9.3 * (8.5 - 4.5) * V_{ft}}{40}$$

V_{ft} **= 139 fpm**

Using the data presented in the example problem and the full transport equation, a chart is constructed which shows the required annular velocities versus hole inclination.

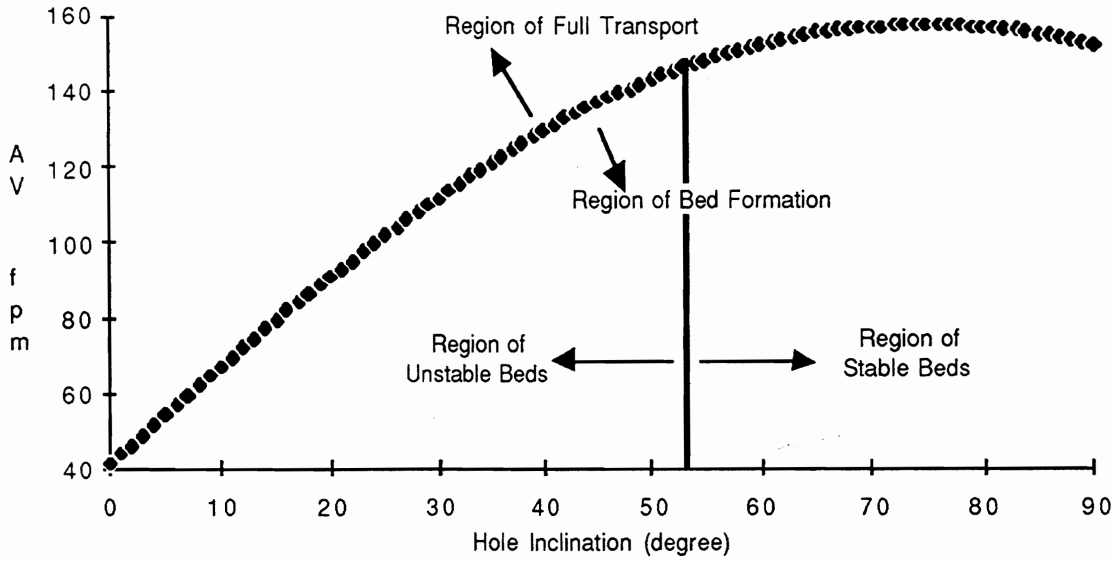
The region above the curved line is the region where full transport of cuttings will occur and a cuttings bed will not form. Below the line a cuttings bed will form. Also shown is the angle at which stable and unstable beds will exist.

Suppose the hole inclination is 45 degrees and the annular velocity is 120 fpm, then an unstable bed will form and could slide down the hole.

However, if the annular velocity is increased to 160 fpm, then the bed will be lifted up the hole and dissipated.

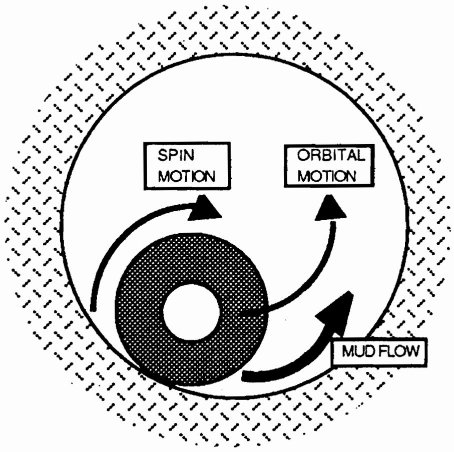
If the annular velocity is 120 fpm and the hole inclination is 70 degrees, then a stable bed which will not slide down the hole will be formed.

Full Transport Annular Velocities



WHIRLING MOTION OF THE DRILL STRING

The rule of not permitting turbulent flow in the open hole annulus may be revoked by considerations surrounding an orbiting drill string. The figure is a view of a cross section of a drill string while looking into the hole. The mud will be displaced by the orbiting drill string in the direction of the arrow captioned with "MUD FLOW". The direction of the flow of the mud is horizontal. However, while this motion is coupled with vertical annular flow, mud will flow in a spiral motion. With no slip of the drill string at the wall of the hole, the whirl rpm of the rotating drillstring is



$$RPM_{whirl} = \frac{OD}{H - OD} * RPM_{spin} \quad \text{(rpm)}$$

and its tangential velocity is

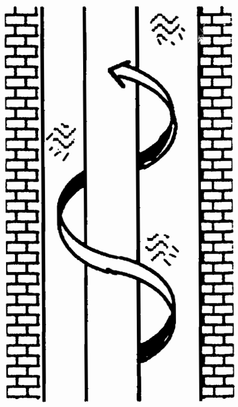
$$V_{tangential} = RPM_{whirl} * \Pi * \frac{OD}{12} \quad \text{(f/m)}$$

OD = OD of the pipe; in   H = diameter of hole; in

This is also the velocity at which the mud will be squeezed out of the crotch of the drill string and the wall. The spiral velocity of the mud is then, the vector sum of these to velocities. It must be noted that common turbulence computations ignore the tangential component of the mud's velocity. This velocity may be significant.

$$V_{spiral} = \sqrt{AV^2 + V_{tangential}{}^2} \quad \text{(f/m)}$$

EXAMPLE



Let 8" collars be rotating at 120 rpm in a 12.25" hole and the vertical annular velocity of the mud be 110 fpm. Compute the average spiral velocity near the wall of the hole.

The whirl rpm is

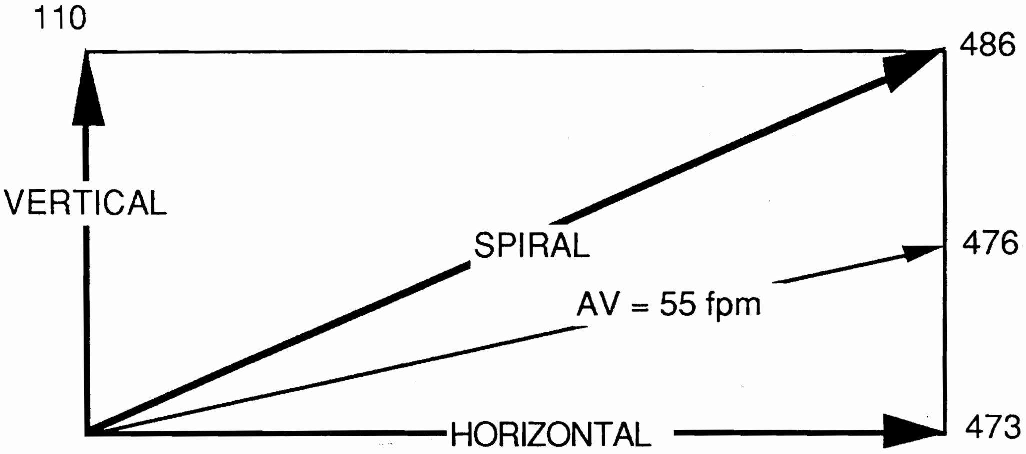
$$RPM_{whirl} = \frac{8}{12.25 - 8} \qquad \mathbf{= 226\ rpm}$$

The tangential velocity near the wall is

$$V_{tang} = 226 * \Pi * 8/12 \qquad \mathbf{= 473\ fpm}$$

The spiral velocity is

$$V_{spiral} = \sqrt{110^2 + 473^2} \qquad \mathbf{= 486\ fpm}$$



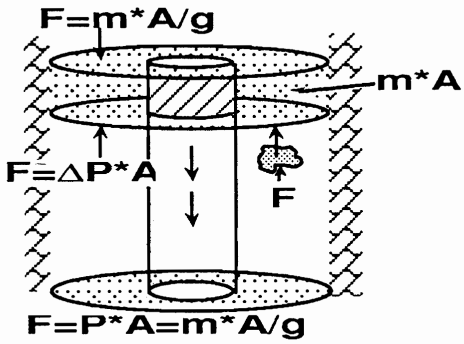
The figure shows the relationships; however, note that if the vertical annular velocity is cut in one-half to 55 fpm the spiral velocity will remain high at

$$V_{spiral} = \sqrt{55^2 + 473^2} \qquad \mathbf{= 476\ fpm}$$

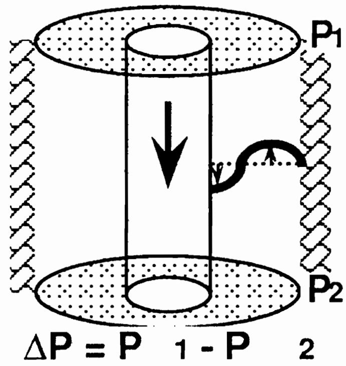
SURGE AND SWAB FOR LONG PIPE STRINGS

Pressure surges within the borehole may be caused by one or more of the listed operations:

1. Friction between the mud and the pipe because of pipe movement.

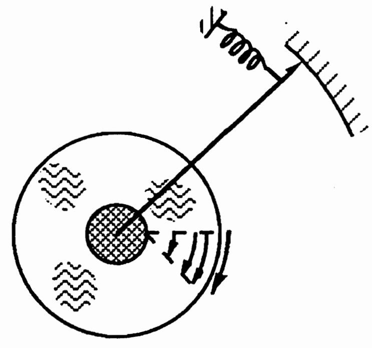


2. Inertial force because of accelerating or decelerating the mud in the hole when initiating or terminating pipe movement or circulation.



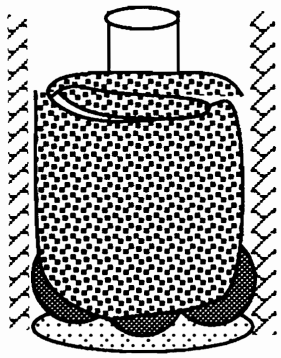
3. Breaking of the thixotropic gel of some muds when initiating or terminating pipe movement or circulation.

4. Balling of the bit or a stabilizer when pulling the drillstring.



GEL

BIT BALLING



If the mud and rig hydraulics are adequate items, 3 and 4 will be reduced to minimal importance. Contrary to popular beliefs, balling can occur and can swab the borehole an appreciable amount if a permeable zone is below the bit and if the mud pressure at the zone only slightly exceeds its pore pressure.

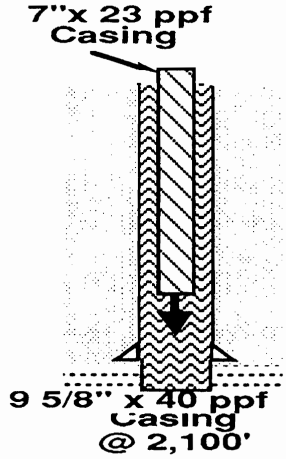
Exxon's bottom hole pressure chart (following page) indicates the relative magnitudes of friction, inertial, and gel pressure surges and swabs in a typical hole.

APPARATUS, CASING AND HOLE

The wellbore was 9 5/8 inches, 40 ppf casing set at a depth of 2,100 feet. The casing for which pressure measurements were recorded while being run was 7 inches, 23 ppf. It was equipped with a differential fillup collar. Five subsurface pressure gauges recorded the running pressures. Two were at the bottom of the 9

5/8 inches casing, two were at the bottom of the 7 inches casing, and one was within the 7 inches casing mounted above the fillup collar. It was reported that most often the gauges were within 10% of one another.

The velocity of the casing was measured with three devices: a telephoto camera photographing one foot marks on the casing, a rig runner pointer and a second camera, and a DC voltage generator driven by the rig runner pointer line.



The mud properties and casing depths during the pressure measurements shown in the figure were

MUD PROPERTIES

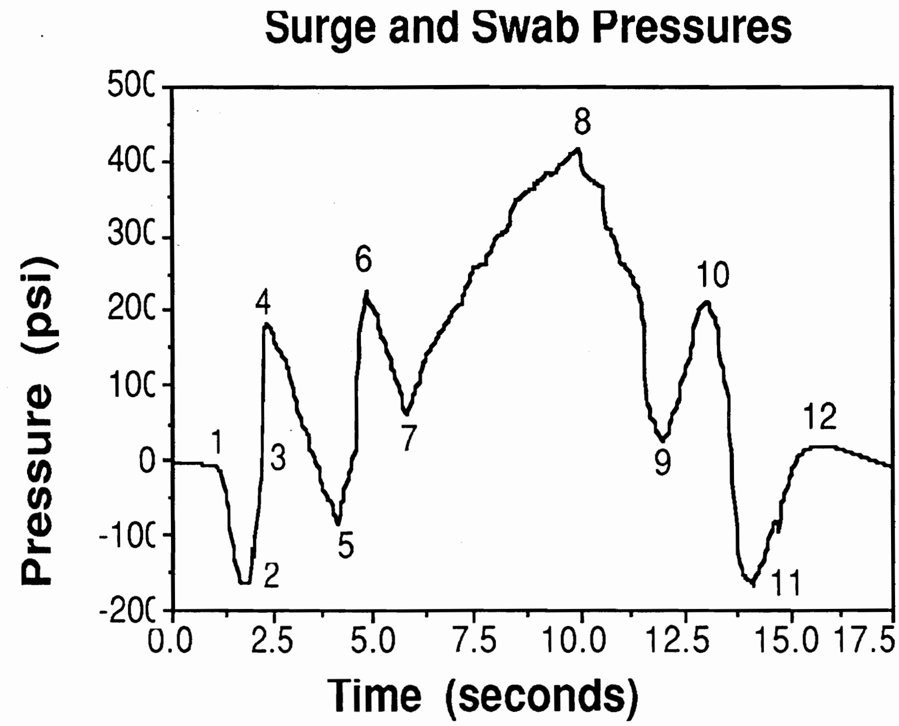
PV	= 12 cp
YP	= 7 lb/100sqft
Density	= 10.9 ppg
Funnel Viscosity	= 41 seconds/qt
Filtration Rate	= 12.5 cc/30 min API
Gel Strengths	
Initial	= 9 lb/100 sqft
3 Minute	= 32 lb/100 sqft

CASING

The 7 inch casing was lowered from a depth of 1,812 to 1,856 feet.

ANALYSIS OF THE EXXON FIGURES

It is important to point out that swabbing can occur while running pipe into the hole as shown at points #5 and #11 in the figure. The maximum swab pressure occurred while lifting the casing to release the slips. Thixotropic gel and the inertial force of the mud played major roles. Points #5, #7, #9, and #11 occurred



while breaking the casing non-uniformly and inertial forces played the major role in these swabs. The magnitude of pressures at points #4, #6, #8, #10 were primarily developed by the velocity of the casing and the attendant friction drag.

The time for running the joint of casing which was 44 feet long was 17.5 seconds. This is equivalent to an average running speed of 151 feet/minute. The equivalent pressure surge at point #8 was 4.4 ppg and the equivalent swab at point #11 was 1.8 ppg.

PRESSURE SURGE COMPUTATIONS

Pressure surge computations call for the solving of two distinct equations for a laminar and a turbulent pressure surge; thereafter, the larger of the two is the applicable surge pressure.

In practice it is convenient and practical to request that the driller time the number of seconds for the lowering of a joint of casing or the middle joint in a stand of drill pipe.

VISCOUS DRAG PRESSURE SURGE EQUATIONS

The pressure surge equations are those published by J. Burkhardt

LAMINAR MUD FLOW IN THE ANNULUS

$$P_L = G * B * PV * VP + 3.3 \frac{YP}{E}$$

TURBULENT MUD FLOW IN THE ANNULUS

$$P_T = F * A * PV^{.21} * MW^{.81} * VP^{1.8}$$

G & F	=	1.0 (if pipe is closed at the bottom)
G & F	=	values from charts if the pipe is open or partially open at the bottom.
P_L	=	laminar pressure surge per 1000 feet of pipe; psi/1000 ft
G & B	=	chart coefficients no units
PV	=	plastic viscosity of the mud; cps
VP	=	lowering or raising velocity of the pipe; feet per minute
YP	=	yield point of the mud; lbf/100 sq. ft
H	=	diameter of the hole (bit diameter); inches
E	=	effective diameter taken from the chart; inches
D	=	outside diameter of the pipe; inches
F & A	=	chart coefficients; no units
P_T	=	turbulent pressure surge per 1000 feet of pipe; psi/1000 ft

The seconds to lower one joint of casing which is L feet long is

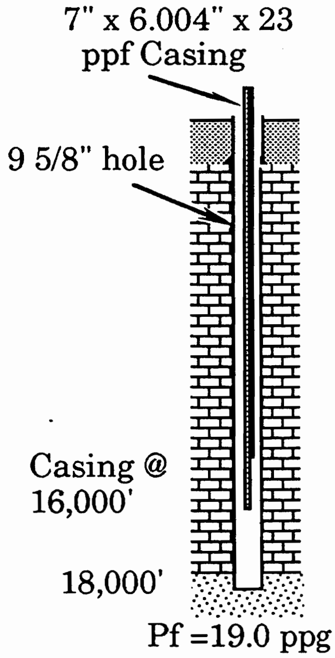
$$T = 2 + \frac{L * 60}{VP} + 2 \qquad L = \text{length of one joint; ft}$$

The first and last "2" in the above equation are the times required for the acceleration and deceleration of the casing.

The following examples illustrate the use of the equations.

EXAMPLE

Casing is being run at a depth of 16,000 feet. Given the data below, how many seconds are required to lower one joint of casing which is 40 feet long?



Hole: drill bit size is 9 5/8 inch
fracture strength = 19 ppg at 18,000 ft and is lowest strength in the hole

Casing: 7", 6.004", 32 ppf
partially open at bottom
Near 16,000 ft the first 10 ft of a joint was lowered in 3 seconds

Mud:		
	PV	= 42 cps
	YP	= 11 lbf/100 sqft
	3 min Gel	= 9 lb/100 sqft
	MW	= 18.5 ppg

SOLUTION:

The chart values for the solution of this problem are

F = 0.26 A = 0.00029 E = 2.15 G = 0.385 B = 0.0055

The maximum permissible surge pressure per 1,000 feet of casing is

$$P_s = .052 * (19\text{-}18.5) * 18{,}000/16 = 416/16 \qquad \mathbf{= 29\ psi/1000\ ft}$$

The laminar surge pressure per 1,000 feet of casing is

$$P_L = G * B * PV * VP + 3.3 \frac{YP}{E}$$

$$29 = 0.385 * 0.0055 * 42 * VP + 3.3 \frac{11}{2.15}$$

VP **= 139 feet/minute (laminar)**

The turbulent surge pressure per 1,000 feet of casing is

$$P_T = F * A * PV^{.21} * MW^{.81} * VP^{1.8}$$

$$29 = 0.26 * 0.00029 * 42^{.21} * 18.5^{.81} * VP^{1.8}$$

VP **= 221 feet/minute (turbulent)**

The smaller of the two values must be chosen. Thus the maximum permissible lowering velocity of the casing is **139 ft/min,**

The seconds to lower one joint of casing which is L feet long is

$$T = 2 + \frac{L * 60}{VP} + 2$$

$$T = 2 + \frac{40 * 60}{139} + 2$$ **= 21 seconds**

THIXOTROPIC GEL PRESSURE SURGE

Many oilwell drilling muds show a pronounced thixotropic property which may place large pressure surges in the wellbore when initializing mud circulation or the initial movement of pipe. The identical equation applies in both cases.

The pressure force at the bottom end of a pipe required to start the movement of the mud in the annulus must be equal to the force required to break the gel strength of the mud in the annulus.

$$P_G = \frac{G * L}{300\,(H - D)}$$

P_G	=	pressure surge required to break gel of the mud; psi
L	=	length of pipe containing mud; ft
G	=	gel strength of mud; lbf/100 sqft
H, D	=	diameter of hole, OD of pipe; inches

EXAMPLE

10,000 feet of 5" by 4.276" drill pipe is run into a 9" drill hole. The 3 minute gel strength of the mud is 38 lbf/100 sq. ft. What is the expected value of the pressure surge if the pipe is to be moved or circulation commenced?

$$P_G = \frac{G * L}{300\,(H - D)} = \frac{38 * 10000}{300\,(9 - 5)}$$ **= 317 psi**

INERTIAL PRESSURE SURGE

While pipe is being lowered into a hole containing mud, the mud must move up the annulus of the hole. The velocity at which the mud moves up the hole depends only on the velocity of the pipe. If the pipe is accelerated then the mud is accelerated and a pressure force at the bottom end of the pipe is required to accelerate the mud.

$$P_A = \frac{MW * A_{mud} * L}{620}$$

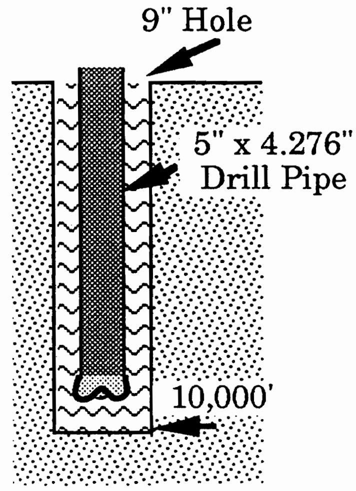
$$A_{mud} = A_{pipe} * \frac{D^2}{H^2 - D^2}$$

$$A_{pipe} = \frac{2\, L_A}{t^2}$$

P_A	=	pressure surge caused by acceleration of pipe; psi
L_A	=	length of pipe during its acceleration; ft
L	=	length of pipe; feet
W	=	mud weight; ppg
t	=	time to accelerate the pipe; seconds
g_c	=	gravitational constant; 32.2 (lbm/lbf) (ft/sec^2)
H, D	=	diameters of hole, OD of pipe, ID of pipe; inches
A_{pipe}	=	acceleration of the pipe; ft/sec
A_{mud}	=	acceleration of the mud in the annulus; ft/sec

EXAMPLE

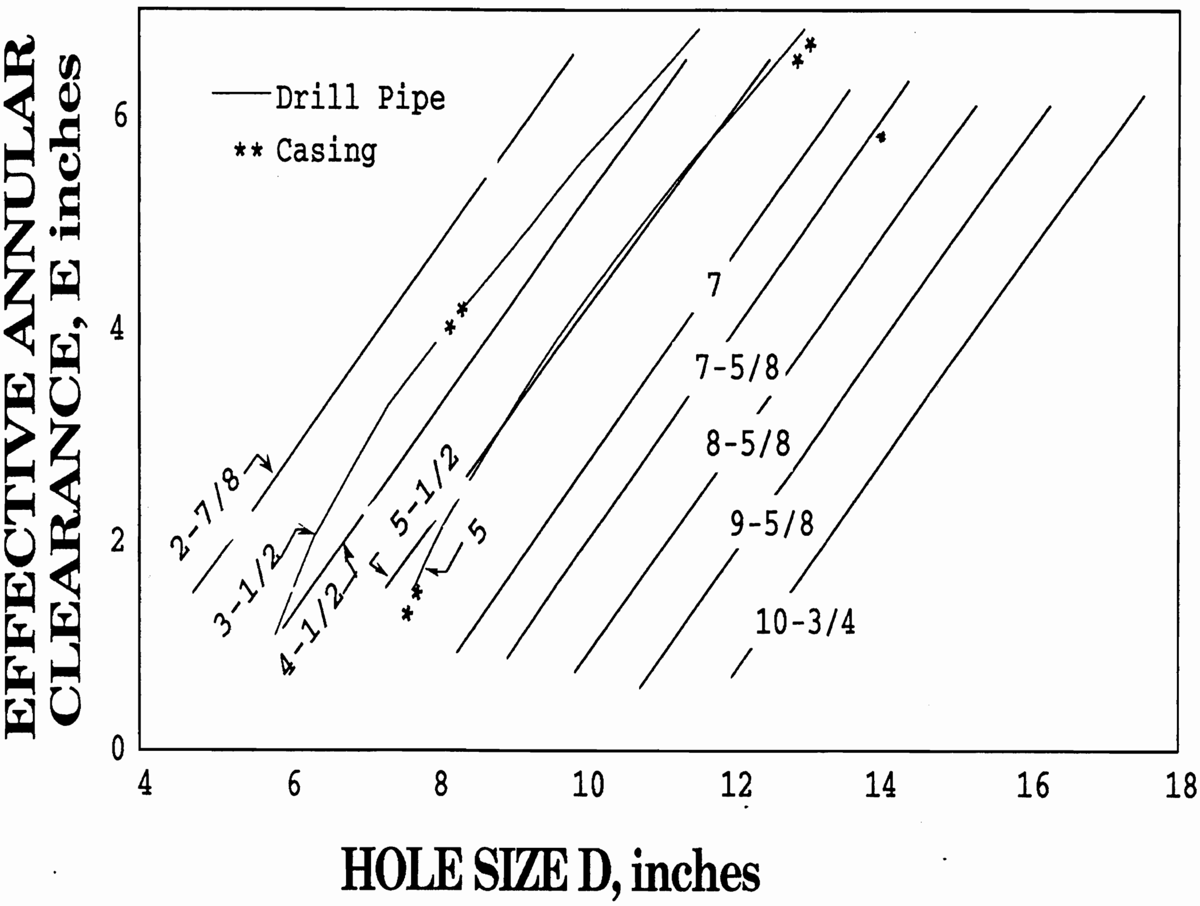
5" by 4.276" drill pipe is being run into a 9" drill hole in 16 ppg. With the pipe at 10,000', and if the first 15' of drillpipe passed through the rotary table in 3 seconds of a stand, what was the value of the acceleration pressure surge?

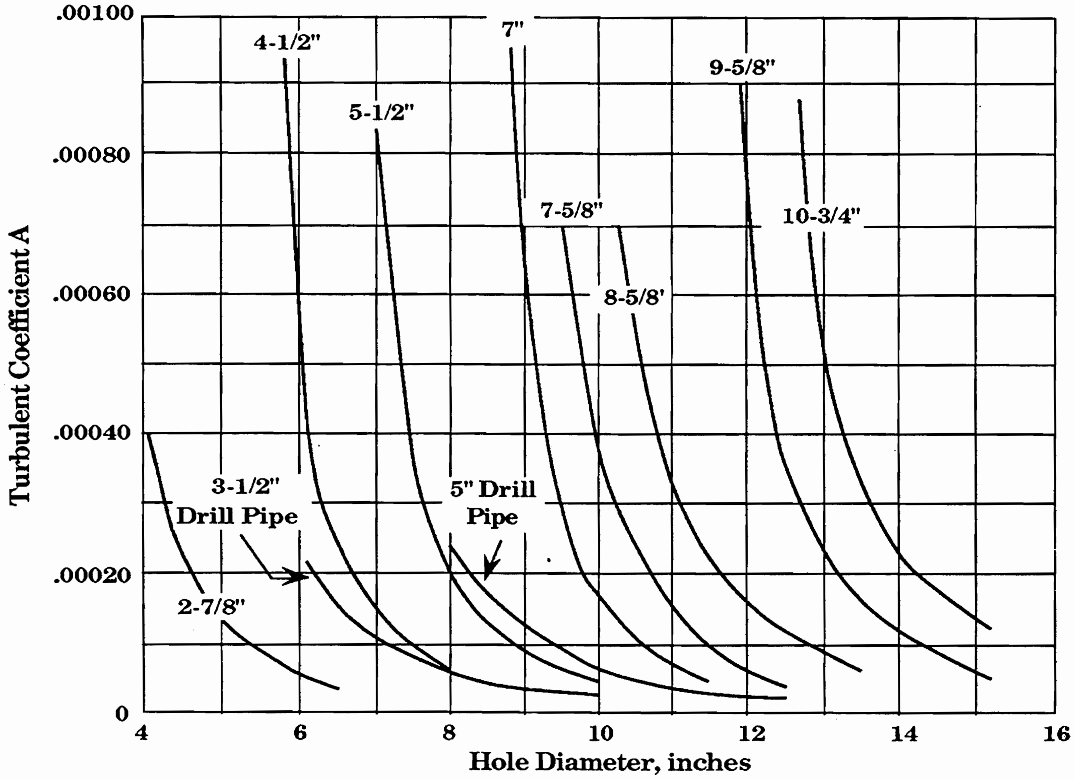


$$A_{pipe} = \frac{2 * 15}{3^2} \qquad \mathbf{= 3.33\ ft/s/s}$$

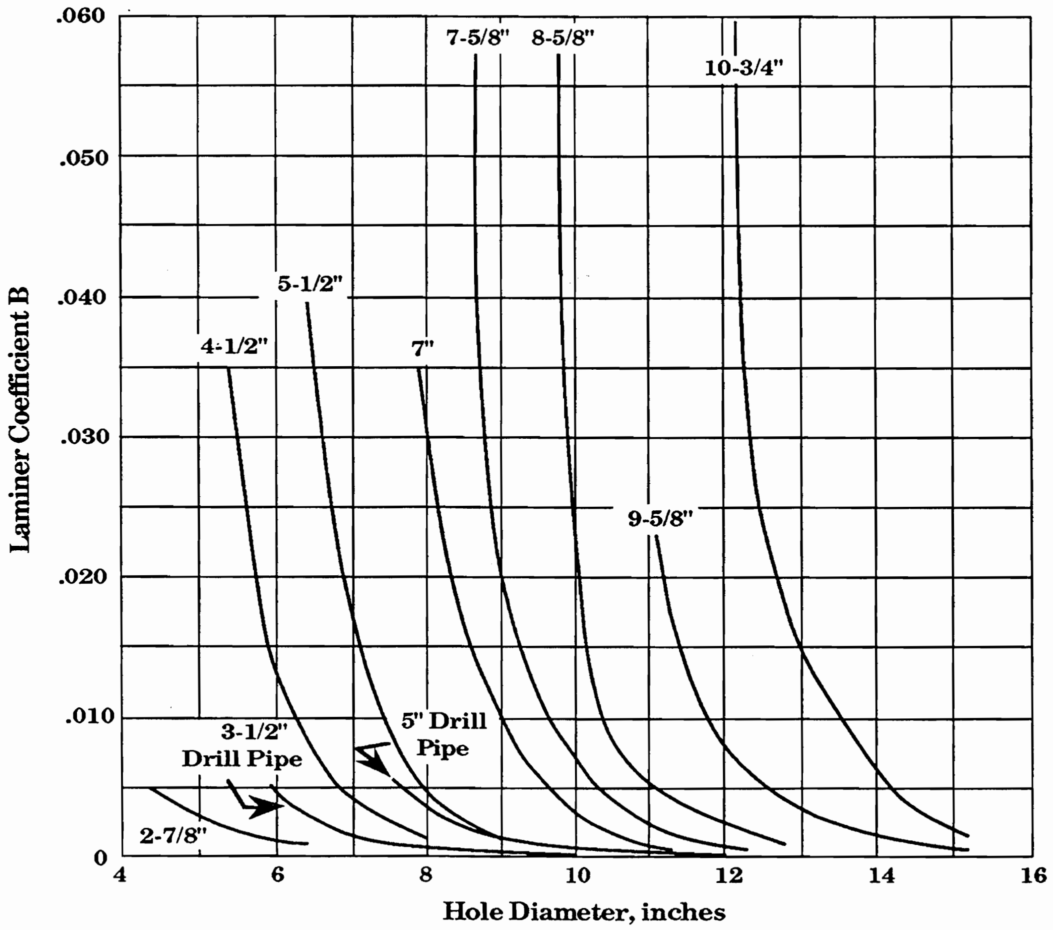
$$A_{mud} = 3.3 * \frac{5^2}{9^2 - 5^2} \qquad \mathbf{= 1.49\ ft/s/s}$$

$$P_A = \frac{16 * 1.49 * 10000}{620} \qquad \mathbf{= 384\ psi}$$

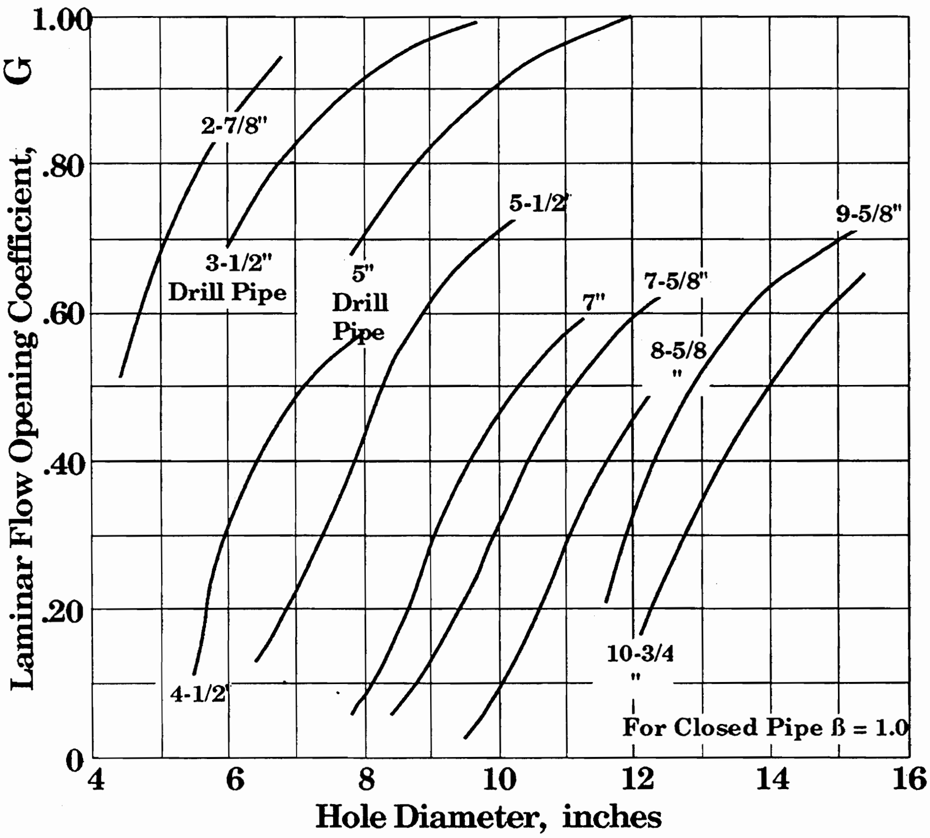

Drill Pipe
** Casing
EFFECTIVE ANNULAR CLEARANCE, E inches
HOLE SIZE D, inches
0
2
4
6
4
6
8
10
12
14
16
18
2-7/8
3-1/2
4-1/2
5-1/2
5
7
7-5/8
8-5/8
9-5/8
10-3/4

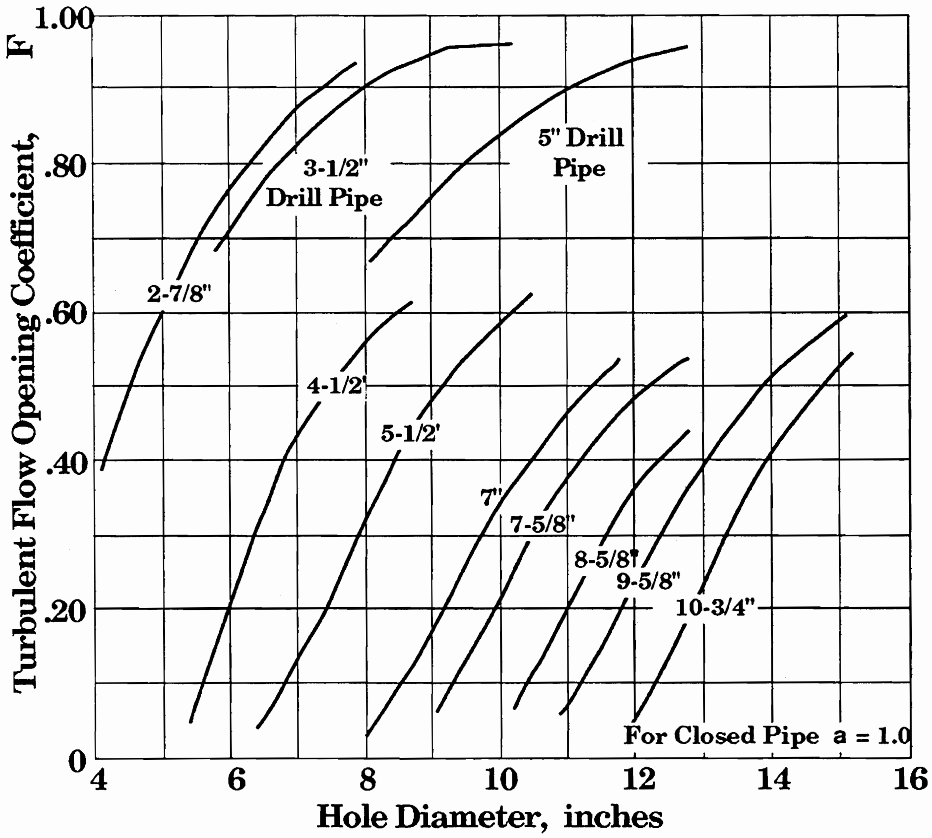


.00100
.00080
.00060
.00040
.00020
0
Turbulent Coefficient A
4-1/2"
7"
9-5/8"
5-1/2"
7-5/8"
10-3/4"
8-5/8'
3-1/2"
Drill Pipe
5" Drill
Pipe
2-7/8"
4
6
8
10
12
14
16
Hole Diameter, inches



.060
.050
.040
.030
.020
.010
0
Laminer Coefficient B
7-5/8"
8-5/8"
10-3/4"
5-1/2"
4-1/2"
7"
9-5/8"
3-1/2"
Drill Pipe
5" Drill
Pipe
2-7/8"
4
6
8
10
12
14
16
Hole Diameter, inches


1.00
.80
.60
.40
.20
0
Laminar Flow Opening Coefficient, G
4
6
8
10
12
14
16
Hole Diameter, inches
2-7/8"
3-1/2"
Drill Pipe
5"
Drill
Pipe
5-1/2'
7"
7-5/8"
8-5/8
"
9-5/8"
10-3/4
"
4-1/2'
For Closed Pipe ß = 1.0


1.00
.80
.60
.40
.20
0
Turbulent Flow Opening Coefficient, F
4
6
8
10
12
14
16
Hole Diameter, inches
3-1/2"
Drill Pipe
5" Drill
Pipe
2-7/8"
4-1/2'
5-1/2'
7"
7-5/8"
8-5/8"
9-5/8"
10-3/4"
For Closed Pipe a = 1.0

SURGE and SWAB PRESSURES of SHORT TOOLS

Swab and surge pressures are critical in close clearance operations. The technical paper by Burkhardt which summarizes the work of many researchers is used to show the effect of clearance on pressure surge or swab. It may be noted in the plot that pressure surge or swab increases rapidly with decreasing clearance. For the purpose of illustration and for tool and hole sizes which are of small clearance, several simplifications may be made reasonably which reduce the complexity of the surge and swab equations to the following. Fluid entrance and exit effects are neglected in the following analysis.

$$V_{ae} = \left(k + \frac{D^2}{H^2 - D^2}\right) * V_p$$

k: clinging constant = .45

$$N_R = \frac{928 * (H - D) * V_{ae} * MW}{PV}$$

$$f = .061 * N_R^{-.22}$$

$$P_f = \frac{f * V_{ae}^2 * MW * L}{25.8 * (H - D)}$$

V_{ae}	=	Effective annular velocity; fps
V_p	=	Casing running velocity; fps
D	=	Tool outside diameter; inch
H	=	Hole diameter; inch
N_R	=	Reynold's number; dimensionless
MW	=	Mud weight; ppg
PV	=	Plastic Viscosity; cp
P_f	=	Pressure surge or swab; psi
f	=	Friction factor; dimensionless
L	=	Length of tool; ft

EXAMPLE

The following example and plot illustrates the effect of clearance on pressure surge and swab. The hole and tool configuration is as follows.

Hole diameter is 8.535". The two tools are 3' and 30' long and have an external diameter of 8.235". Its velocity is 300 fpm (5 fps). The mud weighs 12 ppg and has a plastic viscosity of 43 cp.

$$V_{ae} = (.45 + \frac{8.235^2}{8.535^2 - 8.235^2}) * 5 \qquad = \mathbf{69.6\ fps}$$

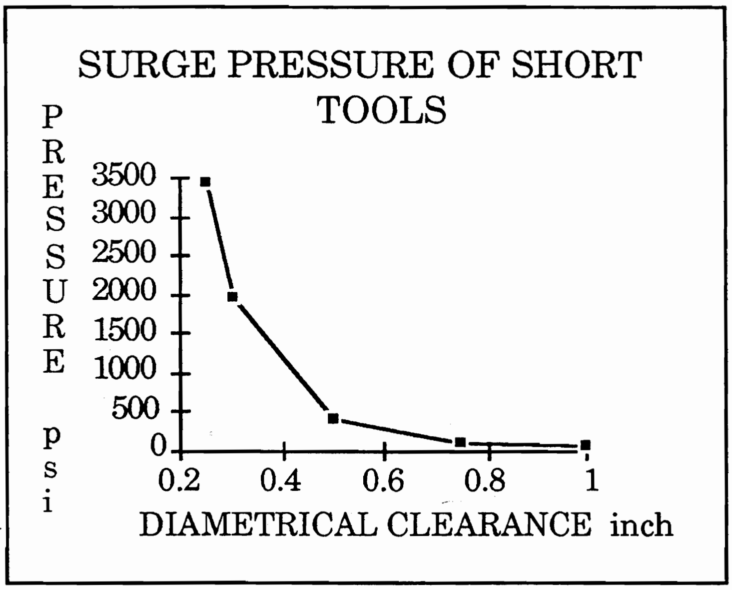
$$N_R = \frac{928 * (8.535 - 8.235) * 69.6 * 12}{43} \qquad = \mathbf{5{,}407}$$

$$f = .061 * 5{,}407^{-.22} \qquad = \mathbf{.00921}$$

$$P_f = \frac{.00921 * 12 * 67.4^2 * 3}{25.8 * (8.535 - 8.235)} \qquad = \mathbf{194.60}$$

P_f = 195 psi for a 3 foot drill collar or tool

P_f = 1,947 psi for a 30 foot drill collar or tool



DATA FOR THE PLOT

Clearance inch	Surge Pressure psi
0.25	3,439
0.30	1,959
0.50	397
0.75	108
1.00	42

CIRCULATING PRESSURES FOR SHORT TOOLS

Annular circulating pressure have been shown to be significantly affected by close clearances. This fact is further supported by the following example and is illustrated in the plot on the following page. The pressure loss equations may be simplified for close clearance tools and holes. Entrance and exit effects are neglected. The hole configuration detailed in the previous section is used here.

The circulation rate of 400 gpm is used to illustrate the effect of clearance on circulation pressures.

$$V_a = \frac{Q}{2.45 * (H^2 - D^2)}$$

$$N_R = \frac{928 * (H - D) * V_a * MW}{PV}$$

$$f = .061 * N_R^{-.22} \qquad P_f = \frac{f * V_a^2 * MW * L}{25.8 * (H - D)}$$

V_a = Annular velocity; fps
Q = Circulation rate; gpm

All other terms are as previously defined.

EXAMPLE

Variables as defined in the previous problem.

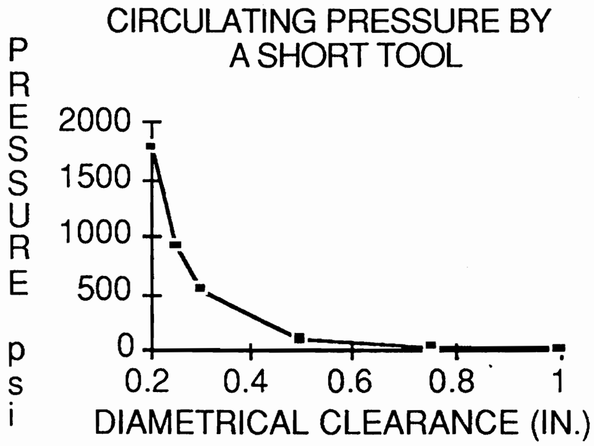
$$V_a = \frac{400}{2.45 * (8.535^2 - 8.235^2)} \qquad \mathbf{= 32.5\ fps}$$

$$N_R = \frac{928 * (8.535 - 8.235) * 32.5 * 12}{43} \qquad \mathbf{= 2{,}525}$$

$$f = .061 * 2{,}525^{-.22} \qquad \mathbf{= .01089}$$

$$P_f = \frac{.01089 * 12 * 32.5^2 * 3}{25.8 * (8.535 - 8.235)} \qquad \mathbf{= 53.5\ psi\ for\ a\ 3\ foot\ tool}$$

P_f **= 535 psi for a 30 foot drill collar or tool**



DATA FOR THE PLOT

Clearance inch	Friction Pressure Loss psi
0.20	1,782
0.25	916
0.30	535
0.50	118
0.75	36
1.00	16

EQUIVALENT CIRCULATING DENSITY

The equivalent circulating pressure, ECP, is the sum of the circulating friction pressure loss in the annulus and the static pressure of the mud above a selected location in the drill hole.

$$ECP = P_{cir.\ fric} + \frac{MW * DEPTH}{19.25}$$

Most often the circulating pressure friction loss is computed with the powerlaw model as illustrated earlier in this chapter.

The equivalent circulating density is thought of as a static mud column from the selected location up to the surface which will create a pressure equal to the ECP.

$$ECD = ECP * \frac{19.25}{DEPTH}$$

EXAMPLE

A mud which has a weight of 11.4 ppg is being circulated within a 9" by 5" annulus which is 15,000 feet long at a rate of 400 gpm. What is the value of the ECD at a depth of 12,000 feet?

The mud properties are given in the powerlaw example of this chapter.

The frictional pressure loss in the annulus was computed to be 0.0128 psi/foot of annulus in that example.

$$\frac{\Delta P_a}{L} \qquad \mathbf{= 0.0128\ psi/ft}$$

and the frictional pressure loss in the annulus at 12,000 feet is

$$\Delta P_a = 0.0128 * 12{,}000 \qquad \mathbf{= 154\ psi}$$

The static mud column pressure at the depth of 12,000 feet is

$$P_{static} = \frac{11.4 * 12000}{19.25} \qquad \mathbf{= 7{,}107\ psi}$$

The ECP and ECD are

$$ECP = 154 + 7{,}107 \qquad \mathbf{= 7{,}260\ psi}$$

$$ECD = \frac{7260 * 19.25}{12000} \qquad \mathbf{= 11.65\ psi}$$

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CHAPTER VI

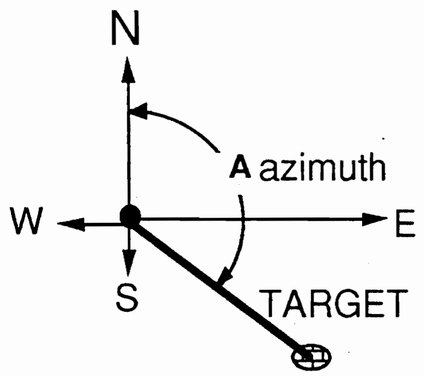
DIRECTIONAL DRILLING

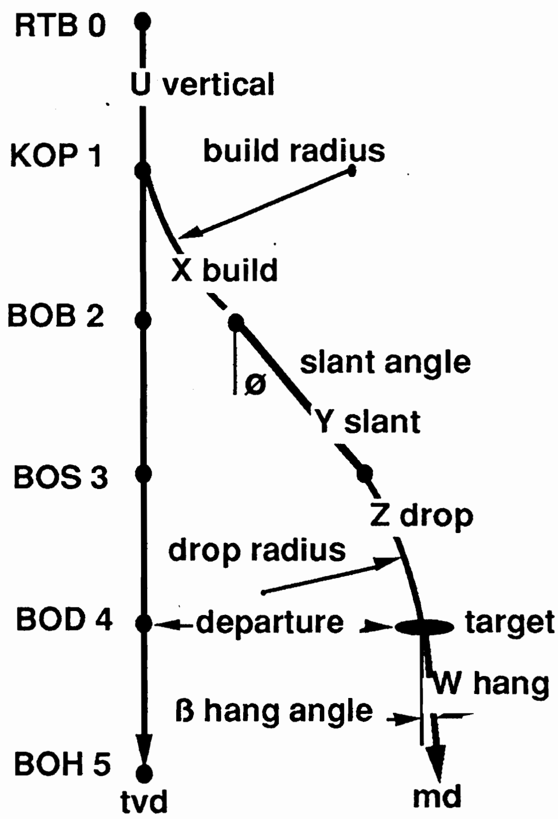
INTRODUCTION

Directional wells are defined as those wells which are to follow a prescribed traverse and intersect a specific objective. The objective is called a target and is usually an enclosed area in a horizontal plane. A target could be a circular area at the top of a producing zone.

If tolerance in the deviations of the well from the planned traverse is critical, the traverse is usually specified as a cylinder surrounding a section of the hole; otherwise, the traverse is given as a line path between the rotary table and the target.

Popular visual presentations of directional well data are on charts called horizontal and section views. The section view is a vertical cross-section drawn through the centers of the rotary table and the target.

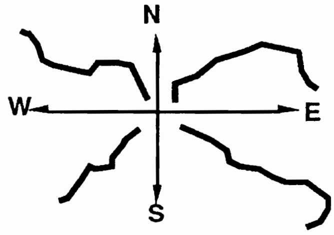




The horizontal view depicts north-south and east-west axis which intersect in the center of the rotary table. The target, the traverse, and directional stations are recorded on the two charts. The axis of the horizontal view may represent magnetic directions if it is desired.

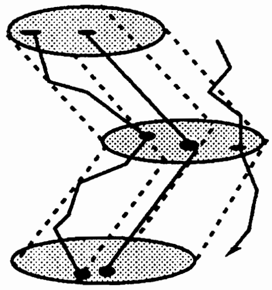
The primary purposes of the two views are to pictorially show deviations of the drilled traverse from the planned traverse and the progress of the hole relative to the target.

Spider views which are of two types are also popular. The regular spider view is a view in which the drill holes of interest are projected to a horizontal plane and the usual plane is the surface. It is a two dimensional chart. The other is a three dimensional view of a set of drill holes. The vertical angle and azimuth at which the drill holes are viewed may be chosen.

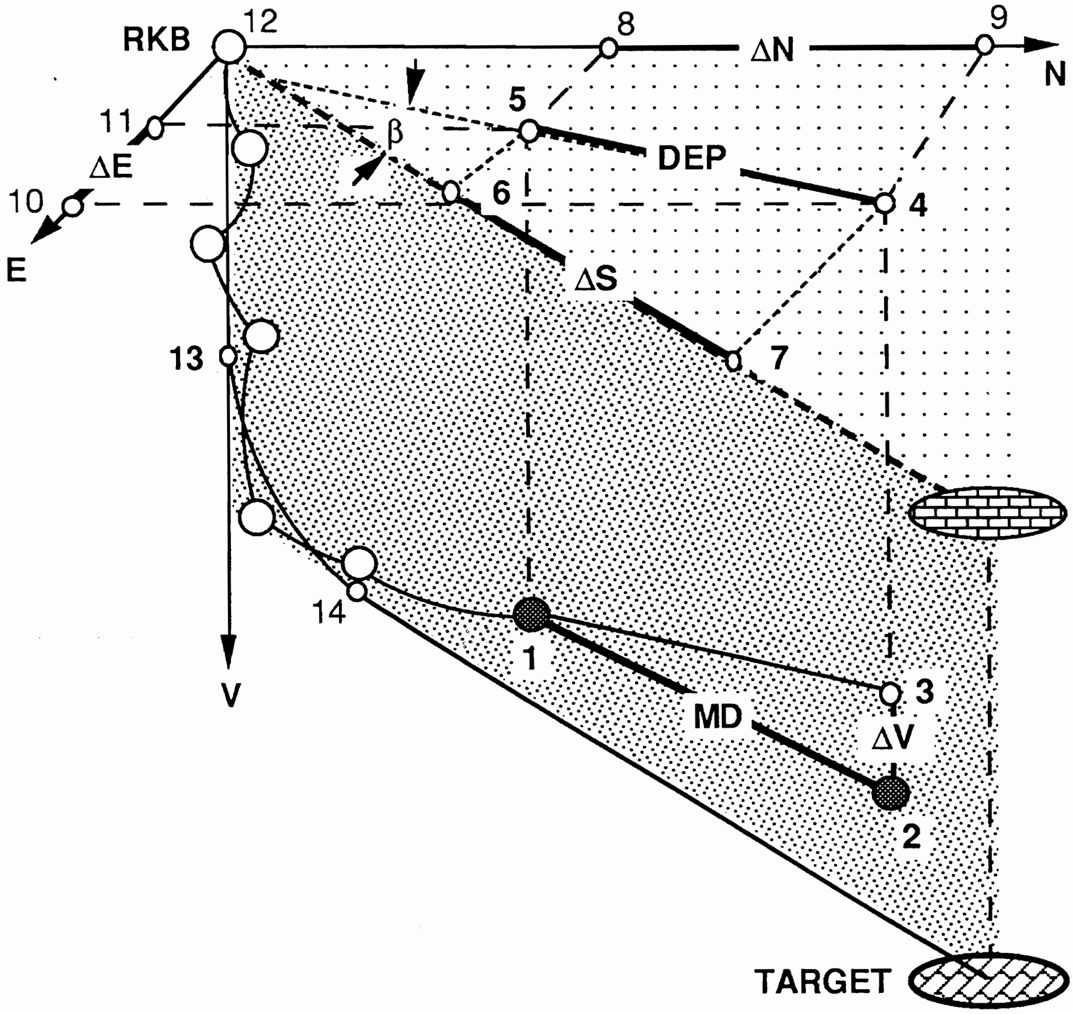


The travelling cylinder is a set of horizontal views. One drill hole of primary interest is at the center of all of the horizontal views.

All selected drill holes in the vicinity of the primary drill hole appear as points at their appropriate locations on the horizontal planes.



DEFINITIONS



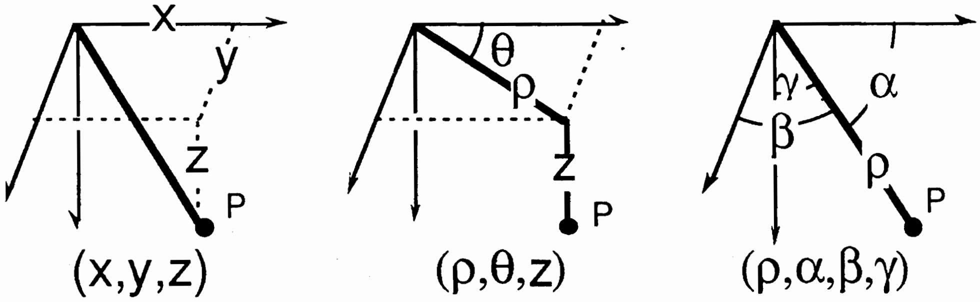
Target Direction = the direction from the rotary table to the target with reference to a meridian: (Ex: N 25.46 E) (Drawing: angle 9-12-7)

Target Azimuth = the clockwise angle between north and the target (Drawing: angle 9-12-7)

Kick off Point = the depth where the traverse departs from the vertical in the direction of the target: (Drawing: point 13)

Build = the curved section of the traverse extending from the kick off point to the slant section: (Drawing: arc 13-14)

Coordinates = Cartesian and spherical orthogonal coordinates are used. Polar coordinates which are often confused with spherical coordinates are not used.



Build Gradient = curvature of the build up section of the traverse: (Ex: 2°/100ft)

Slant = linear section of the traverse: (Drawing: line 14-target)

Slant Angle = angle between the slant and the vertical: (Ex: = 25.74°)

Drop Point = point in the drill hole at the end of the slant and the top of the drop: (not on drawing)

Drop = curved section of the traverse between the slant and the hang: (not on drawing)

Drop Gradient = curvature of the drop off: (Ex: 3°/100ft)

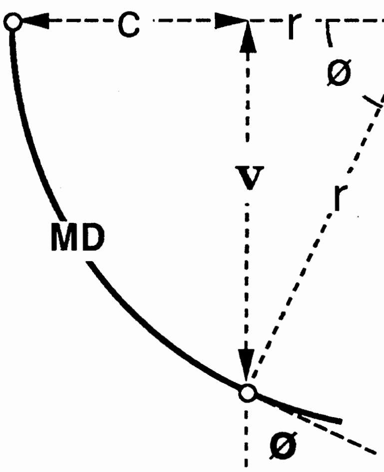
Hang = linear sections of the traverse below the drop off: (not on drawing)

Hang Angle = angle between the hang down and the vertical: (not on drawing)

Station	= location within the wellbore where direction and inclination are recorded by a survey instrument. Measured depth is recorded and is taken from drill string measurements.
Section	= horizontal distance between two stations which has been projected into the target azimuth: (Drawing: line 6-7)
Departure	= horizontal distance between two stations: (Drawing: line 5-4)
Difference Angle	= angle between the departure direction and the target direction: (Drawing: β angle: intersection of line 4-5 and line 6-7)
Closure	= horizontal distance between the rotary table and the drill hole: (Drawing: line 12-5)
Measured Depth	= length along the drill hole: (Drawing: line 1-2)
Vertical Depth	= vertical distance between the drill hole and the horizontal plane containing the rotary table (also called true vertical depth): (Drawing: line 2-4)
Coordinates	= distance along coordinate axes: Cartesian (N, E, V) and spherical (MD, Φ, θ)

EQUATIONS

Analytical geometry gives the following equations for the computations of TVD, departures, and MD, in directional planning.



The slant angle, Ø, is selected to intersect the target. The arc, MD, is a segment of a circle. The upper Ø is equal to the lower Ø because the similar side of the two angles are perpendicular.

The build gradient, BG, is usually given in degrees per 100 feet of measured depth. The sector equation of a circle gives

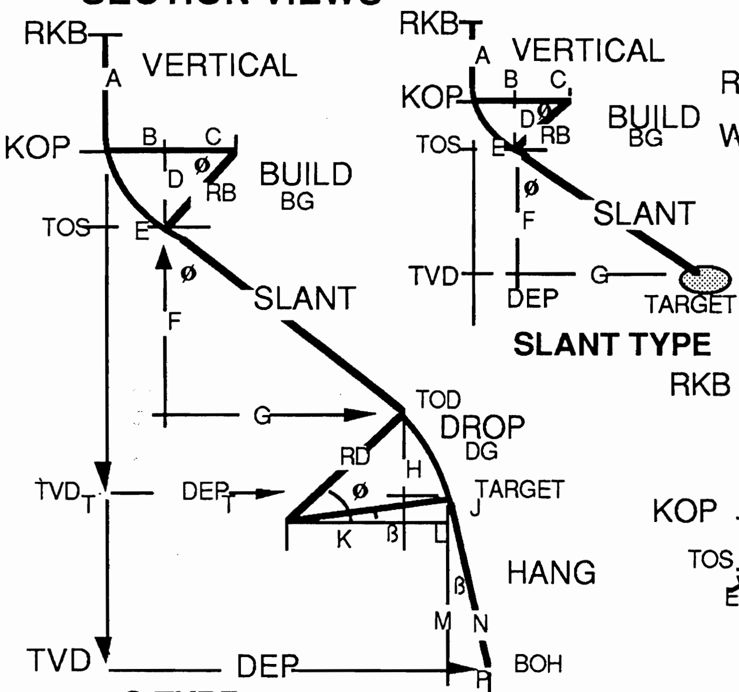
$$BG = \frac{\text{Ø}}{MD} \quad \text{and} \quad r = \frac{MD}{\text{Ø}\frac{\pi}{180}}$$

The value of V and C are given by

$$V = r * \sin\text{Ø} \qquad \text{and} \qquad C = r(1 - \cos\text{Ø})$$

DIRECTIONAL WELL PLANNING

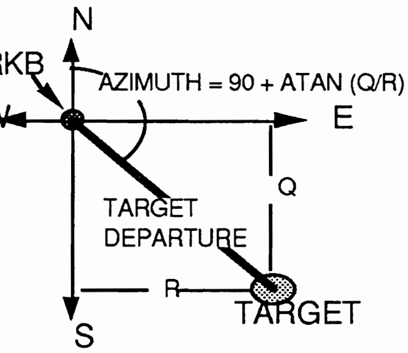
SECTION VIEWS

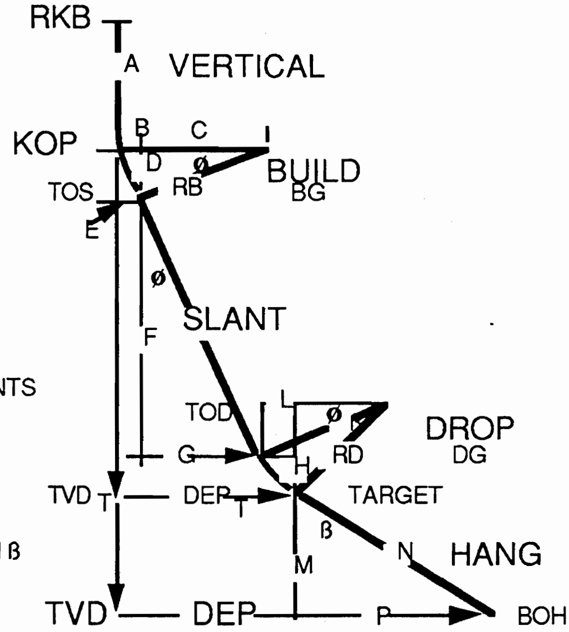


S TYPE

SLANT TYPE

PLANE VIEW





DOUBLE BUILD

TVD'S AT END POINTS

$V_{TOS} = A + D$

$V_{TOD} = A + D + F$

$V_{TOH} = A + D + F + ABS(H)$

DEPARTURES AT END POINTS

$D_{TOS} = E$

$D_{TOD} = D_{TOS} + G$

$D_{TOH} = D_{TOD} + ABS(L)$

$D_{BOH} = D_{TOH} + M \ TAN\ ß$

BG = SELECTED; DEG/FT
DG = SELECTED; DEG/FT
TVD = SELECTED; FT
DEP = SELECTED; FT
KOP = SELECTED; FT
HANG = SELECTED; FT
HANG ANGLE = SELECTED; DEG
TARGET TVD = SELECTED; FT
TARGET DEP = SELECTED; FT
RB = 180/(π BG); FT
RD = 180/(π DG); FT
RD = 0.0 FOR SLANT WELL
A = KOP - RKB
B = RB (1 - COS ø)
C = RB COS ø
D = RB SIN ø
E = B
N = BOH - TOH
M = N COS ß
P = N SIN ß
K = RD COS ø
L = RD (COS ß - COS ø)
J = RD SIN ß
H = RD (SIN ø - SIN ß)

MEASURED DEPHTS AT END POINTS

$M_{TOS} = A + \ ø\ \pi\ RB/180$

$M_{TOD} = M_{TOS} + F/COS \ ø$

$M_{TOH} = M_{TOD} \pm (ø - ß)\ \pi\ RD/180$

\+ S type
\- Double Build

with TARGET data

$F = TVD_T - A - D - ABS(H)$

$G = DEP_T - E - ABS(L)$

with BOTTOM HOLE data

F = TVD - A - D - ABS(H) - M

G = DEP - E - ABS(L) - P

SOLVE BY TRIAL AND ERROR FOR ø

TAN ø = G/F

$$TAN\ ø = \frac{DEP_T - RB * (1 - COS\ ø) - RD * ABS(COS\ ß - COS\ ø)}{TVD_T - (KOP - RKB) - RB * SIN\ ø - RD * ABS(SIN\ ø - SIN\ ß)}$$

RD = 0.0 FOR SLANT TYPE

A directional well plan calls for the enumeration of values shown on the directional plan or the directional drilling plan figures.

Three types of directional wells are popular: (1) the slant which is the most simple and is shown in the directional plan figure, (2) the S type, and (3) the horizontal or double build type which are shown on the directional drilling plan sketch. Some rules of thumb for well planning are that the slant type is less expensive and more feasible to drill than the other two types. In any case, locate the rig such that the drill hole will be drilled up dip for directional control. Selecting high kick off points; however, will require more overall directional footage.

Slant angles between 15 and 40 degrees are optimum. Directional control is difficult for smaller slant angles than 15 degrees and slant angles over 40 degrees cause problems in running some tools. Slant angles over 60 degrees usually require very time consuming techniques for running some tools.

Build gradients are doglegs. In deep holes build gradients in excess of 3 degrees per 100 feet could cause acute problems; while 10 degrees per 100 feet may be tolerable in shallow holes. Lubinski's equation which is published by the API in RP7G follows.

In a slant hole the target departure and the true vertical depth will be assigned by those requesting the drilling of the hole; and the three variables (a) kick off point, (2) build up gradient, and (3) slant angle may be selected. However, only two of the three are independent; in that, the third will be fixed by the appropriate geometric equations.

Experience has shown that the kick off point is the most critical of the three variables followed by the build gradient. Thus the most utilitarian of equations is the one given on the directional drilling plan sketch. Also this one equation solves for the slant angle for all three types of holes. A trial and error technique is required; however, the equation converges to a solution precise to six significant digits after two iterations for most problems.

LUBINSKI'S MAXIMUM PERMISSIBLE DOGLEG SEVERITY EQUATION

Lubinski's equation for the computation of the maximum allowable dogleg severity which will not fatigue drill pipe is the following.

$$MDLS_f = 137{,}500 \frac{\sigma}{E\,D\,K\,L}$$

$$\sigma = 20{,}000\ [\ 1 - \frac{BW/A_s}{145000}]$$ for grade S-135 drill pipe

$$K = \sqrt{\frac{BW}{E\,I}}$$

$$A_s = \frac{\pi}{4}(D^2 - d^2) \qquad I = \frac{\pi}{64}(D^4 - d^4)$$

Lubinski's equation for the computation of the maximum allowable dogleg severity which will not excessively wear the tool joints of the drill pipe is the following.

$$MDLS_w = 34{,}400\frac{\text{Lateral Force}}{L * BW}$$

BW	=	Bouyed weight of drill string below the top of the dogleg; lbf
D, d	=	OD and ID of the drill pipe; inch
Lateral Force	=	an acceptable lateral force on the tooljoints; lbf
L	=	one-half the length of one drill pipe; inch
E	=	Young's Modulus; 3E7 for steel; psi

CODES FOR THE EQUATIONS

The design sketch, shown at the beginning of this section, has two sets of code: 1 through 5 and U through Z. The numeric code refers to the beginning and end of segments of the drill hole; such as, the kick off point and drop point. The alphabetic code refers to the segments; such as, the build and hang segments.

Suppose we wanted to calculate the measured depth within the slant segment if we had been given the true vertical depth; then, the correct choice would be the equation which has the subscript 'm'.

EXPLICIT DIRECTIONAL DRILLING EQUATIONS

$$B = \frac{180}{\pi\, BG} \qquad D = \frac{180}{\pi\, DG} \qquad C = \frac{a}{\cos\mu}$$

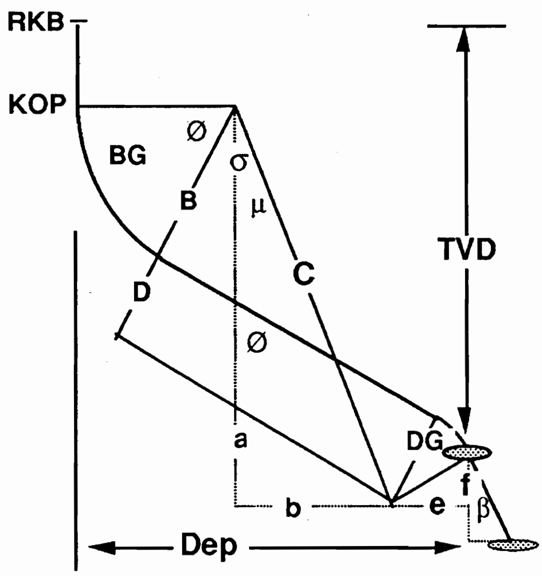
$$f = D \sin \beta \qquad e = D \cos \beta$$

$$\mu = \text{atan}\left[\frac{b}{a}\right]$$

$$\mu = \text{atan}\left[\frac{Dep - e - B}{TVD + f - KOP}\right]$$

$$\sigma = \text{acos}\left[\frac{B + D}{C}\right]$$

$$\emptyset = 90 + \mu - \sigma$$



$$\emptyset = 90 + \text{atan}\left[\frac{Dep - D\cos\beta - B}{TVD + D\sin\beta - KOP}\right] - \text{acos}\left[\frac{B + D}{C}\right] \qquad 0 \leq \emptyset \leq 90$$

Explicit Equations

Explicit solutions, similar to those developed below, exist; but because they have geometrical constraints which are not easily recognized, they are not recommended.

Equations Valid Within Build-Up Section of Hole

$$DEP_x = B * \left[1 - \cos \left[\text{asin} \left(\frac{TVD_x - KOP}{B} \right) \right] \right]$$

$$MD_x = KOP + \frac{1}{BG} * \text{asin} \left[\frac{TVD_x - KOP}{B} \right]$$

Equation Used To Calculate Slant Angle

For a slant well (no hang section):

$$\varnothing = 90 + \text{atan} \left[\frac{DEP_4 - B}{TVD_4 - KOP} \right] - \text{acos} \left[\frac{B * \cos\left[\text{atan}\left(\frac{DEP_4 - B}{TVD_4 - KOP}\right)\right]}{TVD_4 - KOP} \right]$$

For an S-type well (hang section is present):

$$\varnothing = 90 + \text{atan} \left[\frac{Dep - D \cos ß - B}{TVD + D \sin ß - KOP} \right] - \text{acos} \left[\frac{B + D}{C} \right]$$

$$0 \leq \varnothing \leq 90$$

Equations Used To Calculate Coordinates of Point 2

$$TVD_2 = KOP + B \sin \varnothing$$

$$DEP_2 = B\,(1 - \cos \varnothing)$$

$$MD_2 = KOP + \frac{\varnothing}{BG}$$

Equations Valid Within Slant Section of the Hole

$$DEP_y = DEP_2 + (TVD_y - TVD_2) * \tan \varnothing$$

$$MD_y = MD_2 + \frac{TVD_y - TVD_2}{\cos \varnothing}$$

Equations Used to Calculate Coordinates of Point 3

$$TVD_3 = TVD_4 - D\,(\sin \varnothing - \sin ß)$$

$$DEP_3 = DEP_4 - D\,(\cos ß - \cos \varnothing)$$

$$MD_3 = MD_2 + \frac{TVD_3 - TVD_2}{\cos \phi}$$

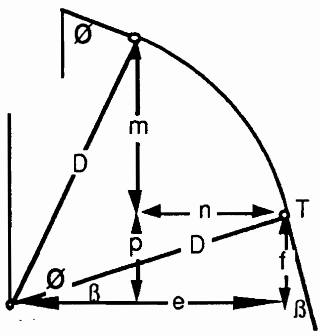
Equations Valid Within Drop-Off Section of the Hole

$$TVD_z = TVD_3 + D * [\sin \phi - \sin(\phi - \frac{MD_3 - MD_z}{D})]$$

$$DEP_z = DEP_3 + D \left[\cos [\operatorname{asin} (\sin \phi - \frac{TVD_z - TVD_3}{D})]\right]$$

$$MD_z = MD_3 + D * [\phi - \operatorname{asin}(\sin \phi - \frac{TVD_z - TVD_3}{D})]$$

Equations Valid Within Hang Section of the Hole



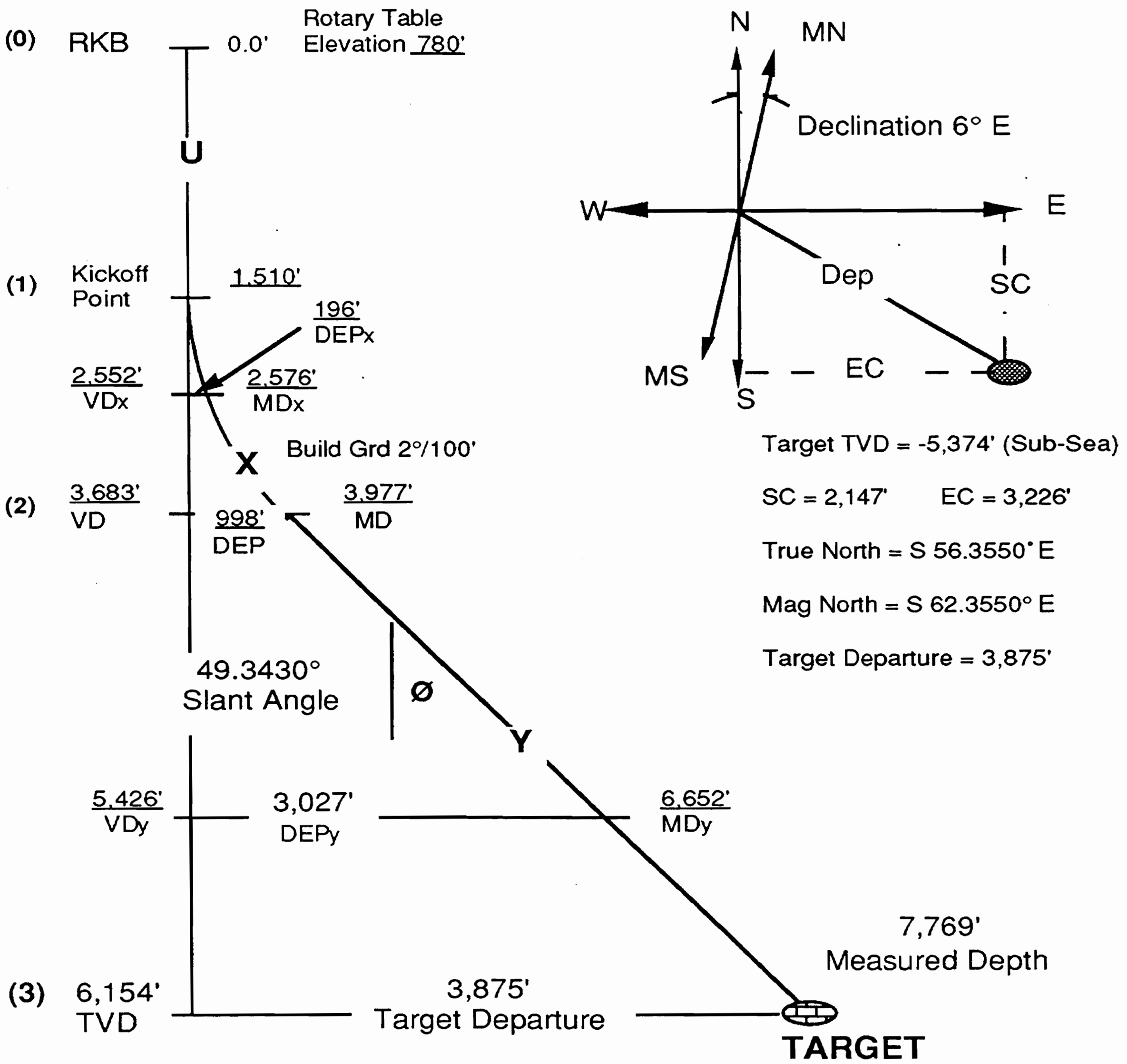
$$DEP_w = (TVD_w - TVD_4) * \tan \beta$$

$$MD_w = MD_4 + \frac{TVD_w - TVD_4}{\cos \beta}$$

EXAMPLE SLANT HOLE PLAN

A slant hole is to have a target depth of -5,374 ft (mean sea level), a south coordinate of 2,147 ft, and east coordinate of 3,226 ft. The rotary bushing is elevation is +780 ft (MSL). The magnetic declination is 6° east. A build gradient of 2 degrees per 100 ft of turn has been selected.

Rotary Table Elevation	= **780'**
Target Depth (sub-sea)	= **-5,374'**
Target True Vertical Depth	= **6,154'**
Target Departure = $(2{,}147^2 + 3{,}226^2)^{.5}$	= **3,875.14'**
Target Azimuth = 90 + atan (2,1447/3,226)	= **123.6450**
Magnetic Azimuth = 123.6450 + 6.0	= **129.6450**
Select Build Gradient = 2°/100ft	= **.02°/ft**



Radius of Curvature $\quad RB = \dfrac{180}{.02 * \pi}$ $\quad$ **= 2,865'**

Slant Angle, ø

$$ø = 90 + \text{atan}\left(\frac{3875 - 2865}{6154 - 1510}\right) - \text{acos}\left[\frac{2865 * \cos\left[\text{atan}\left(\frac{3875 - 2865}{6154 - 1510}\right)\right]}{6154 - 1510}\right] \quad \mathbf{= 49.3430^\circ}$$

$$VD_2 = 1{,}510 + 2{,}865 * \sin 49.3430 \quad \mathbf{= 3{,}683.46'}$$

$$DEP_2 = 2{,}865 * (1 - \cos 49.3430) \quad \mathbf{= 998.37'}$$

$$MD_2 = 1{,}510 + \frac{49.3430}{.02} \quad \mathbf{= 3{,}977.15'}$$

$$MD_3 = 1{,}510 + \frac{49.3430}{.02} + \frac{6154 - 3683.46}{\cos 49.3430} \quad \mathbf{= 7{,}769.06'}$$

$$MD_x = 1{,}510 + \frac{1}{.02} * \text{asin}\left[\frac{2552 - 1510}{2865}\right] \quad \mathbf{= 2{,}576.38'}$$

$$DEP_x = 2{,}865 * \left[1 - \cos\left(\text{asin}\,\frac{2552 - 1510}{2865}\right)\right] \quad \mathbf{= 196.21'}$$

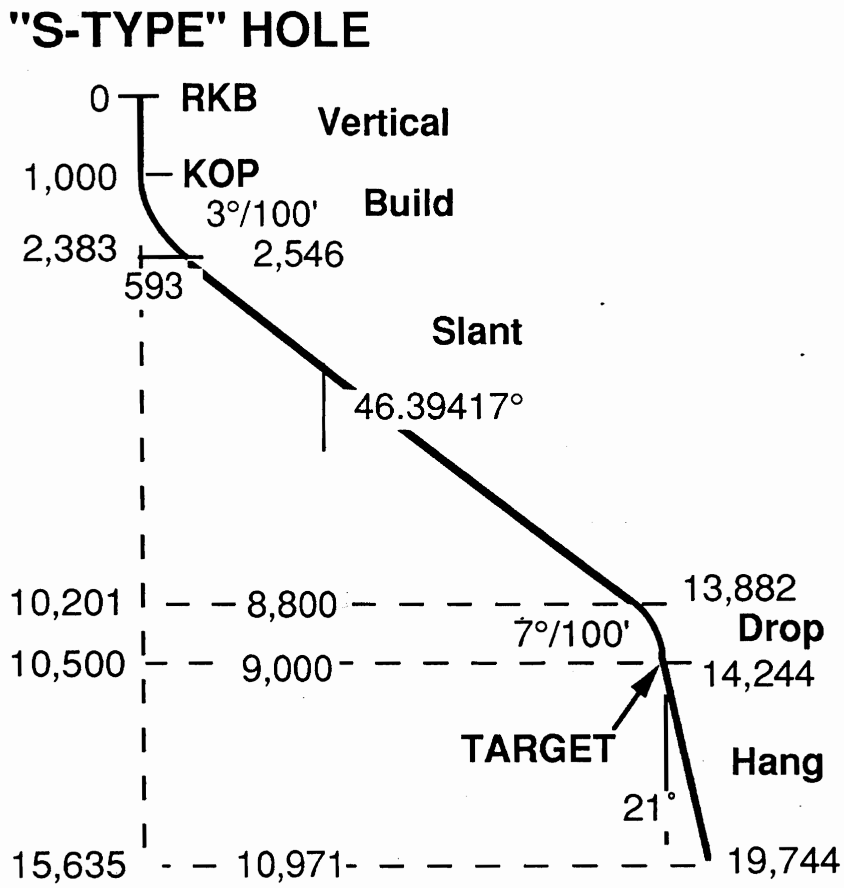
$$MD_y = 3{,}977.15 + \frac{5426 - 3683.46}{\cos 49.3430} \quad \mathbf{= 6{,}651.69'}$$

$$DEP_y = 998.37 + (5{,}426 - 3{,}683.46) * \tan 49.3430 \quad \mathbf{= 3{,}027.34'}$$

EXAMPLE S -TYPE PLAN

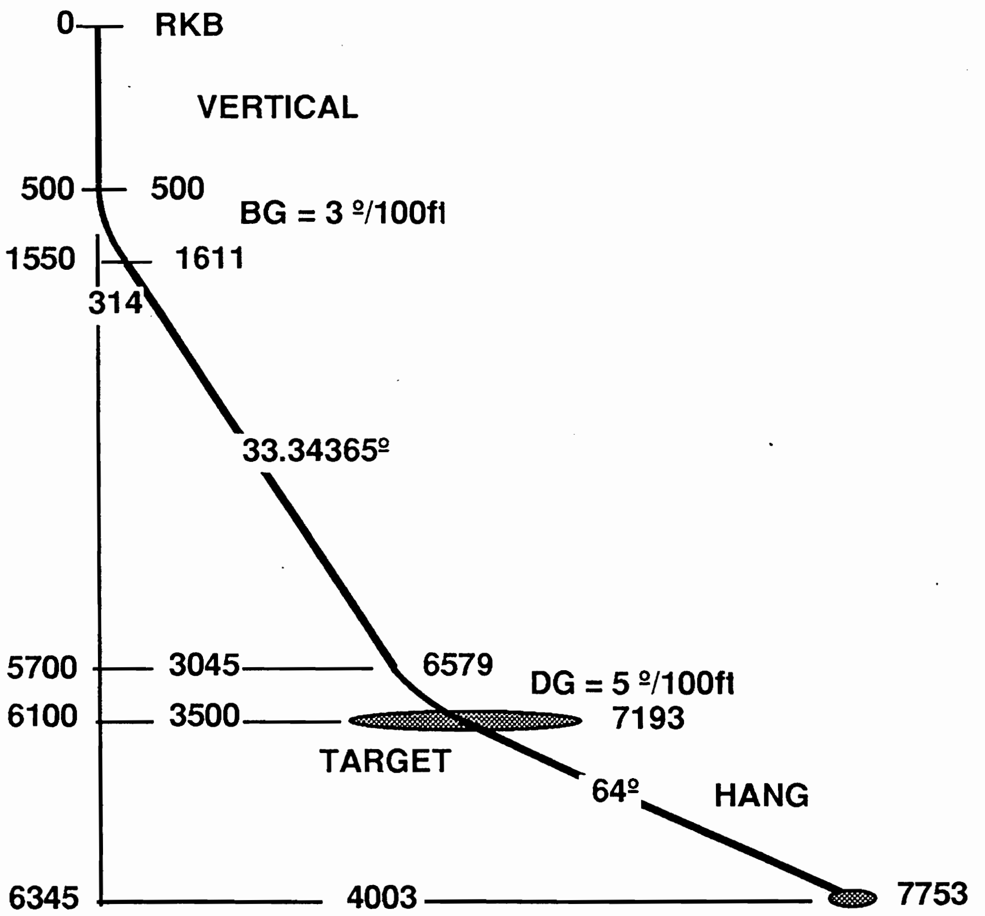
The requirements for an "S-TYPE" hole are (1) a kick off point of 1,000', (2) a build gradient of 3°/100', (3) a target TVD of 10,500', (4) a target departure of 9,000', (4) a drop gradient of 7 degrees per 100', (5) a hang length of 5,135'TVD, and (6) a hang angle of 21°.

In most cases, the driller will only participate in the selection of the build gradient and the slant angle.



EXAMPLE DOUBLE BUILD PLAN

The requirements for a "DOUBLE BUILD" hole are (1) a kick off point of 500', (2) a build gradient of 3°/100', (3) a target TVD of 6,100', (4) a target departure of 3,500', (4) a drop gradient of 5 degrees per 100', (5) a hang length of 245' TVD, and (6) a hang angle of 64°.



SYMBOLS

The Symbols on the preceding figures are the following.

M and MD	=	measured depth
V and VD	=	vertical depth
D and DEP	=	departure
S	=	south
E	=	east
S.L.	=	sea level
ø	=	slant angle

θ	=	direction (angle)
W	=	west
cos	=	cosine
sin	=	sine
BG	=	build gradient
DG	=	drop gradient
tan	=	tangent
KOP	=	kick off point
PI	=	3.141592654
beta	=	hang angle
W, X, Y, and Z	=	segments of the drill hole
1, 2, 3, 4, and 5	=	end of segments of the drill hole
TD	=	target departure
SC	=	south coordinate
EC	=	east coordinate
TD'	=	target direction
MN	=	magnetic north
N	=	north
Declination	=	the angle between the directions of north and magnetic north
TVD	=	true vertical depth

TRANSPOSING MD TO TVD AND TVD TO MD

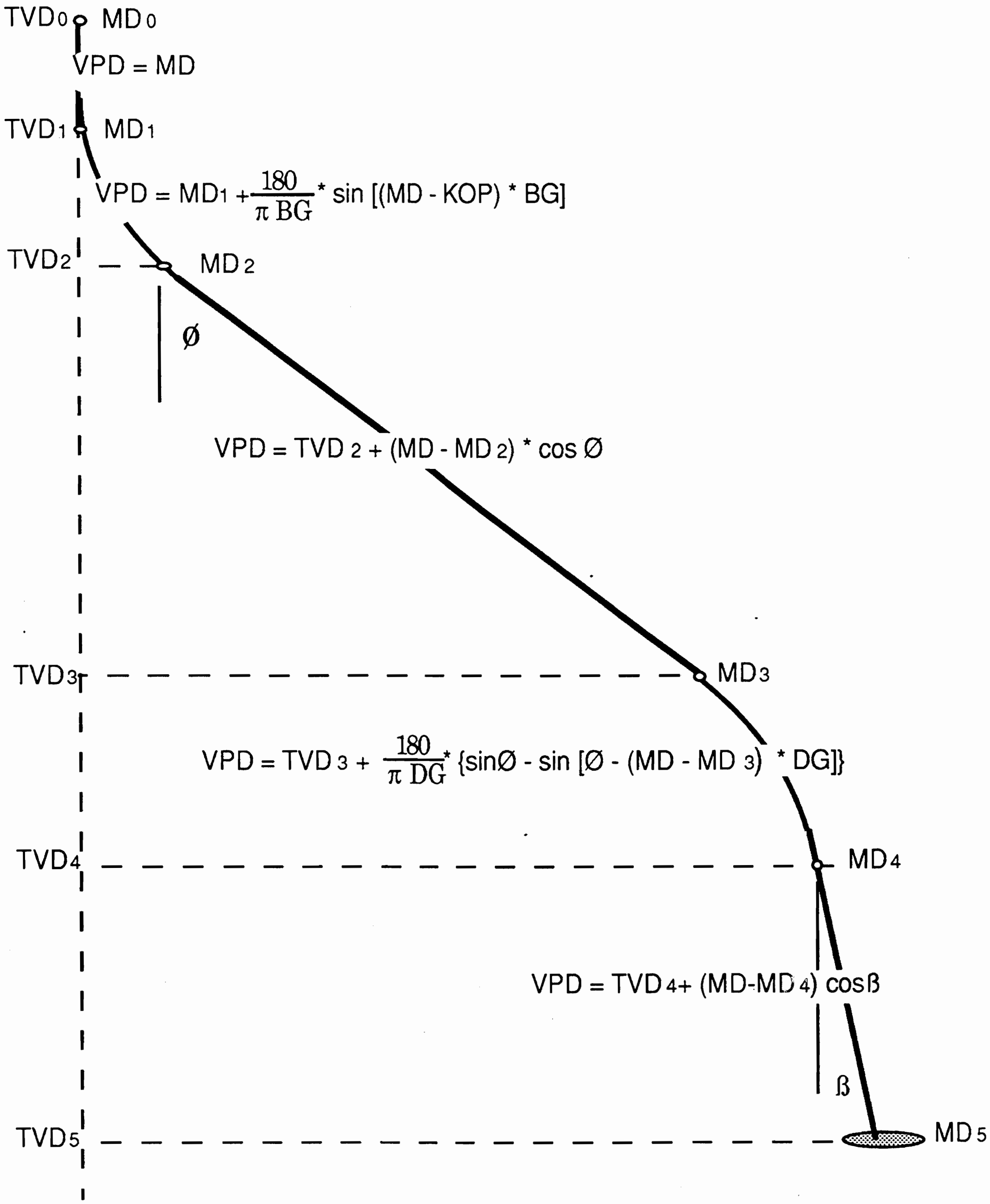
It is often desirable to project or transpose true vertical depths to measured depths from a directional well plan. This comes about if one wants to ascertain the quantity of casing required to reach certain zones or formations. Usually these zones will be specified by the geological or engineering departments in terms of true vertical depths. The figure provides an accurate and quick method of ascertaining any desired measured depth in commonly drilled directional wells.

The equations shown in the figure apply within the portion of the hole in which they are written.

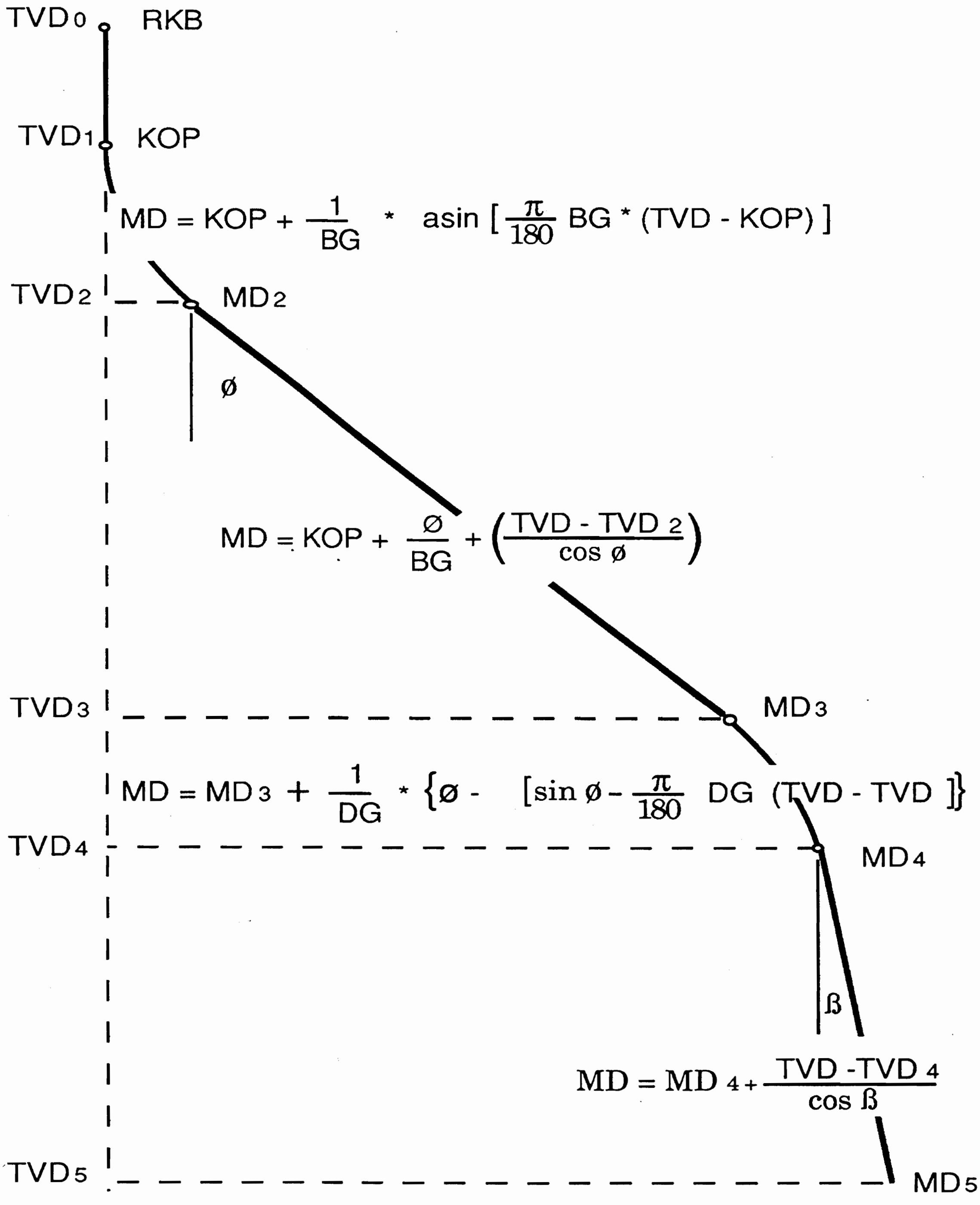
The terms have the usual meanings:

MD	=	The unknown measured depth for which we are searching; feet
TVD	=	The known true vertical depth which we wish to transpose to the measured depth; feet

Equations for Vertical Projected Depths (VPD)



Equations for Measured Depths (MD)



$$MD = KOP + \frac{1}{BG} * \operatorname{asin}\left[\frac{\pi}{180} BG * (TVD - KOP)\right]$$

$$MD = KOP + \frac{\varnothing}{BG} + \left(\frac{TVD - TVD_2}{\cos \varnothing}\right)$$

$$MD = MD_3 + \frac{1}{DG} * \left\{\varnothing - \left[\sin \varnothing - \frac{\pi}{180} DG\ (TVD - TVD\ \right]\right\}$$

$$MD = MD_4 + \frac{TVD - TVD_4}{\cos ß}$$

EXAMPLE TRANSPOSING TVD TO MD

The measured depth at the top of the hang is 14,244' in the preceding example problem for the "S" type directional well.

For a true vertical depth of 10,000', compute the measured depth.

$$MD = 1000 + \frac{46.39417}{.03} + \frac{10000 - 2383}{\cos 46.39417} \qquad \mathbf{= 13{,}591\ ft}$$

The departure is 8,590 feet.

EXAMPLE TRANSPOSING MD TO TVD

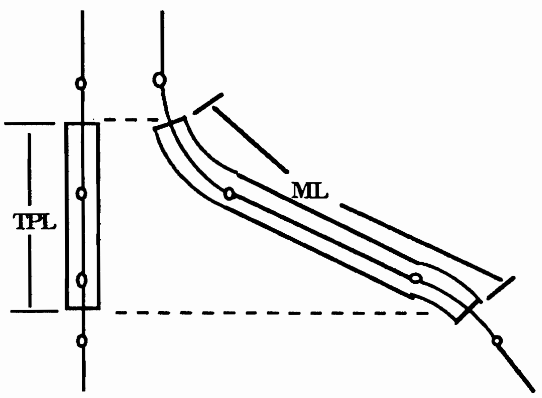
Find the VPL of a section of 7" casing if it lies between 1,500' MD and 14,000' MD. Once again use the data given for the preceding example "S" type well.

$$VPD_{top} = 1000 + (\frac{57.296}{.03}) \sin [(1500 - 1000) * .03] \qquad \mathbf{= 1{,}494\ ft}$$

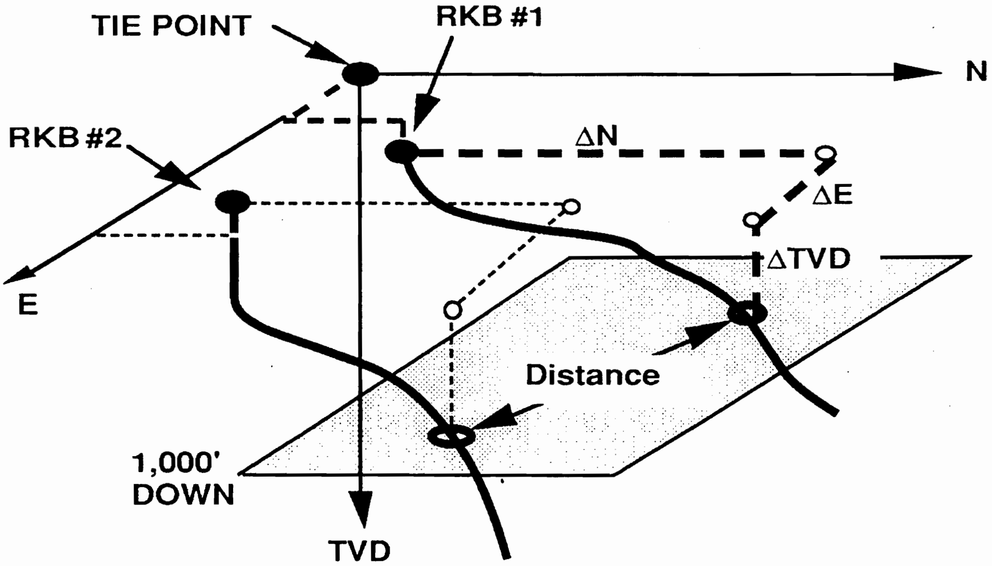
$$VPD_{bot} = 10{,}201 + \frac{57.296}{.07} \left[\sin 46.39417 - \sin(46.3917 - .07\ (14000 - 13882))\right]$$

$$\mathbf{= 10{,}288\ ft}$$

$$VPL_{7''} = 10{,}288 - 1{,}494 \qquad \mathbf{= 8{,}794\ ft}$$

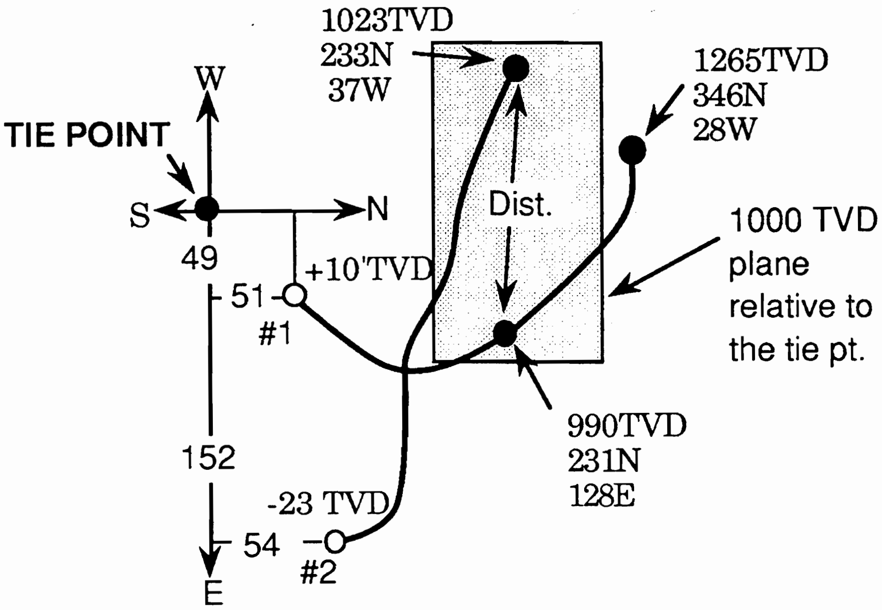


TIE POINT and COLLISION



The collision problem is the problem of ascertaining the three dimensional distance between drill holes. Because surveys are normally referenced to the rotary kelly bushing, rather than a common point called a tie point, the problem begins by determining the distances and heights of the rotary kelly bushings (RKB) from the tie point. (Any point could be the tie point.) Next the distances and heights of each survey stations in each drill hole are computed referenced to the tie point.

Finally, the distances are computed with the distance formula of analytical geometry. The RKB coordinates are ascertained by distances from the tie point to RKB. Imagine you are standing at the tie point and remember that down is positive. The following sketches illustrates these points.



The coordinates of RKB #1 and RKB#2 from the tie point, TP, are

$N/S_{TP \rightarrow RKB\#1}$ = **+51**

$E/W_{TP \rightarrow RKB\#1}$ = **+49**

$TVD_{TP \rightarrow RKB\#1}$ = **+10** (RKB#1 is 10 below the tie point)

$N/S_{TP \rightarrow RKB\#2}$ = **+54**

$E/W_{TP \rightarrow RKB\#2}$ = **+152**

$TVD_{TP \rightarrow RKB\#2}$ = **-23**

The coordinates of a station in a drill hole referenced to the tie point is computed with the distance from the tie point to the RKB added to the station's coordinates. The coordinates of a survey station referenced to the tie point are

$N/S_{TP \rightarrow station} = N/S_{TP \rightarrow RKB} + N/S_{RKB \rightarrow station}$
$E/W_{TP \rightarrow station} = E/W_{TP \rightarrow RKB} + E/W_{RKB \rightarrow station}$
$TVD_{TP \rightarrow station} = TVD_{TP \rightarrow RKB} + TVD_{RKB \rightarrow station}$

The coordinates of the upper station in the first borehole relative to the tie point are

$N/S_{TP \rightarrow station\#1}$ = +51 + 231 **= 282 ft**
$E/W_{TP \rightarrow station\#1}$ = +49 + 128 **= 177 ft**
$TVD_{TP \rightarrow station\#1}$ = +10 + 990 **= 1,000 ft**

The coordinates of the lower station in the first borehole relative to the tie point are

$N/S_{TP \rightarrow station\#1}$ = +51 + 346 **= 397 ft**
$E/W_{TP \rightarrow station\#1}$ = +49 - 28 **= 21 ft**
$TVD_{TP \rightarrow station\#1}$ = +10 + 1,265 **= 1,275 ft**

The coordinates of the station in the second borehole relative to the tie point are

$N/S_{TP \rightarrow station\#2}$ = +54 + 233 **= 287 ft**
$E/W_{TP \rightarrow station\#2}$ = +152 - 37 **= 115 ft**
$TVD_{TP \rightarrow station\#2}$ = -23 + 1,023 **= 1,000 ft**

The horizontal distance between the drill holes at 1,000 TVD relative to the tie point is

$$D_H = \sqrt{((N+/S-)_{tp\,1} - (N+/S-)_{tp\,2})^2 + ((E+/W-)_{tp\,1} - (E+/W-)_{tp\,2})^2}$$

$$D_H = \sqrt{(282 - 287)^2 + (177 - 115)^2} \qquad \mathbf{= 62.20\ ft}$$

The 3 dimensional distance between the lower station in borehole#1 and the station in borehole#2 is

$$3D = \sqrt{((N+/S-)_{tp\ 1} - (N+/S-)_{tp\ 2})^2 + ((E+/W-)_{tp\ 1} - (E+/W-)_{tp\ 2})^2 + (TVD_{tp\ 1} - TVD_{tp\ 2})^2}$$

$$3D = \sqrt{(287 - 397)^2 + (115 - 21)^2 + (1000 - 1275)^2} \qquad \mathbf{= 310.74\ ft}$$

EXAMPLE HORIZONTAL SEPARATION

The horizontal distance between two drill holes are to be computed at a true vertical depth of 1,000 feet relative to a tie point.The coordinates of the RKB's relative to the tie point are given in the table.

Coordinates RKB'			
	N	E	V
Tie Point	0.00	0.00	0.00
RKB_1	51.00	49.00	-8.00
RKB_2	54.00	152.00	10.00

The coordinates of the borehole which are interpolated with the Sectional method, referenced to the tie point, and shown on the following pages are

Coordinates of Hole #1		
N	E	V
108.98	276.63	1000.00

Coordinates of Hole #2		
N	E	V
108.51	276.03	1000.00

The horizontal distance between the two drill holes at a true vertical depth of 1,000 feet is

$$D_H = \sqrt{(108.98 - 108.51)^2 + (276.63 - 276.03)^2} \qquad \mathbf{= 0.76\ feet}$$

KILL WELL DESIGN

A kill well is to be designed for directional considerations. The kill borehole is to intersect the well which is blowing out (its design is given below) 200 ft TVD above the top perforations in the blowout well and then follow the blowout well to its bottom. Later the casing of the kill well will be perforated opposite the perforations of the blowout well and the well killed with mud. Completing the table will provide the directional considerations required.

TABLE OF DIRECTIONAL VARIABLES

WELL NAME	RKB ELEV ft	BOTTOM HOLE TVD ft	BOTTOM HOLE DEP ft	PERF'S TOP-BOT ft	KOP ft	BUILD GRAD. °/100	SLANT HANG deg	DROP GRAD. °/100	HANG LENGTH ft
BLOWOUT	0	6154	3875	6000 6100	1510	2	49.3418	--	--
KILLWELL	+26	6128	5195	none	900	3	50.0436	4	544

TARGET LOCATION

	MD ft	TVD ft	DEP ft
BLOWOUT	7726	5800	3463
KILL WELL	7774	5774	4783

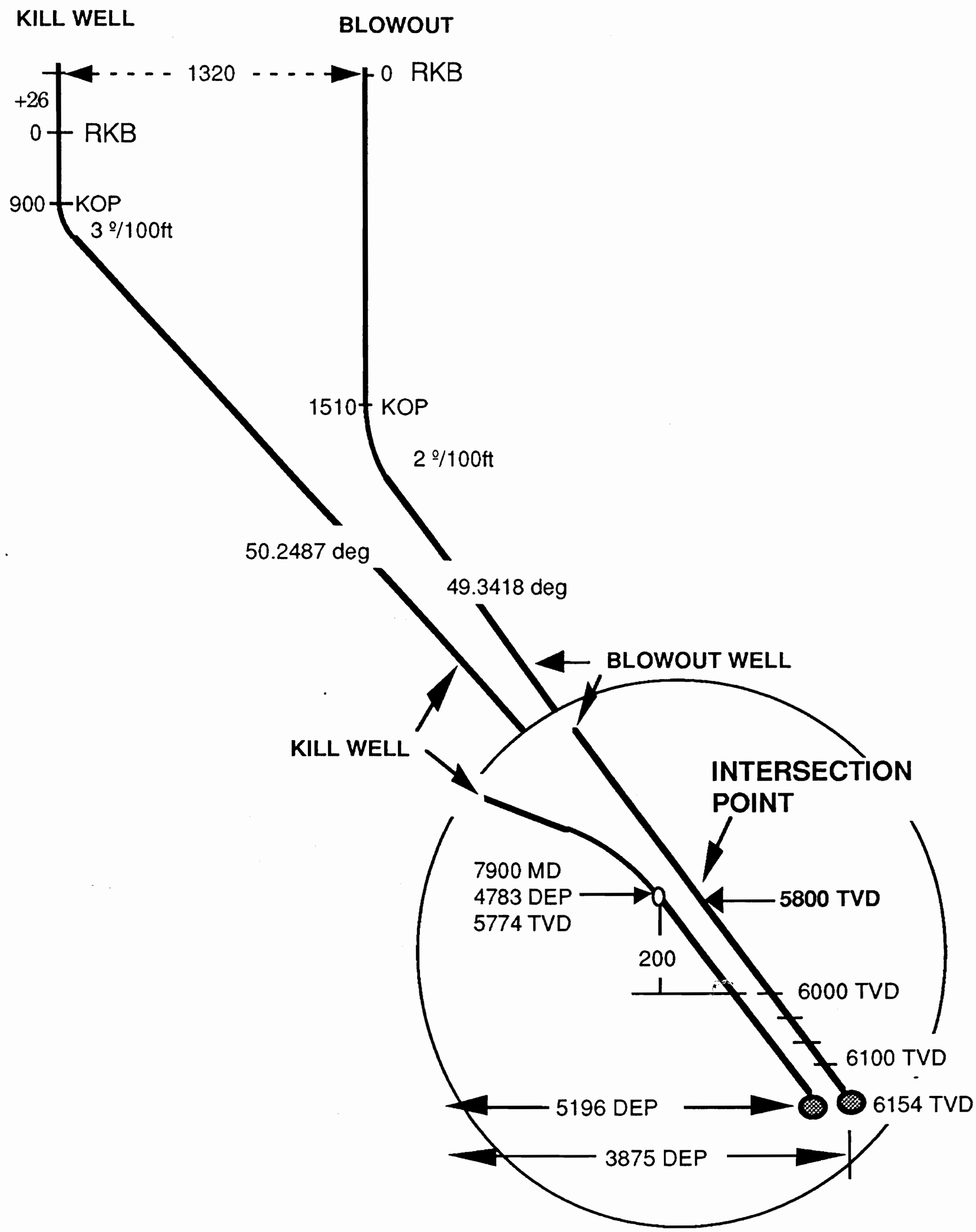
In the blowout well

Target TVD	= 6000 - 200	= 5800 ft TVD
Target DEP	= 3875 - (6154 - 5800) TAN(49.3418)	= 3462.83 ft DEP
Target MD	= 7769 - (6154 - 5800)/COS(49.3418)	= 7225.68 ft MD
Hang Length	= (6154 - 5800)/COS(49.3418)	= 543.32 ft MD

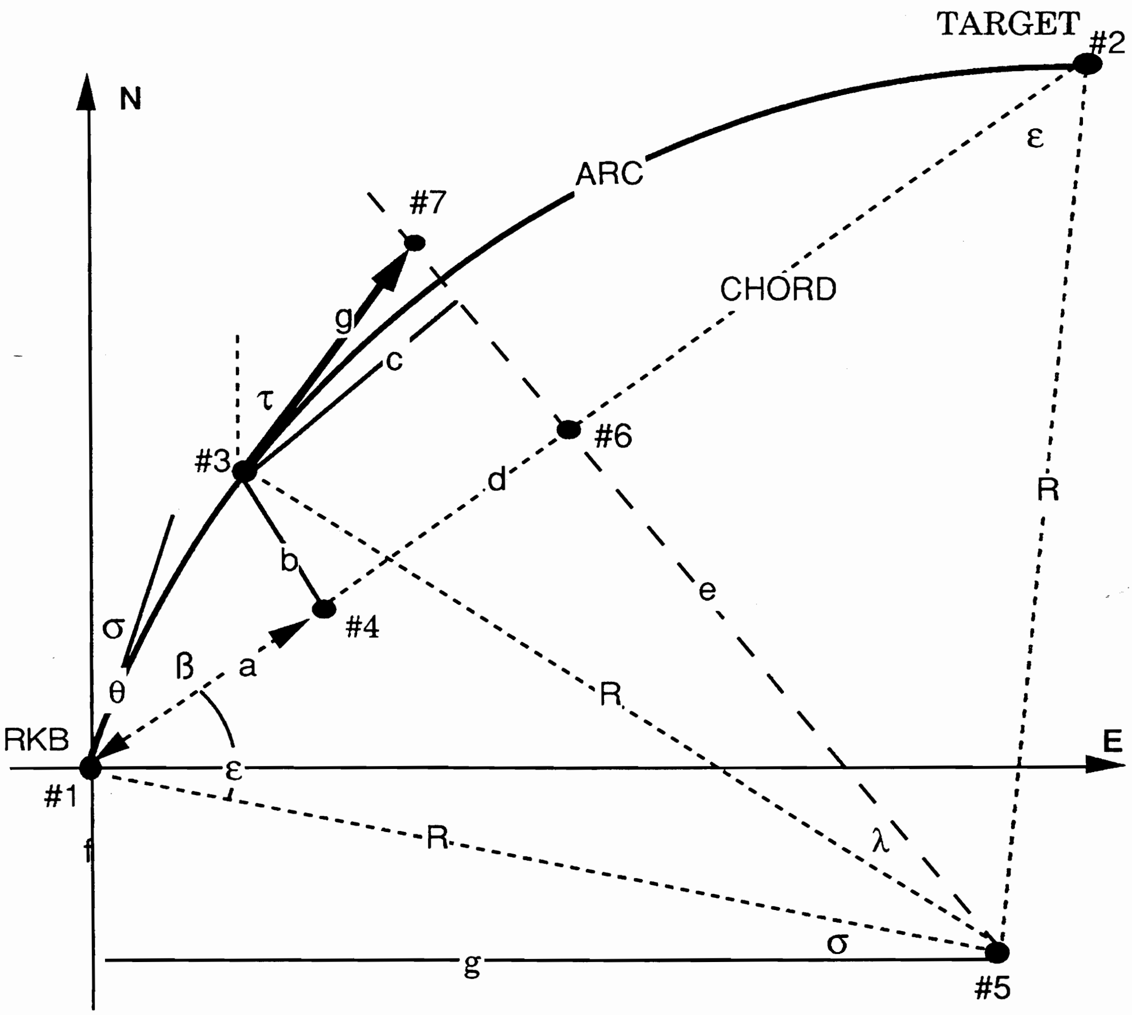
In the kill well

Target TVD	= 5800 - 26	= 5774 ft TVD
Target DEP	= 1320 + 3462.83	= 4782.83 ft DEP
Target MD		= 7900 ft MD
Hang Length	= (6128 - 5774)/COS(49.3418)	= 543.32 ft MD

BLOWOUT WELL INTERSECTION



LEADING THE TARGET WITH PLANNED WALK



The problem is to find the coordinates of any point, such as point #3, on the trajectory arc and the azimuth, angle τ, of the tangent line, g, to the arc at that point. Also, the turn gradient will be ascertained for excessive severity. Most often the lead angle, β, the coordinates of the target, N_2 & E_2. and the distance, a, along the chord to a point opposite the point on the arc are given. This means that in the diagram the values to be found are τ, N_3 & E_3, and the turn gradient (i.e., curvature).

$$\text{Chord length} = \sqrt{N_2^2 + E_2^2}$$

$$R = \frac{\text{Chord}}{2 \cos \varepsilon}$$

$$\varepsilon = 90 - \beta$$

Radius of the trajectory arc

Turn Gradient of the trajectory Arc

$$\text{ARC Gradient} = \frac{100}{R\frac{\Pi}{180}}$$

$$\text{Length of the arc} = R\ (180-2\varepsilon)\frac{\pi}{180}$$

Lengths and Angles

$e = R \sin \varepsilon$

$d = R \cos \varepsilon$

$a = \text{given}$

$c = d - a$

$\lambda = \text{asin}\,\frac{c}{R}$

$b = R\,(\cos \lambda\ -\ \sin \varepsilon)$

$\theta = \text{atan}\,\frac{E2}{N2}$

$\sigma = \theta - \beta$

Coordinates of Point 5

$N_5 = R \sin \sigma$

$E_5 = R \cos \sigma$

$$\cos \alpha_{1\text{-}2} = \frac{N_2}{\text{chord}}$$

$$\cos \beta_{1\text{-}2} = \frac{E_2}{\text{chord}}$$

Coordinates of Point 6

$N_6 = d \cos \alpha_{1\text{-}2}$

$E_6 = d \cos \beta_{1\text{-}2}$

$$\cos \alpha_{5\text{-}6} = \frac{N_6 - N_5}{e}$$

$$\cos \beta_{5\text{-}6} = \frac{E_6 - E_5}{e}$$

Coordinates of Point 4

$N_4 = a \cos \alpha_{1\text{-}2}$

$E_4 = a \cos \beta_{1\text{-}2}$

Coordinates of Point 3

$N_3 = b \cos \alpha_{5\text{-}6} + N_4$

$E_3 = b \cos \beta_{5\text{-}6} + E_4$

$$\cos \alpha_{3\text{-}5} = \frac{N_5 - N_3}{R}$$

$$\cos \beta_{3\text{-}5} = \frac{E_5 - E_3}{R}$$

$$\text{Length 5-7} = \frac{R}{\cos \lambda}$$

$\text{Length 3-7} = R \tan \lambda$

Coordinates of Point 7

$N_7 = \text{Length 5-7} * \cos \alpha_{5\text{-}6} + N_5$

$E_7 = \text{Length 5-7} * \cos \beta_{5\text{-}6} + E_5$

$$\cos \alpha_{3\text{-}7} = \frac{N_7 - N_3}{\text{Length 3-7}}$$

$$\cos \beta_{3\text{-}7} = \frac{E_7 - E_3}{\text{Length 3-7}}$$

Azimuth of the arc at point 3

$$Az = \text{atan}\,\frac{\cos \beta_{3\text{-}7}}{\cos \alpha_{3\text{-}7}}$$

EXAMPLE of LEADING THE TARGET

The north/east coordinates of the target relative to the rotary kelly bushing are 2,178N and 3,216E. The lead angle is 20^0 left of the target azimuth for right hand walk. Determine the coordinates of a point on the arc opposite a point on the target azimuth line which is 524 feet out from the RKB.

Chord length = $\sqrt{2178^2 + 3216^2}$	**= 3,884.11 ft**
$\varepsilon = 90 - 20$	**= 70^0**
Radius of the trajectory arc, $R = \dfrac{\text{Chord}}{2 \cos 70}$	**= 5,678.19 ft**
Turn Gradient of the trajectory arc = $\dfrac{100}{5678.19\frac{\Pi}{180}}$	**= 1.01 ^{0}/100'**
Length of the arc = $5678.19 * 40 * \dfrac{\pi}{180}$	**= 3,964.13 ft**
$e = 5{,}678.19 \sin 70$	**= 5,335.76 ft**
$d = 5{,}678.19 \cos 70$	**= 1,942.06 ft**
$c = 1{,}942.06 - 524$	**= 1,418.06 ft**
$\lambda = \text{asin}\ \dfrac{1418.06}{5678.19}$	**= 14.4620^0**
$b = 5678.19\,(\cos 14.4620 - \sin 70)$	**= 162.52 ft**
$\theta = \text{atan}(\dfrac{3216}{2178})$	**= 55.89^0**
$\sigma = 55.89 - 20$	**= 35.89^0**
Coordinates of point 5	
$N_5 = 5678.19 \sin 35.89$	**= - 3,328.94 ft**
$E_5 = 5678.19 \cos 35.89$	**= 4,600.00 ft**
$\cos \alpha_{1\text{-}2} = \dfrac{2178}{3884.11}$	**= 0.5607**
$\cos \beta_{1\text{-}2} = \dfrac{3216}{3884.11}$	**= 0.8280**
Coordinates of point 6	
$N_6 = 1942.06 * 0.5607$	**= 1,089.00 ft**
$E_6 = 1942.06 * 0.0.8280$	**= 1,608.00 ft**
$\cos \alpha_{5\text{-}6} = \dfrac{1089.00 + 3328.94}{5335.76}$	**= 0.8280**
$\cos \beta_{5\text{-}6} = \dfrac{1608.00 - 4600.00}{5335.76}$	**= - 0.5607**

Coordinates of Point 4

$N_4 = 524 * 0.5607$ **= 293.83 ft**

$E_4 = 524 * 0.8280$ **= 433.87 ft**

Coordinates of Point 3

$N_3 = 162.52 * 0.8280 + 293.83$ **= 428.39 ft**

$E_3 = 162.52 * 0.5607 + 433.87$ **= 342.74 ft**

$$\cos \alpha_{3\text{-}5} = \frac{-3328.94 - 428.39}{5678.19} = \mathbf{0.6617}$$

$$\cos \beta_{3\text{-}5} = \frac{4600.00 - 342.74}{5678.19} = \mathbf{-0.7498}$$

$$\text{Length 5-7} = \frac{5678.19}{\cos 14.4620} = \mathbf{5{,}864.00\ ft}$$

Length 3-7 = 5678.19 * tan 14.4620 **= 1,464.46 ft**

Coordinates of Point 7

$N_7 = 5864.00 * 0.8280 + (-3328.94)$ **= 1,526.38 ft**

$E_7 = 5864.00 * (-0.5607 + 4600.00$ **= 1,311.79 ft**

$$\cos \alpha_{3\text{-}7} = \frac{1526.38 - 428.39}{1464.46} = \mathbf{0.7498}$$

$$\cos \beta_{3\text{-}7} = \frac{1311.79 - 342.74}{1464.46} = \mathbf{0.6617}$$

Azimuth of the arc at point 3

$$Az = \text{atan}\,\frac{0.6617}{0.7498} = \mathbf{41.43^{0}}$$

DOGLEG SEVERITY OF WELL BORES

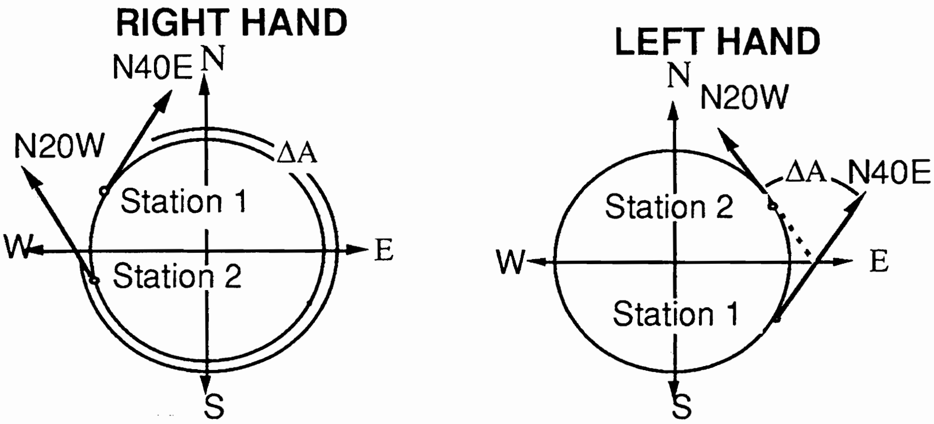
Dogleg severity is the curvature of borehole. Its common units are degrees per 100 feet of borehole length (^{0}/100 ft). The borehole is thought of as a three dimensional space curve which coincides with the center line of the borehole. Further, the space curve is depicted as a circular arc with its extremities terminated by stations.

The stations are inclinational survey stations as measured with single shot, multi-shot, or MWD survey instruments. Instrument depths are normally measured with drill strings. Instrument depths are normally measured with drill strings; and therefore, the distance between stations is the borehole length less off bottom distance for routine engineering purposes.

If the distance between two survey stations is extreme, then a dogleg of greater severity could exist between the two stations than that which may be thought to exist.

Thus, dogleg severities within a borehole are the composite curvatures of the borehole in direction and inclination between all juxtapositional stations.

TRUE DOGLEG AND TRUE AZIMUTH

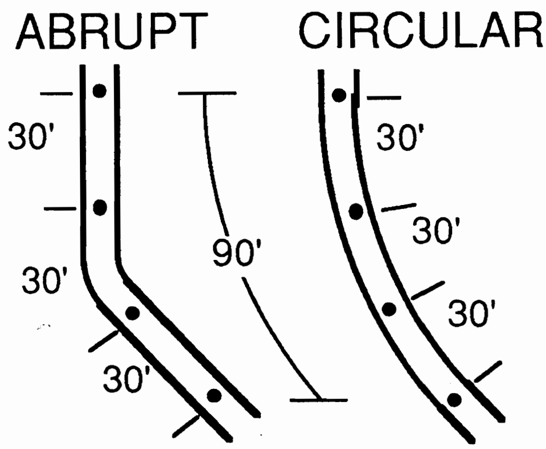


If stations occur in different azimuthal quadrants as shown in the figures below, then the angle difference, A_2 - A_1, is the value of the total angular change between the two stations.

In the figure on the left, ΔA is the angular difference, A_2 - A_1, and has the value of 60°. In the figure on the right, ΔA is 300°.

ABRUPT DOGLEG

The only method of accurately measuring abrupt doglegs is to survey at closer spacings. The sketch illustrates the problem. Note, that if the survey spacing is 90 feet rather than 30 feet, the computed dogleg severities will differ by a factor of three. The tour drillers in most cases will be able to assist in identifying intervals in the drill hole where abrupt and severe doglegs exists and plans can be made to reduce spacings in those intervals.



Dogleg Severity Equation

Wilson's equation for computing dogleg severity is as follows:

$$DLS = 100 * \left[\left(\frac{I_2 - I_1}{MD_2 - MD_1}\right)^2 + \left(\frac{A_2 - A_1}{MD_2 - MD_1} \sin I_a\right)^2\right]^{\frac{1}{2}}$$

$$I_a = \frac{I_2 + I_1}{2}$$

DLS = dogleg severity; ^{o}/100 ft

MD_1	=	measured depth to first (upper) station; ft
MD_2	=	measured depth to second (lower) station; ft
I_1	=	inclinational angle of first station; ^{o}
I_2	=	inclinational angle of second station; ^{o}
I_a	=	average inclinational angle of second station; ^{o}
A_1	=	azimuth of first station; ^{o}
A_2	=	azimuth of second station;^{o}

The following is a more compact form of the above equation.

$$DLS = \frac{100}{\Delta MD} * [\Delta I^2 + (\Delta A * \sin I_a)^2]^{1/2}$$

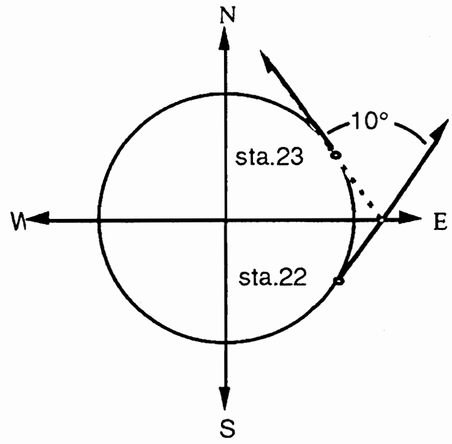
ΔMD	=	difference in measured depth between the two stations; feet
ΔI	=	change in inclination angle; ^{o}
ΔA	=	change in azimuth; ^{o}

EXAMPLE DOGLEG SEVERITIES COMPUTATION

TABLE OF DOGLEG SEVERITIES

Station #	MD ft	I ^{o}	A ^{o}	DLS °/100ft
-------	-----	---	----	-------
21	6,121	32	14	----
22	6,332	36	8	2.47
23	6,542	34	358	????

1. Consider only inclinational curvature for station number 22.



$$DLS = 100 * \frac{36-32}{6332-6121}$$

$$\mathbf{= 1.90\ °/100\ ft}$$

2. Consider only directional curvature for station number 22.

$$DLS = 100 * \frac{(8 - 14) * \sin 34}{(6332 - 6121)} \qquad \mathbf{= 1.59\ °/100\ ft}$$

3. Composite curvature between station numbers 21 & 23.
Station 21-22:

$$DLS = 100 * \left[\left(\frac{36 - 32}{6332 - 6121}\right)^2 + \left(\frac{8 - 14}{6332 - 6121} \sin 34\right)^2\right]^{1/2} \qquad \mathbf{= 2.47\ °/100ft}$$

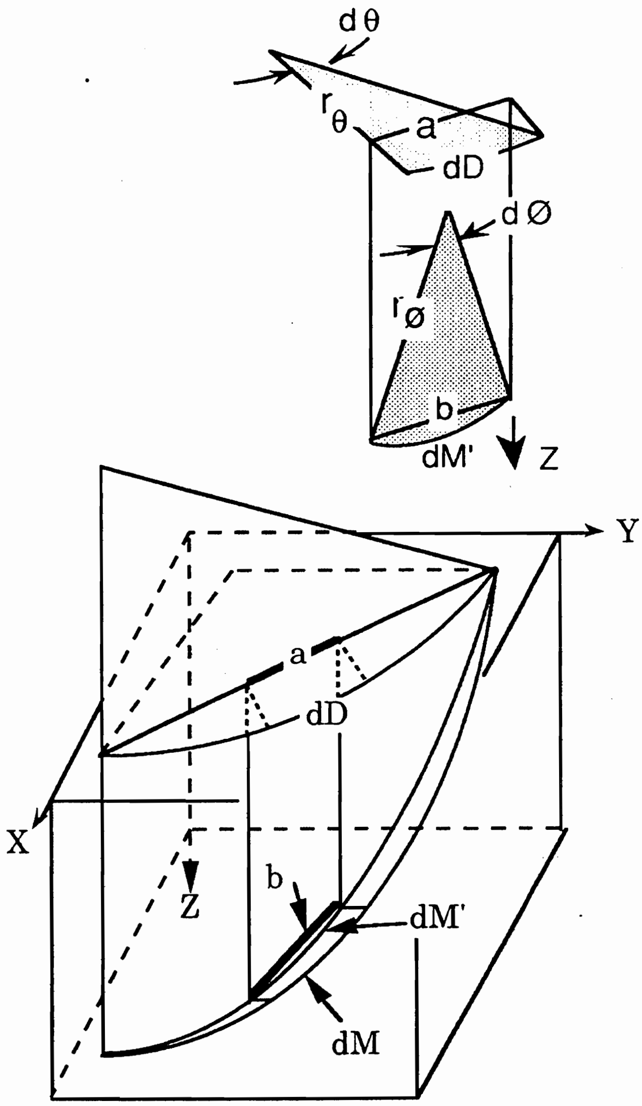
Station 22-23:

$$DLS = 100 * \left[\left(\frac{36 - 34}{6542 - 6332}\right)^2 + \left(\frac{8 - (-2)}{6542 - 6332} \sin 35\right)^2 \right]^{1/2} \qquad \mathbf{= 2.89\ °/100ft}$$

DERIVATION OF WILSON'S DOGLEG SEVERITY EQUATION

There are many dogleg severity formulae and charts and they give almost equal values: Lubinski, Long, and Wilson wrote the more popular ones. The following derivation is Wilson's. He combines the build in a vertical plane with the turn in the horizontal plane to give a three dimensional dogleg severity equation.

More precisely, he projected the drill hole into a vertical plane and set the length of the projection equal to the measured depth of the drill hole. He then projected the drill hole into a horizontal plane and set the length of the resulting arc equal to the departure of the drill hole. Finally he used the equation for the curvature of an arc in rectangular coordinates to get his dogleg severity equation.



Observations of the sketch reveal

$a \approx dD \qquad b \approx dM' \approx dM \qquad a = b \sin \phi$

$dM' = r_\phi \, d\phi \approx dM$

$dD = r_\theta d\theta \qquad \frac{dy}{dD} = \sin \theta \qquad \frac{dx}{dD} = \cos \theta$

$\frac{dy}{dM} = \sin \phi \sin \theta \qquad \frac{dx}{dM} = \sin \phi \cos \theta$

$\frac{dz}{dM} = \cos \phi$ further note that $\frac{d}{dM}\left(\frac{dz}{dM}\right) = - \sin\phi \frac{d\phi}{dM} = - \frac{1}{r_\phi} \sin\phi$

thus, differentiation of the last three equations yields

$$\frac{d^2z}{dM^2} = - \frac{1}{r_\phi} \sin \phi \qquad \frac{d^2y}{dM^2} = \frac{1}{r_\phi} \cos \phi \sin \theta + \frac{1}{r_\theta} \sin^2 \phi \cos \theta$$

$$\frac{d^2x}{dM^2} = \frac{1}{r_\phi} \cos \phi \cos \theta - \frac{1}{r_\theta} \sin^2 \phi \sin \theta$$

curvature of an arc in rectangular coordinates is

$$C' = \left[\left(\frac{d^2E}{dM^2}\right)^2 + \left(\frac{d^2N}{dM^2}\right)^2 + \left(\frac{d^2V}{dM^2}\right)^2 \right]^{\frac{1}{2}}$$

Substitution, squaring and simplification gives

$$C' = \left[\frac{1}{r_\phi^{\ 2}} + \frac{1}{r_\theta^{\ 2}} \sin^4 \phi \right]^{\frac{1}{2}}$$

substitution again yields

$$C' = \left[\left(\frac{\phi_2 - \phi_1}{M_2 - M_1} \right)^2 + \left(\frac{\theta_2 - \theta_1}{M_2 - M_1} \sin \phi \right)^2 \right]^{\frac{1}{2}}$$

Dogleg severity expressed in ^{o}/100 feet is

$$DLS = 100 \left[\left(\frac{\phi_2 - \phi_1}{M_2 - M_1} \right)^2 + \left(\frac{\theta_2 - \theta_1}{M_2 - M_1} \sin \phi_a \right)^2 \right]^{\frac{1}{2}}$$

$$DLS = \frac{100}{\Delta M} \left[(\Delta\phi)^2 + (\Delta\theta \sin \phi)^2 \right]^{\frac{1}{2}}$$

DLS	=	Dogleg severity; ^{o}/100 feet
ϕ	=	Inclination Angle; ^{o}
θ	=	Azimuth Angle; ^{o}
M	=	Drilled Hole (measured depth); feet
ϕ_a	=	Average Inclination Angle; ^{o}

EXAMPLE DOGLEG SEVERITY

Let the measured depths, inclinations, and directions of two stations be P_1 (4112, 29^{o}, 60^{o}) and P_2 (4380, 31^{o}, 63^{o}). Calculate the dogleg severity between the stations.

The dogleg severity equation is

$$DLS = 100 \left[\left(\frac{31 - 29}{4380 - 4112} \right)^2 + \left(\frac{63 - 60}{4380 - 4112} \sin \frac{31 + 29}{2} \right)^2 \right]^{1/2}$$

DLS **= 0.93 °/100ft**

MONITORING OF A DIRECTIONAL WELL

Monitoring of a directional well is the recording, computing, and plotting of the directional data from the surveys of the drill hole. The common data which are analyzed and recorded on directional worksheets such as the one later illustrated are (1) measured depth, inclination angle, and direction of the wellbore, (2) segment length (3) change in depth and cumulative depth (4) departure, (5) change in section and cumulative section (6) change in meridian and longitudinal coordinates and cumulative coordinates. Seldom are analyses made which yield (1) dogleg severity of the wellbore, (2) horizontal, vertical, and perpendicular distances of the wellbore from its planned path, (3) projected location of the intersection of the wellbore and the plane containing the target, and (4) minimum dogleg severity required for wellbore and target intersection. Data commonly plotted at each station in the wellbore are section versus depth on the section view chart and coordinates on the horizontal view chart.

A directional station is the location (measured depth) within the wellbore where a directional and inclinational indicating and recording instrument is positioned. Direction is detected or resolved with either a magnetic compass or a gyrocompass. Inclination is detected with a plumb within the instrument. Depth is taken with the drillstring.

SURVEYING

The more popular instruments photograph onto one film both inclination and direction. Solid state instruments are also popular. It is assumed the instrument is axially aligned with the drill collar where it rests and the drill collar must be of nonmagnetic materials only if the magnetic compass is used. The measured depth is the length of the drillstring between the rotary table and the instrument, not the bottom of the hole. A common distance between the bottom of the hole and the instrument is forty feet.

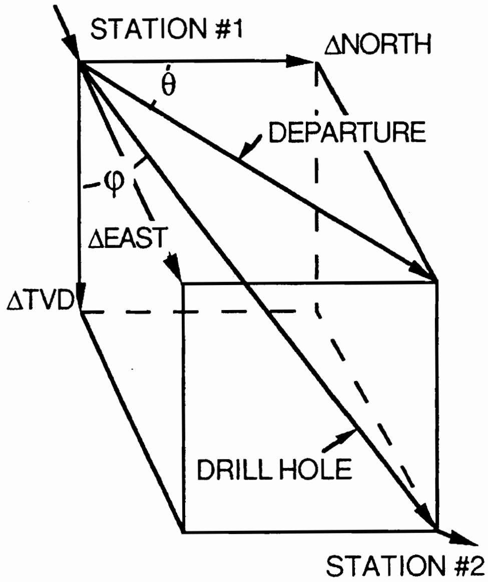
SURVEY CALCULATION METHODS

Because a device or instrument for locating a drill hole at practical depths within the earth does not exist, the method of dead reckoning must be employed for this purpose. The problem is that the instruments measure inclination and azimuth, and drillers report drill string lengths and these measurements are in the spherical coordinate system which must be converted to Cartesian coordinates for every day use. The transformation most often used in practice are called tangential, radius of curvature, and minimum curvature. However, the best of all may be the new "Long" method. These transforms are called "survey calculation methods" in the industry.

The tangential method calls for a straight line approximation of the wellbore between two stations and the inclination and direction assigned to the line are those recorded at the latter station. Similar assumptions appear in the average tangential method; however, the inclination and direction assigned are the average of those of the two stations. The radius of curvature method calls for a vertical and a horizontal arc of constant curvatures between two stations. The minimum curvature method connects two stations with a circular arc in an oblique plane.

TANGENTIAL METHOD

The backward tangential survey calculation method is the easiest of any method to compute. However, it is the least precise. The model assumes a drill hole is a series of straight segments with abrupt changes in direction and inclination at the ends of each segment. The sketch depicts the model.



The equations for the changes in displacement variables are

$\Delta TVD = MD \cos ø$

$Departure = MD \sin ø$

$\Delta N = Departure * \cos \theta = MD \sin ø \cos \theta$

$\Delta E = Departure * \sin \theta = MD \sin ø \sin \theta$

EXAMPLE TRAJECTORY DISPLACEMENTS

Compute the displacements between the two stations with the tangential method.

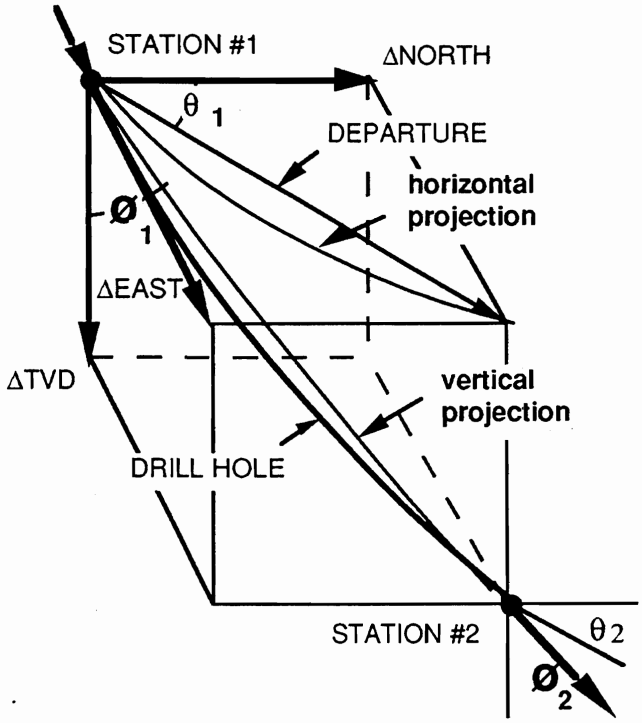
Sta. No.	Mea. Depth	Inclin.	Azimuth	ΔNorth	ΔEast	ΔTVD
31	821.09	23	56	---	---	---
32	912.28	25	59	19.85	33.03	82.65

MD	= 912.28 - 821.09	**= 91.19**
ΔTVD	= 91.19 * cos 25	**= 82.65**
ΔN	= 91.19 sin 25 * cos 59	**= 19.85**
ΔE	= 91.19 sin 25 * sin 59	**= 33.03**

NOTE: Other quadrants may required altered equations.

RADIUS OF CURVATURE

Wilson invented the "Radius of Curvature" survey calculation method to improve the precision of the modelling location of the drill hole. The model assumes a drill hole is a series of circular arc segments with abrupt changes in direction and inclination at the ends of each segment. The sketch depicts the model.

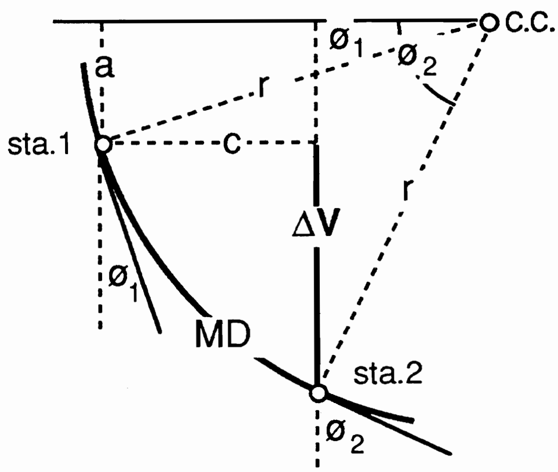


DERIVATION OF ROC

The following sketch is helps derive equation for computing the change in true vertical depth with the Radius of Curvature method. The sketch lies in a vertical plane. The projection of the drill hole, MD, lies between the two stations and is assumed to be a circular arc whose length is equal to the measured depth between the two stations.

The radius of curvature r is

$$r = \frac{180}{\pi}\frac{MD}{\Delta\phi} \qquad \Delta\phi = \phi_2 - \phi_1$$



The change in true vertical depth ΔV is

$$\Delta V = (a + \Delta V) - a$$

$$a = r \sin \phi_1$$

$$(a + \Delta V) = r \sin \phi_2$$

$$\Delta V = \frac{180}{\pi}\frac{MD}{\Delta\phi}(\sin \phi_2 - \sin \phi_1)$$

The equations for north/south N/S and east/west E/W displacements between stations may be derived in a similar manner and are

$$Dep = \frac{180}{\pi}\frac{MD}{\Delta\phi}(\cos\phi_2 - \cos\phi_1)$$

$$N/S = \frac{180}{\pi}\frac{Dep}{\Delta\theta}(\sin\theta_2 - \sin\theta_1)$$

$$E/W = \frac{180}{\pi}\frac{Dep}{\Delta\theta}(\cos\theta_2 - \cos\theta_1)$$

ϕ	= inclination at a station; degree
θ	= azimuth at a station; degree
r	= radius of curvature; feet
MD	= measured depth between stations; feet
Dep	= departure; feet
N/S	= north/south displacement; feet
E/W	= east/west displacement; feet
ΔV	= true vertical depth displacement; feet
$\Delta\theta$	$= \theta_2 - \theta_1$
$\Delta\phi$	$= \phi_2 - \phi_1$

EXAMPLE TRAJECTORY DISPLACEMENTS

Compute the displacements between the two stations with the Radius of Curvature method.

Sta. No.	Mea. Depth	Inclin.	Azimuth	ΔNorth	ΔEast	ΔTVD
31	821.09	23	56	---	---	---
32	912.28	25	59	19.93	31.28	83.30

$\Delta\theta = 59 - 56$ **= 3**

$\Delta\phi = 25 - 23$ **= 2**

$MD = 912.28 - 821.09$ **= 91.19**

$$Dep = \frac{180}{\pi}\frac{91.19}{2}(\cos 25 - \cos 23) \quad \mathbf{= 37.09}$$

$$\Delta TVD = \frac{180}{\pi}\frac{91.19}{2}(\sin 25 - \sin 23) \quad \mathbf{= 83.30}$$

$$\Delta N = \frac{180}{\pi}\frac{37.09}{3}(\sin 59 - \sin 56) \quad \mathbf{= 19.93}$$

$$\Delta E = \frac{180}{\pi}\frac{37.09}{3}(\cos 59 - \cos 56) \quad \mathbf{= 31.28}$$

NOTE: Other quadrants may require altered equations.

SURVEY TABLE

Declination 12 Target Azimuth N 71.42 E

St'n Nº	Meas'd Depth	Course Length	Drift Angle	Vert'l Depth	TrueVert Depth	Vert'l Section	Sect'n Diff.	Course Depart	Drift Direct'n	Diff. Angle	Coordinate Diff. North	South	East	West	Rect. Coordinates North	South	East	West
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19
TAN	4112		29		3982 61	1495 88			N 60 E						863 26		1287 92	
	4380	268	31	229 72	4214 33	1632 42	136 54	138 03	N 63 E	8.42	62 66		122 99		925 92		1410 91	
	4907		27.25						N 67 E	4.42								
	5326	419	27.25	372 50	5053 34	2063 59	190 59	191 85	N 78 E	6.58	39 89		187 66		1060 09		1820 69	
	5748	422	30	365 46	5418 80	2273 20	209 61	211 00	N 78 E	6.58	43 87		206 39		1103 96		2027 08	
	6001	253	31	216 86	5635 66	2395 18	121 98	130 30	S 88 E	20.58		4 55	130 22		1099 41		2157 30	
ROC	4112		29		3982 61	1495 88			N 60 E						863 26		1287 92	
	4380	268	31	232 08	4214 69	1627 85	131 97	133 99	N 63 E		63 93		117 74		927 19		1405 66	
	4907		27.25						N 67 E									
	5326	419	27.25	372 50	5047 20	2074 16	191 52	191 85	N 78 E		57 60		182 69		1093 33		1821 22	
	5748	422	30	370 39	5417 50	2274 98	200 82	202 15	N 78 E		42 03		197 73		1135 40		2019 11	
	6001	253	31	217 99	5635 49	2399 49	124 51	128 41	S 88 E		15 63		127 60		1151 3		2146 40	
Ave.	4112		29		3982 61	1495 88			N 60 E						863 26		1287 92	
TAN	4380	268	31	232 09	4214 70	1627 88	132 00	134 00	N 63 E		63 94		117 76		927 20		1405 68	
	4907		27.25						N 67 E									
	5326	419	27.25	372 50	5047 57	2047 59	191 82	191 85	N 78 E		57 69		182 97		1093 29		1821 12	
	5748	422	30	370 42	5417 99	2275 43	200 84	202 17	N 78 E		42 03		197 75		1135 32		2018 87	
	6001	253	31	217 99	5635 98	2400 25	124 82	128 41	S 88 E		11 19		127 92		1146 51		2146 79	

LONG'S METHOD AND MINIMUM CURVATURE METHOD

Roy Long invented a survey calculational method based on an oblique circular arc. He called it the "sectional" method. His final equations are identical to those of Taylor's "minimum curvature" method and give the same computed values as those of Zaremba's "circular arc" method.

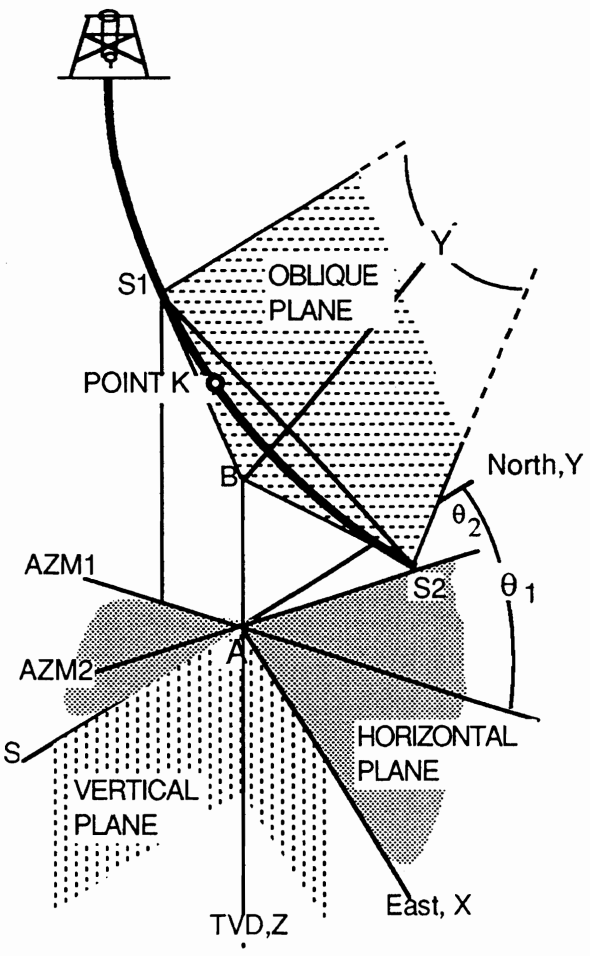
Long's equations are derived with solid analytical geometry whereas Taylor's method is derived with Calculus of Variation and Zaremba's uses matrix transformations.

He also presented a procedure for interpolating the directional variables of measured depth, azimuth, inclination, true vertical depth, and north and east coordinates between survey stations. He defined yet another dogleg severity equation which is most likely the best of all of them. API within their BULLETIN D20, "DIRECTIONAL DRILLING SURVEY CALCULATION METHODS AND TERMINOLOGY," publishes equations for the "minimum curvature" method.

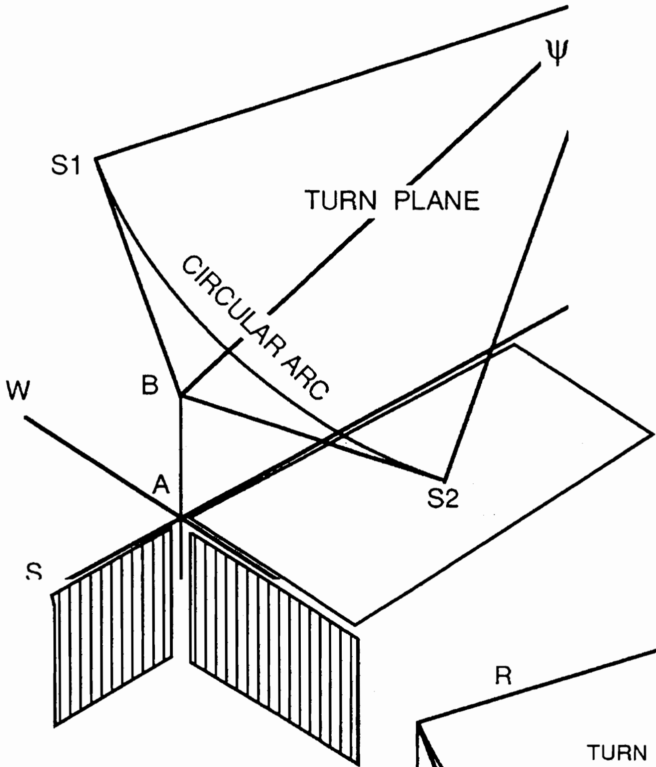
These equations are expected to give slightly different answers than Long's method because API sets a term which it calls a ratio factor equal to unity for angles between surveys which are less than 1/4 degree.

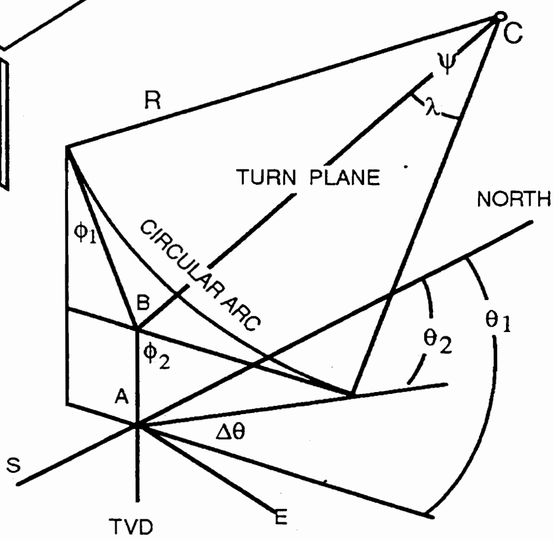
A complete explanation of the derivation of Long's method may be found in his paper; however, the following diagrams give the rudiments of the method. It is assumed that two survey stations, S1 and S2, at the ends of a section of drill hole which form a circular arc in an oblique plane, reside on co-planar tangents, S1-B and B-S2, to the drill hole. At the point, S1, the values of measured depth, azimuth, inclination, true vertical depth, north and east coordinates are known. At S2 only the values of measured depth, azimuth, and inclination are known. C is the center of curvature and Ψ is the angle between the radii.

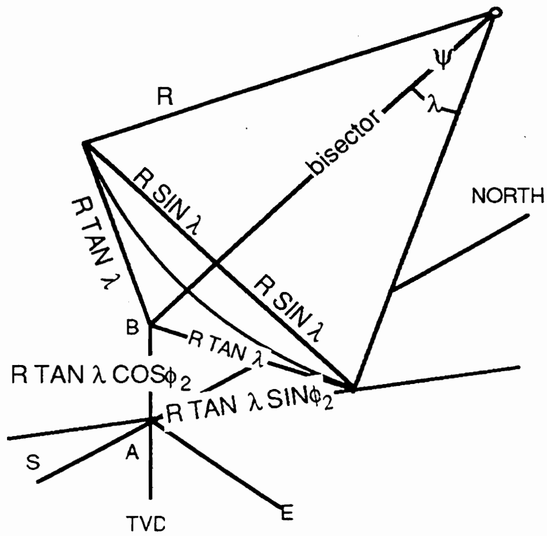
Long's interpolation equations gives the values of measured depth, azimuth, inclination, true vertical depth, north and east coordinates for any point, K, selected on the circular arc between the survey stations, S1 and S2.

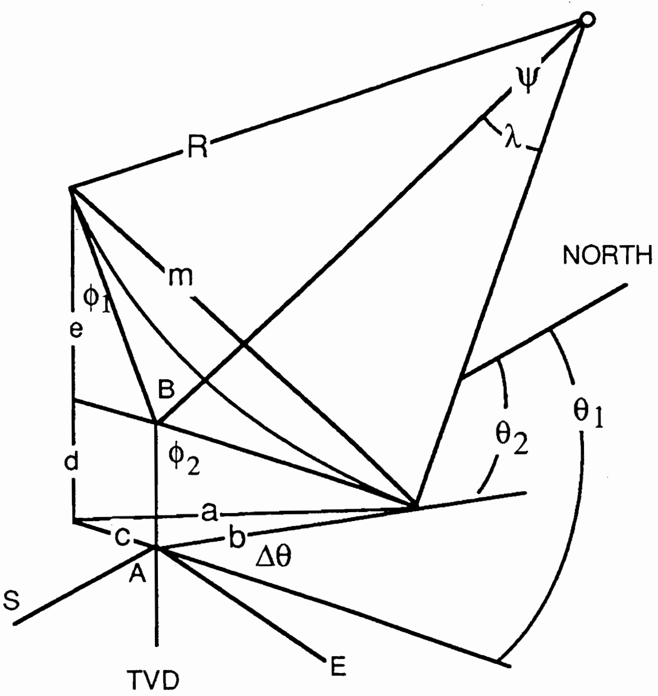


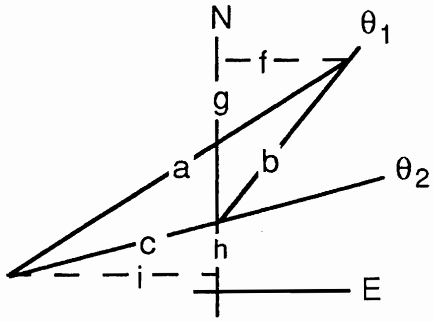
DERIVATION OF LONG'S EQUATION











$\Delta N = h + g$ $\qquad$ $\Delta E = f + i$

$g = b \sin\theta_1$ $\qquad$ $f = b \cos\theta_1$

$h = c \sin\theta_2$ $\qquad$ $i = c \cos\theta_2$

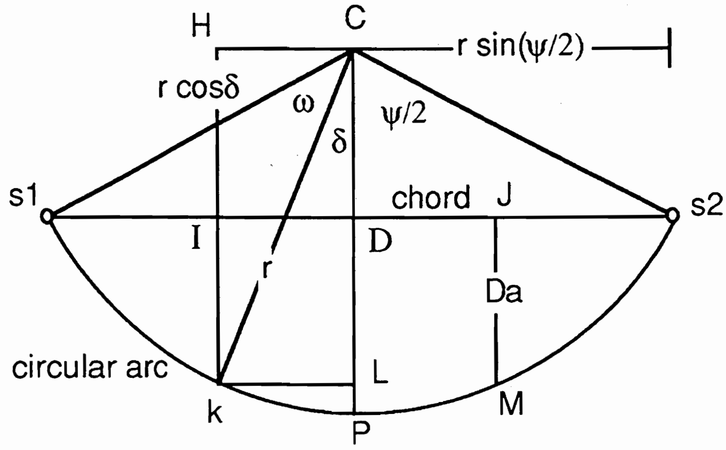
$d = R \tan\lambda \cos\phi_2$ $\qquad$ $e = R \tan\lambda \cos\phi_1$

$b = R \tan\lambda \sin\phi_2$ $\qquad$ $c = R \tan\lambda \sin\phi_1$

$$a = R \tan\lambda \sqrt{\sin^2\phi_1 + \sin^2\phi_2 + 2 \sin\phi_2 \sin\phi_1 \cos\Delta\theta}$$

$R = \Delta MD/\Psi$ $\qquad$ $\Delta TVD = d + e$

λ is found with the equation: $a^2 + (d + e)^2 = (2\,m)^2$



LONG'S COMPUTATIONAL EQUATIONS

The Long method equations for computing the coordinates at S2 are

$$\Psi = 2 \operatorname{acos} \sqrt{\frac{1 + \cos\phi_1 \cos\phi_2 + \sin\phi_1 \sin\phi_2 \cos|\Delta\theta|}{2}}$$

$$\Delta N/S = \frac{180\ \Delta MD \tan(\Psi/2)}{\pi\ \Psi} [\sin\phi_1 \cos\theta_1 + \sin\phi_2 \cos\theta_2]$$

$$\Delta E/W = \frac{180\ \Delta MD \tan(\Psi/2)}{\pi\ \Psi} [\sin\phi_1 \sin\theta_1 + \sin\phi_a \sin\theta_2]$$

$$\Delta TVD = \frac{180\ \Delta MD \tan(\Psi/2)}{\pi\ \Psi} [\cos\phi_1 + \cos\phi_2]$$

The dogleg severity equation is

$$DLS = \frac{200}{\Delta MD} \operatorname{acos} \sqrt{\frac{1 + \cos\phi_1 \cos\phi_2 + \sin\phi_1 \sin\phi_2 \cos|\Delta\theta|}{2}}$$

The equations for interpolating measured depth, azimuth, inclination, true vertical depth, north and east coordinates at any point K on the circular arc are

$$g = \frac{MD_K - MD_{S1}}{MD_{S2} - MD_{S1}}$$

$$h = \frac{1}{2} \left(1 - \frac{\sin[\Psi (0.5 - g)]}{\sin(\frac{\Psi}{2})} \right)$$

$$j = \frac{180\ \Delta MD}{\pi\ \Psi} \left[\cos\{ \operatorname{asin}[(1-2h)*\sin(\frac{\Psi}{2})] \} - \cos(\frac{\Psi}{2}) \right]$$

$$k = \sqrt{2 (1 - \cos\phi_1 \cos\phi_2 - \sin\phi_1 \sin\phi_2 \cos\Delta\theta)}$$

$$a = \frac{\sin\phi_1 \sin\theta_1 - \sin\phi_2 \sin\theta_2}{k}$$

$$b = \frac{\sin\phi_1\cos\theta_1 - \sin\phi_2\cos\theta_2}{k}$$

$$c = \frac{\cos\phi_1 - \cos\phi_2}{k}$$

$$\Delta N_k = h\,\Delta N + jb \qquad \Delta E_k = h\,\Delta E + ja \qquad \Delta TVD_k = h\,\Delta TVD + j\,c$$

$$\phi_k = \text{acos}\left[\frac{\Delta TVD_k}{\dfrac{180\,\Delta MD\,\tan(g\,\Psi/2)}{\pi\,\Psi}} - \cos\phi_1\right]$$

$$\theta_k = \theta_{S1} + \text{acos}\left[\frac{\left\{\dfrac{H}{\dfrac{180\,\Delta MD\,\tan(g\,\Psi/2)}{\pi\,\Psi}}\right\}^2 - \sin^2\phi_1\,\sin^2\phi_2}{2\,\sin\phi_1\,\sin\phi_2}\right]$$

Ψ	=	angle subtended by the circular arc; degree
ϕ_1	=	inclination at station #1; degree
ϕ_2	=	inclination at station #2; degree
θ_1	=	azimuth at station #1; degree
θ_2	=	azimuth at station #2; degree
$\Delta\theta$	=	change in azimuth between stations #1 and #2; degree
$\Delta N/S$	=	change in north or south coordinate; feet
$\Delta E/W$	=	change in east or west coordinate; feet
ΔMD	=	change in measured depth from stations #1 and #2; feet
ΔTVD	=	change in true vertical depth from stations #1 and #2; feet
DLS	=	dogleg severity between stations #1 and #2; deg/100ft
k	=	an interpolation point on the arc
ΔN_k	=	change in the north or south coordinate between station #1 and point k; feet
ΔE_k	=	change in the east or west coordinate between station #1 and point k; feet
ΔTVD_k	=	change in the true vertical depth between station #1 and point k; feet
ϕ_k	=	inclination at point k; deg
θ_k	=	azimuth at point k; deg
π	=	ratio of the circumference of a circle to its diameter

EXAMPLE of LONG'S METHOD

Compute the displacements and dogleg severity between the two stations with Long's method.

Sta. No.	Mea. Depth	Inclin.	Azimuth	ΔNorth	ΔEast	ΔTVD
31	900	10.75	225	-----	-----	-----
32	950	13.50	231	21.55	31.95	82.65

The value of pertinent variables are

$\Delta MD = 950 - 900$ $= \mathbf{50}$

$\Delta\theta = 231 - 225$ $= \mathbf{6^\circ}$

$\Delta\phi = 13.50 - 10.75$ $= \mathbf{2.75^\circ}$

The dogleg severity DLS is

$$DLS = \frac{200}{50}\,\mathrm{acos}\left(\sqrt{\frac{1}{2}[1+\cos(10.75)\cos(13.5)+\sin(10.75)\sin(13.5)\cos(6)]}\right)$$

DLS $= \mathbf{6.0429^\circ/100ft}$

The turn angle Y is

$$Y = \frac{6.0429 * 50}{100} \qquad = \mathbf{3.0215^\circ}$$

The displacement in the vertical direction ΔTVD is

$$\Delta TVD = \frac{180 * 50}{\pi * 3.0215}\tan\left(\frac{3.0215}{2}\right)[\cos(10.75) + \cos(13.5)]$$

ΔTVD $= \mathbf{48.88\ feet}$

The north/south displacement ΔN/S is

$$\Delta N/S = \frac{180*50}{\pi *3.0215}\tan\left(\frac{3.0215}{2}\right)[\sin(10.75)\cos(225) + \sin(13.5)\cos(231)]$$

ΔN/S $= \mathbf{-6.97\ feet}$

The east/west displacement $\Delta E/W$ is

$$\Delta E/W = \frac{180*50}{\pi *3.0215}\tan(\frac{3.0215}{2})[\sin(10.75)\sin(225)+\sin(13.5)\sin(231]$$

$\Delta E/W$ **= - 7.83 feet**

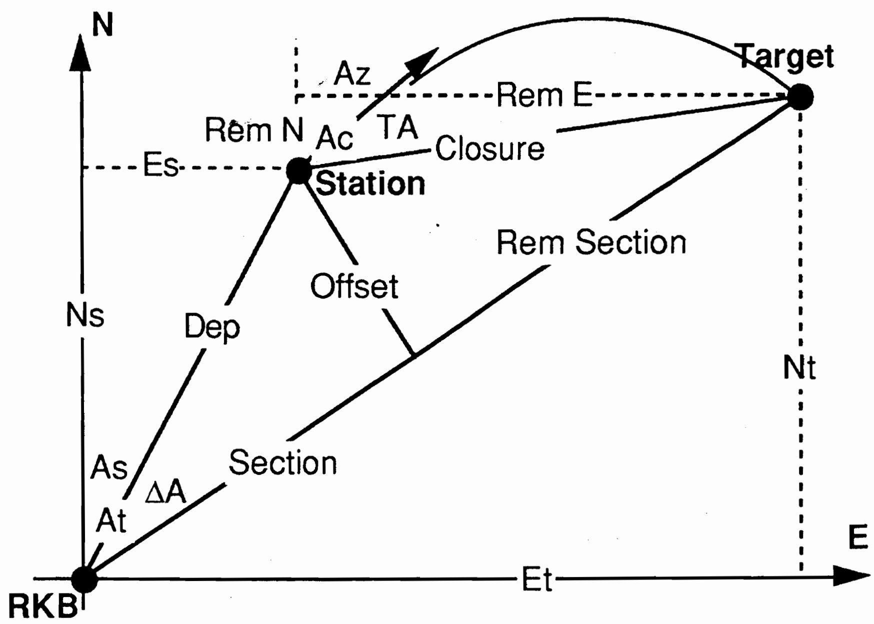
The following table is an example of the Long's method of computing the Cartesian coordinates of the stations of a drill hole.

TABLE OF SURVEY

STA	MD	INCL	AZIM	TVD	N+/S-	E+/W-	DLS
#	SURVEY TIE	POINT	====>	0.00	0.00	0.00	deg/100
1	0	0	0	0.00	0.00	0.00	
2	200	1	238	199.99	-0.92	-1.48	0.50
3	400	5	110	399.76	-4.83	5.24	2.84
4	600	7	208	599.05	-18.60	7.71	4.57
5	800	15	280	795.94	-24.90	-23.67	7.21
6	1000	19	165	991.17	-52.43	-41.09	14.32
7	1200	23	140	1178.22	-113.96	-7.47	4.86
8	1400	27	355	1370.54	-97.72	14.99	23.78
9	1600	31	358	1545.44	-0.98	9.24	2.13
10	1800	35	7	1713.24	107.53	14.43	3.16
11	2000	39	349	1873.41	226.64	9.40	5.75
12	2200	48	350	2018.34	361.88	-15.57	4.51
13	2400	55	353	2142.78	516.59	-38.48	3.69
14	2600	65	350	2242.67	687.61	-64.28	5.17
15	2800	68	351	2322.41	868.48	-94.52	1.57
16	2900	70	355	2358.26	961.12	-105.88	4.24

MISCELLANEOUS DIRECTIONAL SURVEY VARIABLES

DIRECTIONAL SURVEY VARIABLES



The following sketch, table, and equations show, present, and explain the computation of often quoted miscellaneous directional survey variables. Of particular interest is the minimum three dimensional turn gradient required to intersect the target, M3dTGTT. The M3dTGTT is a plane circular turn totally in an oblique plane between a station and the target. For example the M3dTGTT required for station number 4 using the Long method is a curvature of 0.47 degrees per 100 feet of arc length along the curve. A lesser curvature can not intersect the target while a greater curvature will fall short of the target.

It is also of interest to note that over a measured depth, MD, of 270 feet that the difference between the true vertical depths with the tangential and the Long methods is 5.25 feet while the differences in the north and east coordinates is a little over two feet.

TABLE OF SURVEY COMPUTATIONS

Target: N = +3,612 E = +4,173 TVD = +5,638

1	2	3	4	5	6	7	8	9	10	11	12
Sta. no.	ΔMD	I	Az	ΔN	ΔE	ΔTVD	N	E	TVD	MD	Dep
#	ft	deg	deg	ft	ft	ft	ft	ft	ft	ft	ft
Tang't		32	42				548.00	341.00	989.00	1684.00	
2	91	34	39	39.55	32.02	75.44	587.55	373.02	1064.44	1775.00	695.96
3	89	36	40	40.07	33.63	72.00	627.62	406.65	1136.44	1864.00	747.84
4	90	37	43	39.61	36.94	71.88	667.23	443.59	1208.32	1954.00	801.23
Long		32	42				548.00	341.00	989.00	1684.00	
2	91	34	39	37.70	32.15	76.32	585.70	373.15	1065.32	1775.00	694.47
3	89	36	40	39.38	32.48	72.90	625.08	405.63	1138.22	1864.00	745.16
4	90	37	43	40.07	35.48	75.35	665.15	441.10	1213.57	1954.00	798.13

The table is continued on the following page.

Sta. no. #	13 Dep Az deg	14 Cum. Dep ft	15 Section Angle deg	16 Section ft	17 Offset ft	18 Rem Sect. ft	19 Clo. Angle deg	20 Clo. to target ft	21 Rem. N ft	22 Rem. E ft
Tang't		717.00								
2	32.4108	767.75	16.7109	666.57	200.12	4852.53	2.3615	4856.66	3024.45	3799.98
3	32.9402	820.20	16.1815	718.22	208.41	4800.88	2.4857	4805.40	2984.38	3766.35
4	33.6167	874.36	15.5050	772.07	214.19	4747.03	2.5834	4757.86	2944.77	3729.41
Long		717.00								
2	32.5019	766.04	16.6198	665.46	198.63	4853.64	2.3435	4857.70	3026.30	3799.84
3	32.9812	816.73	16.1405	715.79	207.15	4803.31	2,4694	4807.77	2986.92	3767.36
4	33.5519	869.70	15.5698	768.84	214.23	4750.26	2.5822	4755.09	2946.85	3731.88

Sta. no. #	23 Clo. Az deg	24 Hor. Turn Angle deg	25 MHTGTT deg/100ft	26 M3dTGTT deg/100ft
Tang't				
2	51.4832	12.4832	0.51	0.45
3	51.6074	11.6074	0.48	0.40
4	51.7051	8.7051	0.36	0.35
Long				
2	51.4652	12.4652	0.51	0.45
3	51.5912	11.5912	0.48	0.41
4	51.7039	8.7039	0.36	0.37

Target Azimuth $= \text{atan}\left(\frac{E}{N}\right)$ $= \text{atan}\left(\frac{4173}{3612}\right)$ $= 49.1217$

Target Section $= \sqrt{N^2 + E^2}$ $= \sqrt{3612^2 + 4173^2}$ $= 5519.10$

ΔMD $= 91$

Inclination $= 34$

Azimuth $= 39$

ΔN = MD sin I cos Az = 91 sin 34 cos 39 = 39.55

ΔE = MD sin I sin Az = 91 sin34 sin 39 = 32.02

ΔTVD = MD cos I = 91 cos 34 = 75.44

N = ΔN + N = 39.54 + 548 = 587.55

E = ΔE + E = 32.02 + 341 = 373.02

TVD = ΔTVD + TVD = 75.44 + 989 = 1064.44

MD = sum of ΔMD = 91 + 1684 = 1775

ΔDep $= \sqrt{(N_2 - N_1)^2 + (E_2 - E_1)^2}$ $= \sqrt{(587.55 - 548)^2 + (373.02 - 341)^2}$ $= 50.72$

Cum. Dep = sum of ΔDep = 717.00 + 50.72 = 767.72

Dep $= \sqrt{N^2 + E^2}$ $= \sqrt{587.55^2 + 373.02^2}$ $= 695.96$

Dep Azimuth	$= \text{atan}\left(\frac{E}{N}\right)$	$= \text{atan}\left(\frac{373.02}{587.55}\right)$	= 32.4108
Section Angle	= | Dep Az - Tar Az |	= 32.4109 - 49.1217	= 16.7109
Section	= Dep * cos (Sec Ang)	= 695.96 * cos(16.7109)	= 666.57
Offset	= Sec tan (Sec Angle)	= 666.57 * tan(16.7109)	= 200.12
Remaining Section	= Tar Sec - Sec	= 5519.10 - 666.57	= 4852.53
Closure Angle to Target	$= \text{atan}\left(\frac{\text{Offset}}{\text{Rem Sec}}\right)$	$= \text{atan}\left(\frac{200.12}{4852.53}\right)$	= 2.3616
Closure to Target	$= \frac{\text{Offset}}{\sin(\text{Clo Angle})}$	$= \frac{200.12}{\sin(2.3616)}$	= 4856.65
Remaining N	= N target - N	= 3612 - 587.55	= 3024.45
Remaining E	= E target - E	= 4173 - 373.02	= 3799.98
Closure Az to Target	$= \text{atan}\left(\frac{\text{Rem E}}{\text{Rem N}}\right)$	$= \text{atan}\left(\frac{3799.98}{3024.45}\right)$	= 51.4833
Hor. Turn Angle	= Hole Az - Clo Az	= 39 - 51.4833	= 12.4832
Min. Hor. Turn Grd. to Target	= Mitchell Engr. Prog.: Lead_Tgt.XLS		= 0.51 deg/100ft
Min. 3d Turn grd. to Target	= Mitchell Engr. Prog.: 3d_Plan.XLS		= 0.45 deg/100ft

The definitions of most of these miscellaneous variables can be ascertained from the sketch. Those definitions which require enhancement are listed in the following.

N and E: North and East coordinates; South is negative N; West is negative E.

ΔDep: Horizontal departure between adjacent stations.

Total Dep: Sum of the individual departures between adjacent stations; ΣΔDep.

Hor. Turn Angle: Required horizontal turn of the hole at a station to point the hole at the target.

Min. Hor. Turn Grd. to Target, Minimum Horizontal Turn Gradient to Target, MHTGTT: Minimum curvature at which the hole must be turned from a station in order to intersect the target. See Mitchell Engineering Programs, LEAD_TGT.XLS.

Min. 3d Turn grd. to Target, Minimum Three Dimensional Turn Gradient to Target; M3dTGTT: Minimum curvature of the hole in an oblique plane from a station in order to intersect the target. See Mitchell Engineering Programs, 3d_PLAN.XLS.

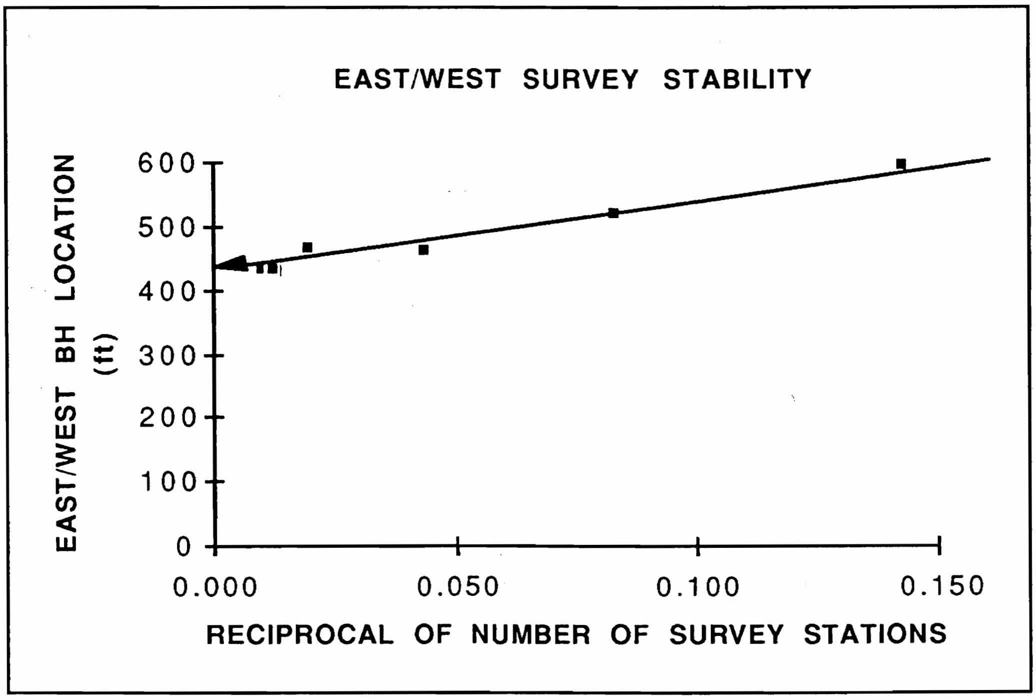
STABILITY OF COMPUTATIONAL SURVEYS

Stability of a trajectory calculation method refers to the change in the location of the trajectory as survey stations are deleted or added to the directional survey. Thus, the higher the stability, the more reliable the prediction of the location of the borehole.

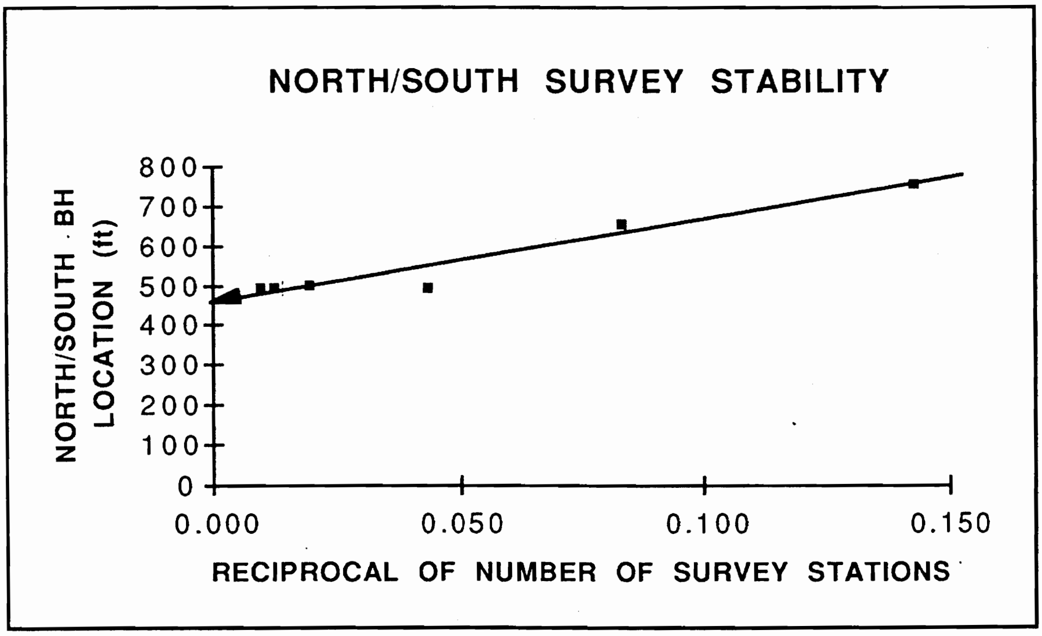
The following tabulated survey data was acquired with an Eastman Whipstock SEEKER tool in a New Mexico borehole. The horizontal view is shown in the figure. Within the table the "No. of Sta." refers to the quantity of stations not deleted from the original number of 101 stations. For example, the values of TVD, N/S, and E/W for the 81 station line of data, means that 20 stations were regularly removed from the original 101 stations. The number within the table all represent the bottom hole location for a measured depth of 10,000 feet.

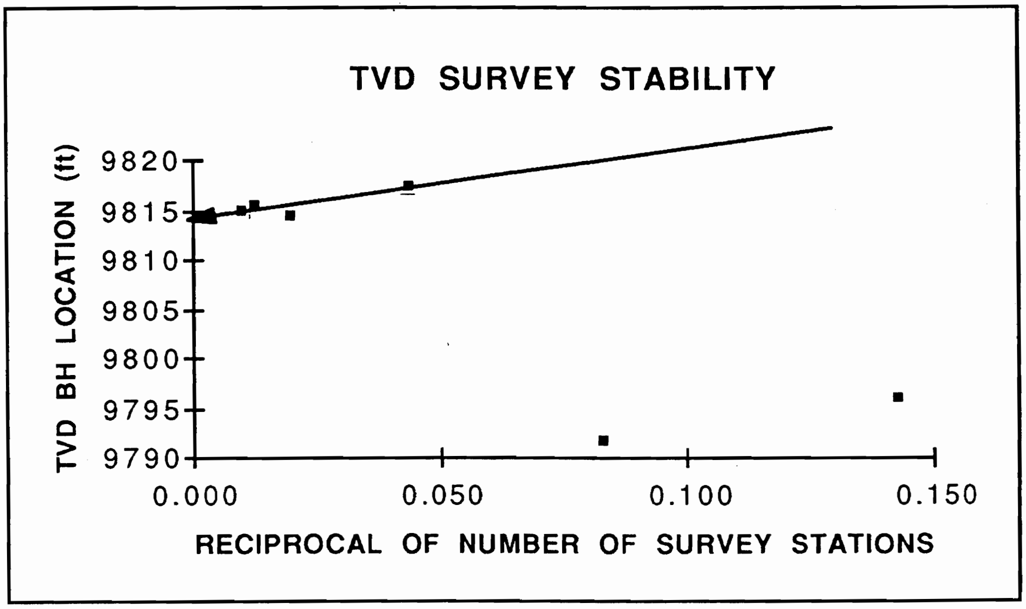
TABLE OF BOTTOM HOLE LOCATIONS

No. of Sta.	TVD	N/S	E/W
101	9814.96	491.09	433.30
81	9815.41	493.59	430.14
51	9814.51	498.67	464.92
23	9817.55	488.22	463.28
12	9791.67	649.75	518.15
7	9795.89	755.66	593.61



Constructing a graph such as the one above, presents a visual reference for the stability of the survey and allows the selection of the most likely location of the survey station at 10,000 feet (the bottom of the hole, in this example). A least squares fit of the 'reciprocal of number of survey stations' and 'east/west bh location' data and extrapolation of the resulting line (y =414.250 + 1245.167 x) to a value of zero, gives the value of 414.25 feet. This is an approximation of the value of the east/west coordinate of the station at 10,000 feet if an infinite number of survey stations had been made with the original survey. The same reasoning applies to the 'best estimates' of TVD and north/south locations.





The following table presents a computation with the Long method of the above survey station data. Columns subscripted with an 'o' contain the original, unaltered data and those columns subscripted with an 'r' contain data which have had alternating survey stations deleted.

TABLE OF REDUCED NUMBER OF STATIONS

MD	TVD(Ft)		EAST(Ft)		NORTH(Ft)		DIFFER	DISP'MENT(N,E)		Λ DISPL.
(Ft)	TVDo	TVDr	EASTo	EASTr	NORTHo	NORTHr	-ENCE	ORIG'L	LESS	(N,E)
===	===	===	===	===	===	===	===	===	===	===
0	0	0	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
200	200	200	-1.48	-1.48	-2.04	-0.92	1.12	2.52	1.74	0.78
400	400	400	2.18	5.24	-5.72	-4.83	3.19	6.12	7.13	1.01
600	599	599	11.41	7.71	-24.01	-18.60	6.58	26.58	20.13	6.45
800	794	796	-23.75	-23.67	-38.42	-24.90	13.63	45.17	34.36	10.81
1000	989	991	-44.81	-41.09	-25.39	-52.43	27.36	51.50	66.61	15.11
1200	1178	1178	1.65	-7.47	-70.04	-113.96	44.86	70.06	114.20	44.15
1400	1364	1371	12.16	14.99	-19.79	-97.72	78.24	23.23	98.86	75.64
1600	1539	1545	15.20	9.24	76.75	-0.98	78.21	78.24	9.29	68.95
1800	1707	1713	20.65	14.43	185.37	107.53	78.35	186.52	108.49	78.02
2000	1867	1873	8.72	9.40	304.31	226.64	77.96	304.43	226.83	77.60
2200	2011	2018	-15.81	-15.57	440.27	361.88	78.73	440.55	362.21	78.34
2400	2138	2143	-37.93	-38.48	593.44	516.59	77.03	594.65	518.02	76.63
2600	2237	2243	-65.85	-64.28	764.13	687.61	76.72	766.96	690.61	76.35
2800	2312	2322	-103.73	-94.52	945.87	868.48	78.69	951.54	873.61	77.93
2900	2347	2358	-115.08	-105.88	1038.50	961.12	78.68	1044.86	966.93	77.92

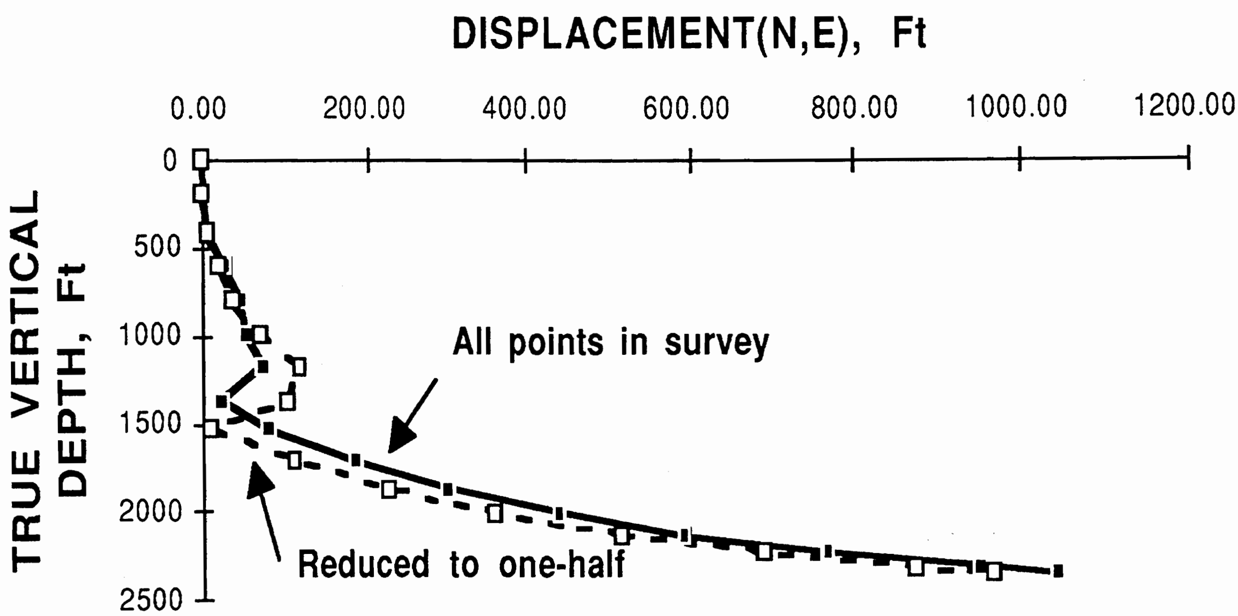
DIFFERENCE $= \sqrt{\Delta N^2 + \Delta E^2 + \Delta V^2}$

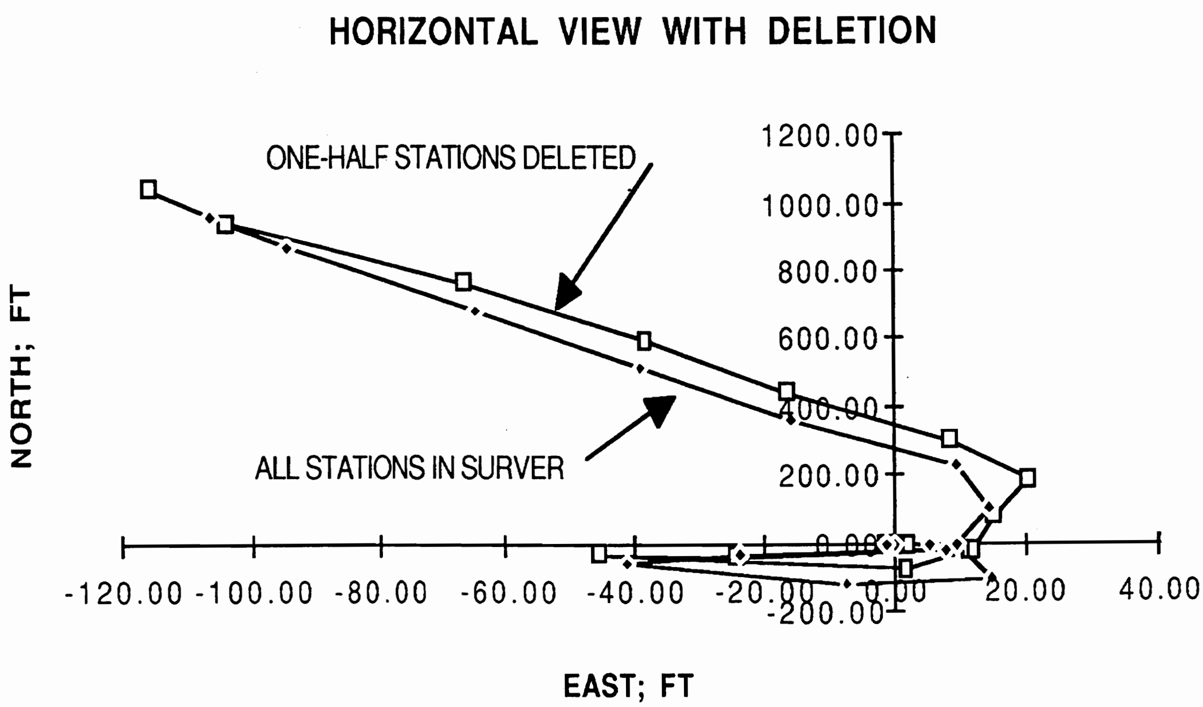
$\Delta N, \Delta E$ = difference in original and reduced surveys

DISPLACEMENT $= \sqrt{\Delta N^2 + \Delta E^2}$

LESS = Survey reduced by deleting alternating stations

DRILL HOLE LOCATION WITH STATION DELETION

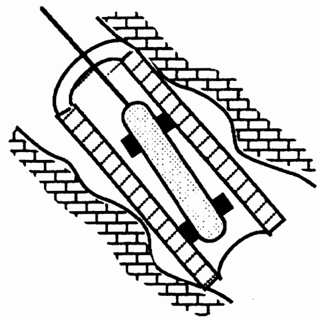




DIRECTIONAL SURVEY

Sta No.	MD ft	INCL deg	AZIM deg
1	0	0	0
2	100	0.167	41.85
3	200	0.317	56.45
4	300	0.37	47.53
5	400	0.25	0.717
6	500	0.233	329.517
7	600	0.38	311.111
8	700	1.517	300.65
9	800	1.8	288.767
10	900	1.867	294.883
11	1000	1.78	288.52
12	1100	1.667	288
13	1200	1.6	286.017
14	1300	1.433	288.55
15	1400	1.35	292.05
16	1500	1.133	284.633
17	1600	0.883	286.933
18	1700	0.7	292.05
19	1800	0.58	290.62
20	1900	0.6	294.467
21	2000	0.533	266.65
22	2100	0.55	247.783
23	2200	0.43	234.53
24	2300	0.167	330.05
25	2400	0.5	15.767
26	2500	0.983	27.083
27	2600	0.85	38.52
28	2700	1	39.8
29	2800	1.067	45.95
30	2900	0.733	34.75
31	3000	0.83	8.32
32	3100	1.017	334.5
33	3200	1.083	326.617
34	3300	1.4	312.25
35	3400	1.783	306.033
36	3500	2.01	299.15
37	3600	2.6	294.383
38	3700	3.483	291.1
39	3800	3.867	291
40	3900	4.15	290.583
41	4000	4.37	289.65
42	4100	4.45	289.733
43	4200	4.5	288.683
44	4300	4.583	288.817
45	4400	4.817	287.85
46	4500	5.45	284.25
47	4600	5.533	285.883
48	4700	5.6	281.017
49	4800	5.7	278.233
50	4900	5.85	277.617
51	5000	5.9	280.45
52	5100	6.067	281.783
53	5200	6.183	279.567
54	5300	6.3	281.7
55	5400	6.333	282.717
56	5500	6.47	284.7
57	5600	6.417	286.167
58	5700	6.583	288.533
59	5800	6.933	287.833
60	5900	7.283	285.8
61	6000	8.32	290.03
62	6100	9.1	292.417
63	6200	9.467	293.517
64	6300	9.85	294.283
65	6400	9.9	295.767
66	6500	9.28	301.52
67	6600	12.333	311.25
68	6700	14.067	314.983
69	6800	14.283	331.85
70	6900	14.467	348.767
71	7000	14.42	5.23
72	7100	13.25	21.3
73	7200	12:367	29.967
74	7300	10.667	44.8
75	7400	8.517	59.4
76	7500	7.28	77.48
77	7600	8.417	75.583
78	7700	10.367	62.2
79	7800	13.567	46.783
80	7900	15.317	46.267
81	8000	16.15	56.95
82	8100	16.567	65.217
83	8200	16.733	78.75
84	8300	16.933	83.183
85	8400	17.167	86.1
86	8500	17.27	87.45
87	8600	17.4	86.983
88	8700	17.317	85.583
89	8800	17.35	84.983
90	8900	19.317	80.667
91	9000	21.83	76.65
92	9100	22.283	77.383
93	9200	22.533	74.55
94	9300	22.367	77.85
95	9400	22.533	69.55
96	9500	23.17	73.75
97	9600	24.183	75.967
98	9700	24.967	78.533
99	9800	24.75	79.5
100	9900	25.611	77.633
101	10000	28.31	78.13

ERRORS IN SURVEYING



Errors intrinsic with directional wellbore surveying may be placed in eight major categories: (1) reading of directional survey film discs for both direction and inclination, (2) mechanical malfunctions and calibration errors of survey instruments, (3) alignment of instruments in a BHA and alignment of BHA's in the drill hole, (4) drillstring length measurements, (5) approximations inherent in the various models of computing surveys, (6) natural magnetic interference and declination errors with survey compasses, (7) magnetic interference caused by "hot spots" in non-magnetic collars and the length of non-magnetic collars, and (8) numerical calculations and recording of data. Poor surface orientation, drift correction, and precession errors are inherent in gyro surveys. Also, errors are further categorized as systematic and random. Warren has noted errors of 1.0 degrees in azimuth of service company master compasses and an error of 2.5° in one single shot tool. Wolff and deWardt report errors of 2° in instruments and errors of .5° in world declination charts. Walstrom introduced the much discussed "ellipse-of-uncertainty."

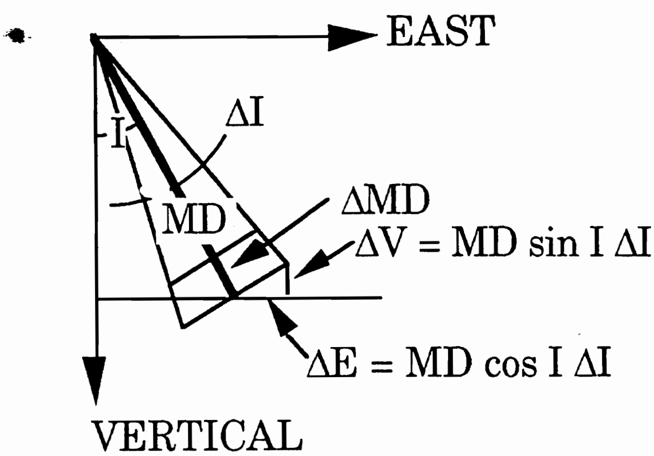
Wolff and deWardt's systematic error categories are

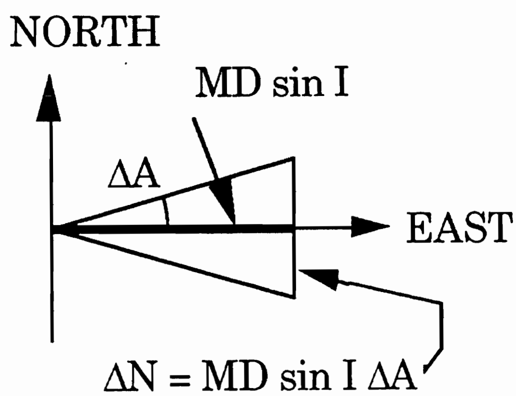
1. relative depth	4. reference
2. misalignment	5. drillstring magnetization
3. true inclination	6. gyro-compass

ELLIPSE OF UNCERTAINTY

The concept of an ellipse at any point in a drill hole to quantify the possible locations of the point is illustrated with the following example.

Suppose the measured depth is 2500 feet as shown in the sketch and that systematic errors (errors which regularly re-occur and are not compensating) at any survey station are **1° in azimuth, 0.5° in inclination,** and 1 part in 500 for measured depth. Further, suppose the inclination and azimuth are thought to be 30° and 90°, respectively.





The expected location of the bottom of the drill hole is

$\Delta V = 2500 * \cos 30$ **= 2165 ft**

$\Delta N = 2500 * \sin 30 \cos 90$ **= 0 ft**

$\Delta E = 2500 * \sin 30 \sin 90$ **= 1250 ft**

The actual location of the bottom of the drill hole could be anywhere within a volume shown as an ellipsoid in the sketch. The dimensions of the ellipsoid are given by the variations of variables of inclination, azimuth, and measured depth. The maximum dimensions of the ellipsoid and consequently the position of uncertainty which is called the "ellipse of uncertainty" are

$\Delta N = 2500 * \sin 30 * \frac{\pi}{180} 1.0$ **= 21.82 ft**

$\Delta E = 2500 * \cos 30 * \frac{\pi}{180} 0.5$ **= 18.89 ft**

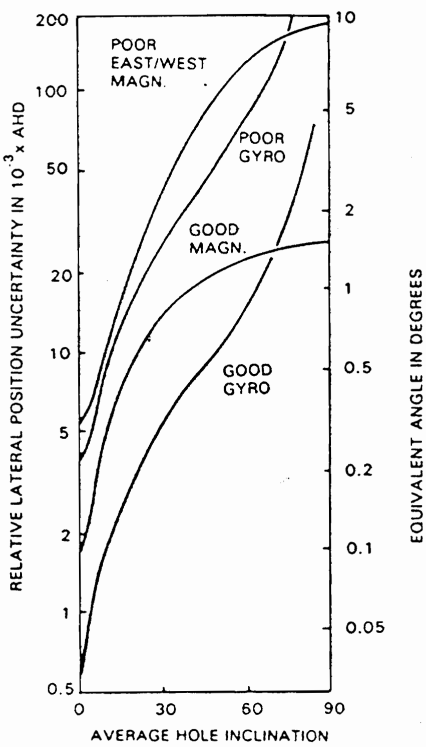
$\Delta V = 2500 * \sin 30 * \frac{\pi}{180} 0.5$ **= 10.91 ft**

SYSTEMATIC AND RANDOM ERRORS

Wolff and deWardt published the following table which presents their finding in the study of surveying errors in the North Sea area. They also wrote that surveying errors are more systematic rather than random. At the time they worked for Shell Oil.

Typical Values For Measuring Errors

	Relative Depth e (10^{-3})	Misalignment ΔI_m (degrees)	True Inclination ΔI_{to} (degrees)	Reference Error ΔC_{10} (degrees)	Drillstring Magnetization ΔC_{20} (degrees)	Gyrocompass ΔC_{30} (degrees)
Good gyro	0.5	0.03	0.2	0.1	-	0.5
Poor gyro	2.0	0.2	0.5	1.0	-	2.5
Good Magnetization	1.0	0.1	0.5	1.5	0.25	-
Poor Magnetization	2.0	0.3	1.0	1.5	5.0 + 5.0	-
Weighting	1	1	sin I	sin I	sin I sin A	$(\cos I)^{-1}$



Typical lateral position uncertainties of inclined wells for poor and good, magnetic and conventional gyro surveys.

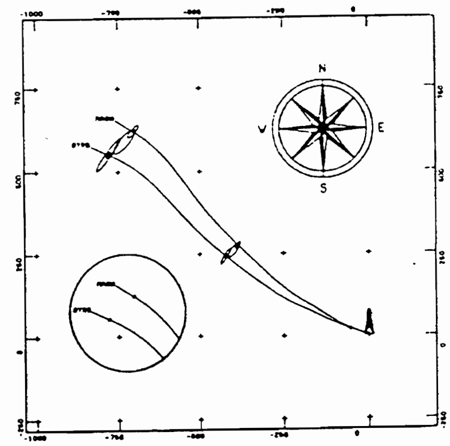


Fig. 4 – Demonstration plot of systematic-error, ellipse-of-uncertainty computer program. (Insert shows random error model results.)

EXAMPLE OF ERROR OF BOREHOLE LOCATION

Use Wolff and deWardt's chart to estimate the lateral location error of a borehole with the following inclinations. Assume a good magnetic multishot survey.

SECTION	DEPTH ft	INCLINATION deg	ERROR ft
Vertical	0 to 3000	0	.0018 * 3000 = 5.40
Build	3000 to 5000	0 to 40	.0095 * 2000 = 19.00
Slant	5000 to 9000	40	.0170 * 4000 = 68.00

LOCATION ERROR AT BOTTOM OF BOREHOLE = **92.40**

SYSTEMATIC AND RANDOM ERRORS BY WARREN

Warren (Amoco) presents analysis of sixteen surveys run on a drill hole which had blown-out and two side track holes near Baton Rouge, Louisiana. The following table and graphs summarize his findings

VALUES OF DIRECTIONAL SURVEYING ERRORS

CONFIDENCE	SYSTEMATIC ERROR INCLINATION	SYSTEMATIC ERROR AZIMUTH	RANDOM ERROR INCLINATION	RANDOM ERROR AZIMUTH
1 STD. DEV.	±0.2°	±1.4°	±0.13°	±0.55°
2 STD. DEV.	±0.4°	±2.9°	±0.26°	±1.10°

The following is a plan view showing ten surveys of the first side track drill hole. Note that Warren places the original drill hole within 25 feet of the MWD survey.

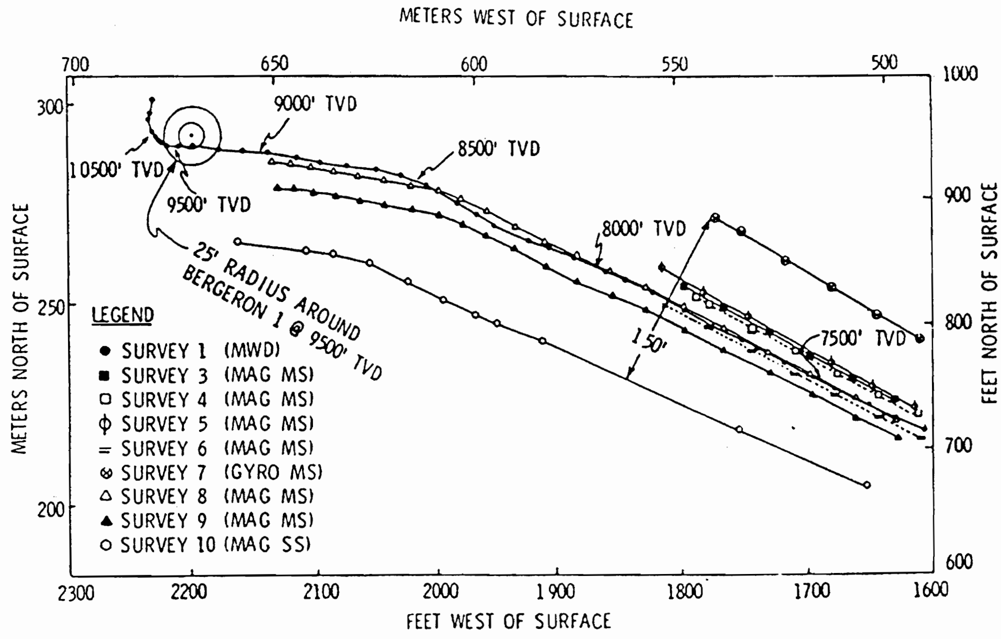
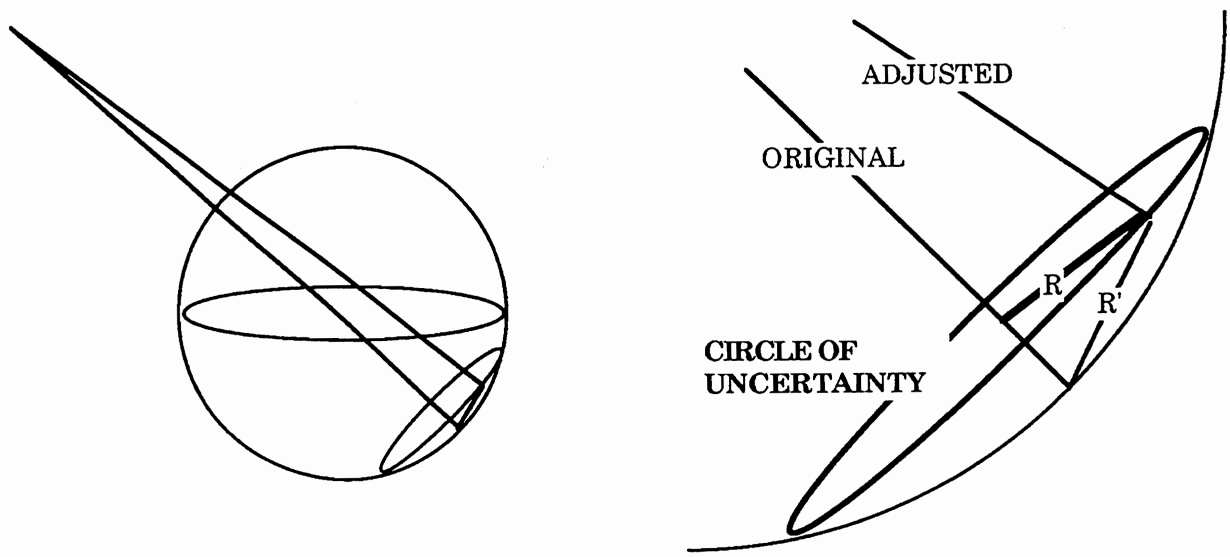


Fig. 3 – Plan view of the Bergeron 2-A from raw surveys

CIRCLE OF UNCERTAINTY

The circle of uncertainty can be generated by first recalculating a directional survey with tolerances of inclination and azimuth added to the values of inclination and azimuth of the original survey. Then the approximation to the radius, R', of the circle of uncertainty is the distance between two similar stations computed with each of the surveys. The radius, R, is within an oblique plane which is perpendicular to the line of borehole just above the original station. A projection of a circle drawn with the radius onto a horizontal plane will be an ellipse. The shorter the spacing between two stations in the original survey and smaller the tolerances, the more precise the approximation of the radius of the circle. The sketch shows the details of the circle.



Suppose the axes of the ellipse of uncertainty is desired at the 45th station from the KOP (3,200 ft MD) in the above survey. Let the expected error in inclination be 0.4 degrees and the error in azimuth be 2.9 degrees as suggested by Warren. The original survey data places the station #45 at 26.55N and 64.25W. The adjusted survey data places station #45 at 33.71N and 64.25W. The value of the radius of the circle of uncertainty at the 45th station is 9.23 feet.

ORIGINAL DIRECTIONAL SURVEY

Sta No.	MD ft	INCL deg	AZIM deg	TVD ft	N/S ft	E/W ft	RADIUS ft
33	3200	1.083	326.617	3200.00	0.00	0.00	0.00
34	3300	1.4	312.25	3299.98	1.61	-1.42	0.00
35	3400	1.783	306.033	3399.94	3.35	-3.59	0.00
36	3500	2.01	299.15	3499.88	5.12	-6.38	0.00
37	3600	2.6	294.383	3599.80	6.91	-9.97	0.00
38	3700	3.483	291.1	3699.66	8.94	-14.87	0.00
39	3800	3.867	291	3799.45	11.24	-20.86	0.00
40	3900	4.15	290.583	3899.21	13.72	-27.39	0.00
41	4000	4.37	289.65	3998.93	16.27	-34.37	0.00
42	4100	4.45	289.733	4098.64	18.86	-41.61	0.00
43	4200	4.5	288.683	4198.33	21.43	-48.97	0.00
44	4300	4.583	288.817	4298.02	23.98	-56.47	0.00
45	4400	4.817	287.85	4397.68	26.55	-64.25	0.00

ADJUSTED POSITIVE DIRECTIONAL SURVEY (+0.4° Incl & +2.9° Azim)

Sta No.	MD ft	INCL deg	AZIM deg	TVD ft	N/S ft	E/W ft	RADIUS ft
33	3200	1.083	326.617	3200.00	0.00	0.00	0.00
34	3300	1.4	312.25	3299.96	2.23	-1.76	0.71
35	3400	1.783	306.033	3399.90	4.54	-4.35	1.42
36	3500	2.01	299.15	3499.82	6.85	-7.62	2.13
37	3600	2.6	294.383	3599.71	9.17	-11.73	2.86
38	3700	3.483	291.1	3699.53	11.84	-17.14	3.61
39	3800	3.867	291	3799.27	14.63	-23.64	4.39
40	3900	4.15	290.583	3898.98	17.72	-30.68	5.18
41	4000	4.37	289.65	3998.65	20.89	-38.16	5.98
42	4100	4.45	289.733	4098.29	24.11	-45.90	6.79
43	4200	4.5	288.683	4197.93	27.31	-53.77	7.60
44	4300	4.583	288.817	4297.56	30.49	-61.78	8.41
45	4400	4.817	287.85	4397.16	33.71	-70.06	9.23

Reading Errors

An experiment conducted with 34 students (each with 3 years of engineering training) who were asked to read and record independently with the aid of a ten power magnifier (standard reader) the direction and inclination on a survey film disc demonstrated that the expected direction value read would be 0.75 degree of the average value read by the group 95% of the time read and 0.5 degree for the inclination angle. The disk could record inclination angle to a maximum of 10 degrees.

A practical application is illustrated in the following problem. Suppose in a slant well as in the figure the planned slant portion to the target is 10,000 feet with and inclination angle of 45 degrees; then if only one survey were taken at the beginning of the slant portion, the displacement in the wellbore at the target caused by error in direction would be

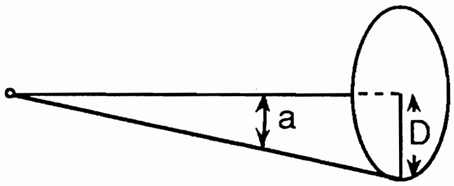
$$e = 10{,}000 * \cos 45 * \sin .75 \qquad = \mathbf{92.6\ feet}$$

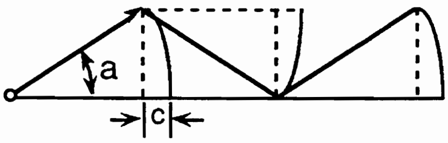
and the displacement caused by inclination error would be

$$e = 10{,}000 * \sin .05 \qquad = \mathbf{87.3\ feet}$$

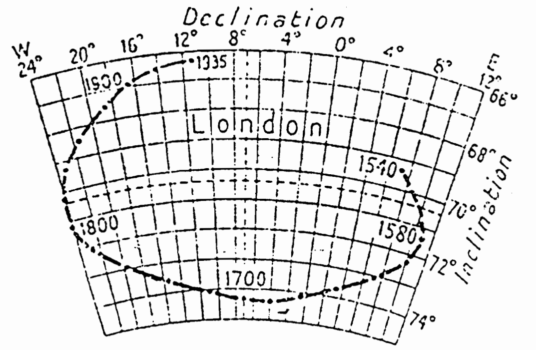
Suppose as in the sketch, surveys are recorded every 100 feet in the above well and that the errors are compensating; that is, there are an equal number of positive and negative errors of equal value and are alternating in sign: Then the largest displacement caused by direction error would be

$$e = \frac{10000}{100} * \sin 0.5 \qquad = \mathbf{0.873\ feet}$$





These values of displacement are acceptable for the precision required for most directionally drilled wellbores; and as expected, the probable error in both direction and inclination decreases with an increase in the number of survey stations.



The direction of the magnetic force at London since A.D. 1580.
(After L. A. Bauer)

DECLINATION CHANGES

Bauer published the sketch showing the direction of the earth's magnetic force at London, England, from year 1540 to 1935. Note the maximum change in declination was 35° and 8° in inclination.

DRILL STRING MEASUREMENTS

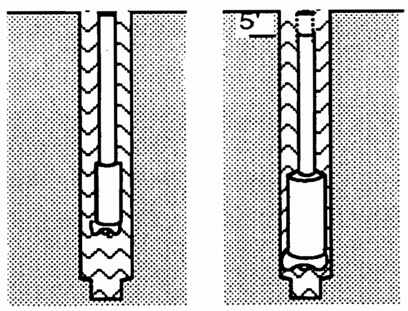
Perhaps the best procedure for measuring the length of a drillstring is to strap the string while pulling it out of the wellbore with the string hanging on the elevators and after rotating each stand free of wellbore drag. In vertical holes this procedure should very accurately give the length of the drillstring and then the measured depth of the wellbore and it should be sufficiently accurate for most directional wellbores if not precisely correct in principle.

EXAMPLE CHANGE IN LENGTH OF BHA & PIPE

Changes in drilling fluid density and in bottom hole assembly will have only a minor effect on drillstring length as demonstrated in the following problem. Suppose a drillstring is composed of 10,000 feet of 5 inches, 19.5 lb. per foot drillpipe and 600 feet of drill collars with 8 inches external and 3 inches internal diameters and is suspended in 8.33 lb. per gallon drilling fluid. The length of the stretch of the string including the drill collars would be

$$e = \frac{72\ Ls^2}{E}\,[\rho_s - 2\rho_f\,(1 - u)] + \frac{40.8\ W_d L_p}{E\ W_p}\ (1 - \frac{\rho_f}{\rho_s})$$

Ls = drillstring length, feet
E = Young's Modulus, $30 * 10^6$ psi
ρ_S = density of steel, lb/in.^{3}
r_f = density of drilling fluid, lb/in.^{3}
u = Poisson's ratio, .3 for steel
W_d = weight of drill collars in air, lb.
L_p = drillpipe length, feet
W_p = drillpipe weight per foot, lb/foot



$$e = \frac{72 * 10600^2}{30 * 10^6}\ [.2833 - 2 * .0361\,(1 - .3)]$$

$$+ \frac{40.8 * 88112 * 10000}{30 * 10^6\ \ 19.5}\ (1 - \frac{.0361}{.2833})$$ **= 116.4 in or 9.7 ft**

If the mud density is changed to 16.7 lb/gallon (.0722 lb/in^3) and the drill collars to 180 feet of length with 7 inches by 3 inches diameters; then the stretch of the string becomes

$$e = \frac{72 * 10180}{30 * 10^6}\ [.2833 - 2 * .0722\,(1 - .3)] + \frac{40.8 * 19224 * 10000}{30 * 10^6 * 19.5}\ (1 - \frac{.0722}{.2833})$$

= 9.995 in or 0.833 ft

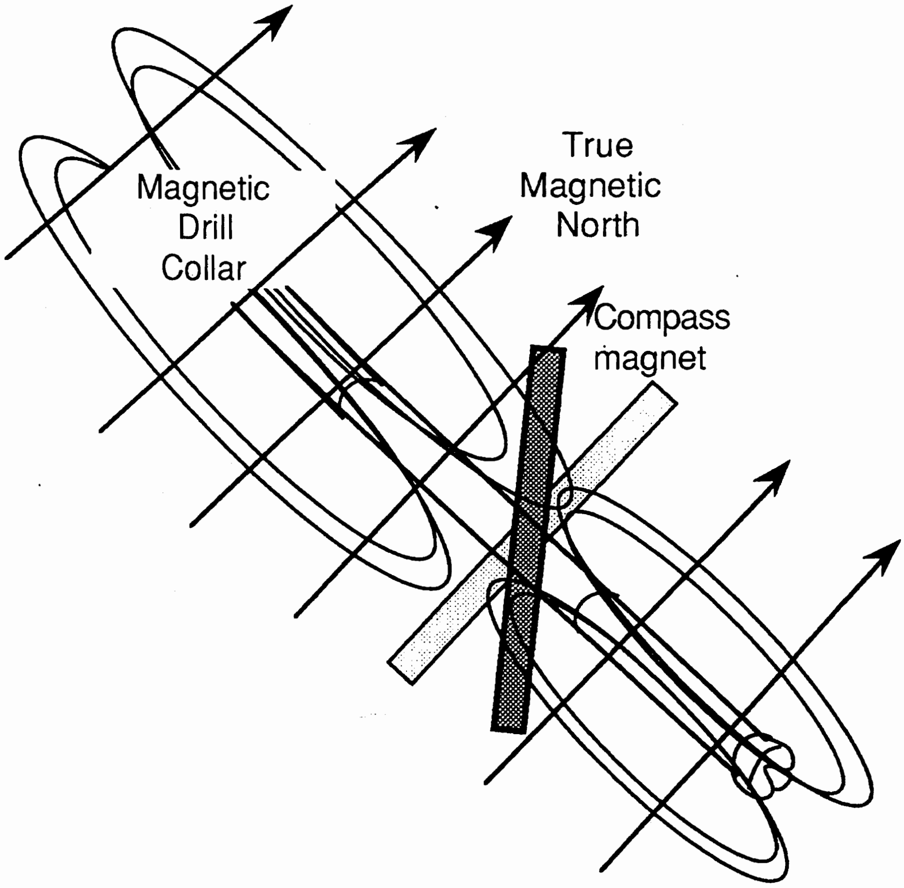
The extreme changes in bottom hole assemblies and drilling fluid densities selected in the above problem caused a difference of 8.9 feet in the stretch of the two strings and this difference would have been smaller in a directional wellbore.

MAGNETIC AND OTHER INTERFERENCES

The magnetic compass is an instrument with a suspended bar magnet which aligns with the resultant direction of all magnetic fields by which it is influenced.

Errors in directional wellbores which occur through the magnetic compass are caused by the lack of compensation for spurious magnetic fields.

The steel in the drillstring is a magnetic material and the affect of its magnetic field is minimized by lengths of non-magnetic drill collars which remove the survey compass from the magnetic steel section of the string.

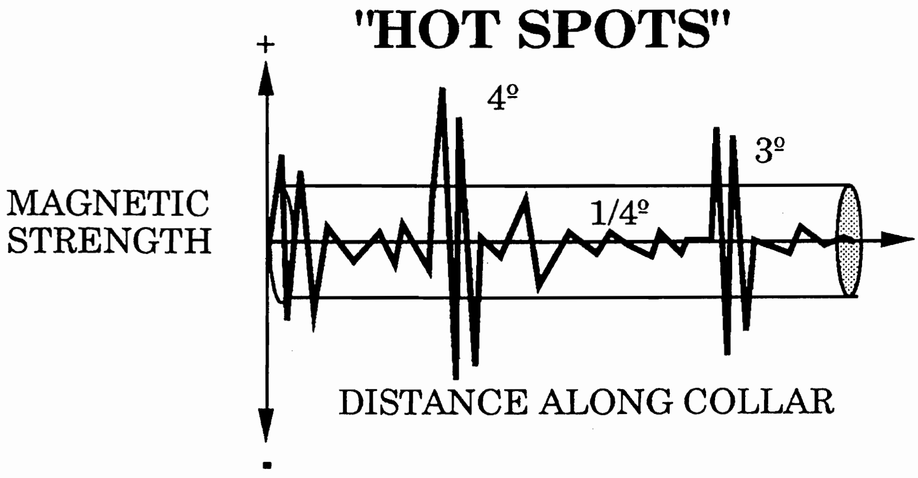


The selection of lengths of non-magnetic drill collars and the most favorable location within the drill collars are shown in the drill collar selection figure.

The lengths of the non-magnetic drill collars must be substantially increased as the north pole is approached.

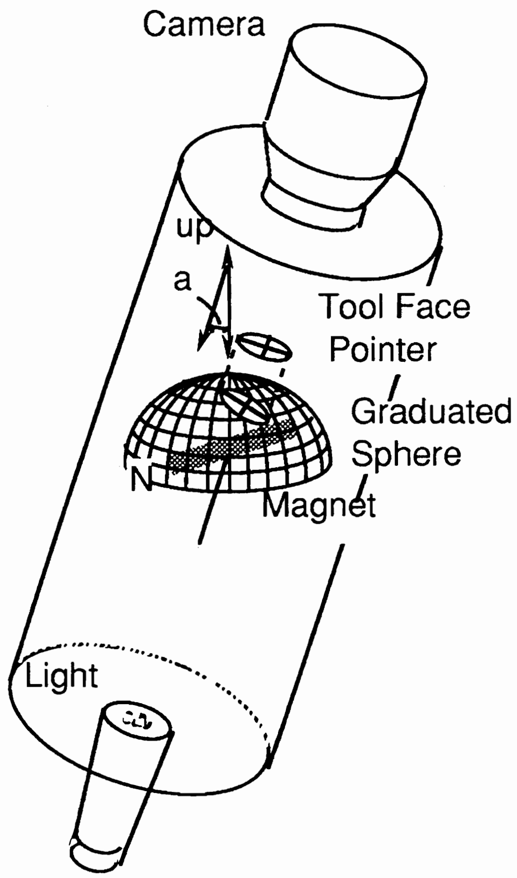
The need arises because the horizontal force component of the magnetic force field of the earth which aligns the bar magnet of the compass with the northerly direction decreases as the magnetic north pole is approached while the magnetic field strength of the steel drill string remains constant.

Extended use will cause the development of a magnetic field of ever increasing strength within non-magnetic drill collars. A test to ascertain the relative strength of a drill collar's magnetic fields versus it length is conducted by running the drill collar over a magnetometer.



The normal range of the deflections from magnetic north along a non-magnetic collar is 1/4 degree. Locations where deflections in the range of 1° are called "HOT SPOTS."

Variations in the direction of the earth's magnetic (declination) with time is a source of error. Charts depicting declination as in the isogonic figure, must be of recent publication.



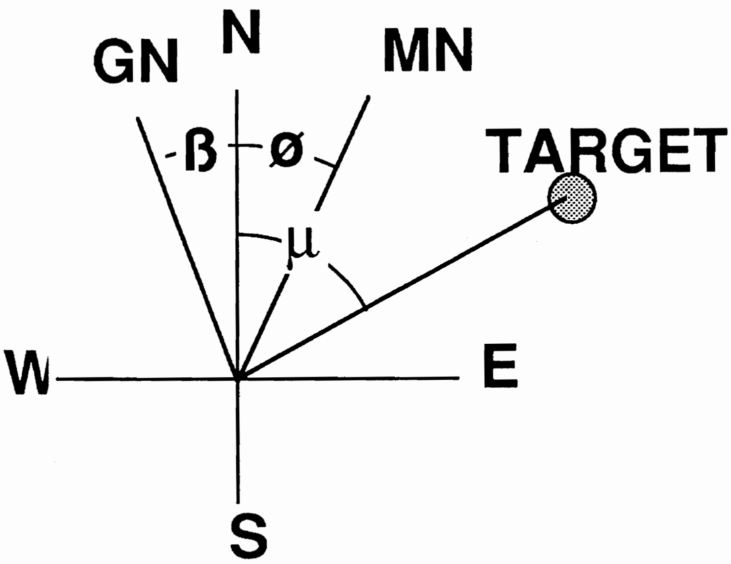
Known variations are (1) secular variation which has an 150 years period and may be of several degrees in magnitude, (2) annual variation which is less than one second per year in magnitude, and (3) solar-diurnal variation which has less than 8 seconds in magnitude.

Other phenomena are thunder storms, aurora-borealis, and magnetic ore bodies.

PEOPLE RECORDING ERRORS

There are many calculations and records made and kept in the process of drilling a directional well and it is recommended that a system be employed by which duplicate and independent readings, calculations, and records are maintained and compared.

NORTH, MAGNETIC NORTH, AND GRID NORTH



On a flat map or chart (not a globe) there is only one vertical line which can be drawn on the chart which will point to both true north and the top of the chart. All other vertical lines point to grid north and not to true north.

The angle between grid north GN and true north N is small.

True north or just north is the location where the spin axis intersects the surface of the earth.

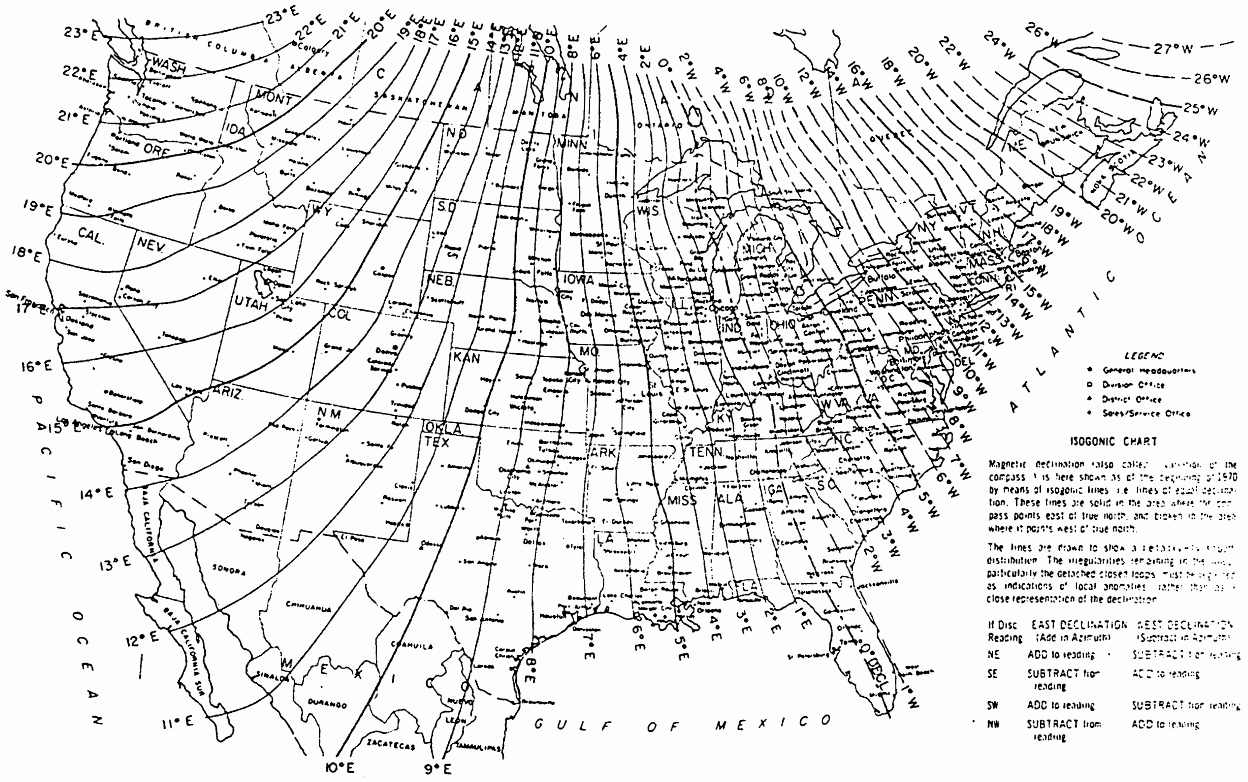
The azimuth of magnetic north is the azimuth indicated by a magnetic compass. The angle from north to magnetic north is called the declination.

EXAMPLES GRID, MAG. NORTH, AND NORTH

A target is on an azimuth of N 63 E. The declination is 7° east. Then the angle on a single shot disk will be 56° if the single shot tool is aligned with the target.

A magnetic single shot disk is read as MN 41° ME at location where the declination is 7° east. Then the azimuth of the single shot is N 48° E.

EASTMAN OIL WELL SURVEY COMPANY

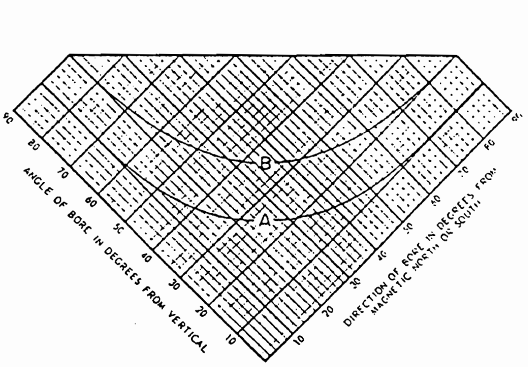


EASTMAN SELECTOGRAPH

FOR NON-MAGNETIC SURVEY DRILL COLLARS

THAT HAVE WELDED-ON OR THREADED STEEL TOOL JOINT ENDS

FOR BOREHOLE SURVEYING UNDER NORMAL CONDITIONS



Use 18' collar (15' non-magnetic section) in area below curve A
Use 25' collar (21' non-magnetic section) in area below curve B
Use tandem collars in area above curve B

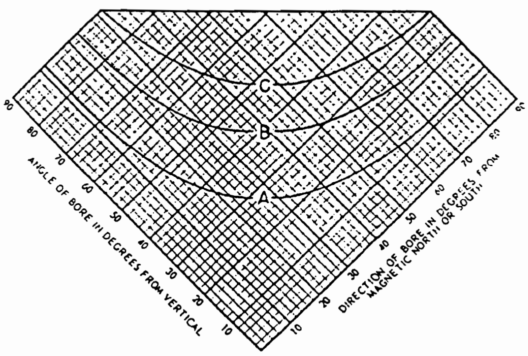
MAGNETIC COMPASS SPACING

18 collar 1' to 2' below center.
25 collar 2' to 3' below center.
Tandem collars, center of bottom collar

EASTMAN SELECTOGRAPH

FOR NON-MAGNETIC SURVEY DRILL COLLARS

WHICH DO NOT HAVE STEEL TOOL JOINT ENDS



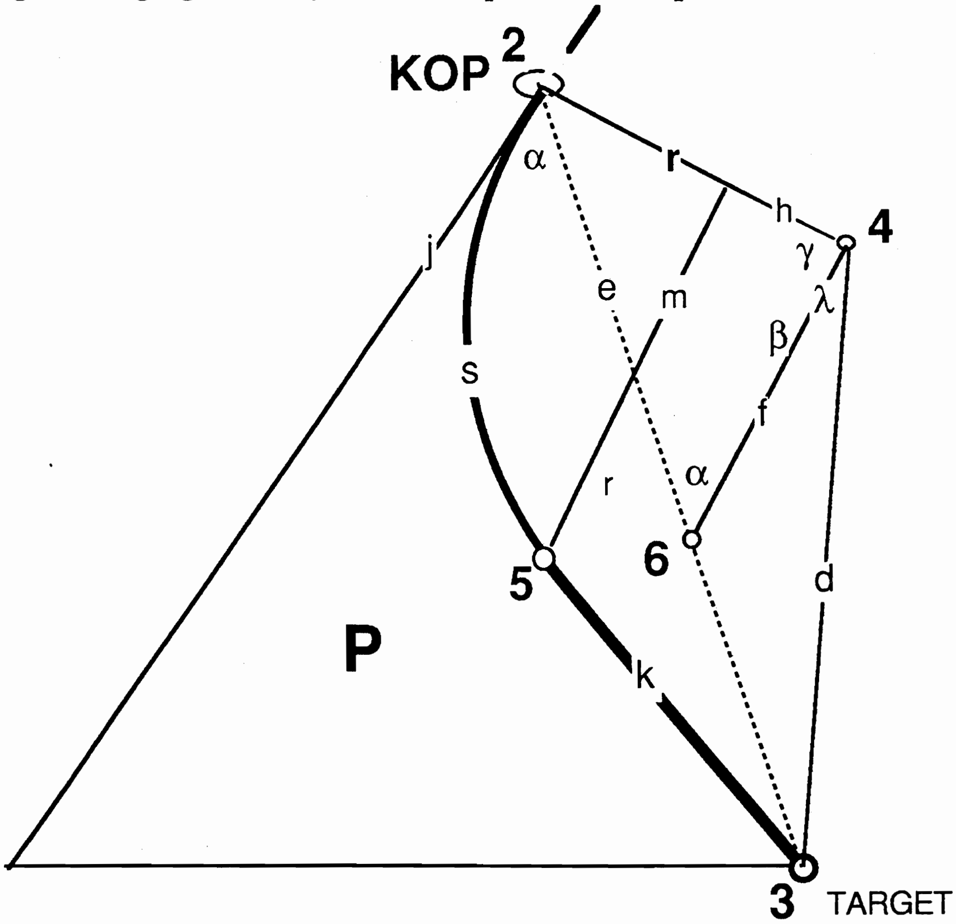
Use 18' collar in area below curve A.
Use 25' collar in area below curve B.
Use 30' collar in area below curve C.
Use tandem collars in area above curve C.

MAGNETIC COMPASS SPACING

18' collar 1' to 2' below center.
25' collar 2' to 3' below center.
30' collar 2' to 3' below center.
Tandem collars, center of bottom collar.

3 DIMENSIONAL DRILL HOLE PLANNING

Four types of directional problems may be solved with the following mathematics: (1) planning of a three dimensional well within an oblique plane, (2) calculation of minimum dogleg severity for the intersection of a non-colinear target with the wellbore, (3) planning the locations of two or more non-colinear targets, (4) selection of a plug-back depth (Sidetrack Depth) in a wellbore for the purpose of reducing the dogleg severity to an acceptable value prior to intersecting the target.



Definition of variables in the sketch:

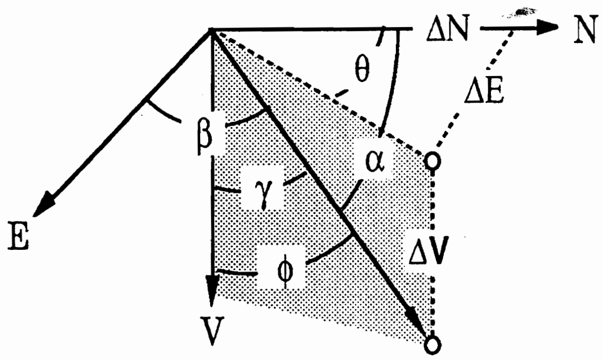
P	=	oblique plane containing points 2, 3, 4, 5, and 6
α, β, γ, λ	=	corresponding angles
d, **e**, **f**, **h**, **j**, k, **m**, **r**, **s**	=	corresponding lengths

In the drilling of a directional well the wellbore may wander off the planned traverse to such a distance that a large displacement of the wellbore toward the target may be required to intersect the target. These large displacements lead to dogleg severities of the wellbore in excess of those acceptable in modern drilling practices.

In order to obviate the excessive dogleg severity problem, a wellbore to target dogleg severity calculation is made at each station. If a plugback depth is desired, the plugback depth is assigned to that station which has an acceptable wellbore to target dogleg severity.

MATHEMATICS

The space is defined with the following sketch. Let the point 2 be a directional station in a drill hole as shown in the sketch with coordinates N_2, E_2, V_2, and let point 3 be a point in the target with coordinates N_3, E_3, V_3. Also, let inclination and azimuth be ϕ and θ respectively at point 2.



The radius of curvature r is

$$r = \frac{18000}{\pi \, DLS}$$

The direction cosines of the hole's direction at the kick off point 2 are equal to those of j and are the following.

$$\cos \alpha_j = \sin \phi \cos \theta \qquad \cos \beta_j = \sin \phi \sin \theta \qquad \cos \gamma_j = \cos \phi$$

The length and direction cosines of line e are

$$e = \sqrt{(N_2 - N_3)^2 + (E_2 - E_3)^2 + (V_2 - V_3)^2}$$

$$\cos \alpha_e = \frac{N_3 - N_2}{e} \qquad \cos \beta_e = \frac{E_3 - E_2}{e} \qquad \cos \gamma_e = \frac{V_3 - V_2}{e}$$

The angle α is

$$\alpha = \mathrm{acos}[\cos \alpha_j \cos \alpha_e + \cos \beta_j \cos \beta_e + \cos \gamma_j \cos \gamma_e]$$

The lengths e, d, f, and k are

$$e = \frac{r}{\sin \alpha} \qquad f = \frac{r}{\tan \alpha}$$

$$d = \sqrt{e^2 + r^2 - 2 e r \sin \alpha} \qquad k = \sqrt{d^2 - r^2}$$

The coordinates of point 6 are

$$N_6 = e \cos \alpha_e + N_2 \qquad E_6 = e \cos \beta_e + E_2 \qquad V_6 = e \cos \gamma_e + V_2$$

The direction cosines of line f are

$$\cos \alpha_f = -\cos \alpha_j \qquad \cos \beta_f = -\cos \beta_j \qquad \cos \gamma_f = -\cos \gamma_j$$

The coordinates of point 4 are

$$N_4 = f \cos \alpha_f + N_6 \qquad E_4 = f \cos \beta_f + E_6 \qquad V_4 = f \cos \gamma_f + V_6$$

The angles λ, β, and γ are

$$\lambda = \operatorname{acos}\left(\frac{r}{d}\right) \qquad \beta = \operatorname{acos}\left[\frac{r^2 + d^2 - e^2}{2\,r\,d}\right]$$

$\gamma = \beta - \lambda$ if f is positive

$\gamma = 2\pi - \beta - \lambda$ if f is negative

The direction cosines of line r from point 4 to point 2 are

$$\cos \alpha_r = \frac{N_2 - N_4}{r} \qquad \cos \beta_r = \frac{E_2 - E_4}{r} \qquad \cos \gamma_r = \frac{V_2 - V_4}{r}$$

The coordinates of point 5 are

$$h = r \cos \gamma \qquad m = r \sin \gamma$$

$$N_5 = N_4 + h \cos \alpha_r + m \cos \alpha_j$$
$$E_5 = E_4 + h \cos \beta_r + m \cos \beta_j$$
$$V_5 = V_4 + h \cos \gamma_r + m \cos \gamma_j$$

The direction cosines of the line k are

$$\cos \alpha_k = \frac{N_3 - N_5}{k} \qquad \cos \beta_k = \frac{E_3 - E_5}{k} \qquad \cos \gamma_k = \frac{V_3 - V_5}{k}$$

The angle of inclination ϕ and azimuth θ into the target is

$$\phi = \operatorname{acos}(\cos \gamma_k) \qquad \theta = \operatorname{atan}\left[\frac{\cos \beta_k}{\cos \alpha_k}\right]$$

The arc length s is $s = r\,\gamma$

POINTS ON THE PLANNED PATH

The values of the variables are continued from the previous section. Point 9 is any point on the arc s. Other variables are defined by the sketch.

The angle ψ is

$$\psi = \frac{\Delta s}{r}$$

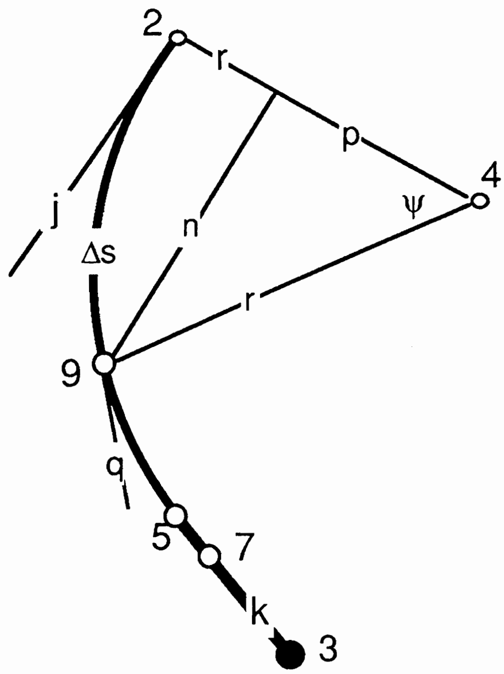
The coordinates of point 9 are

$$p = r \cos \psi \qquad n = r \sin \psi$$

$$N_9 = N_4 + p \cos \alpha_r + n \cos \alpha_j$$

$$E_9 = E_4 + p \cos \beta_r + n \cos \beta_j$$

$$V_9 = V_4 + p \cos \gamma_r + n \cos \gamma_j$$



The direction cosines of the line q are the slopes of the arc at point 9. These direction consines at point 9 are equal to the derivatives of the coordinate equations with respect to the angle ψ and dividing by r.

$$\cos \alpha_q = - \sin \psi \cos \alpha_r + \cos \psi \cos \alpha_j$$

$$\cos \beta_q = - \sin \psi \cos \beta_r + \cos \psi \cos \beta_j$$

$$\cos \gamma_q = - \sin \psi \cos \gamma_r + \cos \psi \cos \gamma_j$$

The inclination angle ϕ and the azimuth θ at point 9 are

$$\phi = \operatorname{acos} (\cos \gamma_q) \qquad \theta = \operatorname{atan} \left[\frac{\cos \beta_q}{\cos \alpha_q}\right]$$

The coordiates of point 7 on the line k at a distance ΔMD from point 5 are

$$N_7 = \Delta MD \cos \alpha_k + N_5 \qquad E_7 = \Delta MD \cos \beta_k + E_5 \qquad V_7 = \Delta MD \cos \gamma_k + V_5$$

The inclination angle ϕ and the azimuth θ at point 7 are

$$\phi = \operatorname{acos} (\cos \gamma_k) \qquad \theta = \operatorname{atan} \left[\frac{\cos \beta_k}{\cos \alpha_k}\right]$$

EXAMPLE 3D PLANNING

The selected coordinates of the kick off point, the inclination, and the azimuth of the hole at point 2 are

$$N_2 = 200 \qquad E_2 = 300 \qquad V_2 = 5000$$

Inclination $\phi = 20°$ Azimuth $\theta = 50°$

The selected coordinates of the target which is at point 3 are

$N_3 = 1500$ $E_3 = 700$ $V_3 = 7000$

The selected dogleg severity of the turn is

DLS = 7.0°/100ft

SOLUTION

The radius of curvature is

$$r = \frac{18000}{\pi\, 7.0} \qquad = \mathbf{818.51}$$

The direction cosines of the hole at the kick off point 2 (equal to those of line j) are

$$\cos\alpha_j = \sin 20 \cos 50 \qquad = \mathbf{0.2198}$$

$$\cos\beta_j = \sin 20 \sin 50 \qquad = \mathbf{0.2620}$$

$$\cos\gamma_j = \cos 20 \qquad = \mathbf{0.9397}$$

The length e is

$$e = \sqrt{(1500 - 200)^2 + (700 - 300)^2 + (7000 - 5000)^2} \qquad = \mathbf{2418.68}$$

The direction cosines of line e are

$$\cos a_e = \frac{1500 - 200}{2418.68} \qquad = \mathbf{0.5374}$$

$$\cos b_e = \frac{700 - 300}{2418.68} \qquad = \mathbf{0.1654}$$

$$\cos\gamma_e = \frac{7000 - 5000}{2418.68} \qquad = \mathbf{0.8269}$$

The angle α is

$$\alpha = \mathrm{acos}[\,0.220 * 0.537 + 0.262 * 0.165 + 0.940 * 0.827] \qquad = \mathbf{20.20}$$

The lengths e, f, and k are

$$e = \frac{818.51}{\sin 20.20} \qquad = \mathbf{2371.02}$$

$$f = \frac{818.51}{\tan 20.20} \qquad = \mathbf{2225.26}$$

$$d = \sqrt{(2418.68)^2 + (818.51)^2 - 2 * 2418.68 * 818.51 \sin 20.20} \qquad = \mathbf{2270.05}$$

$$k = \sqrt{2270.05^2 - 818.51^2} \qquad = \mathbf{2117.35}$$

The coordinates of point 6 are

$$N_6 = (2371.02)(0.537) + 200 \qquad = \mathbf{1474.39}$$

$$E_6 = (2371.02)(0.165) + 300 \qquad = \mathbf{692.12}$$

$$V_6 = (2371.02)(0.827) + 5000 \qquad = \mathbf{6960.60}$$

The direction cosines of line f are

$$\cos \alpha_f = -0.220 \qquad = \mathbf{-0.2198}$$

$$\cos \beta_f = -0.262 \qquad = \mathbf{-0.2620}$$

$$\cos \gamma_f = -0.940 \qquad = \mathbf{-0.9400}$$

The coordinates of point 4 are

$$N_4 = (2225.26)(-0.2198) + 1474.39 \qquad = \mathbf{985.17}$$

$$E_4 = (2225.26)(-0.2620) + 692.12 \qquad = \mathbf{109.09}$$

$$V_4 = (2225.26)(-0.9400) + 6960.60 \qquad = \mathbf{4869.53}$$

The angles λ, β, and γ are

$$\lambda = \operatorname{acos}\left(\frac{818.51}{2270.05}\right) \qquad = \mathbf{68.86}$$

$$\beta = \operatorname{acos}\left[\frac{818.51^2 * 2270.05^2 - 2418.68^2}{2 * 818.51 * 2270.05}\right] \qquad = \mathbf{90.41}$$

$$\gamma = 90.42 - 68.86 \qquad = \mathbf{21.55}$$

The direction cosines of line r (from point 4 to point 2) are

$$\cos \alpha_r = -\frac{200 - 985.17}{818.51} \qquad = \mathbf{-0.9592}$$

$$\cos \beta_r = -\frac{300 - 109.09}{818.51} \qquad = \mathbf{0.2332}$$

$$\cos \gamma_r = -\frac{5000 - 4869.53}{818.51} \qquad = \mathbf{0.1594}$$

The coordinates of point 5 are

$$h = 818.51 \cos 21.55 \qquad = \mathbf{761.29}$$

$$m = 818.51 \sin 21.55 \qquad = \mathbf{300.65}$$

$$N_5 = 985.17 + 761.29 * (-0.9593) + 300.65 * 0.2198 \qquad = \mathbf{320.99}$$

$$E_5 = 109.09 + 761.29 * 0.2332 + 300.65 * 0.2620 \qquad = \mathbf{365.43}$$

$$V_5 = 4869.53 + 761.29 * 0.1593 + 300.65 * 0.9396 \qquad = \mathbf{5273.40}$$

The direction cosines of line k are

$$\cos \alpha_k = \frac{1500 - 320.98}{2117.34} \qquad = \mathbf{0.5568}$$

$$\cos \beta_k = \frac{700 - 365.42}{2117.34} \qquad = \mathbf{0.1580}$$

$$\cos \gamma_k = \frac{7000 - 5273.40}{2117.34} \qquad = \mathbf{0.8154}$$

The angle of inclination ϕ and azimuth θ into target is

$$\phi = \text{acos}\,(\,0.815\,) \qquad = \mathbf{35.37}$$

$$\theta = \text{atan}\left[\frac{0.1580}{0.5568}\right] \qquad = \mathbf{15.84}$$

Arc length S

$$S = (818.51)\left(\frac{21.55\,\pi}{180}\right) \qquad = \mathbf{307.86}$$

Find the coordinates, inclination, and azimuth of point 9 which is 200 feet along the arc down from point 2.

$$\psi = \frac{200}{818.51} \qquad = \mathbf{14.00}$$

$$p = 818.51 * \cos 14 \qquad = \mathbf{794.19}$$

$$n = 818.51 * \sin 14 \qquad = \mathbf{198.01}$$

$$N_9 = 985.17 + 794.19 * (-0.9593) + 198.01 * 0.2198 \qquad = \mathbf{266.85}$$

$$E_9 = 109.09 + 794.19 * 0.2332 + 198.01 * 0.2620 \qquad = \mathbf{346.21}$$

$$V_9 = 4869.53 + 794.19 * 0.1593 + 198.01 * 0.9396 \qquad = \mathbf{5{,}182.19}$$

The direction cosines of the line q are

$$\cos \alpha_q = -\sin 14 * (-0.9593) + \cos 14 * 0.2198 \qquad = \mathbf{0.4453}$$
$$\cos \beta_q = -\sin 14 * 0.2332 + \cos 14 * 0.2620 \qquad = \mathbf{0.1977}$$
$$\cos \gamma_q = -\sin 14 * 0.1593 + \cos 14 * 0.9396 \qquad = \mathbf{0.8732}$$

The inclination and azimuth of the planned hole at point 9 are

$$\phi = \text{acos } 0.8732 \qquad = \mathbf{29.16}$$

$$\theta = \text{atan } \frac{0.1977}{0.4453} \qquad = \mathbf{23.94}$$

The coordinates of point 7 on line k which is 100 feet down from point 5 are

$$N_7 = 100 * 0.5568 + 320.98 \qquad = \mathbf{376.66}$$

$$E_7 = 100 * 0.1580 + 365.42 \qquad = \mathbf{381.22}$$

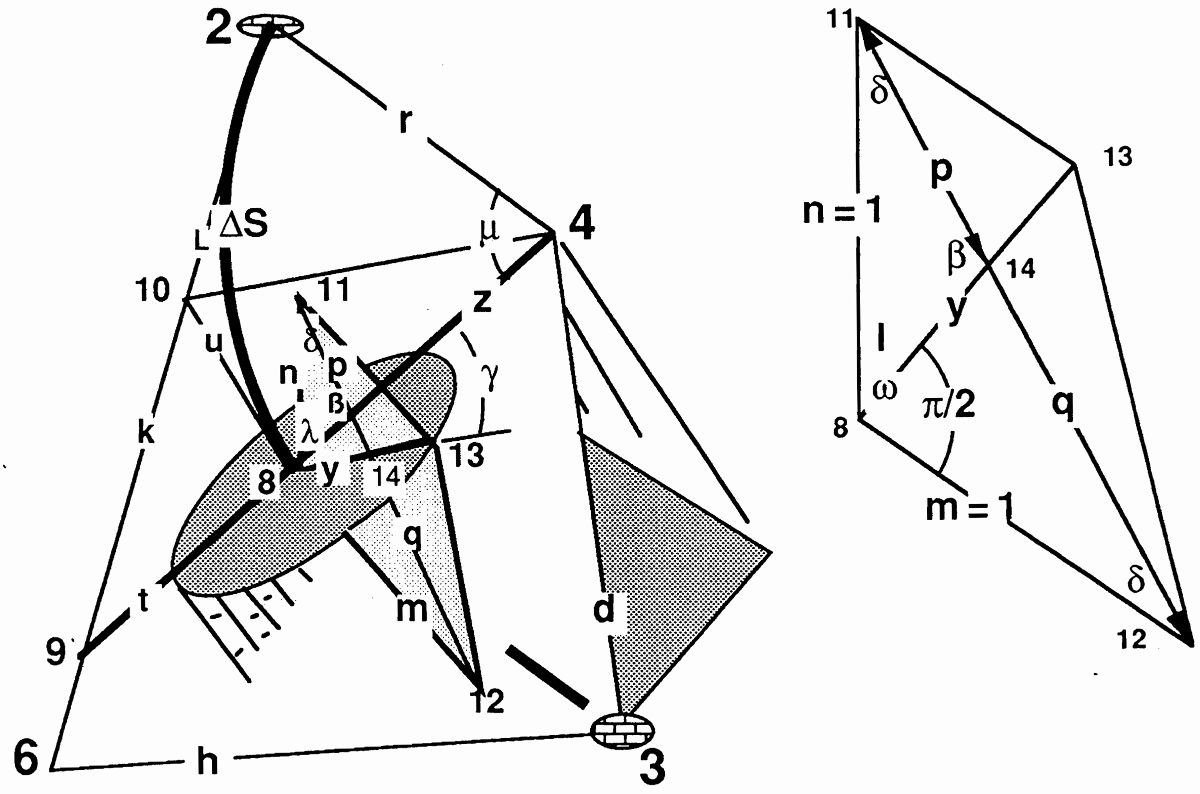
$$V_7 = 100 * 0.8154 + 5273.40 \qquad = \mathbf{5{,}354.94}$$

The inclination and azimuth of planned hole at point 7 are

$$\phi = \text{acos } 0.8154 \qquad = \mathbf{56.16}$$

$$\theta = \text{atan } \frac{0.1580}{0.5568} \qquad = \mathbf{15.84}$$

TOOL FACE ROTATION

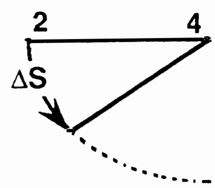


A companion problem to the 3 dimensional planning problem is the "TOOL FACE ROTATION" problem. The problem is the computations of the azimuth and inclination of a drill hole after simultaneously turning the hole with a bottom hole motor and bent sub or another device and drilling a specific distance. The newly drilled hole will be a smooth circular arc in an oblique plane.

The methods of calculating tool face rotation angles by Mitchell (in an earlier book), Millheim, and the Ouija board **assume a straight drill hole during the turning of the hole rather than an arc,** as does this solution.

If the tool face rotation angle is held constant during the turning of the hole, a smooth circular arc will not be drilled. **The tool face rotation angle must be adjusted during the drilling of the turn.**

The mathematics for the tool face rotation angle are a continuation of the 3D planning section.



The length of a segment of the arc ΔS is selected.

ΔS = selected value

The angle μ is

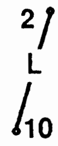
$$\mu = \frac{\Delta S}{r}$$

The length of the line from point 2 to point 9, k is

$$k = r \tan(\mu)$$

The length of the line from point 2 to point 10, L is

$$L = r \tan(\frac{\mu}{2})$$



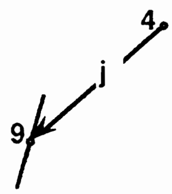
The coordinates of point 9 are

$$N_9 = (k+L) * \cos \alpha_{a+j} + N_2$$
$$E_9 = (k+L) * \cos \beta_{a+j} + E_2$$
$$V_9 = (k+L) * \cos \gamma_{a+j} + V_2$$

The coordinates of point 10 are

$$N_{10} = L * \cos \alpha_{a+j} + N_2 \qquad E_{10} = L * \cos \beta_{a+j} + E_2 \qquad V_{10} = L * \cos \gamma_{a+j} + V_2$$

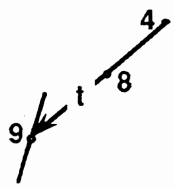
The length of line t+z and its direction cosines are



$$t+z = \sqrt{(N_9 - N_4)^2 + (E_9 - E_4)^2 + (V_9 - V_4)^2}$$

$$\cos \alpha_{t+z} = \frac{N_9 - N_4}{j} \qquad \cos \beta_{t+z} = \frac{E_9 - E_4}{j} \qquad \cos \gamma_{t+z} = \frac{V_9 - V_4}{j}$$

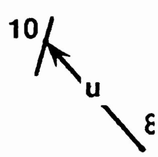
The distance from point 9 to point 8, t, and the coordinates of point 8 are



$$t = (t+z) - z$$

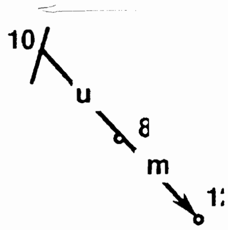
$$N_8 = t * \cos \alpha_{t+z} + N_9 \qquad E_8 = t * \cos \beta_{t+z} + E_9 \qquad V_8 = t * \cos \gamma_{t+z} + V_9$$

The length of line u from point 10 to point 8 and the direction cosines of u and m are



$$u = \sqrt{(N_8 - N_{10})^2 + (E_8 - E_{10})^2 + (V_8 - V_{10})^2}$$

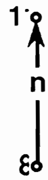
$$\cos \alpha_u = \frac{N_8 - N_{10}}{u} \qquad \cos \beta_u = \frac{E_8 - E_{10}}{u} \qquad \cos \gamma_u = \frac{V_8 - V_{10}}{u}$$



Selecting the length of line m to be unity, the coordinates of point 12 are

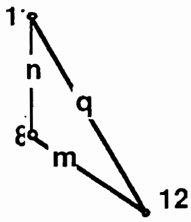
$$N_{12} = 1 * \cos \alpha_j + N_8 \qquad E_{12} = 1 * \cos \beta_j + E_8 \qquad V_{12} = 1 * \cos \gamma_j + V_8$$

The coordinates of point 11 which lies one unit directly above point 8 are



$$N_{11} = N_8 \qquad E_{11} = E_8 \qquad V_{11} = - 1 + V_8$$

The length of line q from point 11 to point 12 and the direction cosines of q are



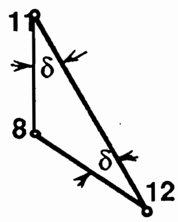
$$q = \sqrt{(N_{12} - N_{11})^2 + (E_{12} - E_{11})^2 + (V_{12} - V_{11})^2}$$

$$\cos \alpha_q = \frac{N_{12} - N_{11}}{q}$$

$$\cos \beta_q = \frac{E_{12} - E_{11}}{q}$$

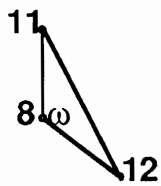
$$\cos \gamma_q = \frac{V_{12} - V_{11}}{q}$$

The angle δ between line n and line q is



$$\delta = \text{acos}(- \frac{1^2 - 1^2 - q^2}{2 * 1 * q})$$

The angle ω, λ, and ß is

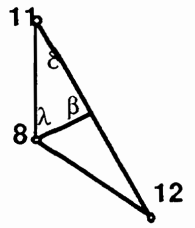


$$\omega = \pi - 2\,\delta$$

$$\lambda = \omega - \frac{\pi}{2}$$

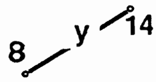
$$ß = \pi - \lambda - \delta$$

The length of y, point 8 to point 14 is



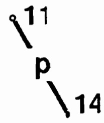
$$y = 1 * \tan(\delta)$$

The law of sines gives the length of the line p, from point 11 to point 14



$$p = \frac{\sin l}{\sin ß}$$

The coordinates of point 14 are



$$N_{14} = p * \cos \alpha_q + N_{11}$$
$$E_{14} = p * \cos \beta_q + E_{11}$$
$$V_{14} = p * \cos \gamma_q + V_{11}$$

The direction cosines of line y are

$$\cos \alpha_y = \frac{N_{14} - N_8}{y}$$

$$\cos \beta_y = \frac{E_{14} - E_8}{y}$$

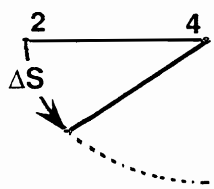
$$\cos \gamma_y = \frac{V_{14} - V_8}{y}$$

The tool face rotation angle γ which is the angle between the z and y is

$$\gamma = \operatorname{acos}(\cos \alpha_y \cos \alpha_j + \cos \beta_y \cos \beta_j + \cos \gamma_y \cos \gamma_j)$$

EXAMPLE TOOL FACE ROTATION

The length of a segment of the arc ΔS is selected.



$$\Delta S = \text{selected value} = 100$$

The angle μ is

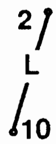
$$\mu = \frac{100}{818.51} \qquad \mathbf{= 0.122\ rad}$$

The length of the line from point 2 to point 9, k is

$$k = 818.51 \tan(.0.122) \qquad = \mathbf{100.37}$$

The length of the line from point 2 to point 10, L is

$$L = 818.51 \tan\left(\frac{.122}{2}\right) \qquad = \mathbf{49.99}$$



The coordinates of point 9 are

$$N_9 = (100.37+49.99) * .220 + 200 \qquad = \mathbf{233.08}$$

$$E_9 = (100.37+49.99) * .262 + 300 \qquad = \mathbf{339.39}$$

$$V_9 = (100.37+49.99) * .940 + 5000 \qquad = \mathbf{5{,}141.34}$$

The coordinates of point 10 are

$$N_{10} = 49.99 * .220 + 200 \qquad = \mathbf{211.00}$$

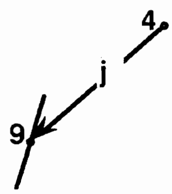
$$E_{10} = 49.99 * .262 + 300 \qquad = \mathbf{313.10}$$

$$V_{10} = 49.99 * .940 + 5000 \qquad = \mathbf{5{,}046.99}$$

The length of line t+z and its direction cosines are

$$t+z = \sqrt{(233.08-985.17)^2 + (339.39-109.09)^2 + (5141.34-4869.53)^2}$$

$$t+z \qquad = \mathbf{832.20}$$



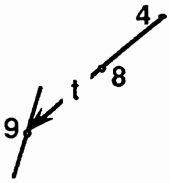
$$\cos \alpha_{t+z} = \frac{233.08-985.17}{832.20} \qquad = \mathbf{-0.9037}$$

$$\cos \beta_{t+z} = \frac{339.39-109.09}{832.20} \qquad = \mathbf{0.2767}$$

$$\cos \gamma_{t+z} = \frac{5141.34-4869.53}{832.20} \qquad = \mathbf{0.3266}$$

The distance from point 9 to point 8, line t and the coordinates of point 8 are

$t = 832.2 - 818.51$ $= \mathbf{13.69}$



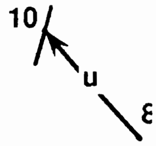
$N_8 = 13.69 * (-0.9037) + 233.08$ $= \mathbf{220.71}$

$E_8 = 13.69 * \ (0.2767) + 339.39$ $= \mathbf{343.18}$

$V_8 = 13.69 * (0.3266) + 5141.34$ $= \mathbf{5145.81}$

The length of line u from point 10 to point 8 and the direction cosines u and m are

$$u = \sqrt{(220.71 - 211.00)^2 + (343.18 - 313.10)^2 + (5145.81 - 5046.99)^2}$$



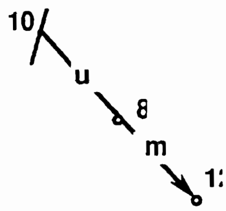
u $= \mathbf{103.75}$

$$\cos \alpha_u = \frac{220.71 - 211.00}{103.75} = \mathbf{.0936}$$

$$\cos \beta_u = \frac{343.18 - 313.10}{103.75} = \mathbf{.2899}$$

$$\cos \gamma_u = \frac{5145.81 - 5046.99}{103.75} = \mathbf{.9525}$$

Selecting the length of line m to be unity, the coordinates of point 12 are

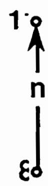


$N_{12} = 1\ (-.9037) + 220.71$ $= \mathbf{219.81}$

$E_{12} = 1 * (.2767) + 343.18$ $= \mathbf{343.46}$

$V_{12} = 1\ (.3266) + 5145.81$ $= \mathbf{5,146.14}$

The coordinates of point 11 which lies one unit directly above point 8 are



N_{11} $= \mathbf{220.71}$

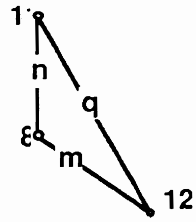
E_{11} $= \mathbf{343.18}$

$V_{11} = -1 + 5145.81$ $= \mathbf{5,144.81}$

The length of line q from point 11 to point 12 and the direction cosines q are

$$q = \sqrt{(219.81 - 220.71)^2 + (343.46 - 343.18)^2 + (5146.14 - 5144.81)^2}$$

$$q \qquad\qquad = \mathbf{1.630}$$

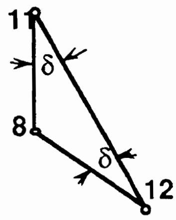


$$\cos \alpha_q = \frac{219.81 - 220.71}{1.630} \qquad = \mathbf{-.5521}$$

$$\cos \beta_q = \frac{343.46 - 343.18}{1.630} \qquad = \mathbf{.1718}$$

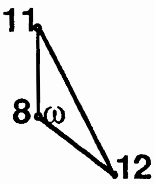
$$\cos \gamma_q = \frac{5146.14 - 5144.81}{1.630} \qquad = \mathbf{.8160}$$

The angle δ between line n and line q is



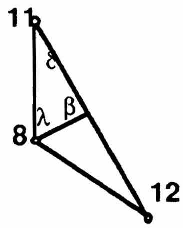
$$\delta = \mathrm{acos}\left(-\frac{1^2 - 1^2 - 1.630^2}{2 * 1 * 1.630}\right) \qquad = \mathbf{.6181}$$

The angle ω is



$$\omega = \pi - 2 * .6181 \qquad = \mathbf{1.9054}$$

The angle λ is

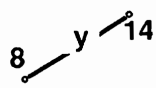


$$\lambda = 1.9054 - \frac{\pi}{2} \qquad = \mathbf{.3346}$$

The angle ß is

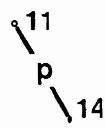
$$ß = \pi - .3346 - .6181 \qquad = \mathbf{2.1889}$$

The length of y, point 8 to point 14 is



$$y = \tan(.6181) \qquad = \mathbf{.7110}$$

The law of sines gives the length of the line p, from point 11 to point 14



$$p = \frac{\sin .3346}{\sin 2.1889} \qquad = \mathbf{.4029}$$

The coordinates of point 14 are

$$N_{14} = .4029 * -.5521 + 220.71 \qquad = \mathbf{220.49}$$

$$E_{14} = .4029 * .1718 + 343.18 \qquad = \mathbf{343.25}$$

$$V_{14} = .4029 * .8160 + 5144.81 \qquad = \mathbf{5,145.14}$$

The direction cosines of line y are

$$\cos \alpha_y = \frac{220.49 - 220.71}{.7110} \qquad = \mathbf{-.3094}$$

$$\cos \beta_y = \frac{343.25 - 343.18}{.7110} \qquad = \mathbf{.0985}$$

$$\cos \gamma_y = \frac{5145.14 - 5145.81}{.7110} \qquad = \mathbf{-.9423}$$

The tool face rotation angle γ which is the angle between the z and y is

$$\gamma = \text{acos}(\ (-.3094)\,(-.9037) + (.0985)\,(.2767) + (-.9423)\,(.3266)\ ) \qquad = \mathbf{90.05^\circ}$$

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CHAPTER VII

HORIZONTAL DRILLING

USES OF HORIZONTAL WELL

Horizontal wells are directional wells drilled with an inclination angle near 90 degrees. The purposes of drilling horizontal wells are not new. However, the application of solid state electronics in directional drilling at long last permits the fulfillment of those purposes. The primary purposes of horizontal wells are the following:

1. Intersect many fractures in a hydrocarbon containing formation. Very popular in limestone and some shale formations.

2. Avoid drilling into water below (or gas above) hydrocarbons or perforating adjacent to water or gas. Either are thought to promote gas and water coning. Popular in formations containing relatively thin oil zones as compared with the underlying water zone.

3. Increase both the drainage area of the well in the reservoir and the lateral surface area of the well bore. The first is thought to increase the cumulative hydrocarbon production, while the second enhances the hydrocarbon production rate. Popular in formations containing heavy oil. These holes may be thought of as drain holes in some cases.

4. Intersect layered reservoirs at high dip angles.

5. Improve coal gas production (degasification).

6. Improve injection of water, gas, steam, chemical, and polymer into formations.

The counter proposal to the drilling of a horizontal well is to drill a vertical well and hydraulically fracture the pay formation. This rarely accomplishes a purpose of a horizontal well, because hydraulic fracturing rarely if ever succeeds in intersecting many fractures in a naturally fractured formation; fractures usually intersect underlying water zones, and fractures filled with propants (sand) are not drain holes.

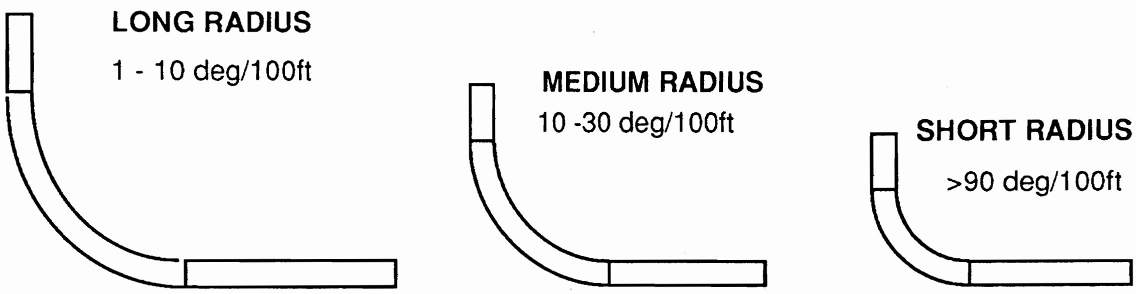
The above purposes stipulate the requirements for the evaluation of a horizontal hole.

1. Hits all targets

2. Smooth turns and builds for promoting long lateral sections

3. Gauge hole allowing fast problem free drilling and good completions
4. Minimum formation damage
5. Reasonable cost

TYPES OF HORIZONTAL WELLS



There are five recognized types of horizontal wells that have evolved for reasons of hydrocarbon reservoir requirements or to designate the equipment required to drill the wells. For example, long-radius wells are drilled with standard directional equipment while short-radius wells require an assortment of special equipment. The types of wells have become known as:

1. Long-Radius
2. Medium-Radius
3. Short-Radius
4. Tangent
5. Combination

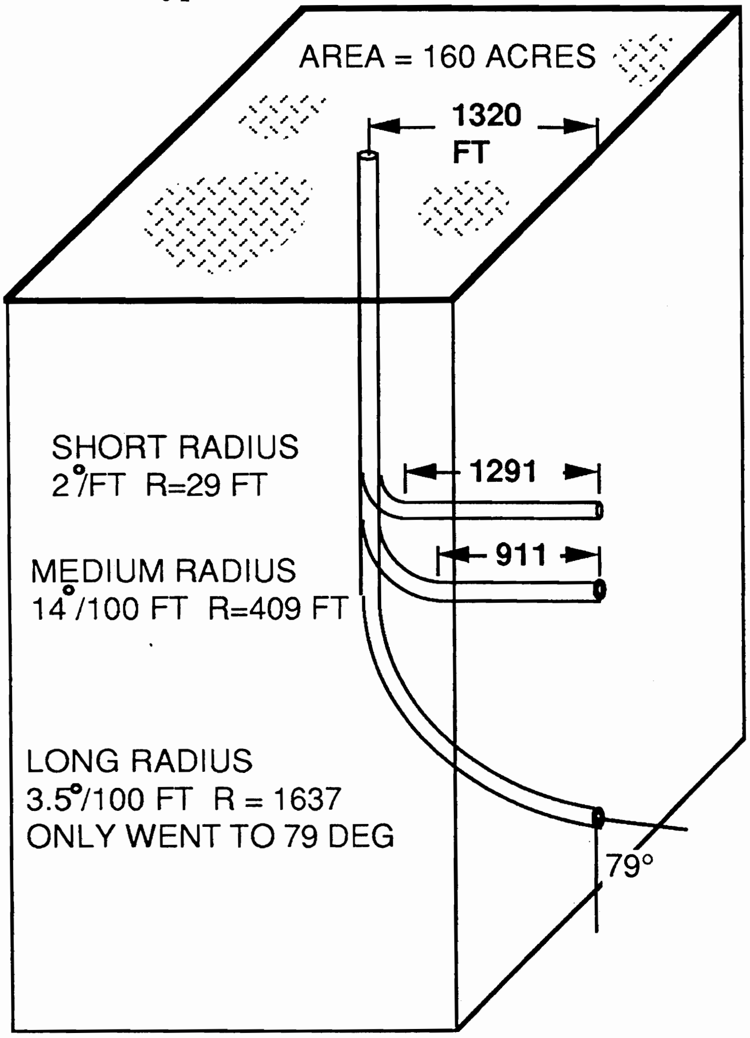
The long-radius, medium-radius, and short-radius wells are shown in the figure. The figure contains their individual attributes. These types have a single build section. It may be noticed that for the same final departure of the well, the short-radius type of well will intersect more pay formation.

However, torque and drag of the drill string may curtail the final departure length of the short-radius well. The horizontal section will, in most cases, be less than 500 feet.

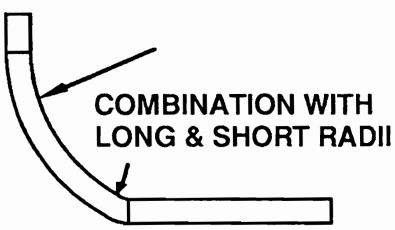
The tangent horizontal well is shown in the figure. Note that the straight section is tangent to both of the circular sections at each of its ends.

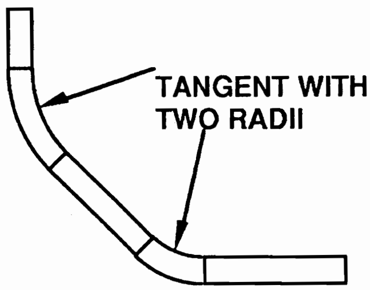
The tangent well is popular because it allows the searching of the formation into which the horizontal section is to be put; that is, the final small turn is not completed until the eminent drilling of the pay formation is assured.

Whereas, the prior three types are a hit or miss proposition once the drill hole is kicked off. However, the tangent well may have a shorter productive interval than the other types.



If a lease boundary or a reservoir terminator exists, the short-radius and the medium-radius wells will give longer horizontal pay sections.





The combination horizontal wells usually contain a long-radius in the upper build, a tangent section, and a medium-radius in the bottom build. The long-radius may reduce costs for the upper and major build while the tangent section in conjunction with the medium-radius or short-radius bottom build gives assurance of drilling the horizontal section in the most favorable location in the pay zone.

HORIZONTAL WELL COSTS

The cost of drilling a horizontal well depends on many factors, contingencies, and circumstances; however, the drilling costs may be reconciled into three sections of the hole:

1. the vertical section
2. the build(s) section
3. the horizontal section

In comparison with vertical holes, horizontal holes most likely will have added costs in the following areas. These costs may be 120% of vertical well costs.

1. Surface location and surface equipment
2. Casing and tubing
3. Rig rental rate and tool rental
4. BHA Equipment rental (excluding directional tools and motors)
5. Mud and mud handling equipment
6. Hole loss and fishing (or sidetracking)

The cost of drilling a horizontal well is given by the following equation:

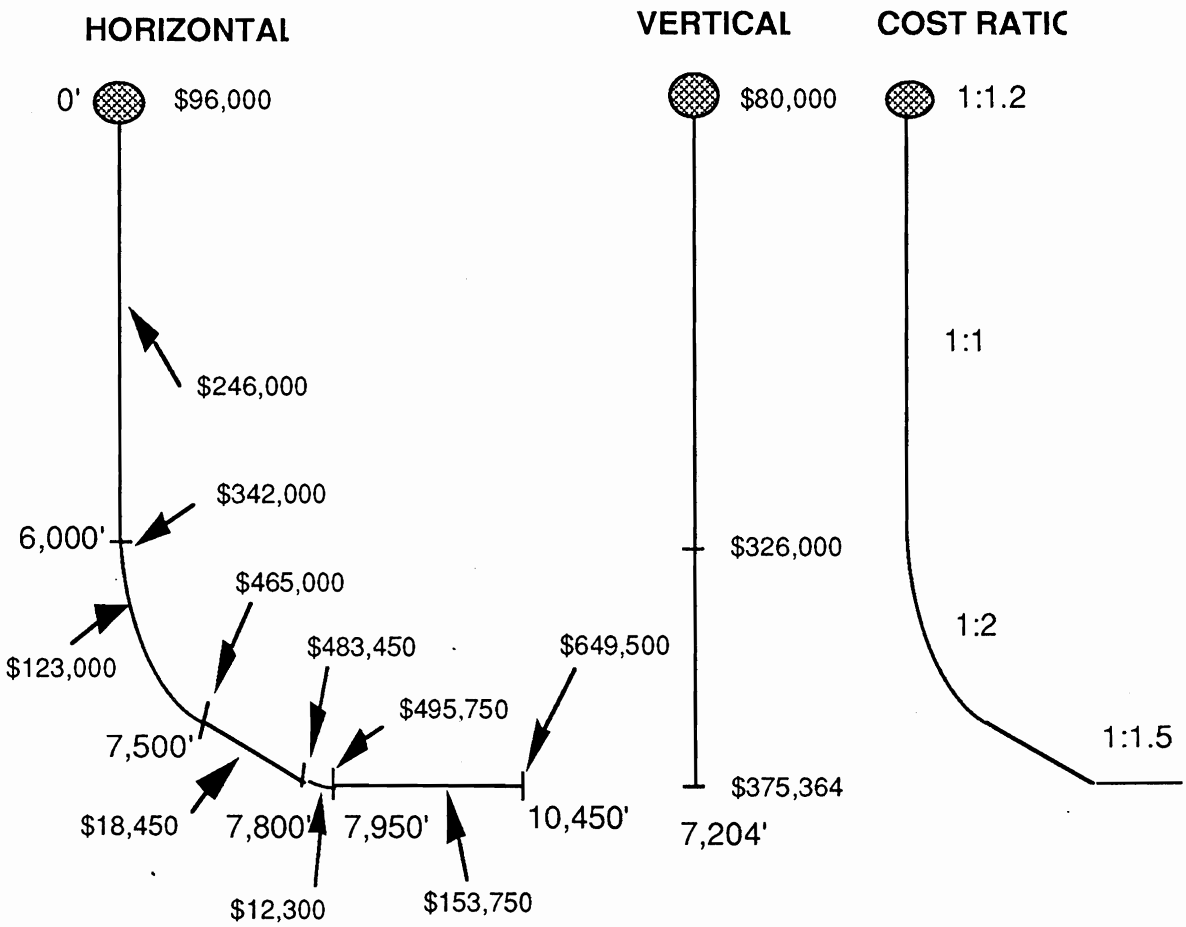
Cost \$ = Location + Casing + Mud + Tool Rental + Directional + Rig Rental + Drilling Time + Logging

Using the above 120% for the lumped costs, and doubling the cost per foot in the build section over the vertical section, and multiplying vertical footage costs by 150% to get inclined and horizontal footage costs, a comparison of a vertical well cost can be made with a horizontal well cost.

Let vertical drilling costs per foot be \$41.00 and let the trajectory plan be a combination type.

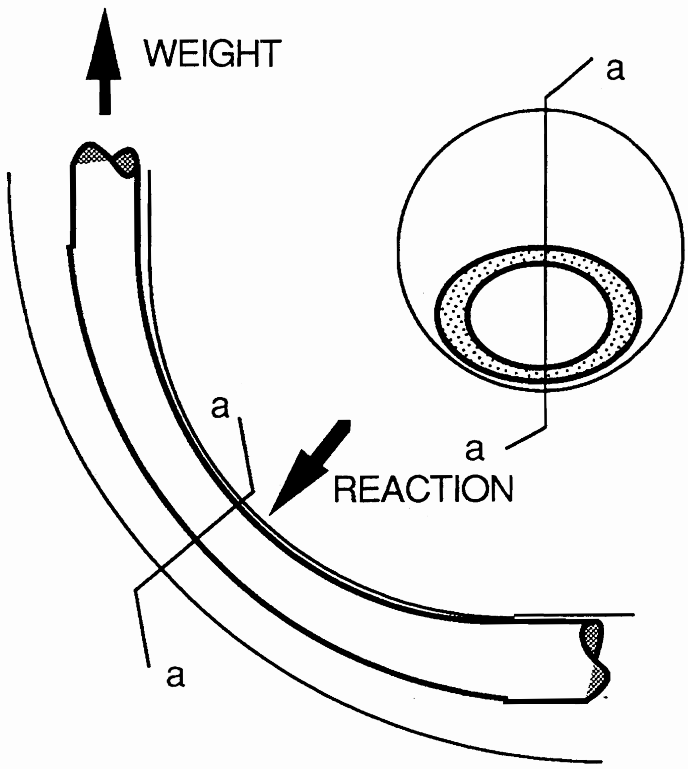
KOP:	6,000 feet
Upper build:	5 deg/100ft to 75 deg
Tangent:	300 feet
Lower build:	10 deg/100ft
Horizontal section:	2,500 feet

COMPARATIVE COST OF WELLS



EVENT	VERTICAL	HORIZONTAL
Surface, etc.	$ 80,000	$ 96,000
Drill to KOP	$326,000	$342,000
Drill to tangent		$465,000
Drill to lower build		$483,450
Drill to horizontal		$495,750
Drill to bottom of hole	$375,364	$649,500

CASING & DRILL BIT SIZES



Akgun used the commercial finite element program call ANSYS to create and publish a set of charts which models casing (any pipe) stresses and deformation.

The stresses are responsible for the failure of the wall of the tube and the deformation leads to ovality. Ovality in casing prevents the passage of tools if overly severe.

The stresses and deformations are all triaxial.

The use of his charts which follow are explained in the following example.

Akgun showed that the classical beam formula derived in mechanics gave stresses of lesser values than his FEM solution and that Lubinski formula gave values of stress in excess of his FEM solution.

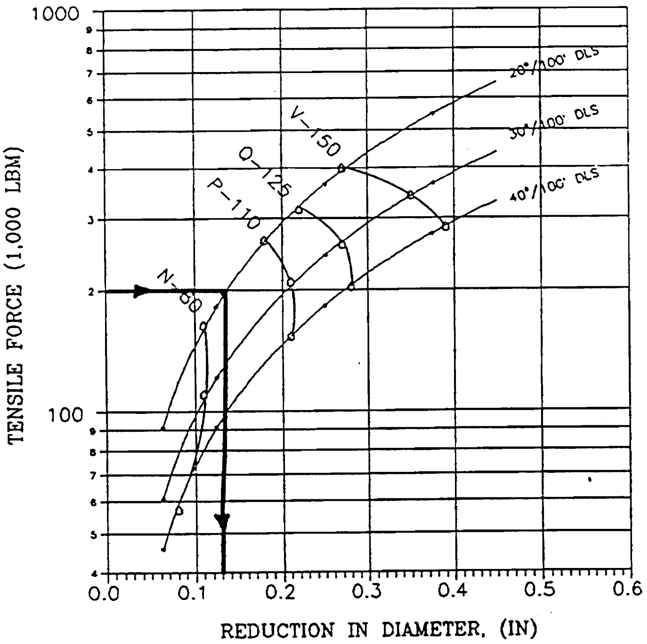
The relative values may be seen in the figure.

EXAMPLE OF CASING STRESS AND DEFORMATION

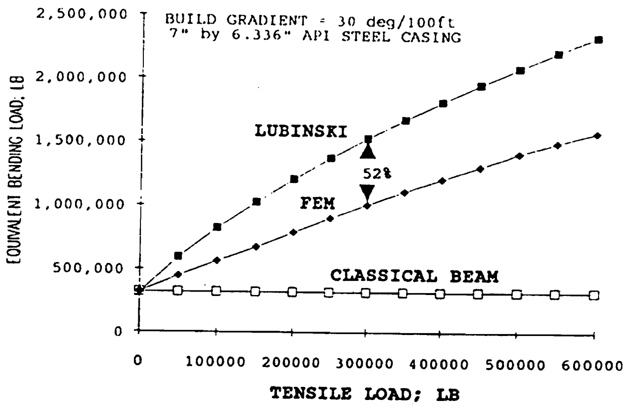
7" casing is in 200,000 lbs of tension in a 20 deg/100ft dogleg. What is the reduction in the diameter of the casing and the maximum axial stress in the wall of the casing?

The 20 deg/100ft and 7" by 6.538" casing chart shows a reduction in diameter of **0.13 inch** and an axial stress of **90,000 psi.**

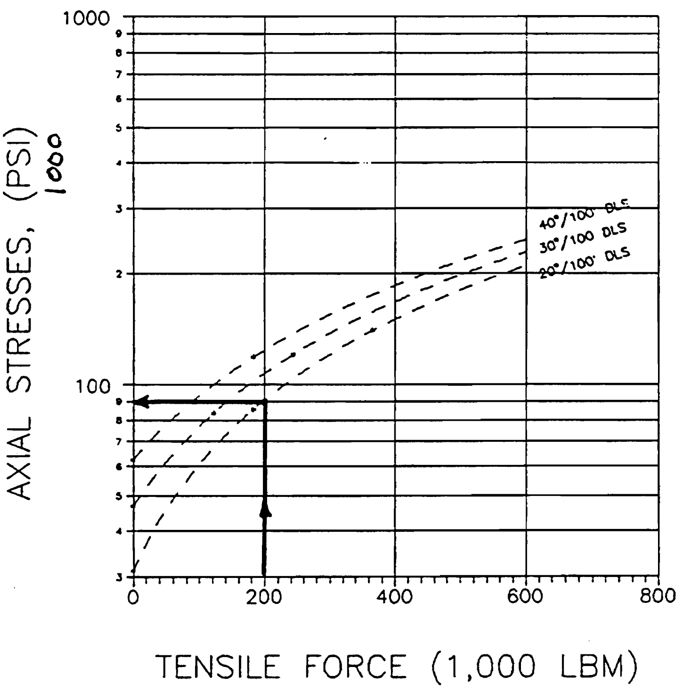
CASING: STEEL, 7*6.336 INCH, 0.332 INCH WALL THICKNESS
COUPLING: STEEL, 7.656 INCH OD, 9 INCH LONG



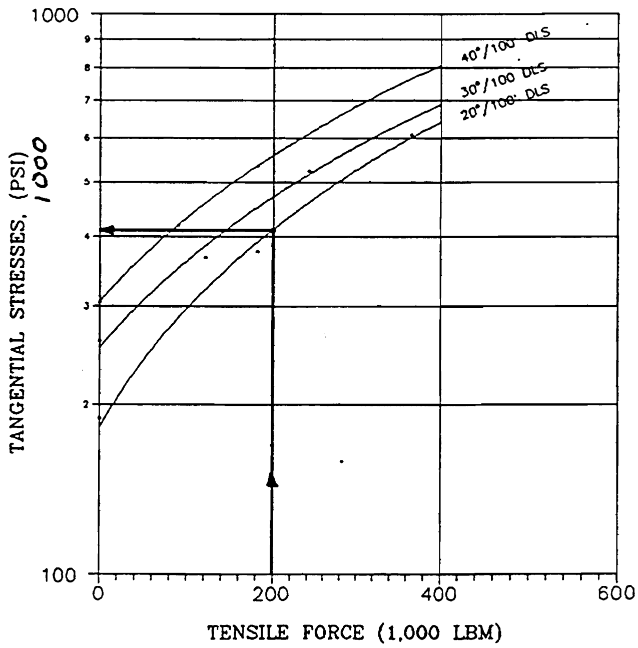
COMPARISON OF FEM, CLASSICAL, & LUBINSKI BENDING LOADS



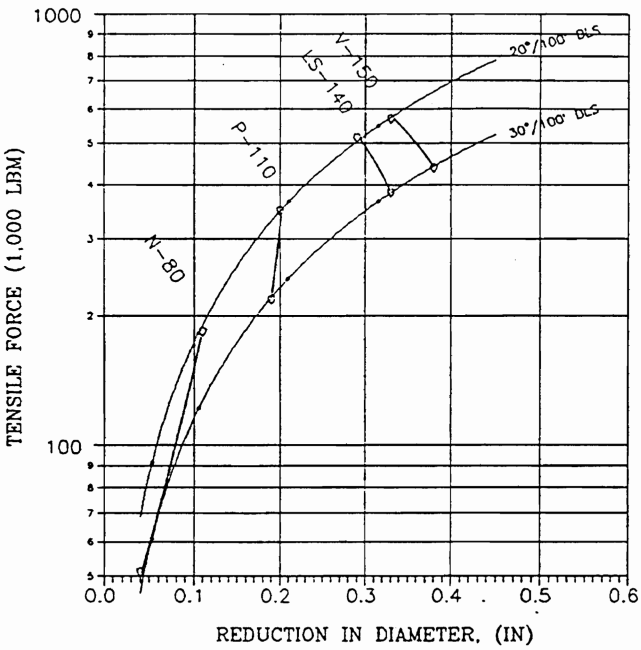
CASING: STEEL, 7*6.336 INCH, 0.332 INCH WALL THICKNESS
COUPLING: STEEL, 7.656 INCH OD, 9 INCH LONG



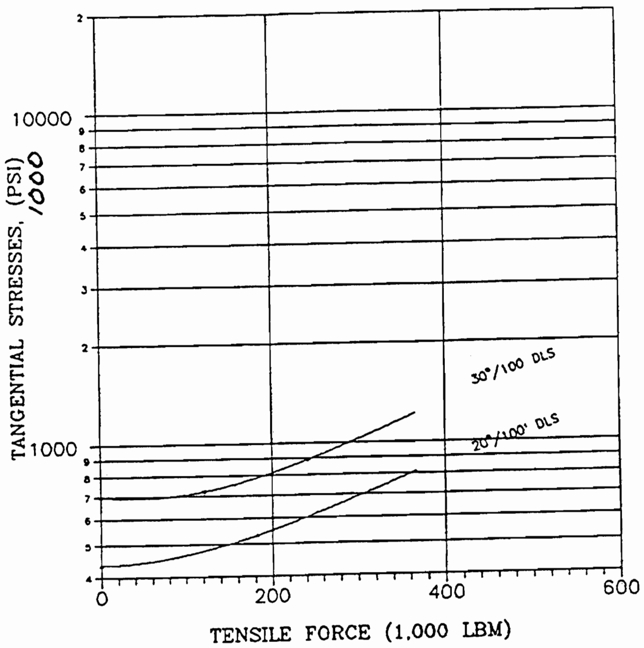
CASING: STEEL, 7*6.336 INCH, 0.332 INCH WALL THICKNESS
COUPLING: STEEL, 7.656 INCH OD, 9 INCH LONG



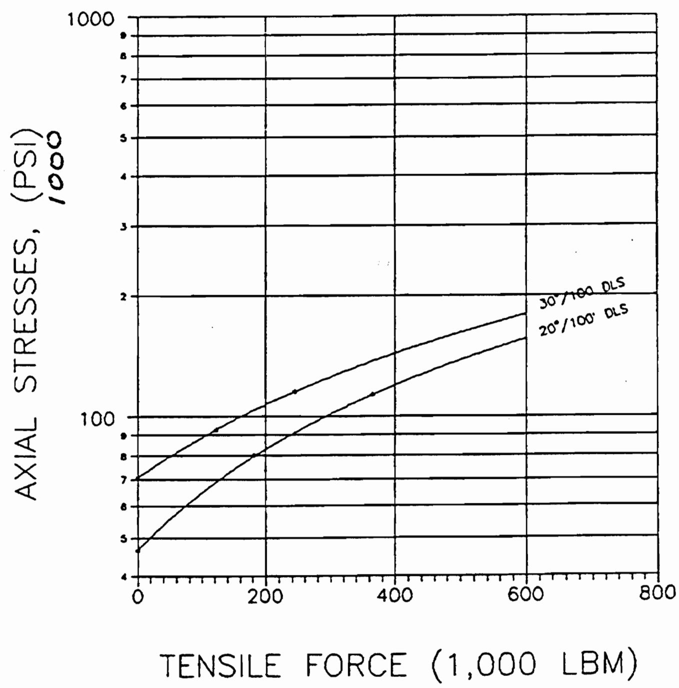
CASING: STEEL, 9.625*8.835 INCH, 0.395 INCH WALL THICKNESS
COUPLING: STEEL, 10.625 INCH OD, 10.5 INCH LONG



CASING: STEEL, 9.625*8.835 INCH, 0.395 INCH WALL THICKN
COUPLING: STEEL, 10.625 INCH OD, 10.5 INCH LONG



CASING: STEEL, 9.625*8.835 INCH, 0.395 INCH WALL THICKNESS
COUPLING: STEEL, 10.625 INCH OD, 10.5 INCH LONG



EQUIPMENT

DRILL COLLARS

ARTICULATED DRILL COLLARS (ADC)
Drill collars which have integral flexible joints. They are used in short radius holes. A rubber hose which is located within the collars carries the drill mud. They are often called flexible pipe.

SPIRAL DRILL COLLARS (SDC)

Spiral drill collars have flats machined on their outer diameter over about 27 feet of their length. The machining of the OD produces a weight loss of about four percent.

NON-MAGNETIC COLLARS (NMDC)

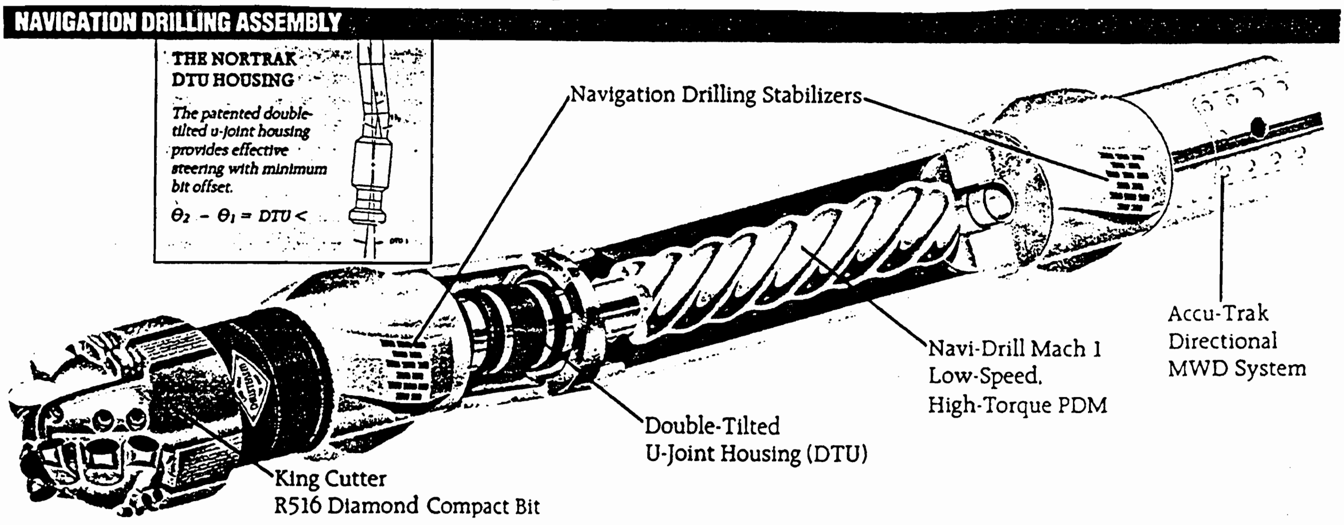
Non-magnetic collars are often called MONEL collars; however, non-magnetic collars are popularly made of stainless steel. Monel is 70% nickle and 30% copper. Non-magnetic collars are manufactured with many types of metal. Selection is based more on corrosion resistance than other criteria. See the BHA design section for length and compass location.

CURVED DRILL GUIDE

A casing which has a manufactured longitudinal curve near its bottom end. It is used in those situations where kick off points are near the surface and where the wells are crowded. The curved guide casing gives an initial azimuth as well as inclination.

POSITIVE DISPLACEMENT ROTARY MOTOR - PDM

This popular rotary positive displacement bottom hole drilling mud and/or air motor was invented by the Frenchman, Moineau. The primary advantage of the Moineau motor over the turbine is its lower operating rpm and higher torque. It is also develops less vibration than the turbine. Moineau motors have straight and bent housings and come in a full range of sizes. The following PDM performance chart is typical.



NORTRAK DIRECTIONAL PERFORMANCE

Tool Size O.D.	Hole Size	UBHS Size O.D.	Typical L ($l_1 + l_2$) MACH 1		Typical L ($l_1 + l_2$) MACH 2		DTU	Bit Offset		Theoretical Dogleg Severity (Degrees per 100 ft/30 m) MACH 1	Theoretical Dogleg Severity (Degrees per 100 ft/30 m) MACH 2
inch	inch	inch	m	ft	m	ft	(°)	mm	inch	MACH 1	MACH 2
4¾	5⅞–7⅞	–⅛	6.6	21.7	7.9	26.0	0.25 0.39 0.52	4.0 5.0 6.9	0.16 0.20 0.28	2.3° 3.6° 4.8°	1.9° 3.0° 4.0°
6¾	8⅜–7⅞	–⅛	7.7	25.1	9.7	31.9	0.32 0.48 0.64	4.9 9.9 10.9	0.20 0.40 0.44	2.5° 3.8° 5.1°	2.0° 3.0° 4.0°
8	9⅞–12¼	–⅛	9.1	29.8	10.2	33.4	0.30 0.64 0.74	5.9 12.9 14.8	0.24 0.52 0.60	2.0° 4.3° 5.0°	1.8° 3.8° 4.4°
9½ (DKO)	12¼–17½	–⅛	9.3	30.5	11.9	38.9	0.38 0.59 (1.00)	7.9 11.9 18.6	0.32 0.48 0.73	2.5° 3.8° 6.0°	2.0° 3.0° 5.0°
11¼	17½–26	–¼	10.8	35.5	12.5	41.0	0.41 0.61 0.78	11.9 14.8 18.8	0.48 0.60 0.76	2.3° 3.5° 4.4°	2.0° 3.0° 3.8°

NORTRAK MOTOR SPECIFICATIONS; MACH 1

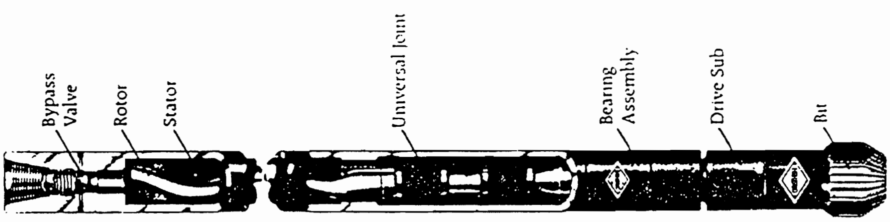
Tool Size O.D.	Recommended Hole Size	Flow Rate*				Bit Speed Range	Operating Differential Pressure		Operating Torque Output**		Maximum Torque		Horsepower Range		Efficiency	Length	
inch	inch	min.		GPM		rpm	bar	psi	Nm	ft-lbs	Nm	ft-lbs	kw	hp	max. %	m	ft.
		min.	max.	min.	max.												
4¾	6 –7⅞	300	900	80	240	100–300	50	725	1600	1180	2560	1890	17–50	22–67	68	5.6	18.5
6¾	8⅜–9⅞	700	1800	185	475	100–260	50	725	3800	2800	6080	4480	40–103	53–139	70	6.6	21.5
8	9½–12¼	1200	2600	315	685	85–190	40	580	6100	4500	9760	7200	54–121	73–163	70	7.9	26.0
9½	12¼–17½	1500	2800	395	740	100–190	55	800	9300	6870	14880	10970	97–185	131–249	72	8.0	26.3
11¼	17½–26	2000	4300	525	1135	80–170	45	655	13200	9770	21120	15570	109–235	149–316	73	9.4	30.6

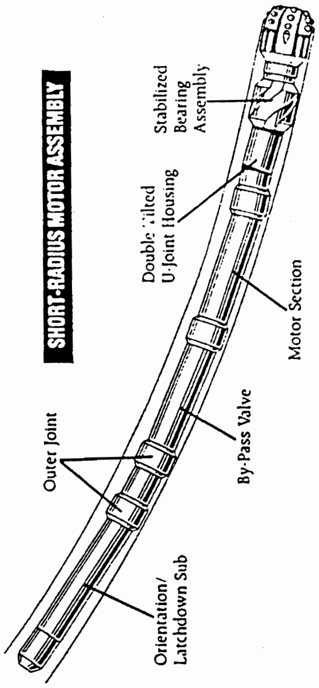
NORTRAK MOTOR SPECIFICATIONS; MACH 2

Tool Size O.D.	Recommended Hole Size	Flow Rate*				Bit Speed Range	Operating Differential Pressure		Operating Torque Output**		Maximum Torque		Horsepower Range		Efficiency	Length	
inch	inch	min.		GPM		rpm	bar	psi	Nm	ft-lbs	Nm	ft-lbs	kw	hp	max. %	m	ft.
		min.	max.	min.	max.												
4¾	6 –7⅞	300	1000	80	265	195–650	50	725	1000	740	1600	1180	21–68	27–92	83	6.9	22.6
6¾	8⅜–9⅞	700	2000	185	530	190–550	50	725	2500	1840	4000	2950	50–144	67–193	86	8.6	28.3
8	9½–12¼	900	2600	240	685	155–450	40	580	3250	2400	5200	3830	53–153	71–206	88	9.0	29.6
9½	12¼–17½	1500	3000	395	790	200–400	60	870	6450	4750	10320	7610	135–270	181–362	90	10.6	34.7
11¼	17½–26	2000	4300	525	1135	155–330	40	580	7300	5530	12000	8850	122–260	163–347	90	11.1	36.2

*Indicated maximum can be increased via installation of rotor nozzle **Operating above this level can shorten tool life.

EASTMAN CHRISTENSEN





TUBULAR SPECIFICATIONS

ARTICULATED DRIVE PIPE (ROTATED THROUGH CURVE)	3¾" System	4¾" System
OD (in.)	3¾"	4½"
ID (in.)	1⁵⁄₁₆"	1⁷⁄₁₆"
Maximum Temperature (° F./°C)	260/127	260/127
Maximum Liner Pressure (psi/Bar)	1500/103	1500/103
Length (ft./m.)	20/6	20/6
Approximate Weight (lb.-ft./Kg./m.)	29.3/44.4	37.3/56.5
CONNECTIONS: Box Up	2⅞" Reg. mod.	3½" FH mod.
CONNECTIONS: Pin Down	2⅞" Reg. mod	3½" FH mod
Recommended Make Up Torque (ft.-lb./Nm)	3500/4745	4500/6100
TUBING (ORIENTED THROUGH CURVE)		
Type	P-105	P-105
OD (in.)	2⅞"	2⅞"
ID (in.)	2.259	2.259
Drift (in.)	2.165	2.165
Length (ft./m.)	30/9	30/9
Approximate Weight (lb.-ft./Kg./m.)	8.7/13.2	8.7/13.2
CONNECTIONS: Box Up	2⅞" PH6	2⅞" PH6
CONNECTIONS: Pin Down	2⅞" PH6	2⅞" PH6
CONNECTIONS: OD (in.)	3½"	3½"
Recommended Make Up Torque (ft.-lb./Nm)	3500/4745	3500/4745

SPECIFICATIONS

	3¾" System	4¾" System
Motor OD (in.)	3¾	4¾
Design Radius (ft./m.)	40/12.2	40 or 60/12.2 or 18.3
Hole Sizes Drilled (in.)	4½ or 4¾	6
Minimum Hole Size Above KOP (in.)	4⅞	6⅛
Minimum Casing Size Above KOP	5½", 17#	7", 26#
Maximum Bottom Hole Temperature (° F./°C)	260/127	260/127

MOTOR DATA

	3¾" System	4¾" System
Speed (rpm)	225-550	150-260
Length (ft./m.)	12-17/3.7-5.2	12-17/3.7-5.2
Weight (lbs./Kgs.)	420/191	650/295
Flow Rate (gpm/lpm)	90-120/341-455	185-230/701-871
Drilling Differential Pressure (psi/Bar)	75-500/5-34	75-720/5-50
Maximum Differential Pressure (psi/Bar)	1000/69	1200/83
Operating Torque (ft.-lb./Nm)	500-700/678-949	350-700/474-949
CONNECTIONS: Box Up	2⅞" PH6	2⅞" PH6
Recommended Make Up Torque (ft.-lb./Nm)	3500/4745	3500/4745
CONNECTIONS: Box Down	2⅜", 2⅞" Reg.	3½" Reg.
Recommended Make Up Torque (ft.-lb./Nm)	3500-5700/4745-7726	9400/12,742

SPECIFICATIONS

	Radius Range	Build Up Rate	Min. Casing in Vertical, in.	Tool Size, in.	Bit Sizes, in.	Max. Casing/ Liner Size, in.
Short-Radius	19-25 ft	3°-2.3°/ft	5½	3¾	4½-4¾	2⅞
	32-38 ft	1.8°-1.5°/ft	7⅝	4¾	6¼-6½	4½
	38-42 ft	1.5°-1.4°/ft	7	4¾	5⅞-6	4½
Medium-Radius[1]	286 ft	20°/100 ft	7	3¾	4¾	2⅞
	716-400 ft	8°-14°/100 ft	7	8	12¼	9⅝
	286-300 ft	20°/100 ft	7	4¾ Mach 1	6-8½	5½-7 FJ*
	716-400 ft	8°-14°/100 ft	9⅝	6¾ Mach 1	8½-9⅞	7 FJ-7⅝
Long-Radius[2]	1900-1000 ft	3°-6°/100 ft	7	4¾	6-8½	5½-7 FJ
	1900-1000 ft	3°-6°/100 ft	9⅝	6¾	8½-9⅞	7 FJ-7⅝

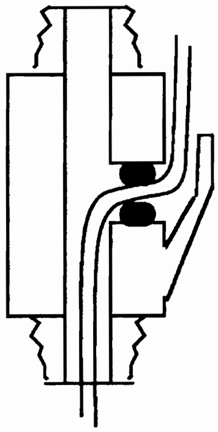
[1]Larger motor sizes/bit diameters/casing sizes are feasible.
[2]Sharper radii possible for special applications.
*Flush-joint

TURBINES

Turbines have not received the acceptance that the PDMs have. However, they maybe beneficial for sidetracking and kicking off a hole in harder rock.

DIRECTIONAL DRILLING SUBS AND STABILIZERS

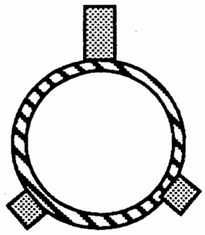
Double Tilt Units (DTU) and Single Tilt Units (STU) have a unit below the PDM which contains universal joints in the drive shaft to the drill bit. Their purpose is to point the drill bit along the path of the drill hole.



SIDE ENTRY SUB

Bent subs have a bend in their bodies. The popular angles for the bend are from 1 degree to 4 degrees. There are permanent and adjustable types of bent subs. Bent subs are very popular. The bend can not be seen.

A side entry sub has a hole in its wall. The purpose of the hole is to allow an electric or other line to be run within a pipe string and to circulate mud while the pipe is in the drill hole. It also allows the additions of drill pipe without pulling the electric cable and instruments.



An offset sub (often called a stabilizer) is a sub which has a wider blade on one side than on the other. Rotating the offset stabilizer allows straight ahead drilling. While motor drilling (not rotating), the path of the drill hole may be altered.

An integral blade stabilizer has the blades machined onto the body of the stabilizer. They are permanent blades and may be helical or straight.

The blades may be replaced on replaceable blade stabilizers. One type has blades machined on a housing while another has "knock out" blades.

The non-rotating stabilizer has rubber blades molded onto a steel housing. Ball bearings between the housing and its mandrel which is screwed into the drill string allows nearly free rotation of the drill string.

The purpose of a near bit stabilizer is to build inclination angle in a hole.

STEERING TOOL

A steering tool is a down hole device which operates within non-magnetic drill collars and gives a surface readout of inclination and azimuth of the drill hole and/or the angle of the tool face from the high side of the hole.

It sends its signal through an electrical cable or a mud pulse. A lubricator sub or a side-entry sub may be used with it. It provides directional surveys of the drill hole.

DMWD

Directional Measurement While Drilling tool (DMWD) is a down hole device which operates within non-magnetic drill collars and gives a surface readout of inclination and azimuth of the drill hole and/or the angle of the tool face from the high side of the hole. It sends its signal to the surface by creating pressure pulses within the mud inside the drill string. A pressure transducer and a computer converts the signals to survey information.

MWD

Measurement While Drilling tool (MWD) is similar to the DMWD described directly above, but may have many more measurement functions. Popular functions are weight on bit, torque, pressure, vibrations at the bit, and formation logging.

Large drillpipe (6-5/8" OD) is popular primarily for the improvement of circulation hydraulics.

COMPRESSION SERVICE DRILLPIPE

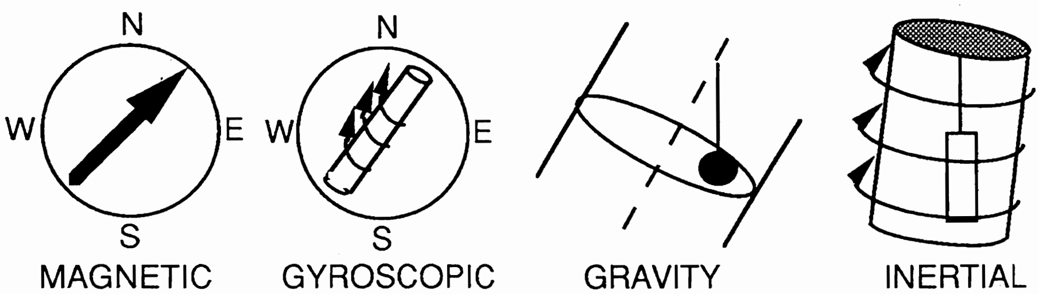
Drillpipe with one, two and three wear pads (often called "knots") between the tooljoints are popular for the purpose of reducing fatigue and erosion of the wall of the drillpipe and its tooljoints. It is called compressive serviceable.

EXTERNAL CASING PACKERS

External casing packers (also known as inflatable packers) are placed on completions strings for the purpose of isolating oil or gas production within the horizontal sections. They are very popular in completion strings which are not cemented. Also they are used to lower the risk of water coning.

SURVEY TOOLS

There are three basic types of survey tools: **magnetic, gyroscopic, and inertial.** Magnetic tools contain an ordinary magnetic north seeking compass for finding azimuth and a liquid level or plumb bob for finding inclination. Non-magnetic drill collars are required.



Gyroscopic tools are based on the law motion that a shaft will not alter its direction in which it is pointing if it is free of forces. A rotating shaft is more stable than a non-rotating shaft. If the gyroscope is pointed north at the surface and then lowered into a hole, the angle change between the housing of the gyroscope and the spinning shaft will find the azimuth of the hole. Non-magnetic drill collars are not required and the tool will operate within ordinary steel tubulars.

Inertial tools are based on the gyroscope, accelerometers, gravity, and the spinning of the earth. Gravitational attraction is toward the center of the earth and the direction is found with the accelerometer. The fact that the earth spins, requires that a shaft pointed in the east-west have the highest forces acting on it if it is turned with the earth, than a shaft pointed in any other direction. A north point shaft would require the least amount of force to turn with the earth. Thus, north is found with the combination of the gyroscope and accelerometer. Eastman Christensen calls their tool the "SEEKER".

SINGLE SHOT

A single shot tool is one in which measurements at a single survey station are taken and the tool is retrieved back to the surface. They are retrieved with a wire line or by pulling the string of pipe.

MULTI-SHOT

A multi-shot tool is one in which a section of the drill hole is surveyed with many surveys. The tool is pulled throughout the section while it resides in a string of pipe or attached to a wireline. Multi-shots may be magnetic, gyroscopic, or inertial tools.

Survey tools are further divided into types which store data in electronic memory, film, tape, or transmit data to a surface readout module through an electric cable.

Surface readout modules are instruments which either receive pressure pulses through the mud within the pipe string or through an electric cable.

PULSER SUB

Pulser sub is a tool which, when commanded to do so from the surface by receiving a signal through the mud, will open a valve allowing mud to flow from within the drill string to the annulus. This in turn produces a short time pressure drop in the drill string which can be detected at the surface. A set sequence of the these pressure drops, which are in fact eight bit binary code, are decoded at the surface and represent data generated by the survey.

BOTTOM HOLE ASSEMBLIES

BHA's for the drilling of horizontal wells may be divided into three categories:

1. Motor BHA's. BHA's in which a bottom hole motor is installed and the motor provides the power of turning the drill bit.

2. Rotary BHA's. The drill string is turned with a rotary table or a power swivel (top drive) at the surface.

3. Steerable BHA's. These have bent subs, tilt sub, offset stabilizer, and a bottom hole motor (curved or straight housing).

All three may have a MWD or a steering tool which in turn requires non-magnetic drill collars.

Horizontal BHA's may not have drill collars other than the non-magnetic type and heavy weight drillpipe.

Typical bottom hole assemblies for long-, medium-, and short-radius wells are the following:

9-7/8" Angle Building	Component	OD inch	Length feet
	Bit	9-7/8	0.90
	Steerable PDM	6-3/4	21.19
	Pony Collar	7	8.17
	Integral Blade Stabilizer	9-3/4	4.46
	MWD Monel	6-1/4	29.24
	MWD Pulser	6-1/4	6.45
	Monel Drill Collar	6-5/16	29.55
	Drill Collars	6-3/8	184.42
	Heavy-Weight Drill Collars	4-1/2	902.86
	Drill Pipe	4-1/2	to surface

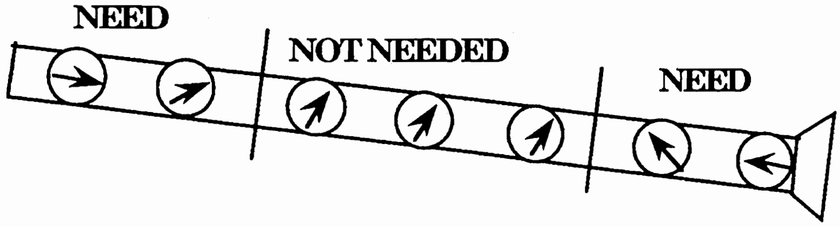
6-1/2" Angle Building	Component	OD inch	Length feet
	PDC Bit	6-1/2	0.80
	Steerable PDM	4-3/4	18.88
	Non-Magnetic Stabilizer	6	5.25
	MWD Monel	4-3/4	31.11
	MWD Pulser Sub	5	3.35
	Float Sub	5-3/8	1.35
	Flexible Non-Magnetic Collars	4-3/4	62.21
	Nipple Sub	5-1/4	1.78
	Heavy-Weight Drill Collars	3-1/2	1748.51
	Drill Collars	4-3/4	448.21
	Heavy-Weight Drill Collars	3-1/2	1265.32
	Drill Pipe	3-1/2	to surface

	Component	OD inch	Length feet
============	=========	=====	======
6-1/2" Short Directional	Bit	6-1/2	0.60
	Directional PDM	4-3/4	10.37
	1-1/2 Degree Bent Sub	4-3/4	1.34
	MWD Monel	4-3/4	31.11
	MWD Pulser Sub	5	3.35
	Float Sub	5-3/8	1.35
	Flexible Non-Magnetic Collars	4-3/4	62.21
	Nipple Sub	5-1/4	1.78
	Heavy-Weight Drill Collars	3-1/2	1748.51
	Drill Collars	4-3/4	448.21
	Heavy-Weight Drill Collars	3-1/2	1265.32
	Drill Pipe	3-1/2	to surface
6-1/2" Lateral Reach	PDC Bit	6-1/2	0.80
	Steerable PDM	4-3/4	18.90
	Pony Collar	4-3/4	6.79
	Integral Blade Stabilizer	6-3/8	2.92
	MWD Monel	4-3/4	31.11
	MWD Pulser Sub	5	3.35
	Float Sub	5-3/8	1.35
	Flexible Non-Magnetic Collars	4-3/4	62.21
	Nipple Sub	5-1/4	1.78
	Heavy-Weight Drill Collars	3-1/2	2923.10
	Drill Collars	4-3/4	448.21
	Heavy-Weight Drill Collars	3-1/2	90.73
	Drill Pipe	3-1/2	to surface

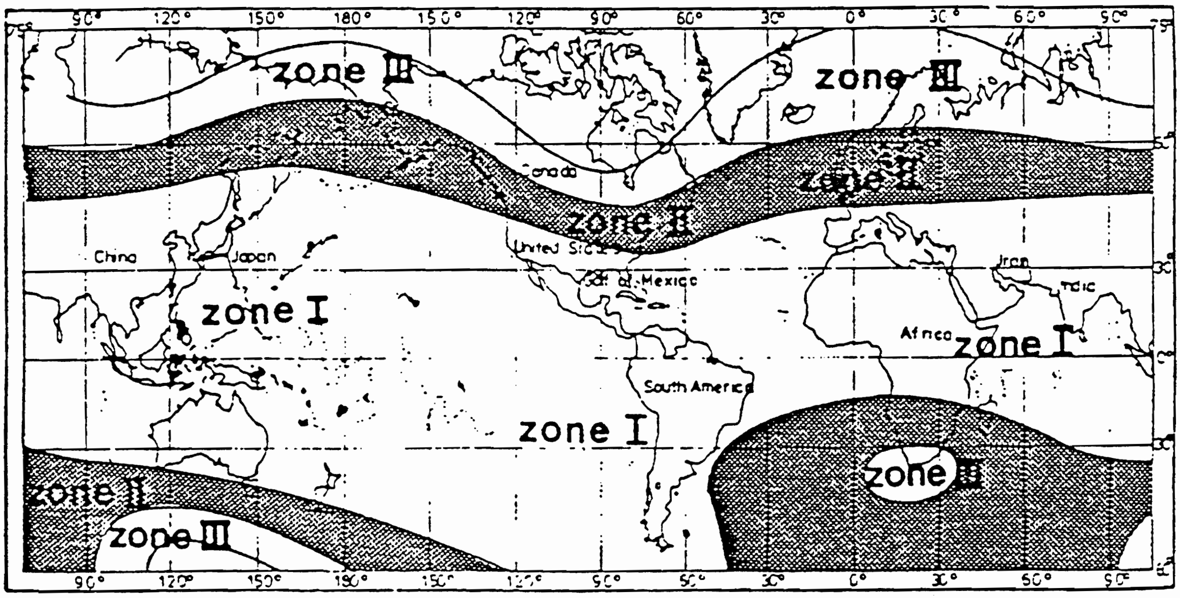
LENGTH OF NON-MAGNETIC DRILL COLLARS

In areas of higher latitudes the horizontal component of the earth's magnetic field is weak. Additional non-magnetic collars are required to remove the magnetic fields of the bit and parts of the bottom hole assembly.

The procedure is to run a drill string which contains extra non-magnetic collars into the bore hole being drilled and then running a magnetic multishot through the length of non-magnetic collars. The length of non-magnetic collars in which the compass is not affect by the magnetization of the parts of the BHA may be removed from the string.

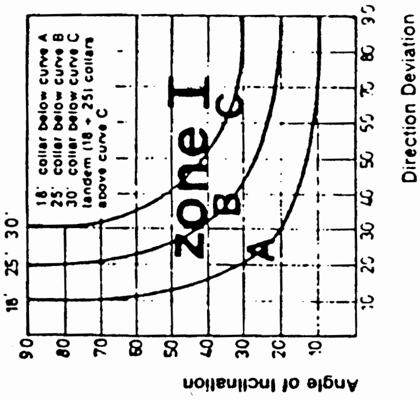


The following charts may be helpful in determining the required length of non-magnetic collars and the location of the surveying compass.



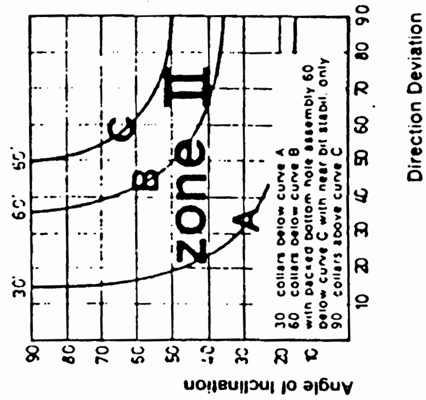
Recommended positioning of compass

Length of drill collars (ft)	Location of compass (ft)
18	1 – 2 under center
25	2 – 3 under center
30	3 – 4 under center
18 + 25	central



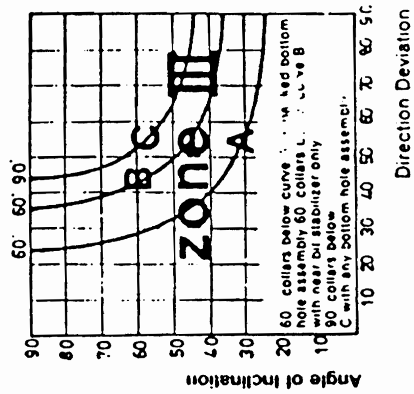
Recommended positioning of compass

Length of drill collars (ft)	Location of compass (ft)
30	3 – 4 under center
60 (B)	central
60 (C)	8 – 10 under center
90	central



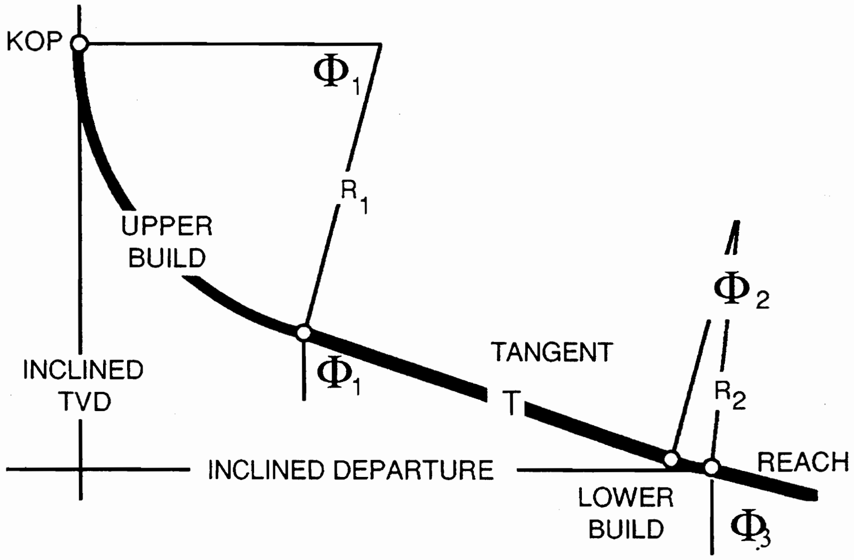
Recommended positioning of compass

Length of drill collars (ft)	Location of compass (ft)
60 (A)	central
60 (B)	8 – 10 under center
90	central



TRAJECTORY PLANNING

The mathematical trajectory planning equations are the same as those used in directional drilling with the exception that the primary unknown in the combination trajectory plan is the depth of the kick off point, KOP. The following development illustrates the equations for locating the KOP.



The inclined portions of the hole without the reach are given by the following three equations. The inclined TVD locates the KOP.

$$\text{Inclined TVD} = R_1 \sin \Phi_1 + T \cos\Phi_1 + R_2(\sin\Phi_3 - \sin\Phi_1)$$

$$\text{Inclined Dep} = R_1(1 - \cos\Phi_1) + T \sin\Phi_1 + R_2(\cos\Phi_1 - \cos\Phi_3)$$

$$\text{Inclined MD} = R_1 * \Phi_1 + T + R_2 * \Phi_2$$

The departure and change in the TVD of the REACH are given by the following two equations. The REACH is the along the hole distance (MD) of portion of the hole which is normal thought to be horizontal.

$$\text{Departure of the REACH} = \text{REACH} * \sin(\Phi_3)$$

$$\text{Change in TVD of the REACH} = \text{REACH} * \cos(\Phi_3)$$

EXAMPLE OF A COMBINATION WITH TANGENT WELL TRAJECTORY

A combination horizontal well with a tangent section is to be planned with the following data. At what depth is the KOP and at what departure does the horizontal section begin. What is the Measured length of the inclined section.

Upper build:	5 deg/100ft to 75 deg
Tangent length:	300 feet
Lower build:	10 deg/100ft to 90 deg
TVD of the horizontal section:	7,204 feet

$\Phi_1 = 75$ deg $\quad\quad \Phi_2 = 15$ deg $\quad\quad \Phi_3 = 90$ deg

$$\text{Inclined TVD} = R_1 \sin \Phi_1 + T \cos\Phi_1 + R_2(\sin\Phi_3 - \sin\Phi_1)$$

$$R_1 = \frac{100}{BG} = \frac{100}{5\frac{\pi}{180}} = 1{,}145.92 \text{ ft} \qquad R_2 = \frac{100}{BG} = \frac{100}{10\frac{\pi}{180}} = 572.96 \text{ ft}$$

$R_1 \sin \Phi_1 = 1{,}145.92 \sin(75)$ **= 1,106.87 ft**

$T \cos\Phi_1 = 300 \cos(75)$ **= 77.65 ft**

$R_2(\sin\Phi_3 - \sin\Phi_1) = 572.96\,(1 - \sin(75))$ **= 19.52 ft**

Inclined TVD = 1,106.87 + 77.65 + 19.52 **= 1,204.04 ft**

Thus, the KOP is at a depth of 6,000 ft (= 7,204 - 1,204).

$$\text{Inclined Dep} = R_1(1 - \cos\Phi_1) + T \sin\Phi_1 + R_2(\cos\Phi_1 - \cos\Phi_3)$$

$R_1(1 - \cos\Phi_1) = 1{,}145.92\,(1 - \cos(75)) = 849.33$ ft

$T \sin\Phi_1 = 300 \sin(75)$ **= 289.78 ft**

$R_2(\cos\Phi_1 - \cos\Phi_3) = 572.96\,(\cos(75) - 0)$ **= 148.29 ft**

Inclined Dep = 849.33 + 289.78 + 148.29 **= 1,287.40 ft**

Thus, the departure of the hole for the inclined section other than the reach is 1,287.40 ft.

$$\text{Inclined MD} = R_1 * \Phi_1 + T + R_2 * \Phi_2$$

$$\text{Inclined MD} = 1{,}145.92 * 75 \frac{\Pi}{180} + 300 + 572.96 * 15 \frac{\Pi}{180}$$

$$= \mathbf{1{,}950\ ft}$$

If the REACH of the hole is 2,000 feet, then its departure and TVD are

$$\text{Departure of the REACH} = \text{REACH} * \sin(\Phi_3)$$

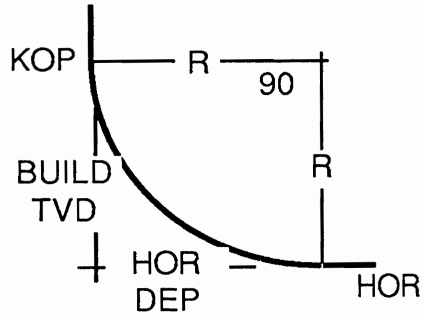
$$\text{Departure of the REACH} = \text{REACH} * \sin(90) \qquad = \mathbf{2{,}000\ ft}$$

$$\text{Change in TVD of the REACH} = \text{REACH} * \cos(\Phi_3)$$

$$\text{Change in TVD of the REACH} = \text{REACH} * \cos(90) \qquad = \mathbf{0\ ft}$$

EXAMPLE OF A SINGLE BUILD WELL TRAJECTORY

A single build horizontal well is to be planned with the following data. At what depth is the KOP and at what departure does the horizontal section begin?



Build Gradient: 12 deg/100ft to 90 deg

TVD of the horizontal section: 7,000 feet

$$\text{Build TVD} = R = \frac{18000}{12\Pi}$$

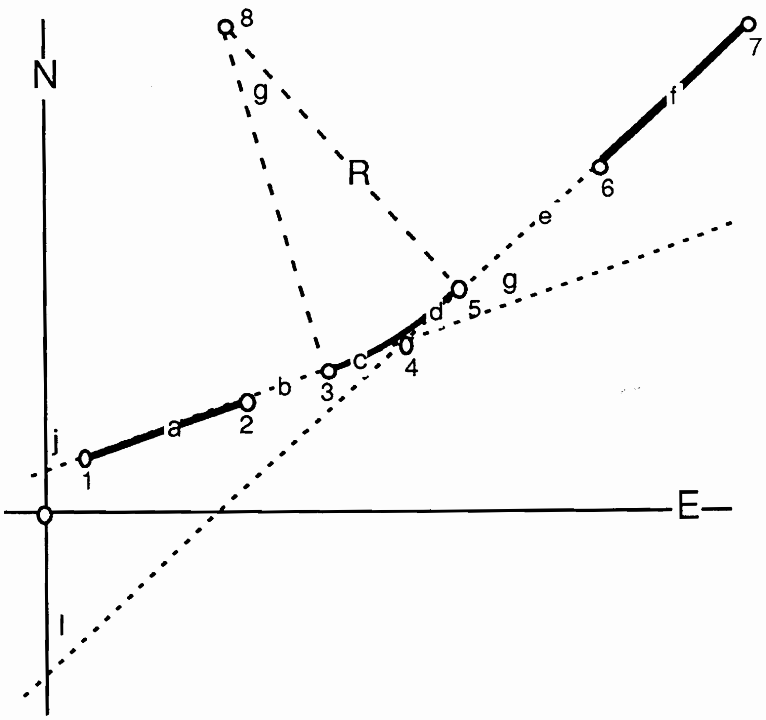
= 477.46 ft

Hor Dep = R **= 477.46 ft**

Thus, the KOP is at a depth **6,522.54** feet (7,000 - 477.46) and the departure of the beginning of the horizontal section is **477.46** feet.

EXAMPLE OF A HORIZONTAL TURN TRAJECTORY

The problem is to plan a horizontal turn in the horizontal section of the planned wellbore. The turn will connect two horizontal sections. The two horizontal sections will be specified by targets at each end of the sections. These ideas are depicted by the sketch. All points and lines in the sketch are in a horizontal plane at the planned horizontal section of the wellbore. The lines and points beginning at point 1 and ending at point 7 represent the planned wellbore. Points 1, 2, 6, and 7 are targets. Lines c and d are the turn arc.



The analytical problem is the computation of the coordinates of points 3 and 5. These are the points where the turn will begin and end.

1. Find the angles φ, λ, and γ

$$\varphi = \text{atan}\left(\frac{E_2 - E_1}{N_2 - N_1}\right) \qquad \varphi = \text{atan}\left(\frac{1210 - 10}{420 - 20}\right) \qquad = \mathbf{71.5651\ deg}$$

$$\lambda = \text{atan}\left(\frac{E_7 - E_6}{N_7 - N_6}\right) \qquad \lambda = \text{atan}\left(\frac{2600 - 2200}{1300 - 1000}\right) \qquad = \mathbf{53.1301\ deg}$$

$$\gamma = \varphi - \lambda = 71.5651 - 53.1301 \qquad = \mathbf{18.4350\ deg}$$

2. Find equations of the lines a and f

$$\text{Slope} = \frac{420 - 20}{1210 - 10} \qquad = \mathbf{0.333}$$

$$N = \frac{1}{\tan\varphi} E + b \quad \text{and} \quad b = N_1 - \frac{1}{\tan\varphi} E_1$$

$$b = 20 - .333 * 10 \qquad = \mathbf{16.667}$$

$N = .333\,E + 16.667$ EQUATION LINE a

$$b = 1000 - .75 * 2200 \qquad = \mathbf{-650.00}$$

$N = .750 * E - 650$ EQUATION LINE f

3. Solve equations for lines a and b to find coordinates of point 4

$$.750\,E_4 - 650 = .333\,E_4 + 16.667$$

$$E_4 \qquad = \mathbf{1{,}600}$$

$$N_4 = .750 * 1600 - 650 \qquad = \mathbf{550}$$

4. Select a turn gradient (TG) for the a turn arc (lines c and d)

TG = 12 deg/100ft along the arc

5. Find the radius of curvature of the turn arc (R)

$$R = \frac{18000}{12\,\Pi} \qquad = \mathbf{477.46}$$

6. Find the length of line d

$$d = R\tan\left(\frac{\gamma}{2}\right) \qquad = 477.46\tan\left(\frac{18.4350}{2}\right) \qquad = \mathbf{77.48}$$

7. Find the combined length of lines d and e

$$d + e = [(1000\text{-}550)^2 + (2200 - 1600)^2]^{1/2} \qquad \mathbf{= 750}$$

NOTE: If d+e < d then a shorter radius of curvature must be selected.

8. Find the coordinates of the outer end of the arc (point 5)

$$N_5 = \frac{d}{d+e}(N_6 - N_4) + N_4$$

$$N_5 = \frac{76.48}{750}(1000 - 550) + 550 \qquad \mathbf{= 595.88}$$

$$E_5 = \frac{595.88 + 650}{.75} \qquad \mathbf{= 1{,}661.18}$$

9. Find the combined length of lines b and c, and note c = d

$$b + c = [(550 - 420)^2 + (1600 - 1210)^2]^{1/2} \qquad \mathbf{= 411.10}$$

NOTE: If b+c < d then a shorter radius of curvature must be selected.

10. Find the coordinates of point 3

$$N_3 = \frac{d}{b+c}(N_2 - N_4) + N_4$$

$$N_3 = \frac{76.48}{411.10}(420 - 550) + 550 \qquad \mathbf{= 525.81}$$

$$E_3 = \frac{525.81 - 16.667}{.333} \qquad \mathbf{= 1{,}527.44}$$

11. Change in measured depth between stations

$$\Delta MD_{3\text{-}2} = \left[(420 - 20)^2 + (1210 - 10)^2\right]^{1/2}$$

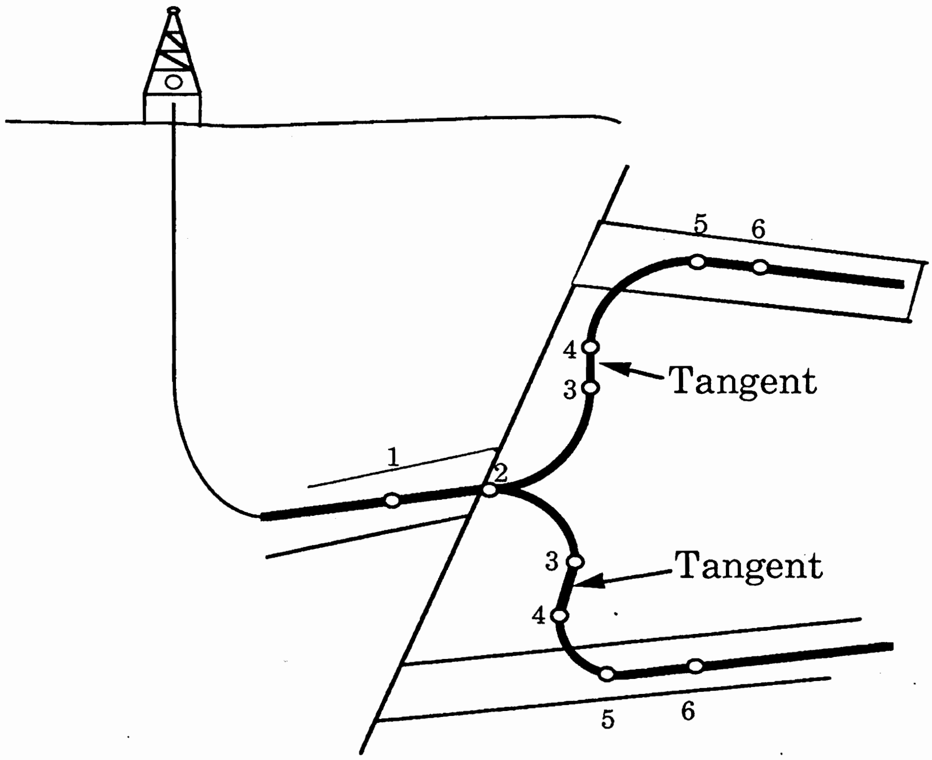
$$\Delta MD_{5\text{-}3} = 477.46 * 18.44$$

The following table summarizes the horizontal turn plan:

HORIZONTAL TURN TABLE

Pt.#	Inc.	Azimuth	TVD	N	E	TG
1	0	71.5651	7000	20	10	0
2	0	71.5651	7000	420	1210	0
3	0	71.5651	7000	525.82	1527.44	12
4	0	62.3476	7000	550	1600	12
5	0	53.1301	7000	595.88	1661.18	12
6	0	53.1301	7000	1000	2200	0
7	0	53.1301	7000	1300	2600	0

VERTICAL TURNS TO A NEW TRACK



As the sketch depicts a vertical turn (build) may be needed to continue to drill in the desired formation.

The Mitchell Engineering computer program "VER_TURN.XLS" can assist in the design of the drill hole. The program requires the following.

1. Inclinations of the drill hole at point 2 and 6
2. Turn gradients (build and drop) between the points 2 and 3 and points 4 and 5
3. Length and inclination of the tangent section between points 3 and 4

The program computes the departures and TVD's of the individual sections and the total departure and TVD of the turn. The measured depth length of the entire turn is also computed.

The procedure for developing a viable design is to enter values into the program until a suitable design is established.

EXAMPLE VERTICAL TURN

VERTICAL TURNS TO NEW TRACK

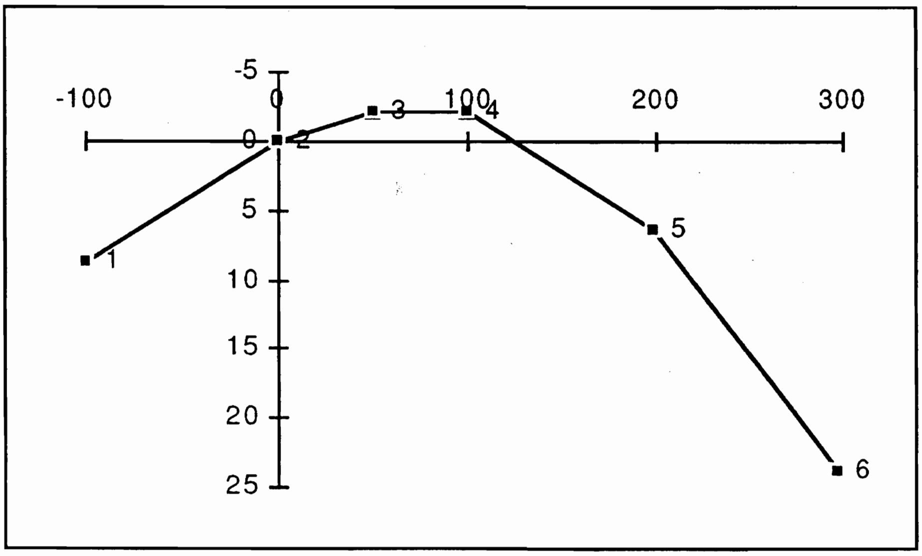
95	Beginning Inclination of Track, (deg)
80	End Inclination of Track, (deg)

10	Begin Turn Gradient, (deg/100ft)
10	End Turn Gradient, (deg/100ft)
50	Length of Tangent Section, (ft)
90	Inclination of Tangent Section, (deg)

57.30

572.96	Radius of Curvature of Begin Turn, (ft)
572.96	Radius of Curvature of End Turn, (ft)
5.00	Angle Change in Begin Turn, (deg)
10.00	Angle Change in End Turn, (deg)
50.00	Length of Begin Arc, (ft)
100.00	Length of End Arc, (ft)
92.50	Chord Angle of Begin Turn, (deg)
85.00	Chord Angle of End Turn, (deg)
49.94	Chord Length of Begin Turn, (ft)
99.49	Chord Length of End Turn, (ft)
49.89	Departure Change in Begin Arc, (ft)
50.00	Departure Change in Tangent Section, (ft)
99.11	Departure Change in End Arc, (ft)
199.00	Total Departure Change, (ft)
-2.18	TVD Change in Begin Arc, (ft)
0.00	TVD Change in Tangent Section, (ft)
8.67	TVD Change in End Arc, (ft)
6.49	Total TVD Change, (ft)
200.00	Change in Measured Depth, (ft md)

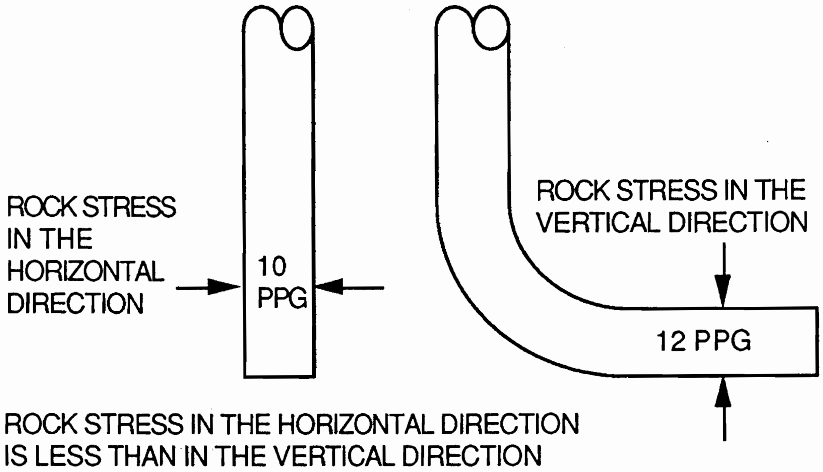
A problem correctly completed will produce the appearance of the figure shown below.



SELECTION OF MUD WEIGHTS FOR HORIZONTAL HOLES

The following equation is suggested for selecting mud weights to stabilize drill holes in horizontal and inclined holes. It is well known that an increase of .1 ppg to .2 ppg in mud weight over that used in vertical wells **will not** stabilize horizontal wells.

HIGHER MUD WEIGHT IN A HORIZONTAL HOLES



$$MW_{horizontal} = MW_{vertical} + (OBW - LOT)\,\frac{1 - \cos 2\varphi}{1.6}$$

$MW_{horizontal}$	= Mud weight to stabilize inclined drill hole; ppg
$MW_{vertical}$	= Mud weight to stabilize vertical drill hole; ppg
OBW	= Overburden weight (overburden stress); ppg
LOT	= Leak Off Test value; ppg
φ	= Drill hole inclination; degrees

EXAMPLE MUD WEIGHT SELECTION IN A HORIZONTAL HOLE

A formation at 7,000 feet has been drilled with vertical holes with little hole instability (maintained gauge) with 11.0 ppg mud. If the leakoff tests have shown values near 16.1 ppg and density logs indicate an overburden weight gradient of 19.0 ppg, what is the mud weight required to stabilize the formation in a horizontal hole?

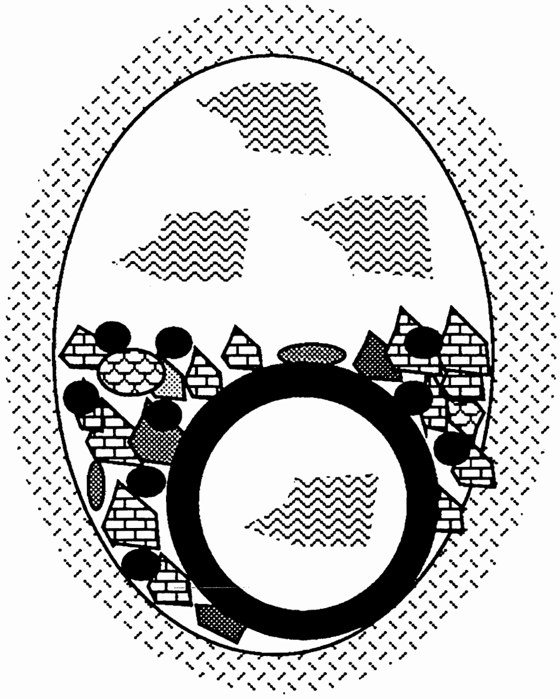
With the above equation,

$$MW_{horizontal} = 11.0 + (19.0 - 16.1)\,\frac{1 - \cos(2 * 90)}{1.6}$$

$MW_{horizontal}$ **= 14.6 ppg**

HYDRAULICS FOR HORIZONTAL HOLES

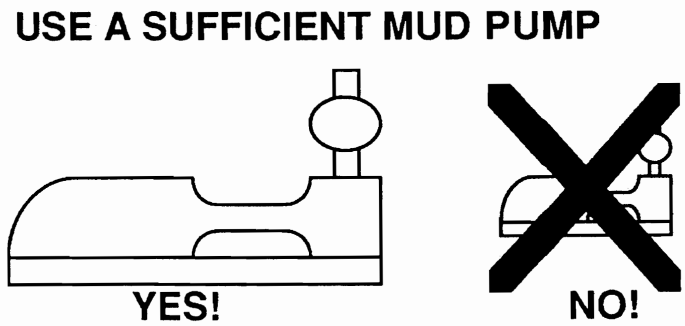
Cleaning of inclined and horizontal holes presents two major concerns which are not present in vertical hole cleaning. One concern is the existence and thickness of a bed of cuttings on the low side of the hole.



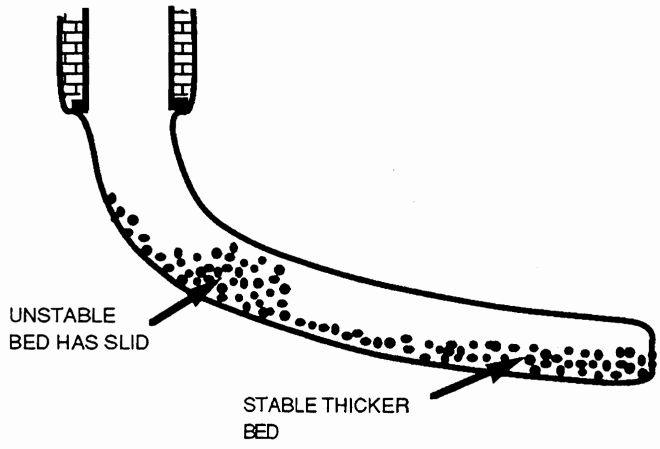
The other is the sliding of a bed of cuttings down hole. In regard to sliding, a bed is said to be stable if it does not have a tendency to slide and unstable if it does. Having a cuttings bed in the hole while drilling places the tripping of the pipe in jeopardy.

Larger mud pumps are required in horizontal boreholes because

1. Much higher circulation rates are required to prevent cuttings beds
2. Mud power is consumed by the bottom hole motor.



Having an unstable bed places the drill string in jeopardy and especially so any time mud circulation is halted. The thicker beds are more likely to form in horizontal hole sections, and unstable beds are more likely in the more vertical hole sections. The sketches show a cross-section of a cuttings bed and stable and unstable hole sections.

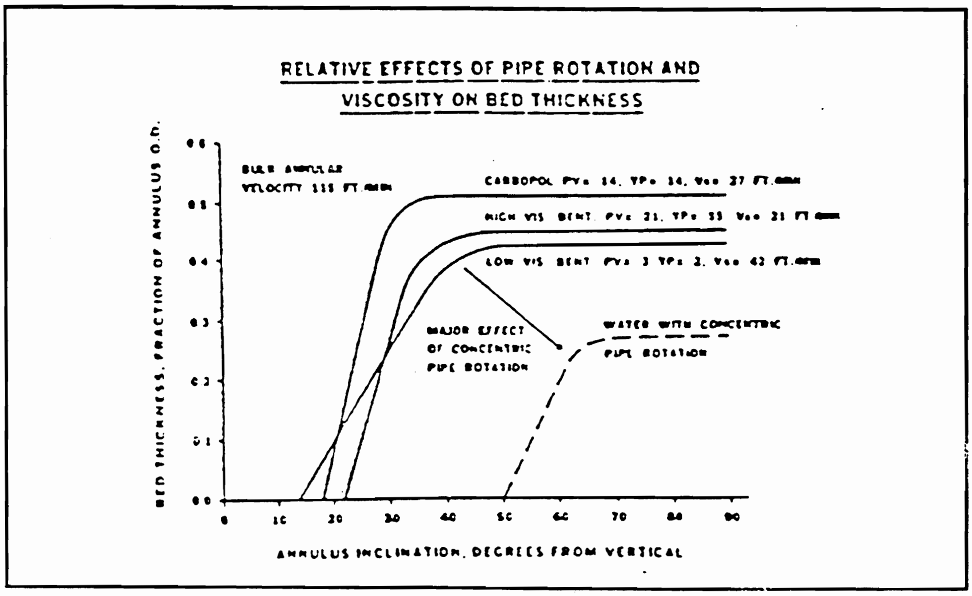


Most cuttings bed problems occur with an oil base mud and normally pressured zones. It is thought that water base muds aid in the disintegration of the cuttings and their subsequent removal. Further, cuttings from over-pressured zones are more buoyant (cuttings less dense and the mud more dense) and have less cohesiveness (internal strength) than cuttings from normally pressured zones and therefore are less likely to form as thick a bed if any at all.

University of Tulsa published a chart which shows that the viscosity of the circulating fluid has little influence on bed thickness; however, circulation rate appears important.

Their data and field data show that a powerlaw Reynold's Number of 1,800 is required to prevent cuttings beds or to remove them.

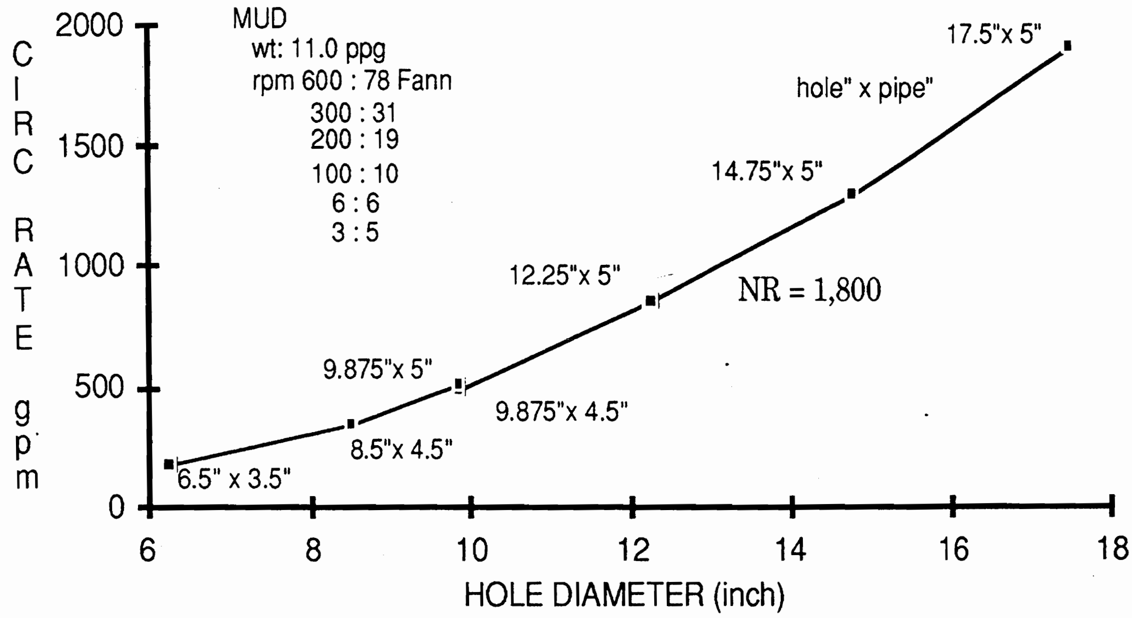
If an oil base mud is being circulated, cuttings beds may not be removed by mud circulation alone and this could be a major problem in cementations and sticking pipe and drillstrings.



N_R = 1800 CLEANS OUT BEDS

The "MINIMUM CIRCULATION RATES" chart was constructed with the Mitchell Engineering computer program "POW_LAW,XLS" and only applies to hole sizes and mud properties as listed in the chart. The chart assumes a Reynold's number of 1,800 for all data points.

MINIMUM CIRCULATION RATES



PD MOTOR and DRILL BIT HYDRAULICS

Maximizing Hydraulic Impact Force

The following steps explain how to maximize hydraulic impact force below the jets of a drill bit with a PDM in the drill string. A PDM uses a percentage of the mud circulation to cool its bearings and this mud does not pass through the jets.

1. The equation, $P_f = j\, Q^m$, satisfactorily models the friction pressure loss in a rig's circulation system. The values of m and j are found with a standard rig pump test. A pump test is run with the bit off bottom while pumping through the MWD, PDM, and the jets. With the bit off bottom the pressure drop through the PDM will be its friction pressure loss because the motor will be doing no work. The pressure drop will be caused by either the friction in the system or the drop across the jets. The friction pressure loss is found by subtracting the jet drop from the pump pressure at each circulation rate. A minimum of two circulation rates and pump pressures are required. Thereafter, the last two equations are solved simultaneously by j and m.

2. Let the circulation rate through the pump be Q and the circulation through the jets be Q_j. The bearing circulation portion will be the difference between these two rates and is

   $Q_b = Q - Q_j$ for the conservation of space; let $\frac{Q_j}{Q} = R$

3. The hydraulic power consumed by friction is

   $$P_f Q = P_p Q - P_j Q_j = j\, Q^{m+1} \quad \text{and} \quad P_f = P_p - P_j \frac{Q_j}{Q} = j\, Q^m$$

4. The values of m and j are given by the equations

   $$\mathrm{Ln}\,(P_p - R\,P_j)_1 = \mathrm{Ln}\, j + m\, \mathrm{Ln}\, Q_1$$

   $$\mathrm{Ln}\,(P_p - R\,P_j)_2 = \mathrm{Ln}\, j + m\, \mathrm{Ln}\, Q_2$$

   $$m = \frac{\mathrm{Ln}\dfrac{(P_p - R\,P_j)_2}{(P_p - R\,P_j)_1}}{\mathrm{Ln}\dfrac{Q_2}{Q_1}} \qquad j = \frac{P_p - R\,P_j}{Q^m}$$

5. Bit hydraulic impact force is given by the equation

   $$IF = \frac{R\, Q\, (MW\, P_j)^{1/2}}{57.66}$$

6. At the condition of maximum hydraulic impact force to the jets of the bit, the two following equations are valid.

$$P_f = \frac{2}{m+2} * P_p$$

$$P_j = \frac{m}{m+2} * P_p$$

7. Substitutions for the variables P_f and P_j into the last equation gives

$$j\,Q^m = \frac{2}{m+2} * P_p$$

$$Q^m = \frac{2}{j(m+2)} P_p$$

8. Solving for Q gives the Q which maximizes the bit impact force with the constraint of a selected pump pressure. This Q is called optimal Q*

$$Q^* = \left[\frac{2}{j(m+2)} P_p\right]^{1/m}$$

9. The optimal jet numbers are ascertained by any trial and error procedure with the equation

$$P_j^* = MW * \left\{\frac{12.51 * R\,Q^*}{J1^2 + J2^2 + J3^a}\right\}^{1/2}$$

The computer program "TFA_HYD.XLS" within the MITCHELL ENGINEERING COMPUTER PROGRAMS accomplishes the tasks outlined above. The final design is a composite of the selected circulation rate and jet sizes.

TORQUE AND DRAG

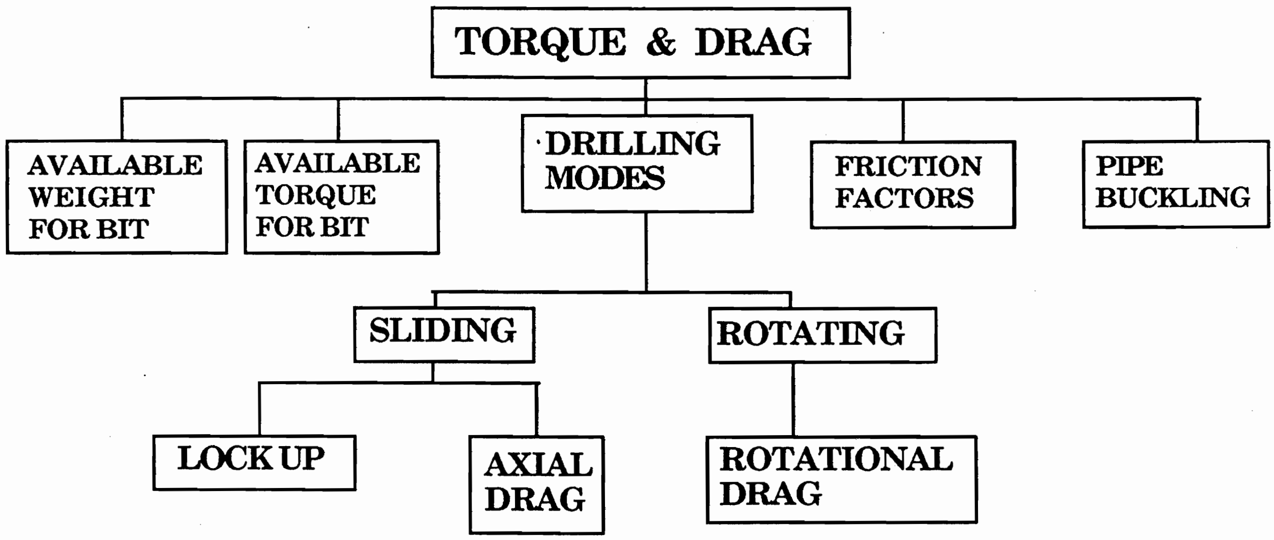
'Torque and drag' is a broad term which refers to the effects which the geometry and other aspects of the drill hole may have on the turning and pulling of the drill string.

In regard to the drill string there are two drilling modes; sliding and rotating. In the 'sliding' mode the drill string is not rotated and torque is low; however axial drag is high and lock-up is possible.

Lock-up is the buckling of a section of the drill string within the drill hole and prevents the transmission of force to the bit or BHA.

In the 'rotational' mode the drill string is rotated at a rate which reduces the drag to a trivial force; lock-up is thought to be impossible; however, torque is high.

Other aspects of torque and drag are the maximum drill string weight available for the drill bit, drill string buckling (lock-up), friction factors, and maximum available torque for the drill bit.



ORIGIN OF DRAG

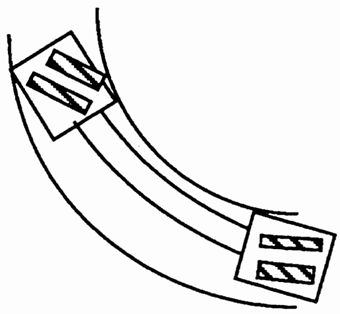
The maximum tolerable drilling drag is governed by the strength of the wall of the drillpipe and tooljoints and connections. The strengths after the consideration of wear of these components may be found in API Recommendation RP 7G. The factors which induce drag on the drill string are the following:

1. Doglegs not only increase drag but consume part of the strength of the drill string by bending it. Doglegs increase the contact force between the drill hole and the drill string.

2. Sharp shoulders on the components of the drill string. Often even in open holes drillpipe rubbers reduce drag and torque.

3. Thick filter cakes and especially if filled with drill cuttings.

4. Ledges and sharp turns in the hole. Especially course correction turns which are not smooth doglegs and are sufficiently short, bind sections of the BHA. The sketch captures the idea. This factor is also thought to cause uneven stabilizer blade wear. Usually the upper or lower end of the blades wear more than the center of the blades.



5. A mud without lubricity. Sometimes such things as walnut hulls help.

6. Cuttings beds and they can be even more troublesome if while pulling out of the hole, the bed is "pushed up" above the BHA. Cuttings beds are classified as stable and unstable. If when circulation is stopped the bed slides down the hole, the bed is said to be unstable; if not, it is stable. The sliding of the bed can cause immediate sticking of the drill string.

7. Swelling beds can be a surprise in horizontal holes if it is not known that higher mud weights must be used over those in vertical holes.

DRILL STRING DRAG MEASUREMENT

The measurement of drag within a drill hole may be ascertained by observing three weights of the drillstring with the rig floor weight indicator.

1. while rotating 1 to 2 feet off bottom
2. while pulling the drillstring at a rate of 10 to 20 feet per minute
3. while lowering the drillstring at a rate of 10 to 20 feet per minute

NET AVAILABLE WEIGHT FOR THE DRILL BIT

The net available weight for the drill bit is the sum of the products of the buoyed weight and the true vertical length of each section of the drill string minus the axial friction which the section generates. The equation is

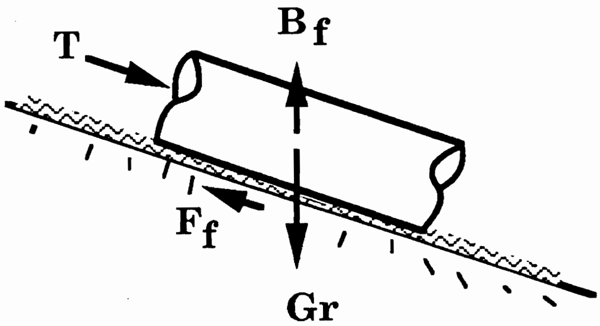
$$\text{NET AWOB} = \Sigma(B_f * W_s * \text{TVL} - F_f)_i$$

AWOB	=	max. available weight for the drill bit; lb
B_f	=	buoyancy factor
W_s	=	air weight of a section of the drill string; lb/ft

TVL	=	true vertical length of a section; ft
F_f	=	axial friction acting on a section; lb
i	=	index representing each section of the drill string

MAXIMUM AVAILABLE WEIGHT FOR THE BIT

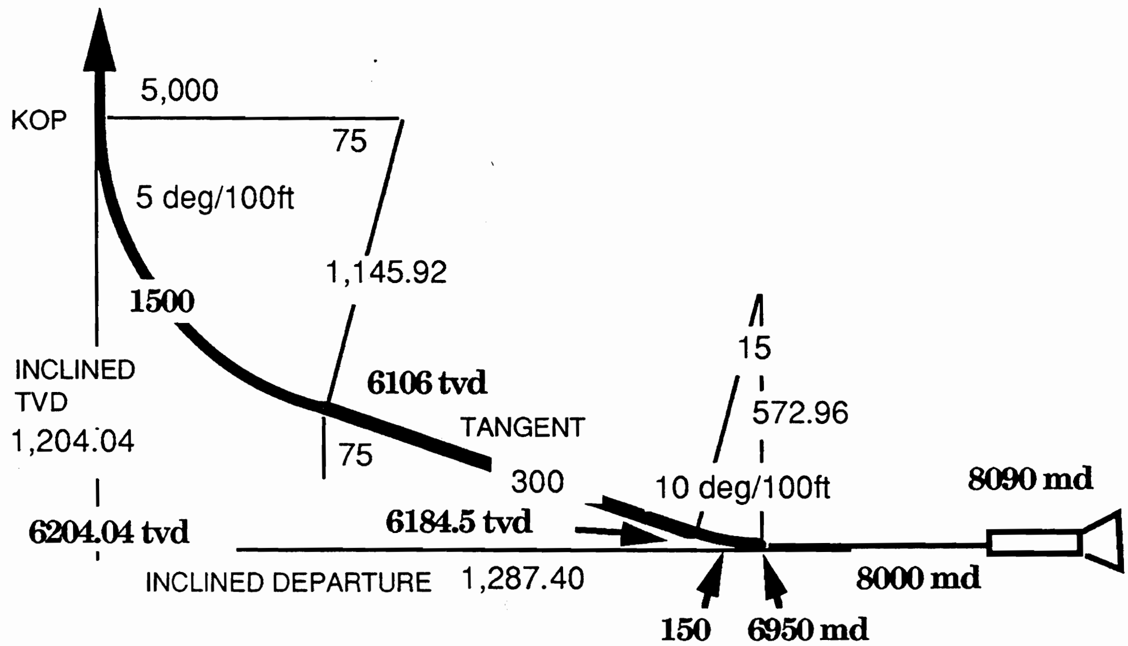
The maximum available weight for the drill bit, which may be generated by the drill string, is the sum of the buoyed weights of all the sections multiplied by their true vertical lengths. The equation is



$$\text{MAX. AWOB} = \Sigma(B_f * W_s * TVL)_i$$

MAX. AWOB	=	max. available weight for the drill bit; lb
B_f	=	buoyancy factor
W_s	=	air weight of a section of the drill string; lb/ft
TVL	=	true vertical length of a section; ft
i	=	index representing each section of the drill string

EXAMPLE MAXIMUM AVAILABLE WOB



The maximum available weight on bit (WOB) for a tangential horizontal well is planned with the drill bit and BHA in the horizontal section, which is the important case. Let the horizontal plan which has been solved in a previous example be as shown in the sketch. Let the drillpipe be composed of 8,000 feet of

5"(OD) by 4.276"(ID) drillpipe with an average weight of 21.6 ppf and the BHA be 90 feet with average diameters of 7" by 2.5"(114 lb/ft). Mud weight is 11.0 ppg.

The gravity weight of the drillpipe or any other section of the drill string is its buoyed weight times its true vertical length.

$$\text{Gr. Wt.} = (1 - \frac{MW}{65.45}) * \text{Air Weight} * \text{TVL}$$

For the above drill string, the gravity weight of the drillpipe is

$$\text{Gr. Wt.} = (1 - \frac{11.0}{65.45}) * 21.6 * (5000 + 1{,}204.04) \qquad \mathbf{= 111{,}484\ lb}$$

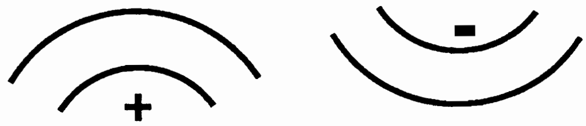
The gravity weight of the drillpipe below and including the tangent section and BHA is

$$\text{Gr. Wt.} = (1 - \frac{11.0}{65.45}) * 21.6 * 98.04 + 0 \qquad \mathbf{= 1{,}761\ lb}$$

DRAG (AXIAL FRICTION)

Axial friction forces may be precipitated by three factors; the buoyed weight of the drill string lying on the low side of the drill hole; the bending of the drill string in a dogleg; and the packing of the drill string in a cuttings bed. The "hanging up" of a component of the drill string on a ledge or other unique obstruction would not be continuously present while drilling and is therefore not thought of as friction.

In the following equation the '+' sign is used if the dogleg is in the shape of an arch and the '-' sign is used if the dogleg is sagging.



The equation for frictional drag is

gravity dogleg packing

$$F_f = \Sigma[\mu * B_f * W_s * DL \ \pm \ \mu * 2 * T * \sin(\frac{DLS * LOS}{2}) \ + \ BP]_i$$

F_f	=	axial friction force; lb
B_f	=	buoyancy factor; lb/lb
W_s	=	air weight per foot of a section of the drill string; lb/ft
DL	=	departure length of a section of the drill string; ft
T	=	axial tension in the section; lb
DLS	=	dogleg severity in which the section lies; deg/100ft
LOS	=	length of the section in the dogleg; ft
BP	=	axial packing force of a cuttings bed; lb

μ	=	coefficient of friction between the drill string and hole; lb/lb
i	=	index representing each section of the drill string

FRICTION FACTORS

Coefficients of friction between the drill string and drill hole with unknown cuttings bed packing, which are often called friction factors, have been reported in the range of 0.24 to 0.28 by north sea operators.

Gravity pulls pipe and BHA's only toward the center of the earth. Friction between the wall of the hole and the drill string vanishes (becomes zero) during the rotation of the drill string. All of the weight of any part of the drill string which is in a hole where the inclination is 90 degrees is supported by the low side of the wall of the hole and none of its weight is placed on the drill bit.

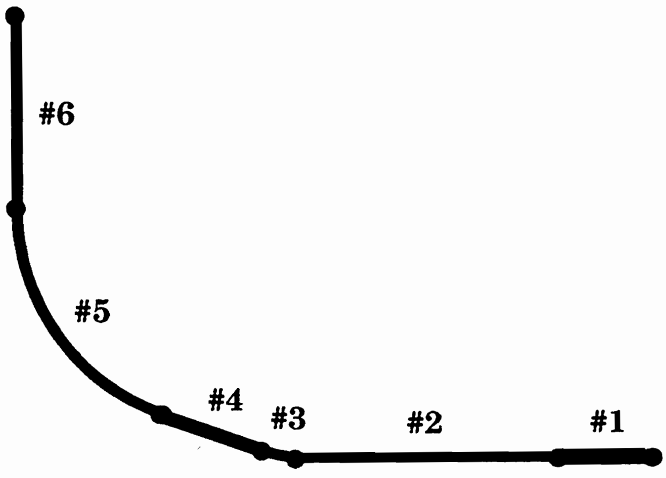
Thus, in a drill hole only the sections of the drill string which reside in the vertical sections or inclined sections can place weight on the drill bit.

Stabilizers and reamer can steal weight away from the drill bit if they are enlarging an under gauge hole, cutting a ledge on the low side, or cocked in the hole.

EXAMPLE DRAG (AXIAL FRICTION)

Compute the drag for the drill string and drill hole in the previous example problem.

Segregate the drill string into six sections: (1) the BHA, and (2) the drillpipe in the reach, (3) the lower build, (4) the tangent section, (5) the upper build, and (6) the vertical section.



The total drag of the drill string will be found by summing the individual drags of each section of the drill string.

The drag (friction force) for any one section of the drill string is given by the formula

$$F_f = \underbrace{\mu * B_f * W_s * DL}_{\text{gravity}} \pm \underbrace{\mu * 2 * T * \sin\left(\frac{DLS * LOS}{2}\right)}_{\text{dogleg}} + \underbrace{BP}_{\text{packing}}$$

DRAG OF THE BHA IN SECTION #1

The BHA is in the horizontal section of the drill hole, the drill hole is straight, and there is no cuttings bed.

$$F_f = .28 * (1 - \frac{11.0}{65.45}) * 114 * 90 \pm .28 * 2 * 0 * \sin(\frac{0 * 90}{2}) + 0$$

$$F_f \qquad = \mathbf{2{,}390\ lb}$$

DRAG OF DRILLPIPE IN SECTION #2

The drillpipe in section #2 is in a horizontal drill hole, the drill hole is straight, and the cuttings bed drag is 5,000 lb.

$$F_f = .28 * (1 - \frac{11.0}{65.45}) * 21.6 * (8000 - 6950) + 5{,}000$$

$$F_f \qquad = \mathbf{10{,}283\ lb}$$

DRAG OF THE DRILLPIPE IN SECTION #3

The drillpipe in section #3 is in the lower build of the drill hole, the drill hole has a dogleg, and the cuttings bed drag is 1,000 lb.

The average tension within the drill pipe through the dogleg must be found before the drag. Calculus gives the following equation for the tension generated within a circular dogleg.

$$T = R * B_f * W_s * [\cos I_1 - \cos I_2]$$

T	=	tension; lb
R	=	radius of curvature of the dogleg; ft
W_s	=	air weight of pipe or tool; lb/ft
L	=	length of section of drill string; ft
B_f	=	buoyancy factor
I_1	=	first inclination angle of the drill hole; deg
I_2	=	second inclination angle of the drill hole; deg

The tension generated by the drill string within the dogleg is

$$T = 572.96 * (1-11/65.45) * 21.6 * [\cos 75 - \cos 90]$$

$$T \qquad = \mathbf{2{,}665\ lb}$$

The average tension within the dogleg is the tension from the sections below the dogleg and one-half of the tension generated within the dogleg.

$$T_{avg} = 2{,}390 + 10{,}283 + \frac{2665}{2} \qquad = \mathbf{14{,}006\ lb}$$

The departure length of the lower build is

$$DL = 572.96 * \sin 15 \qquad = \mathbf{148.29\ ft}$$

The drag generated in section #3 is

$$F_f = .28 * (1 - \frac{11.0}{65.45}) * 21.6 * 148.29 - .28 * 2 * 14006 * \sin(\frac{.1 * 150}{2}) + 1{,}000$$

$$F_f \qquad = \mathbf{722\ lb}$$

DRAG OF THE DRILLPIPE IN SECTION #4

The drillpipe in section #4 is in the tangent part of the drill hole, the drill hole has no dogleg, and has no cuttings bed.

The departure length of the tangent section is

$$DL = 300 * \sin 75 \qquad = \mathbf{289.78\ ft}$$

$$F_f = .28 * (1 - \frac{11.0}{65.45}) * 21.6 * 289.78 - .28 * 2 * 0 * \sin(\frac{0 * 300}{2}) + 0$$

$$F_f \qquad = \mathbf{1{,}458\ lb}$$

DRAG OF THE DRILLPIPE IN SECTION #5

The drillpipe in section #5 is in the upper build of the drill hole, the drill hole has a dogleg, and does not have a cuttings bed.

The tension generated by the drill string within the dogleg is

$$T = 1145.92 * (1-11/65.45) * 21.6 * [\cos 0 - \cos 75]$$

$$T \qquad = \mathbf{15{,}262\ lb}$$

The average tension within the dogleg is the tension from the sections below the dogleg and one-half of the tension generated within the dogleg.

$$T_{avg} = 2{,}390 + 10{,}283 + 722 + 1{,}458 + \frac{15262}{2} \qquad = \mathbf{22{,}484\ lb}$$

The departure length of the upper build is

$$DL = 1{,}145.92 * (1 - \cos 75) \qquad = \mathbf{849.33\ ft}$$

The drag generated in section #5 is

$$F_f = .28 * (1 - \frac{11.0}{65.45}) * 21.6 * 849.33 - .28 * 2 * 22{,}484 * \sin(\frac{.05 * 1500}{2}) + 0$$

F_f = **-3,392 lb** (The minus sign means that the friction is acting on the upper side of the drill hole.)

DRAG OF THE DRILLPIPE IN SECTION #6

The drillpipe in section #6 is in the vertical part of the drill hole, the drill hole has no departure length, the drill hole has no dogleg, and does not have a cuttings bed. Consequently, there is no drag.

TOTAL DRAG OF THE DRILL STRING

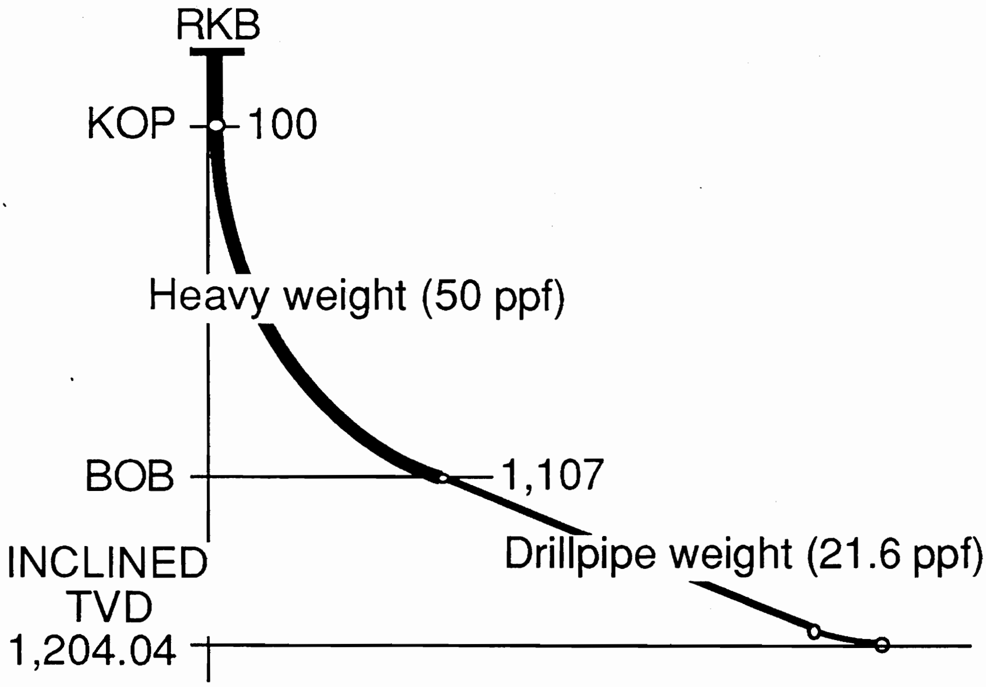
The total drag of the drill string is the sum of the drags in the sections

Drag = 2,390 + 10,283 + 722 + 1,458 + 3,392 + 0 = **18,245 lb**

EXAMPLE AVAILABLE WEIGHT FOR BIT

Let the KOP point be raised to a depth of 100 feet in the previous example and let the drag in the hole be 20,000 pounds while not rotating. Other variables are unchanged.

$$\text{Wt.} = (1 - \frac{11.0}{65.45}) * 21.6 * (100 + 1{,}204.04) \qquad \mathbf{= 23{,}432\ lb}$$



The available weight for the drill bit is 23,432 pounds while rotating and may be sufficient; but while not rotating only 3,432 pounds are available which is not sufficient after deducting the drag.

A solution is to run heavy weight drill pipe (50 lb/ft). The heavy weight drillpipe will provide the most effective weight if it is run in the most steep sections of the hole and, in this case, this is the top of the hole.

The true vertical length of the upper build is 1,107 feet (see a previous example). If heavy weight drillpipe were run from the surface to the bottom of the upper build, the following gravity weight would be available from this section.

$$\text{Gr.Wt.} = (1 - \frac{11.0}{65.45}) * 50 * (100 + 1{,}107) \qquad \mathbf{= 50{,}207\ lb}$$

The gravity weight of the 5" drillpipe below the heavy weight is

$$\text{Gr. Wt.} = (1 - \frac{11.0}{65.45}) * 21.6 * (1{,}204 - 1{,}107) \qquad \mathbf{= 1{,}743\ lb}$$

The total available weight for the drill bit is the sum of the drillpipe and the heavy weight

$$\text{Gr. Wt.} = 50{,}207 + 1{,}743$$

= 51,950 lb

BUCKLING OF THE DRILL STRING

R.F. Mitchell wrote an equation which predicts the buckling of pipe in an inclined hole. The gist of the equation is that gravity pulls the pipe to the low side of the hole which tends to straighten the pipe, the stiffness of the pipe tends to keep it straight, and loads on the ends of the pipe tend to buckle it. His equation is

$$BL = 1{,}617 \left[\frac{B_f * (D^2 - d^2)\ (D^4 - d^4) * SIN(\beta)}{H - D}\right]^{.5}$$

BL	=	Load required to buckle the pipe; lbs
B_f	=	Buoyancy factor
$W_{air\ BHA}$	=	Air weight of the BHA; lbs
ß	=	Inclination angle of the hole; deg
D	=	Outside diameter of the drillpipe; inch
d	=	Inside diameter of the drillpipe; inch
H	=	Hole diameter (not bit diameter); inch

EXAMPLE DRILLSTRING BUCKLING

5" by 4.276" drillpipe is in a horizontal section of a 12.25" hole. Mud weight is 11.0 ppg. What is the maximum load which may be placed on the ends of the drillpipe?

$$B_f = 1 - \frac{11}{65.45} \qquad \mathbf{= .832}$$

Available loading on the drillpipe is

$$(D^2 - d^2) = (5^2 - 4.276^2) \qquad \mathbf{= 6.716}$$

$$(D^4 - d^4) = (5^4 - 4.276^4) \qquad \mathbf{= 290.7}$$

$$SIN(\beta) = SIN(90) \qquad \mathbf{= 1.0}$$

$$\frac{B_f * (D^2 - d^2)(D^4 - d^4) * SIN(\beta)}{H-D} = \frac{.832 * 6.716 * 290.7 * 1.00}{12.25 - 5} \qquad = \mathbf{224}$$

$$1{,}617\left\{\frac{B_f * (D^2 - d^2)(D^4 - d^4) * SIN(\beta)}{H-D}\right\}^{.5} = 1{,}617 * 224^{.5} \qquad = \mathbf{24{,}204\ lb}$$

The load which may be placed on the drillpipe without damaging buckles is 24,204 lb.

LOCK-UP OF THE DRILL STRING

Lock-up occurs when the axial effective compression at any location within the drill string is equal to or greater than drill string's resistance to buckling at that location. The drill string must be in the sliding mode (not rotating). The axial effective compression is the weight on bit, the drag of the drill string from the bit to the location, and is reduced by the gravity weight.

EXAMPLE LOCK-UP

In the example problem on drag of the components of the drill string, it was computed that the expected drag from the BHA, the drillpipe in the reach, in the lower build, and the tangent section was 14,853 lb. The available gravity weight for the bit for that same section of drillpipe was computed to be 1,761.

In the above example it was learned that the resistance to buckling of 5" drillpipe for the conditions of that example was 24,204 lb.

Thus, the maximum weight which can be placed on the bit without lock-up is

Lock-up WOB = 24,204 - 14,853 + 1,761 = **11,112 lb**

AVAILABLE TORQUE FOR THE DRILL BIT

Excessive torque may limit the length of the horizontal section. The torque available to turn the drill bit while drilling in the rotary mode will be constrained by one of the following.

1. The maximum torque possible by the rotary table
2. The torsional strength of the weakest connection or tooljoint
3. The torsional strength of the wall of the thinnest pipe

The torsional strengths of the drillpipe and tooljoints are written in Tables 2.2 to 2.12 and for rotary shoulder connections in Table 3.2 of RECOMMENDED PRACTICE FOR DRILL STEM DESIGN AND OPERATING LIMITS, API recommended Practice 7G (RP 7G), April 1, 1989.

API recommends reducing the torsional strength of drillpipe while it is in tension. The API formula (see p.66 RP 7G) is

$$Q = \frac{.096167\ J}{D}\left[Y^2 - \frac{T^2}{A^2}\right]^{.5}$$

Q	=	Minimum torsional yield strength under tension; lb-ft
J	=	Polar moment of inertia; $\frac{\Pi}{32}(OD^2 - ID^2)$; in^4
D	=	Outside diameter; in
Y	=	Minimum unit yield strength; psi
T	=	Tensile load; lb
A	=	Area of Wall of tube; in^2

EXAMPLE OF ALLOWABLE DRILLING TORQUE

While drilling in the rotary mode with 5" by 4.276" drillpipe and a tension of 100,000 pounds, what is the maximum allowable rotary torque? Let the BHA's connections be 5.5" API REGULAR on 7" by 2.5" collars.

The make-up torque for rotary shouldered connections stresses the steel to 62,500 psi which is about 50% of the steel's unit yield strength. The make-up torque for a 5.5" API REGULAR connection is **39,000** lb-ft .

The make-up torque of 5.5" Full Hole, premium class, tooljoint is **28,800** lb-ft .

The torsional strength of 5", E grade, premium class drillpipe is **32,285** lb-ft .

The torsional strength of 5.5" API Regular Rotary shouldered connection on 7" by 2.5" collar is 31,500 lb-ft (page 35 of RP 7G).

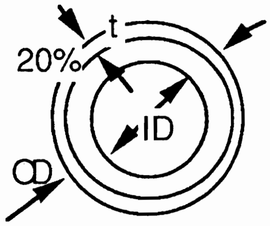
The allowable torque with the API formula is

$$Q = \frac{.096167 * 28.538}{4.855}\left[75000^2 - \frac{100000^2}{4.154^2}\right]^{.5} \qquad \mathbf{= 40{,}152\ lb\text{-}ft}$$

$$J = \frac{\Pi}{32}(5^4 - 4.276^4) \qquad \mathbf{= 28.538\ in^4}$$

The API premium class assumes a 20% reduction in the wall thickness of the drillpipe and it is uniformly worn off the outside diameter. The worn OD is

$$OD_{worn} = ID + t * (1 - wear) * 2$$



$$OD_{worn} = 4.276 + .362 * (1 -.2) * 2 \quad = \mathbf{4.855\ in}$$

$$A = \frac{\Pi}{4}(4.855^2 - 4.276^2) \quad = \mathbf{4.154\ in^2}$$

The allowable torque must be the lowest of the group and is **28,800** lb-ft belonging to the tooljoint of the drillpipe.

CEMENTING CASING: SPECIFIC PROBLEMS

Two cementing problems specific to horizontal wells have been identified. Sabins, et al., have shown that cement particles settle out of the cement slurry within the casing while the slurry is being pumped. Moran, et al., have shown that there is a "critical settling factor" at which particles settle out of cementing spacers and form beds in the annulus. This bed prevents a complete cementation around the circumference of the pipe in the annulus.

SPACER PUMP RATE (SETTLING IN ANNULUS)

Moran posed the following equations for the calculation of pumping rates for weighted (barite and hematite) cement spacers based on his "critical settling factor". His full scale laboratory model showed that a value of the "critical settling factor",CDSF, of 600 prevents settling of barite and also re-suspends barite which has previously settled. A CDSF of 800 prevents the settling of sand and hemitite and re-suspends if a bed has formed. He also found that at borehole angles of 65 degrees or less, settling did not occur.

$$A = -10^{[0.37 - 0.01(\mathrm{LogN})1 + 0.21(\mathrm{LogN})2 - 4.16(\mathrm{LogN})3]}$$

$$B = 10^{[0.06 - 0.01(\mathrm{LogN})1 + 0.14(\mathrm{LogN})2 - 4.03(\mathrm{LogN})3]}$$

$$C = 10^{[-1.88 - 0.171(\mathrm{LogN})1 + 4.82(\mathrm{LogN})2 - 0.74(\mathrm{LogN})3]}$$

$$D = 10^{[A + B\,(\mathrm{Log(Stdoff)})1 + C\,(\mathrm{Log(Stdoff)})2]}$$

$$E = \frac{164640\ D\ (2N + 1)}{N\ \mathrm{Stdoff}\ (H - P)\ (H^2 - P^2)}$$

$$\text{Pump rate} = \frac{10^{\ [0.368 - 0.245\ (H - P) + \mathrm{Log(CDSF)}]}}{E}$$

N	=	n' power law index of flow behavior
Stdoff	=	Standoff; %
H	=	Hole diameter; inch
P	=	Pipe OD; inch
CDSF	=	Critical dynamic Settling Factor
Pump rate	=	Minimum pump rate in bbl/min required to maintain solids suspension

CEMENT SHEATH WITHIN CASING (SETTLING INSIDE PIPE)

Sabins found cement sheaths of 5/16" in thickness within the casing in horizontal wells and that cement plugs bypassed the cement solids build-up. The sheath was primarily composed of cement solids which were thought to have settled out of the cement slurry and formed a bed (or sheath) on low side of the casing.

He found that cements with viscometer readings greater than 11 at both the 3 rpm and 6 rpm values and yield points greater than 13 lb/100sq.ft. at room temperatures eliminated the cement sheaths which he studied. The viscometer readings were raised to 11 or more with additions of bentonite to the cement slurries.

CONVEYED LOGGING; ALSO CALLED WET CONNECT

A popular logging system in horizontal wells is the pipe conveyed tool system. The pipe conveyed system makes use of a side entry sub and a device for attaching the logging tool to the end of the pipe. The pipe is usually drillpipe. The device is called a release tool. The following steps explain the procedure.

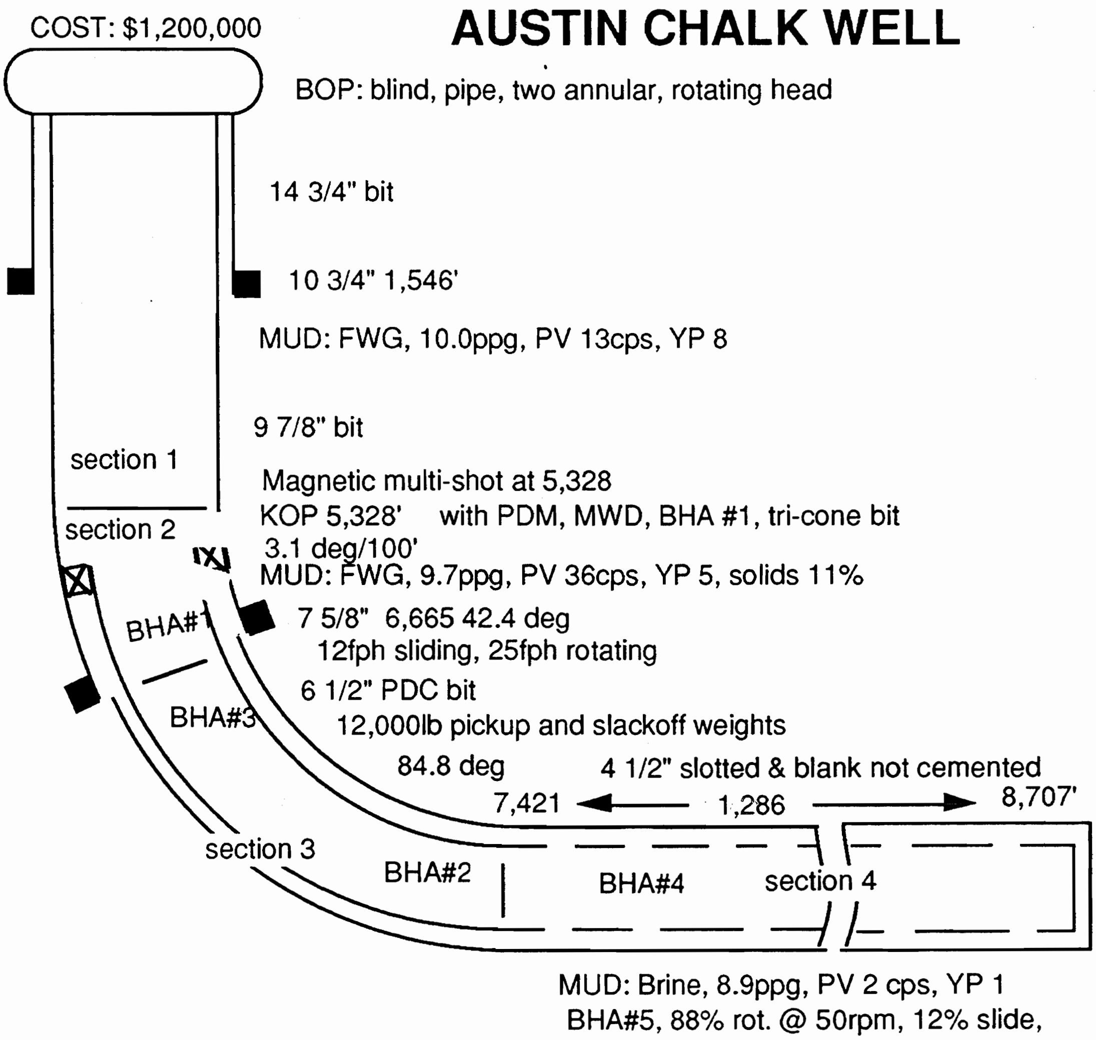
1. The logging tool with the logging line release tool is attached to the drillpipe and run into the hole.
2. The logging tool is run to the top of the interval to be logged.
3. A side entry sub with the logging line inserted and a device to lock it onto the top of the logging tool is installed on the drillpipe at the surface.
4. The remainder of the drillpipe is run into the hole with the logging line outside the drillpipe above the side entry sub.
5. The line and the locking device, which is also an electrical connection, is pumped from the side entry sub to the logging tool and attached to the locking device.
6. As the portion of the drillpipe above the side entry is pulled from the hole, the hole is logged and the side entry sub comes to the surface.
7. The section has been logged.

8. The logging line is pulled from the locking device, through the drillpipe and is removed along with the side entry sub.

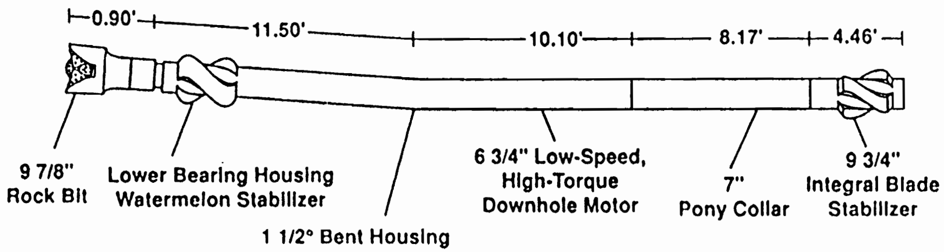
CASE HISTORIES

AUSTIN CHALK WELL - DIMMIT COUNTY, TEXAS, USA

The following is a description of the typical Austin Chalk well in Dimmit county, Texas. Vertical wells in the area produced from no oil to 5 barrels of oil per day. The described well initially flowed 1,660 bopd. The Austin Chalk is a highly, vertically fractured formation with no matrix permeability. The horizontal section of the well was drilled under balanced in order to assess the production rate as the well was drilled and to reduce plugging of fractures with drill solids. The 7 5/8" casing was set into the Austin Chalk to case-off upper problem zones and to provide kick tolerance while drilling under balanced.



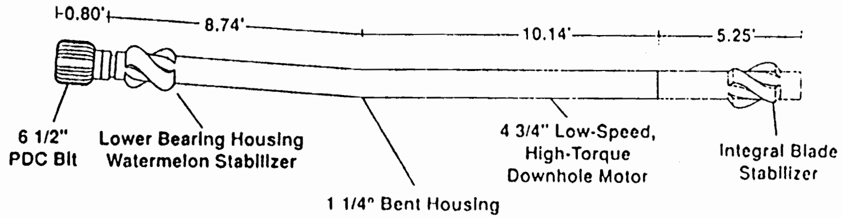
7 5/8" casing drilled out with slick BHA and leakoff test was run.
therafter, BHA#2 was run. Brine mud used.



Performance Summary

Depth In (Ft)	Depth Out (Ft)	Rates of Penetration (Ft/Hr)	Inclination at Start of Run (°)	Inclination at End of Run (°)	Achieved Dogleg While In Sliding Mode (°/100 Ft)
5328	6665	11-25 Rotating 8-15 Sliding	0.2	43.0	3.6-5.1

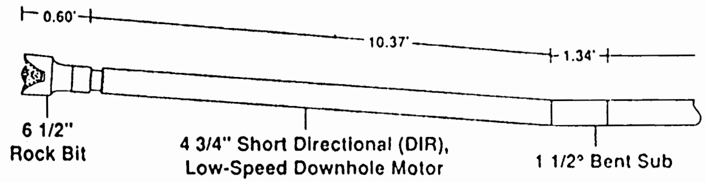
BHA # 1



Performance Summary

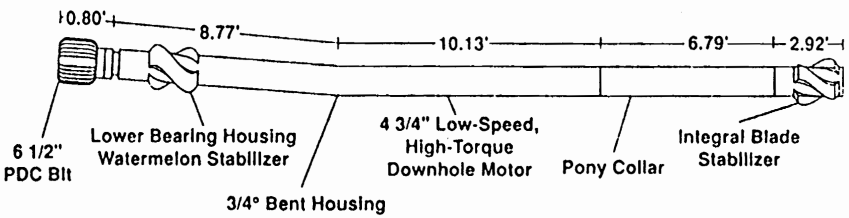
Depth In (Ft)	Depth Out (Ft)	Rates of Penetration (Ft/Hr)	Inclination at Start of Run (°)	Inclination at End of Run (°)	Achieved Dogleg While In Sliding Mode (°/100 Ft)
6805	7254	9-10 Rotating 8-9 Sliding	49.9	74.2	5.8

BHA # 2



Depth In (Ft)	Depth Out (Ft)	Rates of Penetration (Ft/Hr)	Inclination at Start of Run (°)	Inclination at End of Run (°)	Achieved Dogleg While In Sliding Mode (°/100 Ft)
6685	6805	6-8	42.4	49.9	6.3
7254	7421	10-11	74.2	84.8	6.4

BHA # 3



Performance Summary

Depth In (Ft)	Depth Out (Ft)	Rates of Penetration (Ft/Hr)	Inclination at Start of Run (°)	Inclination at End of Run (°)	Achieved Dogleg While In Sliding Mode (°/100 Ft)
7421	8707	14-16 Rotating 12-14 Sliding	84.8	90.5	1.9

BHA # 4

TEB-1 WELL in the TYRA FIELD OFFSHORE DENMARK

The drilling objectives of the TEB-1 were to have a total departure of 10,000 feet which includes a 3,000 foot horizontal section and to cement with rotation a 6 5/8" production liner across the horizontal section. The drilling implemented both long and medium radius builds, steerable medium radius motors, and thermally stable diamond drill bits.

The Tyra reservoir is a moderately hard chalk containing sub-vertical hairline fractures and some chert.

Casing Program

24" driven to 400' (260' water + 140' sea bed)

18 5/8" conductor set in 21" hole at 1,388' TVD or 1,412' MD

13 3/8" surface casing set in 17 1/2" hole at the top of an over-pressured shale at 3,838' TVD or 5,160' MD

9 5/8" intermediate casing set in 12 1/4" hole at the top of the Danian D1 top at 6,470' TVD or 10,127' MD

6 5/8" liner set in 8 1/2" hole at end of drain hole at 6,537' TVD or 11,818' MD

Drilling Mud

21" hole: pre-hydrated bentonite unweighted seawater mud

17 1/2" hole: native clay, lime, caustic soda, seawater weighted to 8.5 to 9.5 ppg

12 1/4" hole: lime-treated water base with asphaltene shale stabilization material

8 1/2" hole: low-toxicity invert-emulsion oil-base, 12.0 to 12.5 ppg 80/20 oil/water ratio, 120,000 to 160,000 mg/L Cl (water phase)

Drill String

Steerable PDM assemblies similar to those used for directional wells in the field were used.

Rather than use heavy weight drillpipe, 5" drillpipe manufactured for compressive service was used to lower torque and drag. The additional resistance to sinusoidal or helical buckling provided by 5" HWDP was not required. Indications of drillpipe buckling, such as, excessive down-drag and torque while drilling, or drillpipe fatigue, were not observed.

Rotary torque while drilling the 8 1/2" horizontal section average approximately 15,000 lb-ft and drag rarely exceeded 10,000 lb.

Drill hole directional surveys

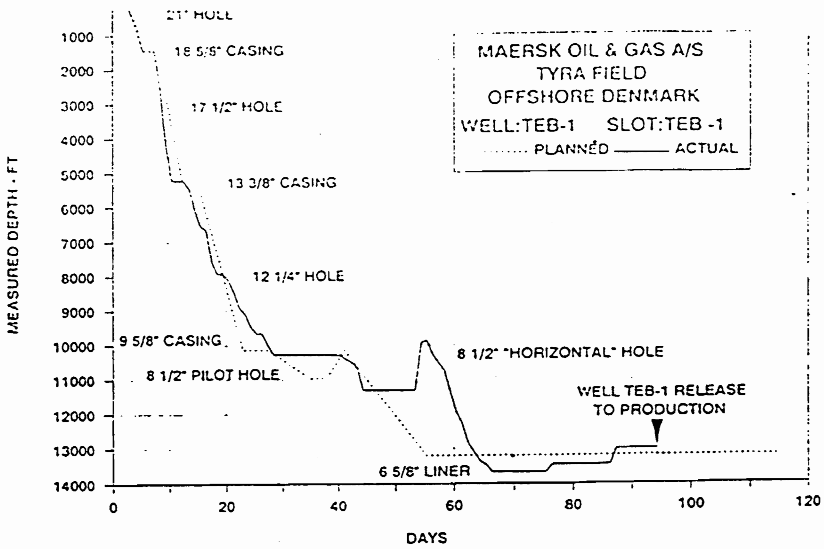
gyro'ed the 24" through drillpipe following the first gyro single shot orientation of the 21" kickoff assembly

gyro's and orientation of 21" kickoff assembly until clear of possible magnetic interference from nearby wells

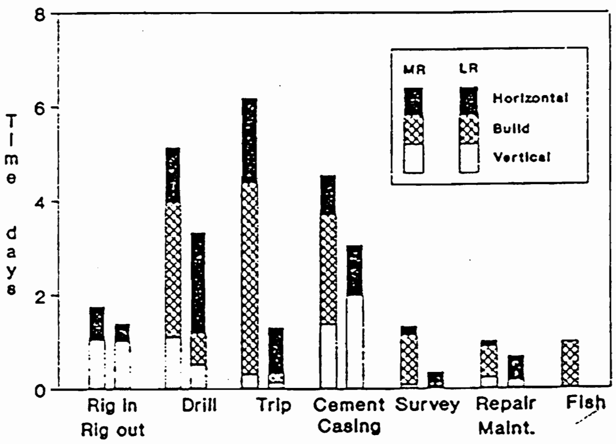
MWD surveys for balance of 21" interval and through the 17 1/2", 12 1/4", 8 1/2" intervals

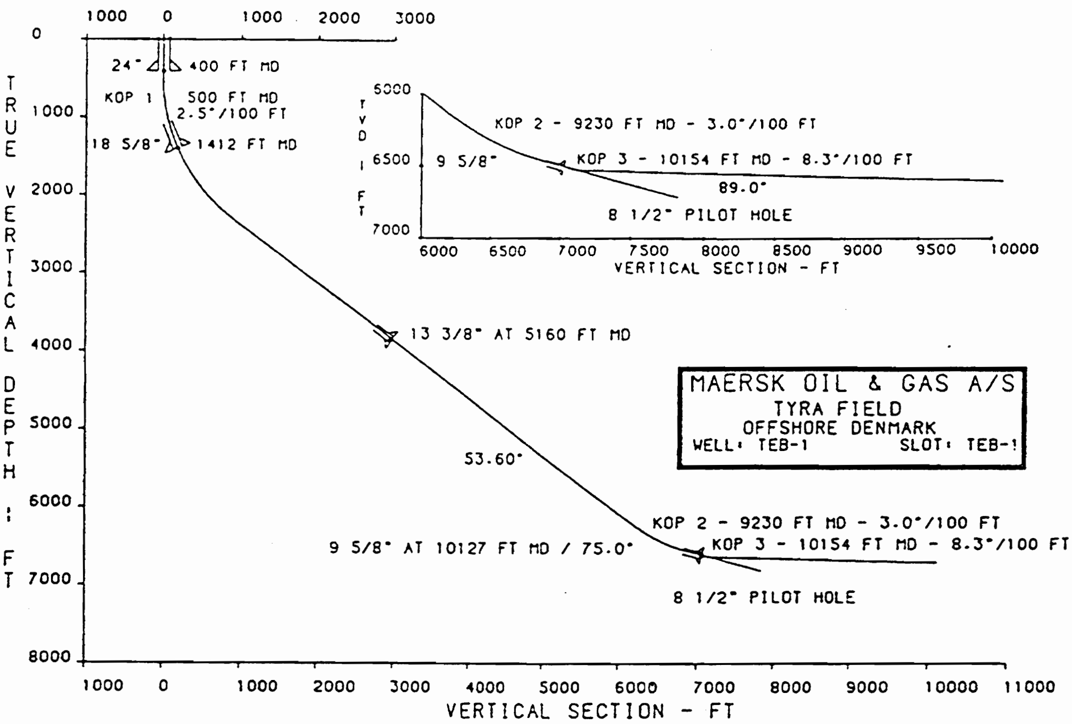
magnetic multi-shot's (electronic) of 17 1/2" and 12 1/4" intervals during trips at the 13 3/8" and 9 5/8" setting depths

gyro multi-shot's inside 18 5/8", 13 3/8", and 9 5/8" casings when set

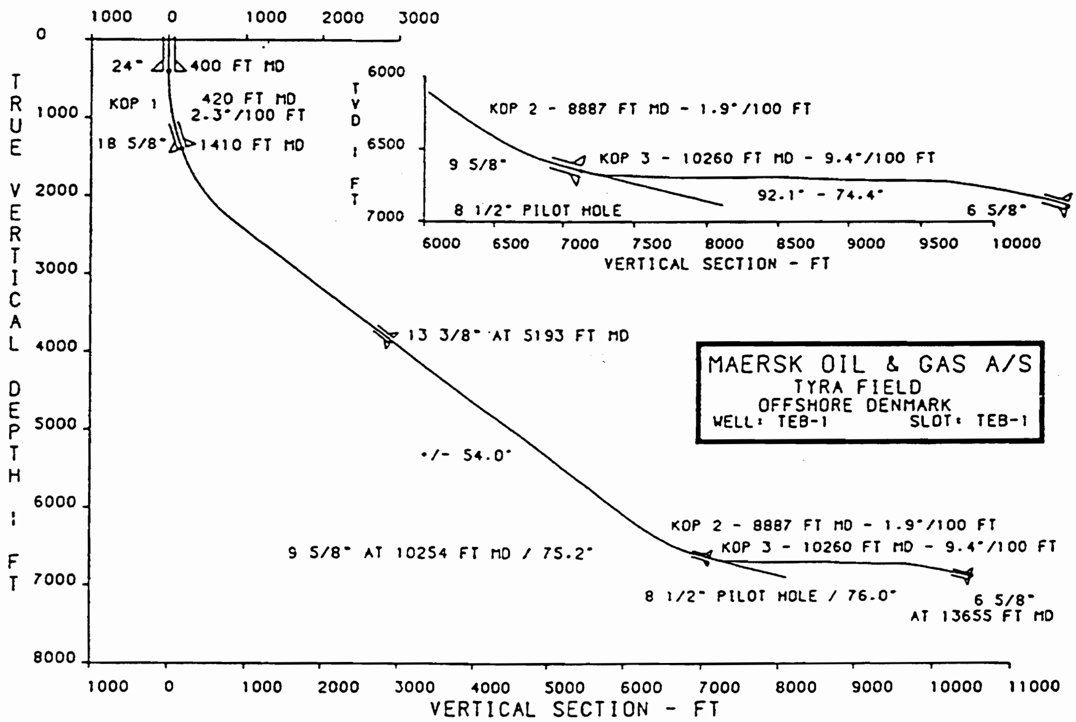


Planned vs. actual drilling time for well TEB-1.





Planned trajectory for well TEB-1.



Actual trajectory for well TEB-1.

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CHAPTER VIII

BOTTOM HOLE ASSEMBLIES

DEFINITION OF BHA

A bottom hole assembly (known as BHA) is a component of a drill string. A BHA resides in the drill string above the drill bit and below the drillpipe. The primary component of the BHA is the drill collar. The following figure shows the possible components of a BHA and their typical location within a BHA.

PURPOSE OF BHA

The purposes of a BHA are as listed in the following.

1. protect the drillpipe in the drill string from excessive bending and torsional loads,
2. control direction and inclination in directional holes,
3. drill more vertical holes,
4. drill straighter holes,
5. reduce severities of doglegs, keyseats, and ledges,
6. assure that casing can be run into a hole,
7. increase drill bit performance,
8. reduce rough drilling, (rig and drill string vibrations),
9. as a tool in fishing, testing, and workover operations,
10. ***not** to place weight on the drill bit*

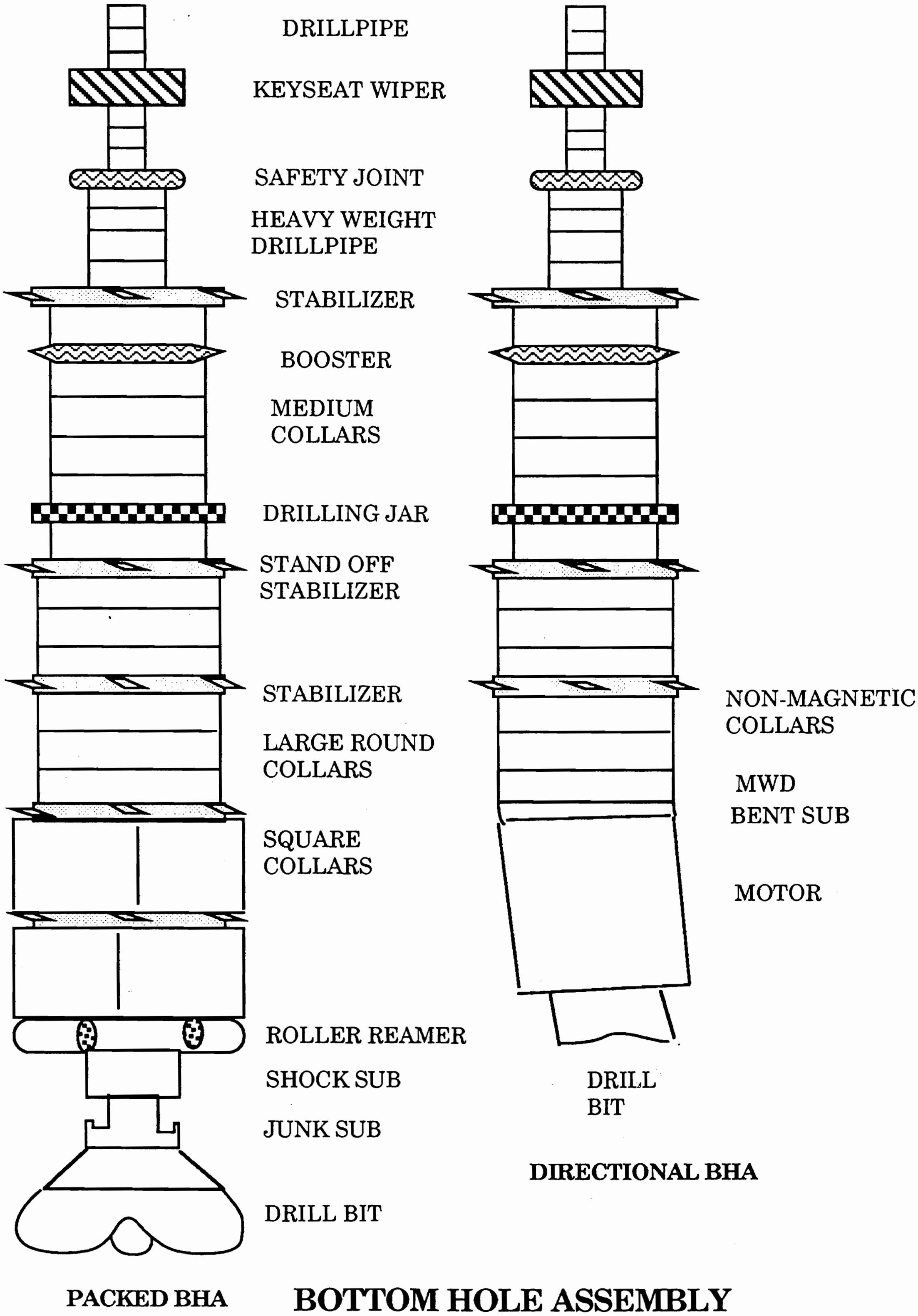
TYPES OF BHA'S

The "SLICK" BHA is composed only of drill collars. It is least expensive and perhaps carries the least risk in regard to fishing and recovery.

The "PENDULUM" BHA is designed to drill holes more vertically and to drop inclination in inclined holes. Lubinski and Woods published tables and charts to locate the lowest most stabilizers in the BHA. Most BHA theories which were intended for vertical holes apply to holes which are inclined 20 degrees or less.

The "PACKED" BHA is designed to drill straight holes and to reduce the severities of doglegs, keyseats, and ledges. It provides the highest assurance that casing can be run into a hole. The theory which supports the packed BHA was developed by Hoch. A packed BHA can be expensive and perhaps carries the highest risk in regard to fishing and recovery.

The "DIRECTIONAL" BHA is designed either to turn the hole to a chosen inclination and direction or to maintain a course selected for the hole. The directional BHA is based on the principles of levers and fulcrums.



BOTTOM HOLE ASSEMBLY COMPONENTS

The "FISHING, TESTING, and WORKOVER" BHA is designed to assist the many tools found in these areas.

Many theories and practices apply equally to all four types of BHA's while others are very restricted to one of the types. The performance of BHA's are affected most by the rugosity (hole enlargements) of the hole. Other factors are

1. formation dip
2. torque while drilling
3. drag during trips
4. stabilizer blade wear
5. drill collar OD wear
6. mud filter cake

DISCUSSION OF COMPONENTS

Drill Bit

Technically, the drill bit is not a component of the BHA; however, it does generate and sends axial and torsional loads to the BHA. Its basic function is to cut rock at the bottom of the hole.

Junk Sub

A narrow trough which exists in the junk sub is designed to collect small pieces of metal; such as, balls and rollers from bearings and broken bit teeth.

Shock Sub

The purpose of shock subs is to dampen the vibration produced by the drill bit and the drill string. It is reasonable to surmise that shock subs prolong the life of drill bits and drill strings and in some cases the rig. Shock sub are not as stiff (resistance to axial bending) as drill collars and, for this fact, often have limited application in straight hole drilling. In addition, large drill collars may be more effective in reducing bottom hole vibrations.

Roller Reamers

A reamer serves two functions: (1) it cuts the wall of the hole to gauge and (2) it centers the BHA in the hole at its location. Reamers with 3 and 6 rollers are most popular. The rollers are on bearings and they contain tungsten carbide inserts which are similar to those placed in drill bits. Roller reamers are used in hard rock where blade reamers cause excessive drill string torque.

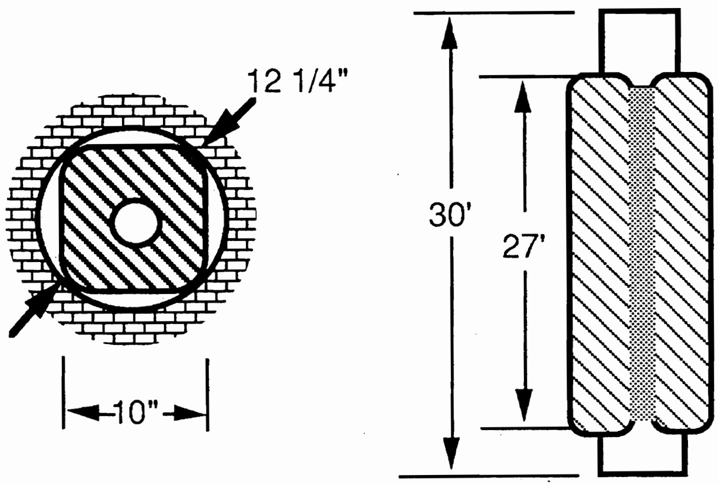
Stabilizers

Stabilizers center the BHA in the hole at their location. In soft rock, a rotating blade type can effectively ream a hole to gauge. There are two basic types: rotating and non-rotating. The blades of the rotating type may be spiraled around the body of the stabilizer or aligned vertically and attached by welding, with a sleeve, or by pins. The rubber blades of the non-rotating type are aligned with the vertical axis of the body of the stabilizer. The rubber blades are popular because they are easily washed-over and do dig into the wall of the hole causing enlargements.

Short stabilizers can pivot in the hole and allow bending moments to be transmitted to adjacent BHA components. They act as fulcrums. Long stabilizers or stacked short ones do not pass bending moments and align themselves in the hole. A BHA which does not transmit bending moments is called a "locked-up" BHA.

Square Drill Collars

Square drill collars accomplish four goals: (1) they provide continuous centralization over their length, (2) they maximize bending resistance (stiffness), (3) they maximize torsional damping, and (4) they minimize axial vibrations. On the down side, they are expensive to buy and to maintain. They usually create high rotary torque. They grind drill bit cuttings and cavings to fines which radically increases mud consumption.



They are difficult to fish. Note that a 11.2 inches round collar has the same stiffness as a 12.125 inches square collar.

Spiral Drill Collar

Spiral drill collars reduce the risk of differential pressure sticking of the BHA. About 4% of the weight of the drill collar is lost because of the machining of the spirals. Thus, the linear weight per foot of an 8 by 3 inches spiral collar is 141 ppf rather than 147 ppf.

Non-magnetic Drill Collars

The primary purpose of non-magnetic drill collars is to reduce the interference of the magnetic fields associated with those sections of the BHA which are both above the below the magnetic compass contained in the survey tool with the earth's magnetic field. The non-magnetic collars reduce this type of interference by moving the BHA sections away from the survey compass.

There are four critical factors in selecting non-magnetic collars: (1) their total length, (2) the location of the survey compass with the non-magnetic collars, (3) the type of material of which the collars are composed, and (4) distinguishing "hot spots".

"Hot spots" are zones of high magnetic field strengths within the material of the collars. These zones are detected with a magnetic permeability probe. The measurements are recorded on a strip chart versus distance into the collar. Hot spots can affect compasses by as much as 4 degrees (2 degrees is common). A "cold" collar will affect a compass by less than 1/4 degree over its entire length.

Non-magnetic collars may be manufactured from many types of material. The selection is based primarily on the corrosion resistance of the material. The most common non-magnetic material is stainless steel while monel, which is 60% nickel and 30% copper, is seldom used.

A practical method of ascertaining the best location for the survey compass and the requisite non-magnetic collar length is to run a BHA into the drill hole and pull a compass through the non-magnetic collars. The portion of the non-magnetic collars which do not show the affects of the fields of the BHA may be removed from the BHA.

Medium and Large Round Collars

The purposes of large round collars are to provide stiffness next to the drill bit and add weight to the BHA.

The medium collars add weight to the BHA and reduce ever present flexure stresses between large collars and drillpipe or other tools of less rigidity than the large collars. Both may be used for jarring weight.

Heavy Weight Drillpipe

Heavy weight drillpipe is small drill collars with drillpipe tooljoints. They serve the same purposes as medium weight collars. They may be operated in a buckled mode while drilling or a part of a fishing BHA.

Keyseat Wiper

The purpose of keyseat wipers is to ream the "key" section out of the wall of the hole. Keyseat wipers are run in the BHA or the drillpipe. The outside diameter of the cutting structure of the keyseat wipers are commonly selected as 1/8 to 1/2 inch larger than the collars or tooljoints in which they are installed.

Keyseat wipers are either single or double acting; that is, the blade section which slides on a mandrel reams either in the up or down position if single acting but not both.

The double acting type act if the blade section is up or down, but not if the section is floating between up and down.

Because most rigs cannot raise and rotate the drill string simultaneously, the single acting wiper which reams in the down position is required.

Bumper Sub

Bumper subs are used to counter the heave of floating drilling vessels by permitting the extension and contraction of the drill string. In this use the bumper subs are placed between the BHA and the drillpipe. They are also used in fishing operations in a manner similar to that of jars.

Safety Joint

Safety Joints are rarely used in a drilling BHA. Their purpose is to provide a means of easily releasing a BHA.

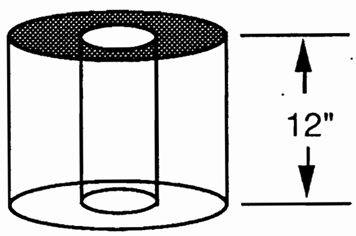
MECHANICAL PROPERTIES OF BHA

The following are properties of BHA's which are prevalent throughout the literature of BHA's.

Cross-sectional area "A"; in^2
Air weight "W_s"; lbs/ft
Buoyed weight"BW"; lbs/ft
Cross-sectional second moment of inertia "I"; in^4
polar moment of inertia "J"; in^4
steel has a density of 0.2832 lbs/in^3 or 65.45 ppg
steel has modulus of elasticity "E" of 30E6 lbs/in^2
steel has modulus of shear "G" of 12E6 lbs/in^2

Air Weight

The air weight of drill collars is the natural attraction of the earth for the steel comprising the collars. A convenient equation for computing the air weight of collars is



$$W_s = 2.67 * (D^2 - d^2) * L$$

W_s	=	air weight of collars; lbs
D	=	outside diameter of collars; inch
d	=	inside diameter of collars; inch
L	=	length of collars; feet

Note that

$$W_s = w * V$$

$$= .2832 * 12 * \frac{\Pi}{4} * (D^2 - d^2)$$

$$= 2.67 * (D^2 - d^2)$$

EXAMPLE

Compute the air weight per foot and the total weight of a 8" * 3" * 30' collar.

$W_s = 2.67 * (8^2 - 3^2)$ **= 146.7 lb/ft**

$W_s = 2.67 * (8^2 - 3^2) * 30$ **= 4,400.6 lb**

Buoyed Weight of BHA

The buoyed weight of drill strings is the weight of a drill string which would be indicated if it were attached to a scale and hanging free. The buoyed weight at a point in a drill string is the computed buoyed weight of the components below that point. and is computed as if it were hanging free. It should be noted that tension is not usually equal to buoyed weight. Buoyed weight is closely associated with moments in the drill string.

Two formulas are useful for computing buoyed weights of drill strings. If the fluid on the inside of the pipe is equal in density to the fluid on the outside of the pipe, the buoyed weight is the product of the buoyancy factor and the air weight of the pipe. If not then the following formula should be used.

$$BW_{\text{per foot}} = W_s + 0.0408 * (MW_i * d^2 - MW_e * D^2)$$

BW	=	buoyed weight per foot; lbs/ft
W_s	=	air weight of the tube per foot; lbs/ft
MW_i	=	weight of fluid inside the tube; ppg
MW_e	=	weight of fluid outside the tube; ppg
D	=	outside diameter of tube; in
d	=	inside diameter of tube; in

If the fluids inside and outside the tube are of equal weight, the equation is

Buoyancy factor

$$B_f = (1 - \frac{MW}{65.45}) \quad \text{or} \quad (1 - .01528 * MW)$$

The buoyed weight of the tube is

$$BW = B_f * W_s$$

EXAMPLE

What is the buoyed weight of 8" * 3" * 147 ppf drill collars which are within and contain 16.0 ppg mud?

$B_f = 1 - \frac{16.0}{65.45}$ **= 0.756**

$$BW = 0.756 * 147 \qquad = \mathbf{111.13 \frac{lb}{ft}}$$

What is the buoyed weight of 8" * 3" * 147 ppf drill collars which are within 16.0 ppg mud and contain 7.5 ppg oil?

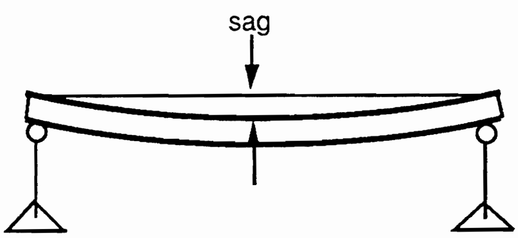
$$BW = 147 + 0.0408 * (7.5 * 3^2 - 16.0 * 8^2) \qquad = \mathbf{103.3 \frac{lb}{ft}}$$

Stiffness

Stiffness in regard to BHA's has many meanings. In the next paragraphs some of those meanings are discussed.

Axial stiffness

One meaning of stiffness of a BHA refers to its resistance to axial bending and is proportional to the cross-sectional second moment of inertia, I. (In mechanics the definition is the product of the modulus of elasticity and the second moment.) Larger diameter BHA's are stiffer than smaller diameter collars. The equation for computing this stiffness is



$$I = \frac{\Pi}{64} * (D^4 - d^4) \text{ for round collars}$$

$$I = \frac{S^4}{12} - d^4 * \frac{\Pi}{64}$$

for square collar with sides of length "S" and a round hole of diameter "d"; in^4

A second meaning of stiffness is the resistance of a BHA to twisting (torsion). (In mechanics the definition is the product of the modulus of shear and the polar moment.) The equation for computing this stiffness is

$$J = \frac{\Pi}{32} * (D^4 - d^4)$$

A third meaning of stiffness is the resistance of a BHA to sag. The sag may be visualized as the deflection of the middle of the BHA as it is resting between two stabilizers. The equation for computing this stiffness is

$$Sag_{6"} = \frac{5 * W * L^4}{384 * E * I}$$

W = weight per linear inch; lbs/in

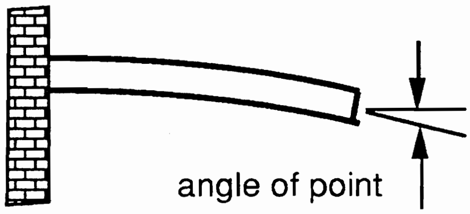
L = length; lbs/in

A fourth meaning of stiffness is the resistance of a BHA droop. Drooping may be visualized as the angle of point of a cantilevered BHA. The equation for computing this stiffness is

$$\text{Angle of point} = \text{atan}\left[\frac{W\,L^3}{6\,E\,I}\right]$$

W = weight per linear inch; lbs/in
L = length; in

A fifth meaning of stiffness is the resistance of a BHA to buckled. There are two types; axial and torsional buckling. Both are discussed elsewhere in this manual.



EXAMPLE

Compare the axial, torsional stiffness, the sag and the angle of point of 6" OD and 10" OD drill collars which are 30' long.

$$I_{6''} = \frac{\pi}{64} * (6^4 - 2.5^4) \qquad \mathbf{= 61.7\ in^4}$$

$$I_{10''} = \frac{\pi}{64} * (10^4 - 3^4) \qquad \mathbf{= 486.9\ in^4}$$

Thus the axial stiffness of 10" drill collars is about 8 times that of 6" collars.

$$J_{6''} = \frac{\pi}{32} * (6^4 - 2.5^4) \qquad \mathbf{= 123.4\ in^4}$$

$$J_{10''} = \frac{\pi}{32} * (10^4 - 3^4) \qquad \mathbf{= 973.8\ in^4}$$

Thus the torsional stiffness of 10" drill collars is about 8 times that of 6" collars.

$$W_{6''} = 2.67 * \frac{6^2 - 2.5^2}{12} \qquad \mathbf{= 6.61\ lb/in}$$

$$Sag_{6''} = \frac{5 * 6.61 * (30*12)^4}{384 * 3e7 * 61.7} \qquad \mathbf{= 0.78\ in}$$

$$W_{10''} = 2.67 * \frac{10^2 - 3^2}{12} \qquad \mathbf{= 20.2\ lb/in}$$

$$\text{Sag}_{10"} = \frac{5 * 20.2 * (30*12)^4}{384 * 3e7 * 486.9} \qquad \mathbf{= 0.30\ in}$$

Thus the sag of 10" drill collars is about $\frac{1}{3}$ times that of 6" collars.

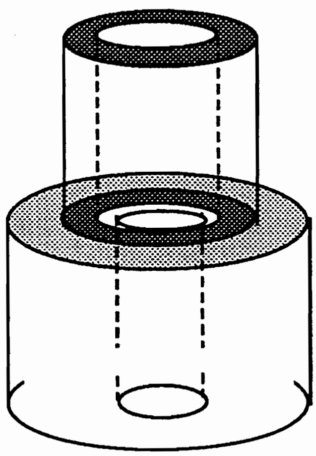
$$\text{Angle of point}_{6"} = \text{atan}\left[\frac{6.61\,(30 * 12)^3}{6\ 3e7\ 61.7}\right] \qquad \mathbf{= 1.59°}$$

$$\text{Angle of point}_{10"} = \text{atan}\left[\frac{20.2\,(30 * 12)^3}{6\ 3e7\ 486.9}\right] \qquad \mathbf{= 0.62°}$$

Thus the angle of point of 10" drill collars is about $\frac{1}{3}$ times that of 6" collars.

TAPERED BHA

In the make-up of BHA's it is often required that components which have different outside or inside diameters be connected. A rule of thumb which has been published by DRILCO, a manufacturer of BHA components, is that the section modulus ratio (SMR) of the bodies of the two connected components be 5.5 or less. Section modulus is the cross-section moment of inertia divided by the radius to the outer fiber. DRILCO further claims that a short sub with an intermediate section modulus will not protect the smaller component from excessive fatigue.



The section modulus of a round component is computed with the equation

$$SM = \frac{\pi}{32} * \frac{D^4 - d^4}{D}$$

EXAMPLE

Should 5" * 4.276" * 19.5 ppf drillpipe be connected to 9" * 3" drill collars?

$$SM_{5"} = \frac{\pi}{32} * \frac{5^4 - 4.276^4}{5} \qquad \mathbf{= 5.71\ (drillpipe)}$$

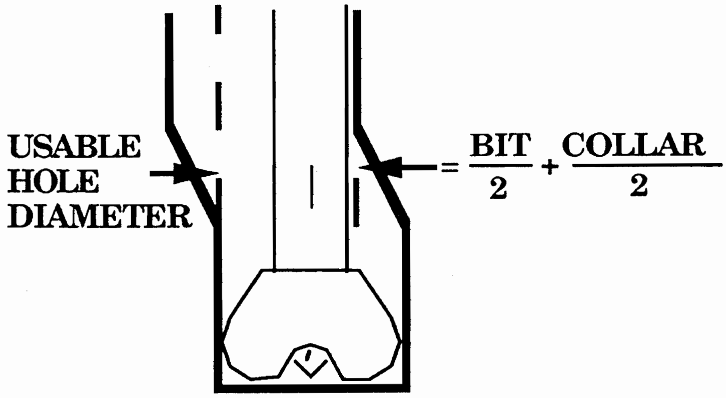
$$SM_{9"} = \frac{\pi}{32} * \frac{9^4 - 3^4}{9} \qquad \mathbf{= 70.7\ (drill\ collars)}$$

$$SMR_{9"/5"} = \frac{70.7}{5.71} \qquad \mathbf{= 12.4}$$

This connection should not be made because the section modulus ratio is greater than 5.5.

USABLE HOLE DIAMETER

Ledges reduce usable hole diameter as is shown in the figure. One solution is to run larger collars above the drill bit. It may be reconciled from the sketch that the usable hole diameter is the sum of one-half the bit and one-half the drill collar diameters.



Thus, a collar diameter yields a specific usable hole diameter. The equation for selection a collar diameter is then

Required Collar Diameter = 2 * Usable Hole Diameter - Drill Bit Diameter

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EXAMPLE

API casing couplings on 9 5/8" casing have a diameter of 10.625". What diameter drill collars must be used to assure that the 9 5/8" casing can be run into a hole drilled with a 12 1/4" drill bit?

Required Collar Diameter = 2 * 10.625 - 12.25 **= 9.0 in**

Minimum Drill Collar Diameter while Drilling with Slick or Pendulum BHA

Bit	Casing	Coupling	Drill Collar
6.500	4.500	5.000	3.750
7.875	5.500	6.050	4.225
8.500	7.000	7.656	6.562
9.500	7.625	8.500	7.500
10.625	8.625	9.625	8.625
12.250	9.625	10.625	9.000
13.750	10.750	11.750	9.750
14.750	11.750	12.750	8.750
17.500	13.375	14.375	11.250
20.000	16.000	17.000	14.000
24.000	18.625	19.750	15.500
26.000	20.000	21.000	16.000

WALL FORCE

Wall force is a lateral force placed on the tool joints of drillpipe by the wall of the hole. Wall force is created by either a dogleg in a hole, buckling of the drill string, centrifugal orbiting of the drill string, or the inclination of the hole.

Wall Force in Doglegs

If drill pipe is within a dogleg, the lateral force will be concentrated at the tool joint if the wall of the drillpipe does not contact the wall of the hole. This is the usual case. The dogleg concept is illustrated in the figure.

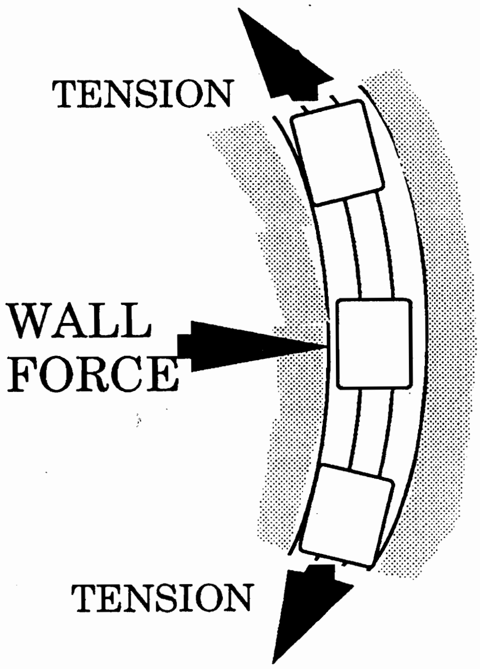
The wall force created by drillpipe tension and a dogleg is computed with the equation

$$WF = 2 * T * \sin [DLS * \frac{LJ}{2}]$$

WF	=	wall force on a single tooljoint; lbs
T	=	tension in the drillpipe at the dogleg; lbs
DLS	=	dogleg severity of the dogleg; deg/ft
LJ	=	joint length of one drillpipe; ft

EXAMPLE

A BHA consists of 500' of 8" * 3" * 147 ppf and 5" * 19.5 ppf (21.0) * range 2 (30') drillpipe. Drilling depth is 14,000' and the mud weight is 11.0 ppg. A dogleg with a severity of 5 degrees/ 100 feet of hole length exists at 2,000 feet. Weight on bit while drilling is 50,000 lbs. What is the wall force on a tooljoint at 2,500 feet?



The effective tension in the drill string at 2,500 feet in the dogleg is

$$T = (1 - \frac{11}{65.45}) * (500 * 147 + 11,000 * 21) - 50,000$$

T = **203,323 lb** (tension in the drillpipe in the dogleg)

The wall force on the tooljoint in the dogleg is

$$WF = 2 * 203,323 * \sin [\frac{5}{100} * \frac{31}{2}] \qquad = \mathbf{5,500\ lb}$$

Practice is to keep the wall force on tooljoints below 2,000 lbs in water base muds and 3,000 lbs in oil base muds in order to reduce wear to an acceptable rate and to prevent the heat checking of the steel.

Wall Force in Inclined Holes

The wall force on any tubular within an inclined hole is the product of its buoyed weight and the sine of the inclination angle of the hole.

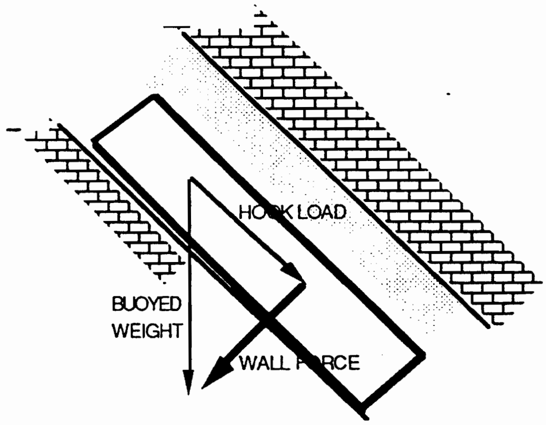
The equation for computing the wall force on a tube in an inclined hole is

$$WF = BW * \sin(I)$$

WF	=	wall force per foot of the tube; lbs
BW	=	buoyed weight per foot of the tube; lbs
I	=	inclination of the hole; deg

EXAMPLE OF WALL FORCE

Compute the wall force per foot of drillpipe created by 5" * 4.276" * 19.5 ppf (21 ppf) drillpipe which is filled with 8.4 ppg water, resting in 17.0 ppg mud, and inclined in a hole at 50 degrees.



The buoyed weight per foot of the drillpipe is

$$BW = 21 + .0408 * (8.4 * 4.276^2 - 17 * 5^2) \quad \mathbf{= 9.93\ ppf}$$

The wall force is

$$WF = 9.93 * \sin(50) \quad \mathbf{= 7.61\ lb/ft}$$

The wall force on a 31 foot joint of drillpipe is

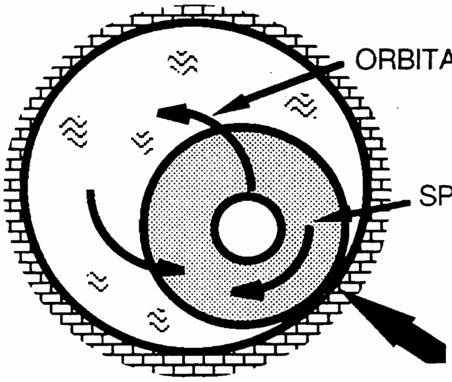
$$WF\ per\ joint = 31 * 7.61 \quad \mathbf{= 235.8\ lb}$$

CENTRIFUGAL FORCE

The rotation of the BHA produces an orbital motion of the BHA if sufficient friction exists at the contact point of the BHA and the wall of the hole. The idea is illustrated in the figure.

The centrifugal wall force equation is

$$WF = \frac{W_s * (H - D) * N^2}{70471}$$



WF	=	wall force per foot; lbs/ft
W_s	=	weight per foot of component; lbs/ft
H	=	hole diameter; in
D	=	outside diameter of tube; in
N	=	revolutions per minute of tube; rpm

EXAMPLE

What is the centrifugal wall force if 9" * 3" * 192 ppf drill collars are rotating at 91 rpm without wall slip in a 17.5" hole?

$$WF = \frac{192 * (17.5 - 9) * 91^2}{70471} \qquad \mathbf{= 192\ lb}$$

TORSIONAL DAMPENING - FLY WHEEL EFFECT

Flywheels are used to dampen torsional vibrations, such as the flywheel in a car engine. They do this by removing the high peak torsional energy variations. In the drilling operation variations in torsional energy within the drill string are caused by the drill bit, stabilizers, and the rotation of the BHA and drillpipe. Large drill collars act very well as flywheels.

The equation for the dampening variable or the flywheel effect is

$$DV = \frac{N * J_c * L_c}{79058}$$

DV	=	dampening variable
N	=	rotary speed; rpm
J_c	=	polar moment of inertia; in^4
L_c	=	length of component; ft

EXAMPLE OF DAMPING

The drilling has been too rough and the rig has been shaking. The bottom hole assembly is composed of 400 ft of 8" * 3" collars and has been rotated a speed of 40 rpm. If two stands (180 ft) of 11" collars were added to the bottom hole assembly and the rotary speed could be increased to 100 rpm, what would the value of the new dampening variable be?

The polar moments of inertia for the 8" and 11" collars are

$$J_c = \frac{\pi}{32} * (D^4 - d^4) \qquad = \frac{\pi}{32} * (8^4 - 3^4) \qquad \mathbf{= 394\ in^4}$$

$$J_{c(11")} = \frac{\pi}{32} * (11^4 - 3^4) \qquad \mathbf{= 1429\ in^4}$$

The dampening variable for the original 8" BHA is

$$DV_{8"} = \frac{40 * 394 * 400}{79058} \qquad \mathbf{= 79.7}$$

$$DV_{11"} = \frac{100 * 1429 * 180}{79058} + 199.4 \qquad = 325.3 + 79.7 \qquad \mathbf{= 529.4}$$

Thus the dampening variable was increased by **five fold** after adding the 11" collars to the 8" collars.

TORQUE OF A SPINNING BHA

The restoring torque placed on the drill bit by the BHA, when and if the bit hangs and stops spinning, is given by the equation

$$\mathbf{T = .795 * N * J_c}$$

T	=	torque transmitted to the drill bit; lbs-ft
N	=	BHA rotational speed just before bit stops; rpm
J_c	=	polar moment of inertia of the BHA; in^4

It should be noted that the rotational speed of the rotary BHA is not necessarily equal to the rotational speed of the rotary table at any time.

The time period for which the restoring torque will act on the drill bit is the time for the torsional torque wave to rise from the bit within the BHA to the top of the BHA and return to the bit. The equation for computing this period time is

$$t = \frac{L_c}{5238}$$

t	=	time torque is applied to the bit; seconds
L_c	=	length of BHA; ft
5,238	=	speed of a torsional wave in steel; fps

EXAMPLE

If a drill bit connected to a BHA which is composed of 11" * 3" collars, which are 180' long, and which is spinning at 200 rpm because of previous bit hang-ups, hangs-up and stops spinning, what will be the value of the restoring force and how long will it last?

$$J_c = \frac{\pi}{32} * (D^4 - d^4) \qquad = \frac{\pi}{32} * (11^4 - 3^4) \qquad \mathbf{= 1429\ in^4}$$

$$T = .795 * 200 * 1{,}429 \qquad \mathbf{= 227{,}211\ lb\text{-}ft}$$

The time of application of the torque to the bit is

$$t = \frac{180}{5238} \qquad \mathbf{= 0.034\ second}$$

TORSIONAL BUCKLING OF A BHA AND DRILLPIPE

Two factors which cause drillpipe helical buckling are investigated. These are drill bit hang-up and BHA rotational drag.

Drill Bit Hang-up

Schuh invented a BHA torsional model which predicts the transmission of torsional loads created by the bit through the drill collars and then into the drillpipe. These torsional loads are dampened or eliminated by large diameter drill collars. Schuh's equation is

$$T = .795 * N * J_p * \left[\frac{2 * J_c}{J_c + J_p}\right]$$

T = torque transmitted: lbs-ft
N = change in rotational speed of the drillcollars; rpm
J_p = polar moment of inertia of the drillpipe; in^4
J_c = polar moment of inertia of the drill collars; in^4

Roark published an equation which predicts the minimum torque required to buckle a tube. His equation is

$$BT = \left[833{,}333\, I_c * (2{,}056{,}168 \frac{I_c}{L_c^2} + P)\right]^{1/2}$$

BT = resistance to buckling of the tube; lbs-ft
I_c = cross-sectional moment of inertia of the tube; in^4

L_c	=	length of the tube; ft
P	=	axial load (compression is - and tension is +); lb

EXAMPLE

If the bit hangs-up while rotating at 100 rpm and the drill string is composed of 400' of 8" * 3" collars and 5" * 4.276" * 19.5 ppf drillpipe, can the drill collars and drillpipe torsionally buckle?

Compute the resistance to buckling of the collars and the drillpipe and compare these values with the torque created by the bit hang-up.

The drill collars would be normally axially compressed at a point near the bit by 40,000 lbs with this BHA. Thus, P is set to 40,000 for the collars and a value of zero for the drillpipe. The length of the drillpipe is set at 10,000 feet. The moments of the collars and drillpipe are

$$Ic_{8"} = \frac{\pi}{64} * (8^4 - 3^4) \qquad Ip_{5"} = \frac{\pi}{64} * (5^4 - 4.276^4)$$

$$Ic_{8"} = \mathbf{197\ in^4} \qquad Ip_{5"} = \mathbf{14.3\ in^4}$$

$$Jc_{8"} = \frac{\pi}{32} * (8^4 - 3^4) \qquad Jp_{5"} = \frac{\pi}{32} * (5^4 - 4.276^4)$$

$$Jc8" = \mathbf{394\ in^4} \qquad Jp5" = \mathbf{28.5\ in^4}$$

The resistance to torsional buckling of the collars and the drillpipe are

$$BT_{8"} = \left[(833{,}333 * 197 * \{\frac{2056168 * 197}{400^2} - 40000\}\right]^{.5}$$

$BT_{8"}$ = **0** at all points where the compression is 2,531 lbs or greater

$$BT_{5"} = \left[(833{,}333 * 14.3 * \{\frac{2056168 * 14.3}{10000^2} + 0\}\right]^{.5} \qquad \mathbf{= 1{,}872\ lb\text{-}ft}$$

The amount of torque placed on the drillpipe by collars is

$$T = 0.795 * 100 * 28.5 \ \frac{2 * 394}{394 + 28.5} \qquad \mathbf{= 4{,}226\ lb\text{-}ft}$$

Thus, the drillpipe will helically buckle by the torque placed on it by the drill collars if the bit hangs-up. However, that portion of the BHA is thought to be in tension, and will not be buckled by bit hang-up.

Buckling by Rotational Drag

Torsional buckling of the drillpipe may occur if the input torque by the rotary table is excessive. Torsional drag created by both the BHA and drillpipe could be responsible. The following problem uses the watts consumed by an electric rotary drive to estimate the drag of the drillstring and subsequent drillpipe buckling.

The horsepower and torque output of an electric motor is

$$HP = \frac{V * I * eff}{746}$$

$$Q = 5{,}252 \frac{HP}{RPM} = 7.04 \frac{V * I * eff}{RPM}$$

HP	=	horsepower output of a motor; hp
V	=	voltage across a motor; volts
I	=	amperes consumed; amps
eff	=	electrical efficiency of a big motor (=.92); hp/hp
Q	=	torque output of a motor; lbs-ft
RPM	=	rotational speed of the motor; rpm

The mechanical efficiency of a rotary table drive system with an electric drive is about 95%. Thus the torque put into the drillpipe is

$$Q_p = 7.04 \frac{V * I * eff * mff}{N}.$$

Q_p	=	drillpipe torque by the rotary system; lbs-ft
mff	=	mechanical efficiency of the rotary system
N	=	drillpipe rotational speed; rpm

EXAMPLE

A drillstring is composed of a BHA and 5" * 4.276" * 19.5 ppf drillpipe. It is thought that 75% of the rotational drag of the drillstring is being created by the BHA. The rotary speed at the rotary table is 80 rpm and the ampere meter shows 150 amperes. The rig voltage is 440 volts.

The torque put into the drillpipe by the rotary table is

$$Q_p = 7.04 \frac{440 * 150 * .92 * .95}{80}$$

Q_p **= 5,076 lb-ft** (at the top of the drillpipe)

The torque at the bottom of the drillpipe is that which is created by the BHA.

Q_p = 5,076 * .75 **= 3,807 lbs-ft** (at the bottom of the drillpipe)

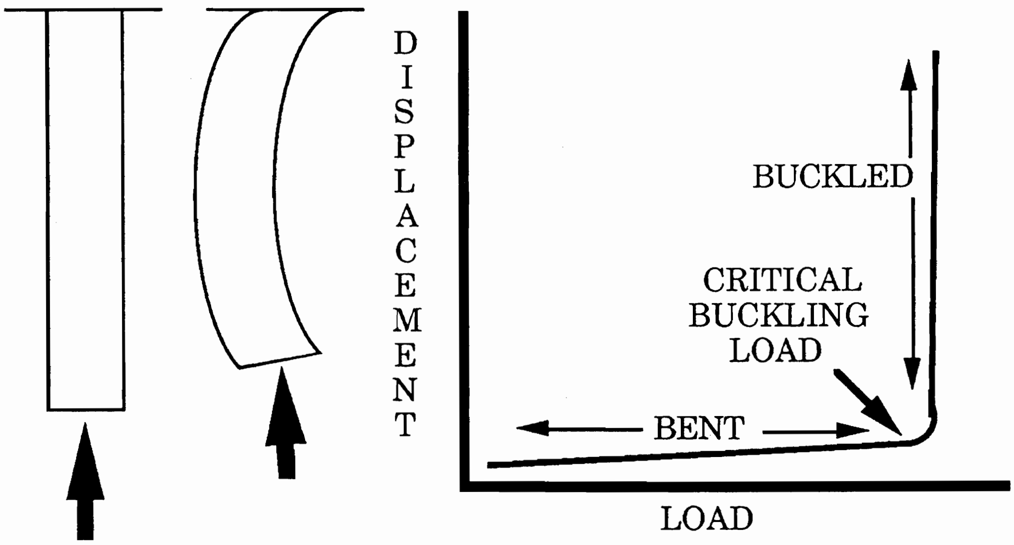
In the previous problem it was shown that the drillpipe's resistance to buckling was 1,872 lbs-ft of torque; thus the drillpipe will be buckled by torsional drag at its bottom end. If the tension in 10,000 feet of drillpipe is 100,000 lbs at the rotary table, then its resistance to buckling will be

$$BT_{5''} = \left[833{,}333*14.3*\left\{\frac{2056168*14.3}{10000^2}+100{,}000\right\}\right]^{.5} \qquad = \mathbf{1{,}091{,}636\ lb^5\text{-}ft}$$

Thus, the drillpipe will not be buckled at its upper end.

CRITICAL BUCKLING LOAD

The critical buckling load is the load which causes large displacements with little increases in load. The figure illustrates axial buckling of a tube.



The minimum load acting on an end of a tube which creates axial buckling (critical buckling load) is

$$B_{crit} = f * \left[B_f^2 * (D^2 + d^2) * (D^2 - d^2)^3 \right]^{1/3}$$

B_{crit}	=	critical buckling load; lbs
B_f	=	buoyancy factor; no units
D	=	outside diameter of the tube; inch
d	=	inside diameter of the tube; inch
f	=	80 for 1st order buckling
f	=	155 for 2nd order buckling

EXAMPLE

What is the critical buckling load of 8" * 3" drill collars within 11.3 ppg mud?

$$B_f = 1 - \frac{11.3}{65.45} \qquad = \mathbf{.827}$$

$$(D^2 + d^2) = (8^2 + 3^2) \qquad = \mathbf{73.0}$$

$$(D^2 - d^2)^3 = (8^2 - 3^2)^3 \qquad = \mathbf{166{,}375}$$

$$\{ B_f^2 * (D^2 + d^2) * (D^2 - d^2)^3 \}^{1/3} \qquad = \mathbf{202.52}$$

B_{crit} = 80 * 202.52 = **16,202 lb** 1st order buckling

B_{crit} = 155 * 202.52 = **31,391 lb** 2nd order buckling

WEIGHT ON DRILL BIT IN VERTICAL AND INCLINED HOLES

The BHA is **not** designed for the purpose of putting weight on the drill bit. The prevalent purpose for a BHA is to eliminate axial and torsional buckling of the drillpipe. This purpose is not achieved in some cases if the buoyed weight of the BHA is only slightly larger than the weight on bit.

One exception to the above statement is the situation in which moderately hard rock is being drilled at a very shallow depth. In this case the weight of the drillpipe would be insufficient to achieve an efficient drilling rate and a heavy BHA would be required. However, if the real purpose of the BHA were to "put weight on the bit", then while drilling at a depth of 5,000 feet for example, the weight of the drillpipe would produce a very efficient drilling rate.

Weight on Bit in Vertical Holes

The statement "only the buoyed weight of the BHA should be placed on the bit" is correct. Only using the buoyed weight of the BHA on the bit will prevent the drillpipe from axially buckling and most likely also prevent it from torsionally buckling. It is not true that the drillpipe is in tension just above the BHA. The normal portions of the drill string which are bent, straight, compressed and tensioned are shown in the figure. Note the section which is in compression and is straight.

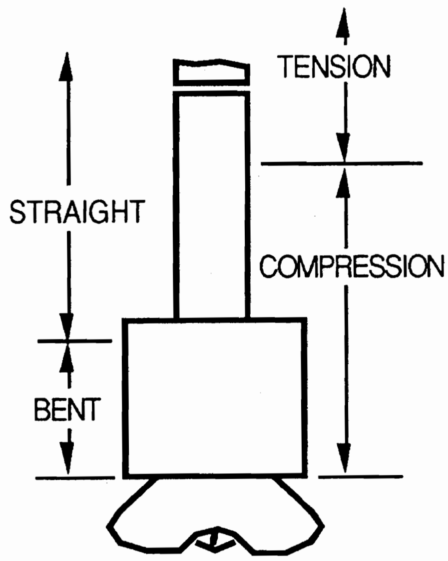
The location of the cross over from the bent to the straight section is called the neutral point of bending (NPB). This point should be kept within the BHA. The neutral point of tension and compression (NPT&C) is normally within the drillpipe and is of no practical use in the design of BHA's.

If practical, all BHA tools should be placed above the neutral point of bending to prevent their buckling. This would prevent their premature failure.

In vertical holes the equation for ascertaining the maximum allowable weight which may be placed on a drill bit from a BHA is

WOB = Buoyancy Factor * Air Weight of the BHA * Safety Factor

Safety factors range from .5 to .9 and depend on how rough the drilling is. The rougher the drilling the lower the safety factor.



The air weight of the BHA is the sum of all of its parts. Perhaps the most accurate procedure for ascertaining the buoyed weight of the BHA is to weigh it with the rig's weight indicator after running only the BHA in the hole. BHA must be vertical and drag eliminated. The formula for computing the air weights of most components which include the drill collars is

$$\text{Air Weight} = 2.67 * (D^2 - d^2)$$

EXAMPLE

A BHA is composed of the following listed components. Its weight as weighed with the rig weight indicator is 61,230 lbs. The hole is vertical for its first 1,000 feet and the mud weight is 11.3 ppg.

BHA COMPONENT	AIR WEIGHT (ppf)	BUOYED WT. (lbs)
9" * 3" * 31'	192.2	4930
Stab 9.5" * 3" * 2.46'	216.9	441
9" * 3" * 149'	192.2	23698
7.5" * 2.75" * 153'	130.0	16455
drilling jar 7.5" * ? * 30'	130.0	3226
7.5" * 2.75" * 90'	130.0	9679
hevi-wt 5" * 3.1" * 90'	41.1	3059
BHA buoyed weight	=	61,488 lbf

Weight which may be applied to the bit is

WOB = 61,230 * .9 **= 55,339 lbs**

Sample calculations of weights and the buoyancy factor are

$$\text{Air Weight}_{9''} = 2.67 * (9^2 - 3^2) \qquad \mathbf{= 192.2\ ppf}$$

$$\text{Buoyancy Factor} = 1 - \frac{11.3}{65.45} \qquad \mathbf{= .827}$$

The buoyed weight is

$$\text{Buoyed Weight}_{9''} = .827 * 192.2 \qquad \mathbf{= 159.0\ ppf}$$

Weight on Bit in Inclined Holes

In inclined holes sources of available weight which may be placed on the drill bit are the BHA and the drillpipe. The available weight from the BHA is reduced because only a fraction of the weight of the BHA acts in the direction of the bit. In inclined holes drillpipe weight can be placed on the drill bit because complete helical buckling of the drillpipe is delayed by the effort required to lift the drillpipe to the high side of the hole. Once the axial load in the drillpipe exceeds its critical buckling load the drillpipe will buckle in an "lazy S". This type of buckling is not bad.

Thus, the equation which accounts for both sources for weight on bit is

$$WOB = B_f * W_{air\ BHA} * COS(\beta) + 1{,}617 \left[\frac{B_f * (D^2 - d^2)(D^4 - d^4) * SIN(\beta)}{H - D}\right]^{.5}$$

WOB	=	available weight to be placed on the bit; lbs
B_f	=	buoyancy factor
$W_{air\ BHA}$	=	air weight of the BHA; lbs
ß	=	inclination angle of the hole; deg
D	=	outside diameter of the drillpipe; inch
d	=	inside diameter of the drillpipe; inch
H	=	hole diameter (not bit diameter); inch

EXAMPLE

The drill string is composed of 90 feet of 8" * 3" collars, three stabilizer (8" * 3" * 3' each), and 9,000 feet of 5" * 4.276" drillpipe. The hole near the bottom of the drillpipe is enlarged to 16 inches. The hole inclination where the BHA and the lower portion of the BHA is located is 55 degrees. The mud weight is 11.3 ppg.

$$B_f = 1 - \frac{11.3}{65.45} \qquad \mathbf{= .827}$$

$W_{air\ BHA} = 147 * (90 + 9)$ **= 14,553 lbs**

$COS(ß) = COS(55)$ **= .574**

Available weight from the BHA is

$B_f * W_{air\ BHA} * COS(ß)$ **= 6,908 lbs**

Available weight from the Drillpipe is

$(D^2 - d^2) = (5^2 - 4.276^2)$ **= 6.716**

$(D^4 - d^4) = (5^4 - 4.276^4)$ **= 290.7**

$SIN(ß) = SIN(55)$ **= .819**

$$\frac{B_f * (D^2 - d^2)(D^4 - d^4) * SIN(ß)}{H - D} = \frac{.827 * 6.716 * 290.7 * .819}{16 - 5} = 120.2$$

$$1{,}617 \left[\frac{B_f * (D^2 - d^2)(D^4 - d^4) * SIN(ß)}{H\text{-}D}\right]^{.5} = 1{,}617 * 120.2^{.5} = 17{,}729 \text{ lb}$$

Thus, the total available weight from both the BHA and the drillpipe is

$WOB = 6{,}908 + 17{,}729$ **= 24,637 lb**

CRITICAL ROTARY SPEEDS OF BHA

Rough drilling which results in the shaking of the rig and drill string is created by the drill bit cutting on the bottom of the hole and the natural vibrational frequencies of the BHA and the drillpipe. The vibrations are of both longitudinal and torsional types.

It is believed that the roughest drilling conditions occur while the longitudinal or torsional vibration frequencies of the bit are matched by those or one of their harmonics of the BHA or the drillpipe. If the frequencies match they are said to be in phase or resonant.

Evidence of this may be seen during a normal practice while rough drilling is occurring. The practice is that in an attempt to reduce rough drilling, the driller raises the bit off bottom, rotates the drill string to a high rpm, and then while continuing the rotation, lowers the bit back onto the bottom of the hole.

It is usually observed during these sequence of events that during the high speed rotation of the drill string and for a short period after the bit is on bottom that the rig does not shake. However, thereafter the shaking commences and builds back to its original level. That is, rotation on bottom is rough, and rotation off bottom is smooth.

A second thought is that shock subs which are located in the BHA directly above the bit dampen rough drilling better than if they are located at any other place.

The longitudinal or torsional vibrational frequencies of a tricone bit is usually found to be three times the rotary speed. It is thought that the bending and orbital motion of the BHA may in part be responsible for these vibrations. In the longitudinal case the three lobes created are similar to a roller coaster track on the bottom of the hole. In the torsional case the three lobes are in a horizontal plane and are similar to the three leafs of a clover.

$$F_{bit} = 3 * \frac{N}{60}$$

$$F_{bit} = .05 * N$$

$$N_{crit\ bit} = 3 * N$$

The equation for computing the critical rotary speed for a BHA which is vibrating longitudinally, and if the frequency of the bit equals the resonant frequency of the BHA, is

$$F_{bit} = F_{long}$$

$$3 * \frac{N}{60} = \frac{4212}{L}$$

$$N_{crit\ long} = N = \frac{84240}{L}$$

Rotary speeds which create resonant frequencies in the BHA and the drillpipe are to be avoided whether they match with those of the bit or not. As with frequencies which match those of the bit, these frequencies may cause fatigue damage to the drill string whether or not they are detectable at the surface. However, for normal BHA's these frequencies are not in the range of normal rotary speeds in current practice.

The serious band widths of the resonant frequencies are approximately 20 rpm on either side of the critical rotary speeds.

The natural frequencies and the critical rotary speeds of the BHA with and without a shock sub are

without shock sub

$$F_{long} = \frac{1}{4L} * (\frac{E}{D})^{.5} * n$$

$$F_{long} = 4{,}212 / L$$

$$N_{crit\ long} = \frac{84240}{L} * n \quad \text{(match with tricone bit)}$$

$$N_{crit\ long} = \frac{252720}{L} * n \quad \text{(no match with tricone bit)}$$

$$F_{tors} = \frac{1}{4L} * \left(\frac{G}{D}\right)^{.5} * n$$

$$F_{tors} = \frac{2663}{L}$$

$$N_{crit\ tors} = \frac{53240}{L} * n \quad \text{(match with tricone bit)}$$

$$N_{crit\ tors} = \frac{159780}{L} * n \quad \text{(no match with tricone bit)}$$

with a shock sub

$$F_{long} = \frac{5.675}{2*\pi} * \left(\frac{K}{M}\right)^{.5}$$

$$N_{crit\ long} = 62.6 * \left(\frac{k}{M}\right)^{.5} \quad \text{(match with tricone bit)}$$

$$N_{crit\ long} = 187.7 * \left(\frac{k}{M}\right)^{.5} \quad \text{(no match with tricone bit)}$$

L	=	length of BHA; ft
E	=	modulus of elasticity; psf
D	=	density of steel; slugs/c.f.
G	=	shear modulus; psf
K	=	spring constant of shock sub; lbs/ft
k	=	spring constant of shock sub; lbs/in
M	=	mass of BHA; lbs
w	=	weight per foot of the BHA; ppf
n	=	1,3,5,- - - (gives higher harmonic frequencies)

Most shock shock subs do not dampen torsional vibrations; and therefore, do not affect torsional vibrations. The lower the value of the spring constant the better the dampening of the shock sub.

EXAMPLE

Compute the practical first harmonic resonant frequencies of a BHA both with and without a shock sub which is composed of 8" * 3" * 400 feet long. The rotary speed is 50 rpm. The spring constant of the shock sub is 50,000 lbs/in.

$M = w * L \qquad = 147 * 400 \qquad \mathbf{= 58{,}800\ lbs}$

without shock sub

$F_{long} = \frac{4212}{L} \qquad = \frac{4212}{400} \qquad \mathbf{= 10.53\ hertz}$

$N_{crit\ long} = \frac{84240}{400} \qquad \mathbf{= 211\ rpm}$

$F_{tors} = \frac{2663}{L} \qquad = \frac{2663}{400} \qquad \mathbf{= 6.66\ hertz}$

$N_{crit\ tors} = \frac{53240}{L} \qquad \mathbf{= 133\ rpm}$

with shock sub

$F_{long} = 0.9 * (\frac{k}{M})^{.5} \qquad = 0.9 * \left(\frac{50000 * 12}{58800}\right)^{.5} \qquad \mathbf{= 2.88\ hertz}$

$N_{crit\ long} = 62.6 * (\frac{k}{M})^{.5} \qquad \mathbf{= 58\ rpm}$

and the bit

$F_{bit} = .05 * N \qquad = .05 * 50 \qquad \mathbf{= 2.5\ hertz\ or\ 150\ cpm}$

PLACEMENT OF THE PENDULUM STABILIZER

One of the most important functions of stabilizers is to assist the bit in the drilling of a more vertical hole. The pendulum BHA as illustrated in the figure is superior to other designs. The principle of the pendulum BHA is to place the centralizer at a location in the BHA such that the point of tangency of the BHA does not contact the wall. This will maximize the pendulum force which swings the BHA back to the vertical. In addition to the location of the stabilizer the other important variable is the weight on bit.

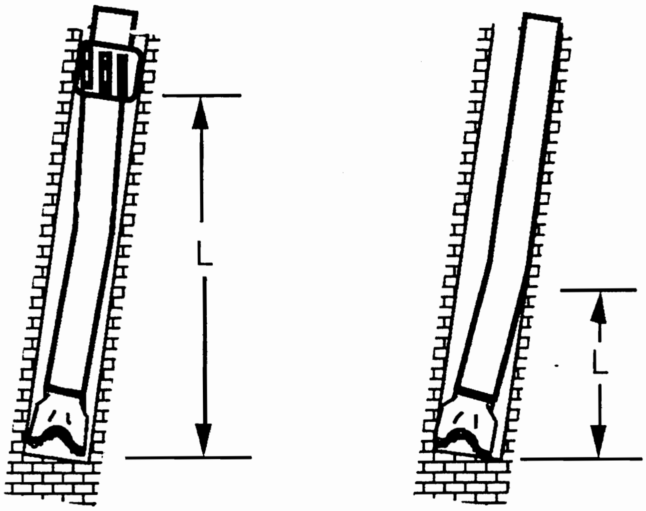
The weight on bit is critical because of the buckling of the BHA and the subsequent angle at which the bit is pointed.

The illustration shows that the pendulum force is increased with a stabilizer in the BHA for two reasons:

1. the lever arm which is the distance from the bit to point of contact on the wall of the hole is increased and

2. the weight of the pendulum is increased because there is more BHA below the point of contact.

Thus, the pendulum BHA requires the selection of the height above the bit to locate the stabilizer and the selection of the weight on bit.



Woods and Lubinski invented a theory for locating the bottom most stabilizer in a BHA to take maximum advantage of the pendulum effect. The theory appeared in charts in an abbreviated format and in tables which incorporated the full power of the theory.

Use of the charts

The following steps within the example problem explain the use of the charts which were published in the Oil & Gas Journal. (Woods and Lubinski, "Charts - - - Hole deviation," OIL AND GAS JOURNAL, April 4, 1954.)

EXAMPLE

Use the charts to determine the ideal stabilizer location and the percent more weight on bit if an inclination angle of 6 degrees has developed in a 12.25" hole while drilling with 60,000 lbs and 9" collars in a slick BHA. After putting the stabilizer in the BHA the inclination angle will be allowed to increase to 10 degrees.

STEPS

1. Locate the chart for a 12.25" hole and 9" collars.

2. On the horizontal axis (WEIGHT WITH NO STABILIZER - THOUSAND POUNDS) of the bottom chart locate the weight on bit of 60,000 lbs.

3. Follow a vertical line up to the 6 degree curve.

4. From the intersection of the 60,000 lbs line and the 6 degree curve follow a horizontal line to the vertical axis (% MORE WEIGHT WITH STABILIZER IN IDEAL POSITION) and read the value of 38%.

5. Compute the weight which may be placed on the bit with a stabilizer in the ideal position.

   Weight with stabilizer = (1 + .38) * 60,000 **= 82,800 lb**

6. On the upper companion 9" collar chart locate the value of 82,800 lbs on the horizontal axis (WEIGHT WITH STABILIZER - THOUSAND POUNDS).

7. Follow a vertical line up until it either intersects the 6 degree line or the dashed line. In this case the 6 degree line is intersected.

8. Follow a horizontal line from the 6 degree line intersection to the vertical axis (IDEAL DISTANCE STABILIZER TO BIT - FEET).

9. Read the value of the distance that the stabilizer must be placed above the bit to gain the maximum pendulum effect. The authors have written that the stabilizer may be placed 10% closer to the bit without losing a significant portion of the pendulum force.

Use of tables

The tables shown in the following figures are examples taken from the DRILCO Publication No. 59, DRILLING STRAIGHT HOLES IN CROOKED HOLE COUNTRY, copyright 1977 by Smith International,Inc.

The tables quantify the effects of collar diameter, stabilizer location, hole size, hole inclination angle, formation dip, and weight on bit on the drilling of more vertical holes.

An overview of the use of the tables is the following: Past drilling data which includes all of the parameters above are used to ascertain the isotropy of the formation drilled. In the tables isotropy of a formation has been assigned an alphabetic letter from "A", the worst formation, to "U", the mildest formation.

The theory presumes that once a letter has been assigned to a formation that the letter never changes.

Thereafter, if one or a combination of the above parameters are altered, then the remaining may be ascertained with the tables.

Several important examples are illustrated with the following problems.

EXAMPLE

A 12.25" bit is drilling a hole which has developed an inclination angle of 10 degrees. The formation has a dip of 20 degrees and the BHA is slick and contains 9" collars. The weight on bit has been 27,000 lbs.

The appropriate table is on page 64 of the DRILCO book. Find the portion of the table which has the listed hole angle of 10 degrees. Under the heading of 9" find the entry of 27,127 lbs and note that the corresponding hole classification is the letter "J".

1. Where should a stabilizer be located in the above BHA and how much weight can be put on the bit if the chosen hole angle is not changed?

Under the heading "9" with Stabilizer at Height Shown" within the 10 degree portion of the table and on the "J" row, note that the answers to the question are the values of 34,600 lbs and 75-83 feet.

2. Where should a stabilizer be located in the BHA and how much weight can put on the bit if the hole angle is to be reduced to 6 degrees?

Under the heading "9" with Stabilizer at Height Shown" within the 6 degree portion of the table and on the "J" row, note that the answers to the question are the values of 18,200 lbs and 86-96 feet.

3. Where should a stabilizer be located in the BHA and how much weight can put on the bit if the hole angle is to be increased to 15 degrees and 10" collars with a stabilizer are to be used?

Under the heading "10" with Stabilizer at Height Shown" within the 15 degree portion of the table and on the "J" row, note that the answers to the question are the values of 100,800 lbs and 63-70 feet.

4. Where should a stabilizer be located in the BHA and how much weight can be put on the bit if the hole angle is to be 10 degrees, and 7" collars with a stabilizer are to be used to drill with an 8.75" bit?

On page 57 of the DRILCO book and under the heading "7" with Stabilizer at Height Shown" within the 10 degree portion of the table and on the "J" row, note that the answers to the question are the values of 16,400 lbs and 57-63 feet.

5. What is the relative rates of penetration of the parameters listed in questions three and four?

TABLE 39 (Cont'd)
12¼" Hole size -- 20 Degree Formation Dip
Drill Collar Sizes

HOLE ANGLE AND CLASS.		8"	8" with Stabilizer at Height Shown		8½"	8½" with Stabilizer at Height Shown		9"	9" with Stabilizer at Height Shown		9½"	9½" with Stabilizer at Height Shown		10"	10" with Stabilizer at Height Shown		10½"	10½" with Stabilizer at Height Shown		11"	11" with Stabilizer at Height Shown	
6°	A	1414	1750	90–100	1597	1990	90–100	1781	2223	88–98	1960	2440	88–98	2124	2643	86–96	2259	2813	83–92	2342	2914	77–86
	B	1759	2175	90–100	1988	2475	90–100	2216	2767	88–98	2438	3038	88–98	2643	3293	85–96	2815	3502	83–92	2924	3640	77–86
	C	2294	2840	89–99	2592	3226	90–100	2890	3610	88–98	3179	3955	88–98	3445	4280	85–95	3666	4570	83–92	3800	4730	77–86
	D	2864	3550	89–99	3234	4025	89–99	3604	4503	88–98	3962	4940	88–98	4291	5340	85–95	4566	5680	83–92	4738	5900	77–86
	E	3597	4460	88–98	4062	5070	88–99	4528	5660	88–98	4980	6200	87–97	5394	6700	85–95	5741	7140	83–92	5958	7420	77–86
	F	4531	5610	88–98	5120	6370	88–98	5711	7130	88–98	6285	7830	87–97	6814	8480	85–94	7247	9020	83–92	7503	9340	77–86
	G	5640	7000	88–98	6385	7950	88–98	7138	8920	88–98	7874	9800	87–97	8561	10670	85–94	9128	11370	82–91	9476	11800	77–86
	H	7233	9000	87–97	8207	10230	88–98	9197	11500	87–97	10173	12720	86–96	11096	13800	85–94	11841	14760	82–91	12276	15330	76–85
	I	9122	11400	86–96	10343	12990	87–97	11581	14530	87–97	12799	16000	85–95	13945	17380	84–93	14903	18580	81–90	15513	19400	76–85
	J	11321	14250	86–96	12864	16170	86–96	14437	18200	86–96	15997	20100	85–94	17481	21770	84–93	18737	23380	80–89	19565	24410	76–84
	K	13860	17500	85–94	15808	20000	85–95	17813	22500	85–95	19829	24900	84–93	21783	27200	83–92	23428	29230	80–89	24513	30740	76–84
	L	17373	22040	84–93	19876	25260	84–93	22472	28640	84–93	25109	31700	84–93	27704	34840	83–92	29943	37300	79–88	31520	39600	75–83
	M	21360	27280	83–92	24584	31540	83–92	27979	35900	83–92	31492	40000	83–92	35044	44250	81–90	38142	47900	78–87	40406	50900	75–83
	N	26150	33950	81–90	30288	39300	81–90	34710	44800	82–91	39374	50300	81–90	44211	56400	79–88	48549	61600	77–86	51943	65900	74–82
	O	32614	43200	78–87	37958	50300	79–88	43729	57300	79–88	49899	64800	79–88	56412	72800	78–87	62274	79700	76–84	66918	85600	73–81
	P	40524	55700	75–83	47580	65600	76–84	55346	74700	76–85	63840	84800	76–85	73080	96500	76–84	81542	106800	75–83	88526	115000	71–79
	Q	47788	68200	69–77	56796	80600	72–80	66949	93700	74–82	78384	107500	74–82	91288	–	–	103708	–	–	114978	–	–
	R	57763	91100	60–67	69599	105300	62–69	83282	123200	66–73	99162	–	–	117756	–	–	136591	–	–	155230	–	–
	S	66840	–	–	81644	–	–	99170	–	–	120077	–	–	145387	–	–	173035	–	–	203683	–	–
	T	78331	–	–	97249	–	–	120244	–	–	148524	–	–	184024	–	–	226769	–	–	280812	–	–
	U	91701	–	–	116917	–	–	148832	–	–	189934	–	–	244380	–	–	320491	–	–	436115	–	–
10°	A	2725	3380	79–88	3077	3830	79–88	3430	4280	78–87	3775	4720	77–86	4094	5090	75–83	4360	5430	72–80	4524	5630	68–76
	B	3522	4370	78–87	3978	4950	79–88	4435	5540	78–87	4880	6100	77–86	5290	6580	75–83	5630	7000	72–80	5839	7260	68–76
	C	4445	5500	78–87	5028	6260	78–87	5613	7020	78–87	6177	7720	76–85	6694	8320	75–83	7122	8870	72–80	7393	9200	68–76
	D	5572	6950	78–87	6314	7860	78–87	7065	8820	78–87	7783	9720	76–85	8441	10500	75–83	8990	11200	72–80	9332	11620	68–76
	E	6988	8700	77–86	7929	9870	78–87	8885	11090	77–86	9790	12230	76–85	10616	13220	75–83	11304	14090	71–79	11708	14620	67–75
	F	8772	10930	77–86	9929	12380	77–86	11144	14000	77–86	12301	15380	76–84	13377	16630	74–82	14293	17800	71–79	14869	18570	67–75
	G	10895	13600	76–85	12392	15500	76–85	13921	17500	76–85	15396	19220	76–84	16773	20860	74–82	17958	22350	71–79	18731	23400	67–75
	H	13620	17100	76–84	15553	19570	76–84	17549	22100	76–84	19446	24400	75–83	21221	26450	74–82	22764	28360	70–78	23772	29760	67–74
	I	16923	21350	75–83	19388	24600	75–83	21955	27800	75–83	24414	30800	75–83	26738	33540	73–81	28812	35830	70–78	30269	38000	67–74
	J	20667	26350	74–82	23810	30400	75–83	27127	34600	75–83	30302	38400	74–82	33341	41900	72–80	36128	45400	70–78	38163	48000	67–74
	K	25010	32200	73–81	28969	37100	74–82	33199	42500	74–82	37267	47600	73–81	41215	52100	71–79	44939	56800	69–77	47761	60500	66–73
	L	30999	40200	71–79	36065	46900	72–80	41534	53700	72–80	46758	60300	71–79	51847	65800	70–78	56709	71900	68–76	60532	77000	66–73
	M	37957	50300	69–77	44450	58500	70–78	51556	67700	70–78	58312	76100	70–78	64942	83500	69–77	71412	91300	67–75	76710	97900	65–72
	N	45302	61700	67–74	53475	71800	68–76	62567	85000	68–76	71360	94600	68–76	80214	105000	68–76	89198	115800	67–74	96940	–	–
	O	53863	74900	66–73	64401	89700	67–73	76414	107000	66–73	88203	121800	66–73	100424	–	–	113411	–	–	125372	–	–
	P	64494	94600	58–65	77915	112000	61–68	93503	–	–	109057	–	–	125586	–	–	143789	–	–	161636	–	–
	Q	73405	–	–	89541	–	–	108597	–	–	128509	–	–	150599	–	–	176111	–	–	202834	–	–
	R	84383	–	–	103925	–	–	127374	–	–	153195	–	–	183165	–	–	219449	–	–	260979	–	–
	S	94359	–	–	117396	–	–	145490	–	–	177801	–	–	216767	–	–	265895	–	–	325939	–	–
	T	104995	–	–	132248	–	–	166122	–	–	207272	–	–	259256	–	–	328018	–	–	420065	–	–
	U	117641	–	–	149990	–	–	190933	–	–	244054	–	–	314747	–	–	413225	–	–	563238	–	–
15°	A	6935	8620	70–78	7843	9760	70–78	8758	10940	70–78	9649	12060	69–77	10474	13060	67–75	11143	13900	65–72	11523	14380	61–68
	B	8858	11070	69–77	10037	12570	70–78	11229	14080	69–77	12399	15500	69–77	13494	16800	67–74	14400	17930	65–72	14952	18630	61–68
	C	11143	13990	69–77	12639	15900	70–78	14156	17780	69–77	15651	19630	68–76	17059	21260	67–74	18219	22700	65–72	18916	23580	61–68
	D	13872	17470	68–76	15765	19800	69–77	17695	22300	68–76	19610	24700	68–76	21433	26750	67–74	22952	28600	65–72	23897	29800	61–68
	E	17009	21550	68–76	19367	24390	68–76	21784	27530	68–76	24200	30500	67–75	26522	33050	66–73	28498	35500	64–71	29805	37150	60–67
	F	20994	26750	67–74	24009	30450	67–75	27134	34400	67–75	30304	38500	67–74	33418	41800	66–73	36045	45100	64–71	37747	47100	60–67
	G	25887	33190	66–73	29688	37830	67–74	33654	42900	67–74	37715	48100	66–73	41755	52700	65–72	45203	56800	63–70	47510	59300	60–67
	H	31806	41200	65–72	36526	47100	66–73	41468	53400	66–73	46550	59600	66–73	51637	65400	65–72	56070	70600	62–69	59201	74500	59–66
	I	38839	50800	64–71	44755	58200	65–72	51002	66400	65–72	57492	74400	65–72	64086	81700	64–71	69917	88700	62–69	74192	93700	59–66
	J	45996	61000	62–69	53329	70400	64–71	61184	80500	64–71	69493	90500	64–71	78142	100800	63–70	85862	110000	61–68	91635	117200	58–65
	K	54237	73700	60–67	63349	86100	62–69	73266	98200	62–69	83969	110900	62–69	95410	–	–	105844	–	–	114019	–	–
	L	64442	89500	59–66	75821	104700	60–67	88397	122800	60–67	102226	–	–	117368	–	–	131523	–	–	143185	–	–
	M	74575	108000	55–61	88314	–	–	103695	–	–	120873	–	–	140059	–	–	158718	–	–	175332	–	–
	N	83959	–	–	100569	–	–	119571	–	–	141347	–	–	166460	–	–	191689	–	–	215428	–	–
	O	93942	–	–	113681	–	–	136676	–	–	163601	–	–	195474	–	–	229289	–	–	264168	–	–
	P	103912	–	–	126997	–	–	154349	–	–	187012	–	–	226605	–	–	270351	–	–	318382	–	–
	Q	113013	–	–	139462	–	–	171302	–	–	210029	–	–	258009	–	–	312857	–	–	376145	–	–
	R	121586	–	–	151581	–	–	188280	–	–	233756	–	–	291347	–	–	360630	–	–	446678	–	–
	S	129055	–	–	162141	–	–	203108	–	–	254568	–	–	320785	–	–	404370	–	–	515394	–	–
	T	136136	–	–	172434	–	–	217931	–	–	275883	–	–	351675	–	–	452665	–	–	596947	–	–
	U	143112	–	–	182764	–	–	233072	–	–	298036	–	–	384365	–	–	505671	–	–	691397	–	–

The response of rate of penetration to weight on bit per inch of bit diameter is linear for most drilling operations.

Thus, the relative penetration rate for the 12.25" hole of question number three and the 8.75" hole of question number four is

$$\text{Relative Pent. rate} = \frac{\left[\dfrac{\text{WOB}}{\text{BIT DIA.}}\right]_1}{\left[\dfrac{\text{WOB}}{\text{BIT DIA.}}\right]_2} = \frac{\dfrac{100800}{12.25}}{\dfrac{16400}{8.75}} = \mathbf{4.4}$$

Thus, it is expected that the penetration rate in this formation will be **4.4 times faster** in the 12.25" hole than in the 8.75" hole.

PACKED BHA

The packed BHA is the following: the lower 90 feet of the BHA is as axially stiff as is possible with maximum wall contact area.

The stiffness of a BHA is primarily set by the outside diameter of drill collars and the wall contact area is set by stabilizers.

Two principles have evolved for the design of packed BHAs: (1) a drill bit drills away from the hole in which it resides in the direction of the resulting lateral forces which are acting on it, and (2) in the direction in which it is pointed. A bit may be pointed out of the path of the hole by a straight BHA lying to one side of the hole or by bit tilting.

The tilt of a bit is mostly controlled by the bend in the BHA above the bit, diametrical clearance of the hole and BHA, and to some extent by the hardness of the formation.

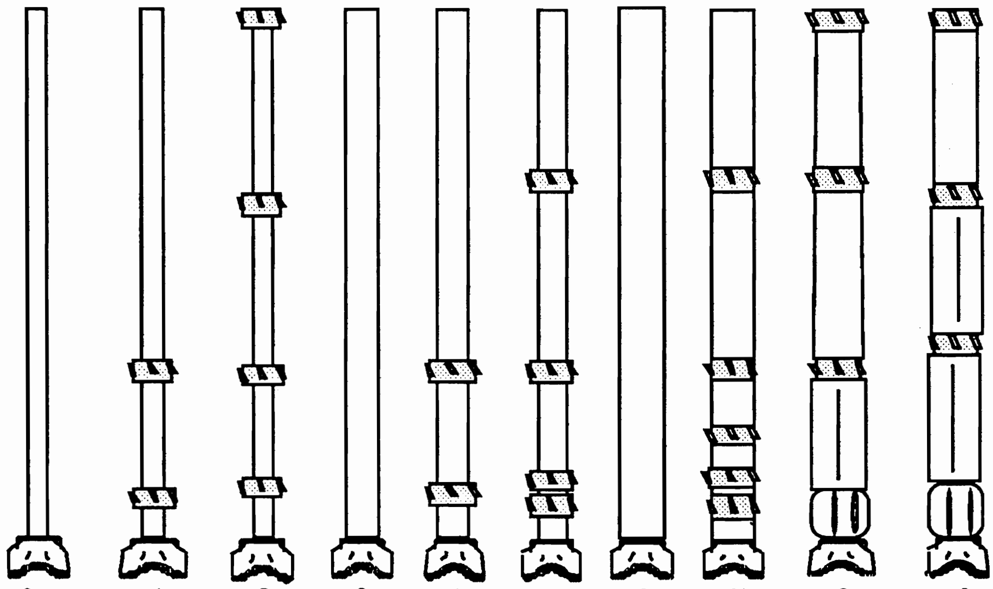
Thus, the higher the weight on bit, the clearance, and the formation strength, the more crooked the hole if all other variables are the same.

The lateral forces are derived from the pendulum force and a poorly understood or defined lateral formation force. Current thinking is that the lateral force sets the direction in soft rock and bit tilt is more important in hard rock.

Regardless of these theories, the wall contact of stabilizers or square collars improve the pointing of the bit and big collars improves the pendulum force and the tilt of the bit.

Packed BHA's must build angle; however, the build will be at very low gradient.

The following sketch presents BHA's ranked on a scale from zero to nine in regard to packness.



BHA's numbered 1, 2, and 3 have small diameter collars. Those assigned numbered 3, 4, and 5, have intermediate size collars, and those numbered 6, 7, 8, 9, have large collars. BHAs numbered 8 and 9 have square collars and a roller reamer below the square collar. Only BHAs 5, 7, 8, and 9, are truly packed BHA's.

If drilling with a 12.25 inches bit, then 7 inches collars are small, 9.5 inches collars are intermediate, and 11 inches collars are large.

In practice the roller reamers protect the square collars from excessive wear rates. In hard rock roller reamers are required because blade stabilizers create excessive torque if the bit drills an under gauge hole.

It has been argued that square drill collars appreciably increase mud bills because they grind cuttings to an unremovable size before cuttings can pass by the square collars.

Square drill collars may increase drill string torque to an unacceptable level. It may be impossible to run a square collar after drilling a hole or a section of a hole with large round collars even if stabilizers were used. Most drillers drill out beneath the last casing string with squares for this reason. Often, it is impossible to run large round collars after drilling a hole with intermediate or small collars.

The packed-pendulum BHA has a packed assembly above the pendulum section. Its purpose is to assure that a packed BHA will follow a pendulum BHA. After the pendulum section is removed, reaming of the hole below the packed assembly may be required.

Fishing for large round collars and squares may be excessively expensive or impossible to recover.

Shock subs are not stiff and because of this they should not be run directly above the bit. A recommended location is above the first drill collar, if doglegs are a potential problem.

Stabilizers, reamers, large diameter collars, and square collars are expensive.

ADVANTAGES

Packed BHA's are very popular because they serve all the purposes of BHA's but two: (1) they don't drill more vertical holes and (2) they don't drill directional holes. However, packed BHAs do the following very well:

1. protect the drillpipe in the drill string from excessive bending loads,
2. drill straighter holes,
3. reduce severities of doglegs, keyseats, and ledges,
4. assure that casing can be run into a hole,
5. increase drill bit performance,
6. reduce rough drilling, (rig and drill string vibrations).

DIRECTIONAL BHA

A directional BHA has many facets. The primary function of a directional BHA is to steer the drill bit onto the planned path of the hole and intersect the targets. In order to accomplish this, directional BHA's must be capable of

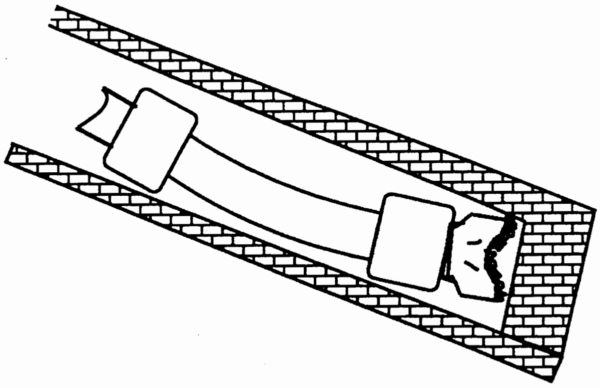
1. building inclination
2. dropping inclination
3. walking right
4. walking left
5. drilling straight ahead

If a bottom hole motor (BHM) is run then the correction of the drilled trajectory back onto the planned path is accomplished with the orientation of the BHM. If a BHM is not in the BHA, then the primary tools for hole corrections are

1. locations of stabilizers
2. rotary speed
3. weight on bit

BUILD

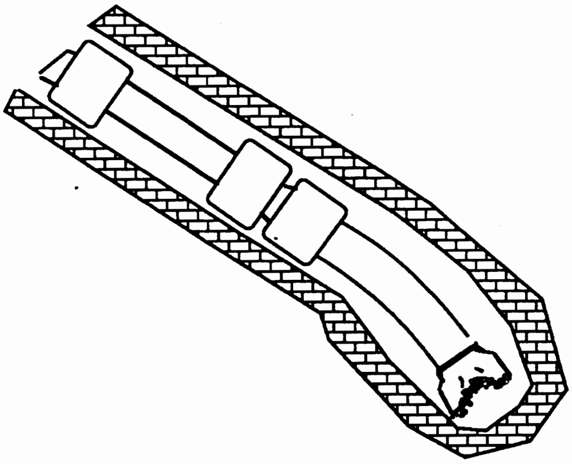
Building of inclination is accomplished with a near bit stabilizer. The near bit stabilizer is placed directly on the top of the bit and the second stabilizer is placed at a distance above the first stabilizer as to allow the sag of the collar between the two to point the bit up.



The OD and length of the collar and the weight put on the bit control the build gradient. The following sketch portrays the BHA.

DROP

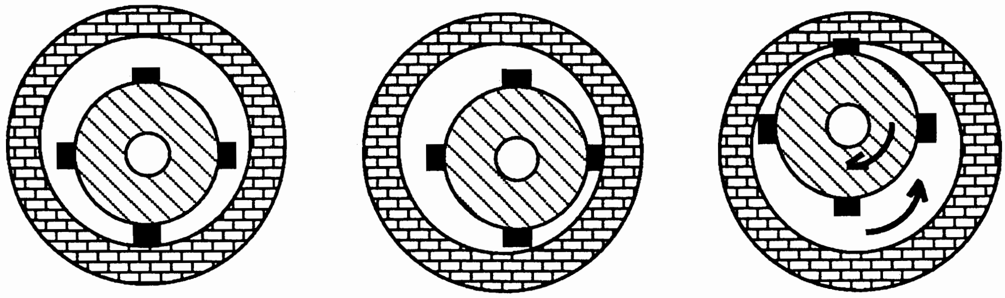
The drop BHA is portrayed in the sketch. The packed-pendulum BHA which is shown is popular as well as the pendulum. The gradient of the drop is controlled by the length of the pendulum and the weight put on the bit.



WALK

Right and left hand walk is controlled by the rotary speed and the location of stabilizers. The normal tendency for a BHA is to walk right. At lower rotary speeds the tendency is for the BHA to climb the right hand side of the hole and drag the bit to the right, and will do so until the BHA begins to whirl at higher rotaty speeds.

Most BHA's will begin to whirl at a rotary speeds of about 90 to 100 rpm in 45 degrees directional holes. The following sketch shows this progression.

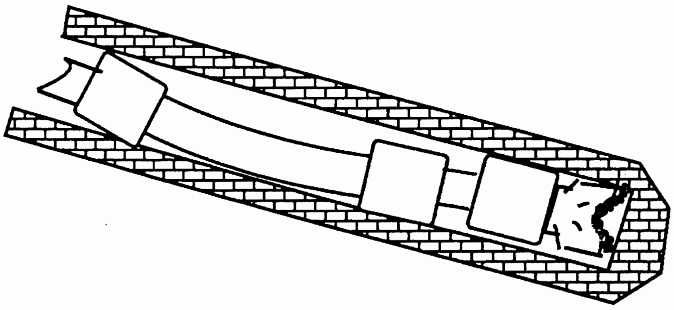


50 rpm 70 rpm 80 rpm

Once the bit and the BHA begins to whirl, the bit drills straight ahead.

Left hand walk is difficult to achieve with a right hand drill string. In order to produce left hand walk the near bit stabilizer and the bit must be raised to the top of the hole and rotated below whirling speed. The BHA in the sketch shows this.

Although the directional BHA is designed by trial and error, an experienced directional driller can call on this experience to locate stabilizers and select weight on bit and rotary speeds, which will cause the hole to intersect the target at a viable cost.

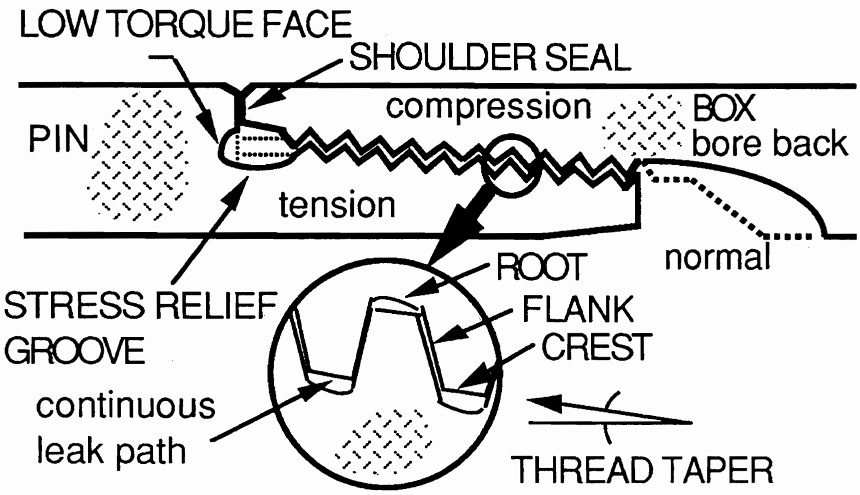


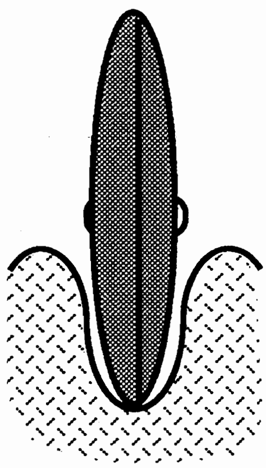
BHA CONNECTIONS

BHA connections are of the rotary shouldered type. This type of connection is strong in both tension and compression and is capable of transmitting high torque loads. Rotary shouldered connections primarily fail from bending fatigue and improper make-up.

The components of proper make-up is cleaning of the connection, lubrication, and sufficient torque. The threads do not seal and have a continuous leak path from the shoulder seal to the inside the tube. The shoulder seal must be in sufficient compressive stress to prevent separation while the tube is in tension and is being bent.

The purpose of the low torque face is to increase the compressive stress at normal make-up torque above that of a normal face. The stress relief groove is to mitigate fatigue cracks where the face and threads would have otherwise joined.





ROTARY SHOULDER CONNECTION

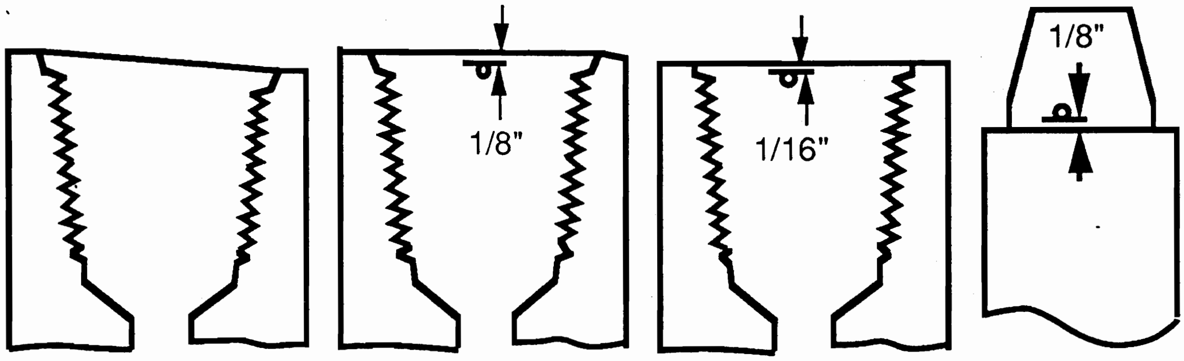
COLD ROLLING

It is above expectation that all elements of the connection be machined without error and to mate perfectly. For this reason new connections should be cleaned, lubricated, and made-up and then broken-out and cleaned again, lubricated, and re-made-up on the rig floor before running them in the hole to drill.

Most if not all fatigue cracks begin in the root of threads or in cutting tool marks. Cold working the root of threads removes the cuts left by machinist's cutting tools.

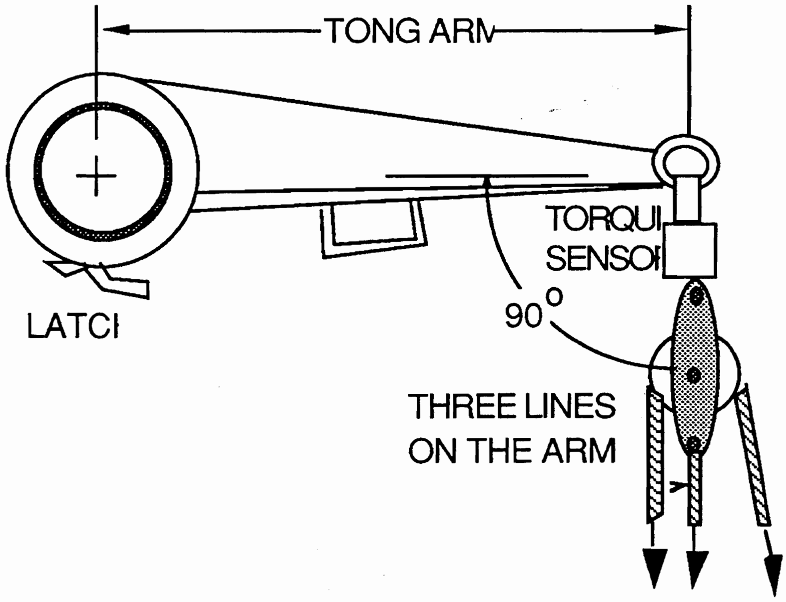
REFACING

Field refacing must be done carefully because close adherence to tolerances of mating surface on the pin and the box is a must. Not more than an accumulated 1/16" should be remove by repeated refacings. The following shows errors which are common in field refacing.



MAKE-UP OF CONNECTIONS

Torque is the length of the tong arm multiplied by the tension in the cathead line multiplied by the number of lines pulling on the tong arm. The length of the tong arm is measured in feet and is the distance from the center of the connection to the clasp (bolt) at the other end.



The line pull is measured in pounds. No more than a 1.5 percent error in make-up torque will occur if the tong arm-line pull angle is between 80 and 100 degrees. Thus, an exact 90 degree angle is not required. The following sketch shows these points.

TORQUE = TONG ARM * CATHEAD LINE PULL * NUMBER OF LINES

IDENTIFICATION OF CONNECTIONS AND DRILLPIPE

DRILCO manufactures a steel connection identification ruler which is excellent for identifying rotary shouldered connections.

A copy of <u>API RECOMMENDED PRACTICE FOR DRILL STEM DESIGN AND OPERATION LIMITS</u> (RP 7G) is mandatory for identifying worn classifications of drillpipe and tool joints.

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CHAPTER IX

AIR DRILLING

INTRODUCTION

Air drilling utilizes air or gas as the borehole circulating fluid. Four categories of air drilling exist. These are straight air, mist, foam, and aerated mud. Straight air drilling requires only that air be compressed and circulated such that bit cuttings are lifted from the borehole. Mist drilling requires the addition of a foaming agent (surfactant) to the compressed air. Small volumes of water are lifted as droplets. The bit cuttings are wet; however, the continuous fluid phase within the wellbore is air. Foam drilling requires the addition of a foaming agent and water or mud to the air. If water is used, the resulting circulating fluid is called foam and if mud is used it is called stiff foam. Large volumes of water may be lifted (30-40 gph). The air appears as tightly compressed bubbles in an ascending liquid stream. In aerated water drilling, water is the primary circulating medium and air is added without a foaming agent. Thus, foam is not created and the continuous fluid phase is water. If mud replaces the water, it is called aerated mud drilling.

ADVANTAGES AND LIMITATIONS OF AIR/GAS

The selection of air drilling systems in preference to normal mud is based on the feasibility of drilling the hole and, of course, economics. The primary advantages of straight air drilling are greatly increased penetration rates (2 to 10 times faster), more bit footage and fewer borehole drilling problems which lead to fishing operations. Water, gas and/or oil flow into the wellbore from porous zones limits the feasibility of straight air drilling. Compressor rental will increase daily costs by 50 %.

Mist drilling is usually selected to follow straight air drilling after a porous water zone is encountered. Generally, drilling rates and bit footage drop and the risk of fishing increases. Large water flows from porous zones usually require a conversion to mud drilling. Compressor costs increase slightly over straight air drilling and chemical costs may become critical.

Foam drilling is a specialty system. Its primary advantages are attributable to its reduced head on formations near or at the bottom of the borehole and high cuttings lifting capacity. It is frequently used for drilling known lost circulation zones and pay zones. Alaska's operations have shown that near-gage holes can be drilled with foam through permafrost and frozen zones because of its low heat capacity and poor heat conductance. Foam disposal can be a problem.

Aerated mud or water is popular in combating lost circulation zones and especially so if they occur in conjunction with prolific water flows. The problem is that overlying water zones require subnormal heads while lower zones require a more normal head. Disposal of excessive produced water can be a problem.

Possible advantages of air drilling are as follows:

1. Penetration rates are increased several fold.
2. Bit footage is increased.
3. Straighter holes are drilled because of lower required bit weight.
4. Tight hole problems may be reduced.
5. Differential pipe sticking is eliminated.
6. Lost circulation may be reduced.
7. Borehole diameter may be controlled more closely; however, erosion of the borehole can be a problem.
8. Water for drilling is reduced.
9. Daily maintenance costs of a mud system are reduced; however, a reserve of drilling mud must be maintained for well control.
10. Formation fluid sampling occurs on a continuous basis.
11. Lag time for circulating up formation samples is insignificant.
12. Permeability damage to producing zones may be substantially reduced.
13. Core recovery percentage is excellent.
14. The low heat capacity and the low heat conductance coefficient of air reduces thawing of frozen zones.

Possible disadvantages of air drilling are as follows:

1. Daily operational costs may be 50% higher because of additional equipment rental and material costs.
2. Downhole explosions or fire is a possible hazard. Nitrogen is safe.
3. The drillpipe and borehole may be lost if the drilling string is dropped. Excessive cork screwing of drillpipe may obviate its recovery.
4. Drillpipe rubber protectors may not be used.
5. String weight is increased because buoyancy is reduced.

6. Milling of metal (steel) with clustrite tools cannot be done economically.

7. The penetration of an undetected water zone may lead to the sticking of the drillstring.

8. Surface fires may occur while drilling with gas or by drilling in gas zones with air.

9. Deviation surveys must be run on wireline.

10. Acelerated casing and drillpipe erosion may occur.

11. Borehole erosion may present cementing problems.

12. Casing designs may be altered to provide for empty casing.

13. Drilling location areal size may be increased to accommodate additional equipment.

14. Drilling jars cannot be used.

15. The operational life of shock subs is greatly reduced.

16. In straight air drilling, reduced cutting size may be a problem in formation sampling.

AIR DRILLING EQUIPMENT

The figure shows a complement of air drilling equipment and accessories and the following table presents a summary of equipment requirements for the various types of air drilling.

Blooey Line

The blooey line carries exhaust air and cuttings to the flare pit. Recommended lengths and diameters are 300 feet or more with a cross-sectional area equivalent to that of the annulus. The outlet end of the line should be crosswind to the prevailing wind. Ells should not be permitted and the line should be anchored.

Bleed-off Line

The bleed-off line bleeds pressure within the stand pipe, rotary hose, kelly, and the drillpipe to the depth of the top float valve. The line connects the stand pipe manifold with the blooey line or mud flow line.

Chemical Pump and Tank

A chemical pump injects liquid foamers, corrosion inhibitors, and water into the high pressure air line.

These are normally of small displacements and high pressure ratings with not more than 10 horsepower. The volume of its associated tank is usually of not more than 10 barrels.

Bits

Recommended bits are those types with either tungsten carbide or short teeth, gage protected, and sealed bearings. Air bits which have a passage way for air inside the bit to circulate through the bearings are popular. Also, air bits have a strengthened web.

There appears no need to have sonic velocity of the air through the jets. Jet bits are preferred because regular bits often erode at a severe rate at their fluid courses. Bit tooth and gauge wear are not discernable while drilling and as a result under-gage holes requiring reaming are common.

Bits are commonly pulled on the basis of rotating time in the hole.

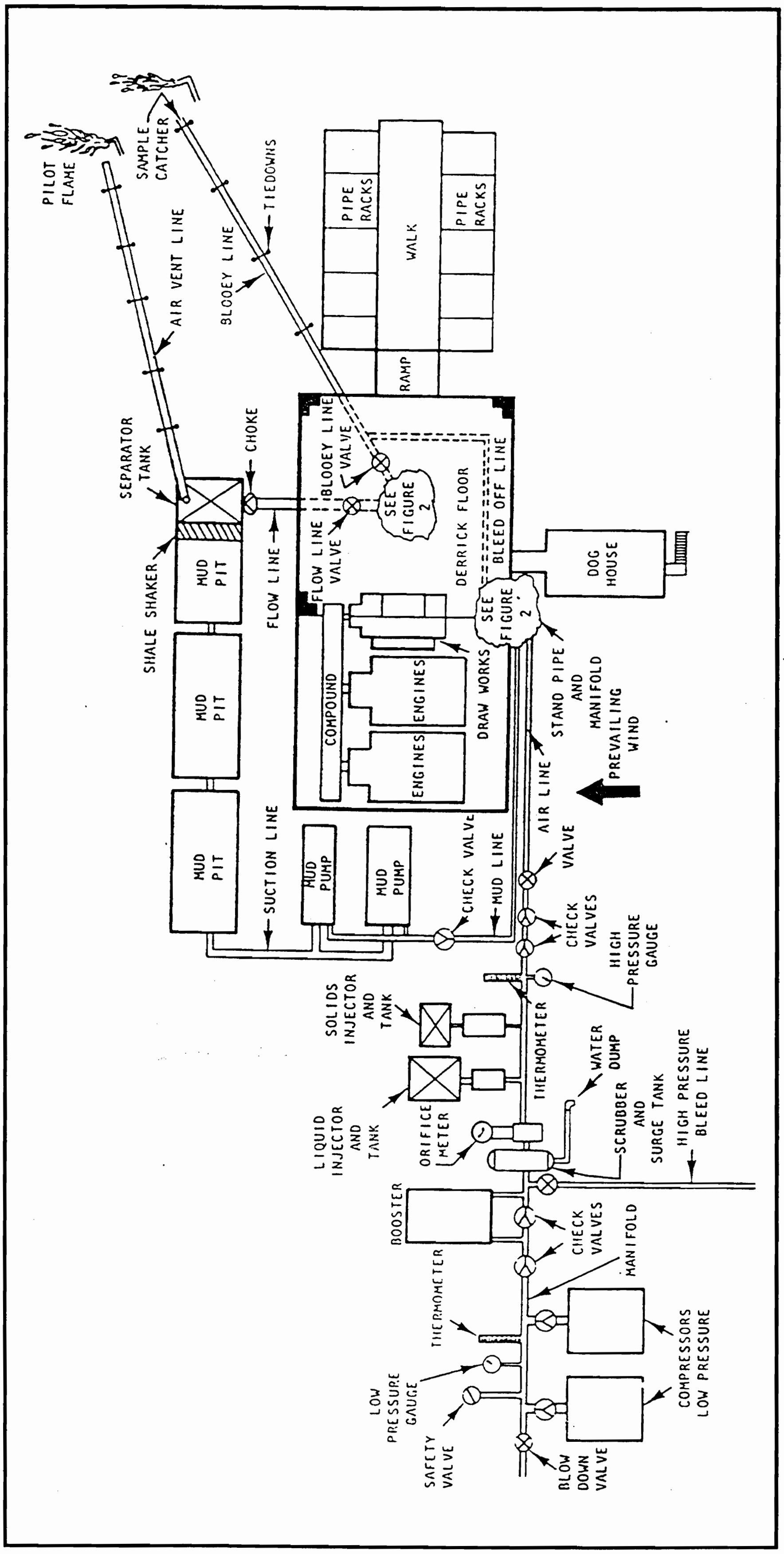
Kelly

A hexagonal kelly is recommended over the square type because of the effectiveness of the seal within the rotating kelly packer.

Floating and Float Valve Sub

Float valves are run at the bottom and top of the string. Either or both may be omitted. The bottom valve presents the backflow of cuttings into the string which could result in a plugged bit.

The top valve retains high pressure air within the drillstring while making connections. Several types are available from Baker Oil Tools and others.



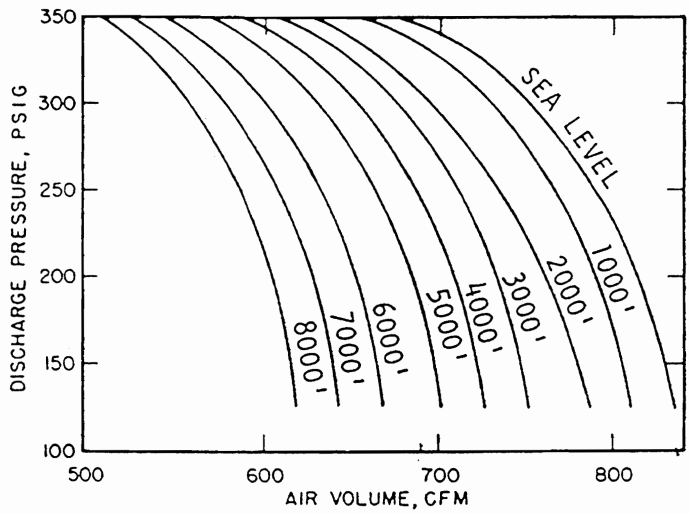
PILOT FLAME
SAMPLE CATCHER
AIR VENT LINE
BLOOEY LINE
TIEDOWNS
PIPE RACKS
WALK
PIPE RACKS
RAMP
SEPARATOR TANK
CHOKE
SHALE SHAKER
MUD PIT
MUD PIT
MUD PIT
FLOW LINE
BLOOEY LINE VALVE
FLOW LINE VALVE
SEE FIGURE 2
DERRICK FLOOR
BLEED OFF LINE
DOG HOUSE
COMPOUND
ENGINES
ENGINES
DRAW WORKS
SEE FIGURE 2
STAND PIPE AND MANIFOLD
PREVAILING WIND
AIR LINE
SUCTION LINE
MUD PUMP
MUD PUMP
CHECK VALVE
MUD LINE
VALVE
CHECK VALVES
HIGH PRESSURE GAUGE
SOLIDS INJECTOR AND TANK
THERMOMETER
WATER DUMP
LIQUID INJECTOR AND TANK
ORIFICE METER
SCRUBBER AND SURGE TANK
HIGH PRESSURE BLEED LINE
BOOSTER
CHECK VALVES
MANIFOLD
THERMOMETER
LOW PRESSURE GAUGE
SAFETY VALVE
BLOW DOWN VALVE
COMPRESSORS LOW PRESSURE

Requirements for Air Drilling (after F. W. Smith)

	Air Volume	Air Pressure	Drill String & Well Head Equipment	Flow Line
Surface Holes 12 1/4" to 22" Wet or Dry	2500 cfm to 4000 cfm	60 - 350 psi	Cutting Deflector Float Valve Bleed Line	9 5/8" Flow Line
Under Surface Pipe to 5000'(dry) 7 7/8" to 9 7/8" Hole	1500 cfm to 2000 cfm	60 - 250 psi	Double BOP Rotating Head Float Valve Bleed Line	100' to 300' 7" or 8 5/8" Flow Line Sample Catcher
Under Surface Pipe to 5000'(wet) 7 7/8" to 9 7/8" Mist Drilling	2500 cfm	200 - 350 psi	Double BOP Rotating Head Float Valve Bleed Valve	100' to 300' 7" Flow Line Sample Catcher
Under intermediate Casing 2500' to 15,000' (dry) 7 7/8" to 9 7/8" Hole	1500 cfm to 3000 cfm	100 - 300 psi	Double BOP Rotating Head Float Valve Bleed Line	200' to 300' 7" Flow Line Sample Catcher
Under intermediate Casing 2500' to 12,000' (wet) 7 7/8" to 9 7/8" Hole Mist Drilling	2500 cfm to 4000 cfm	300 - 1500 psi	Double BOP Rotating Head Float Valve Bleed Line	200' to 300' 7" Flow Line Sample Catcher
Casing in Soft Formation	1000 cfm to 2000 cfm	250 - 500 psi	Double BOP Rotating Head Bleed Line	200' to 300' 7" Flow Line
Casing in Hard Formation	1000 cfm to 2000 cfm	250 - 500 psi	Double BOP Rotating Head Bleed Line	200' to 300' 7" Flow Line
Aerated Mud Fast Penetration 5 3/4" to 8"	1500 cfm to 2000 cfm	500 - 1250 psi	Double BOP Rotating Head Float Valve Bleed Line	7" Flow Line Mud - Air Separator
Aerated Mud Loss Circulation	50 to 150 cfm per bbl of fluid circulated	500 - 1000 psi	Double BOP Rotating Head Float Valve Bleed Line	7" Flow Line Mud - Air Separator

Compressors

The compressors provide low pressure air (about 125 psig) for drilling or for charging boosters. Example performance curves are given in the following figure.



Compressor Performance Curves for a Gardner-Denver WEN

Boosters

Boosters provide high pressure air (1200 psig). See the compressor section for a discussion of boosters.

Instrumentation

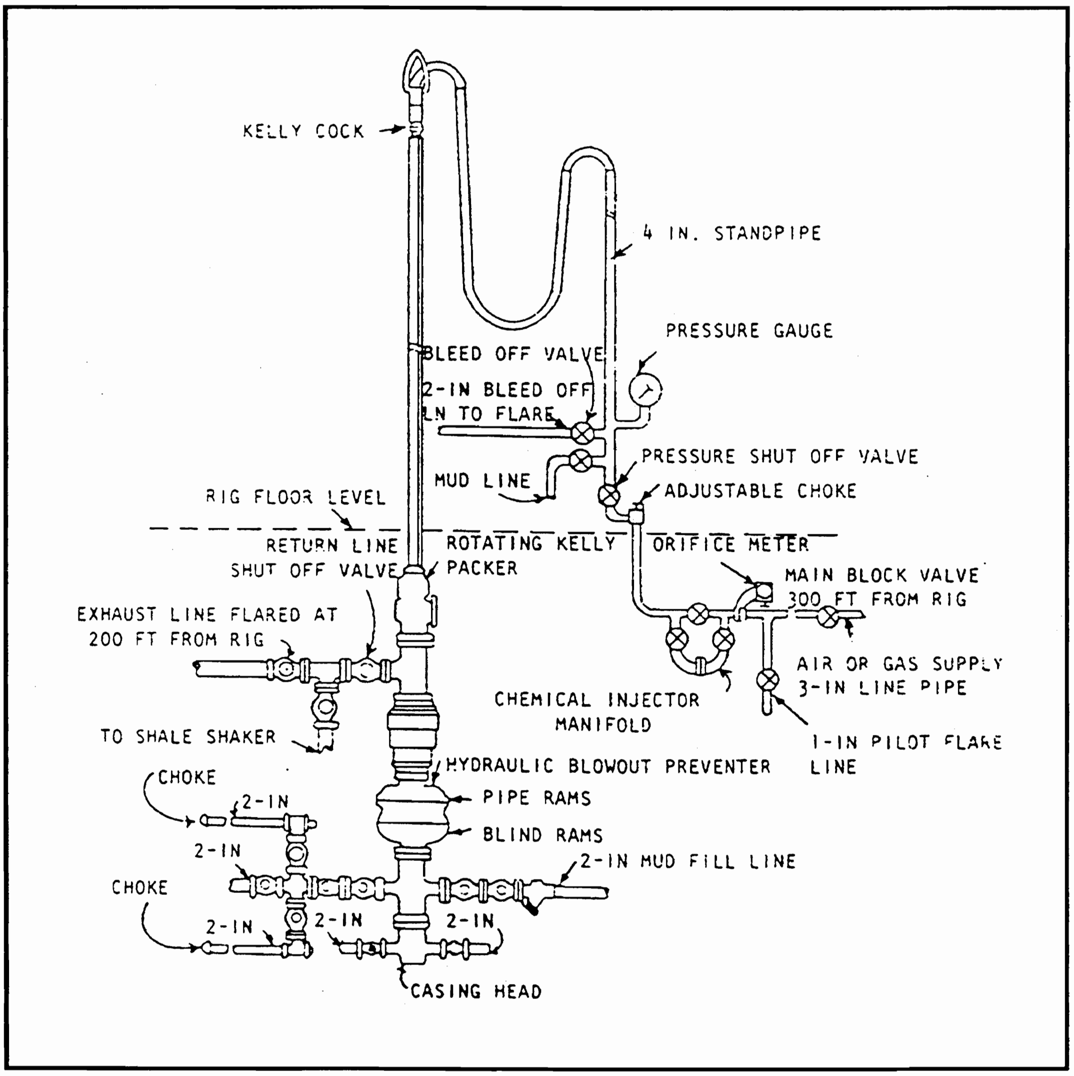
Normal rig instrumentation is sufficient. Pressure gauges are recommended to be installed in the low and high pressure air lines, in the stand pipe, and if aerated drilling, in the mud flow line (back pressure gauge). A thermometer is recommended in the air line near the stand pipe. An automatic driller may be helpful in maintaining the normally low bit weights of air drilling.

Check Valves, Shut-in Valves, and Safety Valves

A safety valve should be placed in the air manifold on the low pressure side of the check valve to prevent damage to the equipment in case high pressure formation fluids are encountered. A check valve must be placed in the air flow line between the booster intake line and discharge line. Shut-in valves should be placed in the air and mud line at their intersection for reasons of safety and self-protection.

Drillpipe and Bottom-hole Assemblies

Drillpipe and bottom-hole assemblies are the same as those required for mud; however, effective string weight will be increased because buoyancy is negligible.



Rotating Kelly Packer

The rotating kelly packer is also called a rotating blow-out preventer or rotating head. It serves to seal the annulus at the top of bell nipple. Air and cuttings are thus shunted through the blooey line or mud flow line.

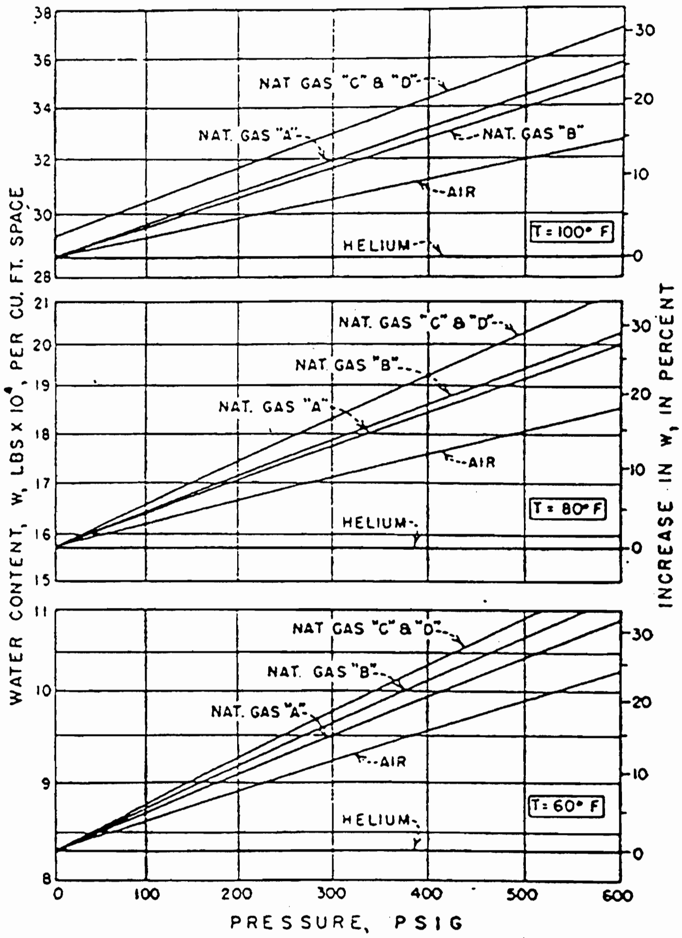
Meter Run for Measuring Air Volumes

A standard orifice plate meter run is popular for measuring air or gas injection volumes. Tables 2 and 3 contain the needed information for volume determinations. See special topics section for an example problem.

Solid Injectors

Solids injectors inject hole-drying powders into the wellbore to dry weeping water zones or to reduce torque in a deep hole. The endless chain or belt type with pistons are most practical.

Scrubber



Water Content of Air and Other Gases at Saturated Conditions (after Deaton and Frost)

The scrubber removes excess water in the air stream to assure that a minimum of moisture is circulated and to protect the booster in some cases. The included figure may be used to determine water condensation temperatures and pressures and water content.

Surge Tank

A surge tank (Figure 5) is only required in aerated water or mud drilling. Its purpose is to prevent the air from blowing water or mud out of the system and onto the location and to aid separation. Back pressure control chokes at the surge tank help control the well downhole pressures and surging.

PNEUMATICS AND HYDRAULICS REQUIREMENTS

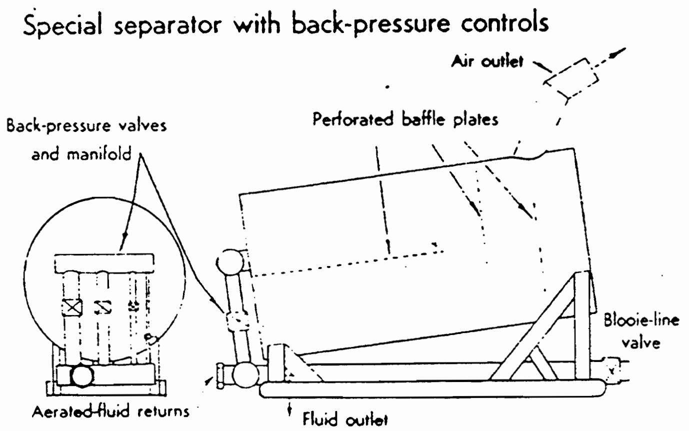
Air Volume Requirements

Air or gas volume requirements for straight air drilling may be predicated on independent studies published by R.R. Angel or K.E. Gray or by experience within an area. Both men consider the effects that borehole size, drillpipe size, drilling rate, depth and gas gravity may have on volume requirements. A major limitation of both studies is that neither consider the effects of injected or produced water.

Angel states that the lifting capacity of air increases with its density and square of its velocity. He assumes that 3000 ft/min of equivalent standard air (adjusted for density) is required to lift cuttings satisfactorily. He also assumes that all solid particles are reduced to molecular size dust.

K.E. Gray studies are based on aerodynamic theory and laboratory experiments. In these experiments, he found characteristic drag coefficients for realistic drilling chips. He, then, coupled these findings with the assumption that a 500 ft/min chip Ascension rate was adequate to clean the hole and concluded volume requirements for straight air drilling.

The published studies of both men agree sufficiently for planning purposes. As an example, consider that for drilling a 7-7/8" hole with a 4-1/2" drillpipe at a rate of 50 ft/hr and a depth of 5000 feet Angel requires 1054 ft3/min and Gray requires 1013 ft^3/min. This is a difference of 3.9%. Angel's values are presented in the following table.



Aerated Mud or Water Separator with Flow Line Pressure Control (after A. H. G. Pinks)

DATA FOR CALCULATING APPROXIMATE CIRCULATION RATES REQUIRED TO PRODUCE A MINIMUM ANNULAR VELOCITY WHICH IS EQUIVALENT IN LIFTING POWER TO A STANDARD AIR VELOCITY OF 3,000 FT/MIN

HOLE	PIPE		AIR				GAS SPECIFIC GRAVITY .60				
			VALUE OF N					VALUE OF N			
SIZE	OD	Q_0	Drilling Rate (ft/hr)				Q	Drilling Rate (ft/hr)			
(in)	(in)	(scf/min)	0	30	60	90	(scf/min)	0	30	60	90
17 1/2	6 3/8	4,209	82.2	131	177	221	5,434	66.3	128	186	240
	5 1/2	4,428	79.8	126	171	213	5,716	61.8	119	174	226
	4 1/2	4,588	78.0	123	166	207	5,924	58.0	113	165	215
15	6 5/8	2,905	71.7	112	151	188	3,751	64.2	118	167	214
	5 1/2	3,124	68.7	107	143	178	4,033	58.6	108	154	197
	4 1/2	3,285	66.0	103	137	171	4,241	54.0	100	144	185
12 1/4	6 5/8	1,700	62.3	97.8	130	160	2,194	63.0	112	155	194
	5 1/2	1,918	58.0	89.5	119	146	2,477	56.3	97.7	137	172
	4 1/2	2,079	55.3	83.6	111	136	2,684	50.8	88.2	124	157
11	6 5/8	1,237	60.6	94.5	124	151	1,597	64.5	112	152	188
	5 1/2	1,456	54.8	83.8	110	135	1,880	55.5	95.4	131	163
	4 1/2	1,616	50.6	76.9	101	124	2,087	50.0	84.4	116	146
9 7/8	5 1/2	1,079	53.0	80.3	104	126	1,393	56.4	94.7	128	157
	5	1,163	50.3	75.5	98.7	120	1,502	52.3	87.7	119	147
	4 1/2	1,240	47.8	71.7	93.3	114	1,600	48.8	81.6	111	138
9	5	898	49.1	73.0	94.4	113	1,160	53.0	87.1	116	141
	4 1/2	975	46.1	68.5	88.5	107	1,258	49.0	80.3	108	132
	3 1/2	1,103	41.5	61.0	79.0	95.5	1,424	42.0	68.9	93	115
8 3/4	5	827	49.0	72.7	93.2	112	1,068	53.5	87.0	115	140
	4 1/2	903	46.0	67.8	87.3	105	1,166	49.1	80.0	107	130
	3 1/2	1,032	40.8	60.0	77.3	93.7	1,332	41.8	68.3	92	114
7 7/8	4 1/2	670	44.7	65.0	82.7	98.3	865	50.1	78.8	104	125
	3 1/2	798	39.2	56.7	72.5	86.9	1,031	41.6	66.3	87.8	107
7 3/8	3 1/2	676	38.5	55.0	69.8	83.2	873	41.6	65.3	85.5	104
6 3/4	3 1/2	535	37.3	52.8	66.1	78.0	690	41.5	63.8	82.3	99.0
6 1/4	3 1/2	430	37.0	51.5	63.6	74.7	555	42.0	63.1	80.0	94.7
	2 7/8	494	32.8	46.0	57.3	67.7	638	37.0	55.1	71.4	85.4
4 3/4	2 7/8	229	31.6	41.3	49.5	56.5	296	37.0	51.3	62.6	72.2
	2 3/8	271	27.8	37.2	44.8	51.6	350	32.3	45.6	56.3	65.5

Q (Required cu ft/min) = Q_0 + N * H (Depth in thousand feet)

Example

Calculate the circulation rate required to air drill 11-in hole with $5\frac{1}{2}$-in. drill pipe at rate of 90 ft/hr at 11,000 ft.

$Q = Q_0 + N * H = 1{,}456 + 135 * 11$ **= 2,941 cu ft/min** (after R.R. Angel)

PRESSURE LOSSES IN PIPE AND FITTINGS

The equations for computing friction pressure losses in rig piping (100 feet and less) and fittings are

$$\Delta P_{air} = \frac{f \rho V^2}{144 D\, 2 g_c} L \qquad \text{(consistent units)}$$

$$f = \left[\frac{1}{1.74 - 2 \log_{10} (2\, \varepsilon/d)}\right]^2$$

$$\varepsilon = 0.00015 \text{ in} \qquad \text{(commercial steel pipe)}$$

$$\rho_{air} = .0764 * \frac{P_{air} + 14.7}{14.7} * \frac{460 + 60}{T_{air} + 460}$$

$$D = \frac{d}{12}$$

$$V_{air} = \frac{Q \dfrac{14.7}{P_{air} + 14.7} \dfrac{T_{air} + 460}{460 + 60}}{\dfrac{\pi}{4} \dfrac{60}{144} d^2}$$

$$\Delta P_{air} = \frac{6.504\text{E-}6}{(4.39 + \text{Log}_{10} d)^2} \frac{T_{air} + 460}{P_{air} + 14.7} \frac{Q^2}{d^5} * L$$

ΔP_{air}	=	friction pressure loss over length L of pipe; psi
L	=	length of pipe; feet
Q	=	flow Rate; Standard cubic feet/min
d	=	internal diameter; inch
D	=	internal diameter; feet

g_c	=	gravitational constant; ft/sec^2
P_{air}	=	gauge pressure of flowing air; psig
T_{air}	=	flowing air temperature; °F
ρ_{air}	=	density of air; lb/cf
ε	=	roughness factor; in
V_{air}	=	velocity of flowing air; ft/sec
f	=	Moody's friction factor

Nominal Pipe Size Inches	Schedule Number	Inside Diameter		Globe Valve*	Angle Valve*	Gate Valve*	Swing Check* Valve**	Plug Cock*	45° Std. Elbow	90° Std. Elbow	90° Long Radius Elbow	Standard Tee		Close Return Bend	90° Welding Elbow	
												Run of Tee	Side Outlet			
		Inches	Feet	L/D = 340	L/D = 145	L/D = 13 *	L/D = 135	L/D = 18	L/D = 16	L/D = 30	L/D = 20	L/D = 20	L/D = 60	L/D = 50	Short Radius	Long Radius
½	40	0.622	0.0518	17.6	7.5	.67	7.0	.93	.83	1.55	1.04	1.04	3.11	2.59		
¾	40	0.824	0.0685	23.3	9.9	.89	9.2	1.23	1.10	2.06	1.37	1.37	4.11	3.43		
1	40	1.049	0.0872	29.7	13.6	1.14	11.8	1.57	1.40	2.62	1.74	1.74	5.2	4.36	1.4	1.1
1½	40	1.610	0.134	45.5	19.4	1.74	18.1	2.41	2.14	4.02	2.68	2.68	8.1	6.7	2.1	1.6
2	40	2.067	0.172	59	25.0	2.24	23.2	3.10	2.75	5.2	3.44	3.44	10.3	8.6	2.8	2.1
2½	40	2.469	0.206	70	29.9	2.68	27.8	3.70	3.30	6.2	4.12	4.12	12.4	10.3	3.3	2.5
3	40	3.068	0.256	87	37.1	3.32	34.6	4.60	4.10	7.7	5.1	5.1	15.4	12.8	4.1	3.1
4	40	4.026	0.335	114	48.5	4.35	45.2	6.0	5.4	10.1	6.7	6.7	20.1	16.8	5.4	4.0
5	40	5.047	0.420	143	61	5.5	57	7.6	6.7	12.6	8.4	8.4	25.2	21.0	6.7	5.1
6	40	6.065	0.505	172	73	6.6	68	9.1	8.1	15.1	10.1	10.1	30.3	25.3	8.1	6.1
8	40	7.981	0.665	226	96	8.7	90	12.0	10.7	19.9	13.3	13.3	40.0	33.3	11	8.0
10	40	10.020	0.836	284	121	10.9	113	15.0	13.4	25.1	16.7	16.7	50.2	41.8	13	10
12	40	11.938	0.995			13.0	134	17.3	15.9	29.8	19.9	19.9	60	50	16	12
14	30	13.250	1.104			14.3	149		17.7	33.2	22.1	22.1	66	55	18	13
16	30	15.250	1.270			16.5	171		20.3	38.2	25.4	25.4	76	64	20	15
18	30	17.124	1.430			18.6	193		22.8	43.2	28.6	28.6	86	72	23	17
20	20	19.250	1.600			20.8	216		25.6	48.0	32.0	32.0	96	80	25	19
24	20	23.250	1.940			25.2	262		31.0	58	38.8	38.8	117	97	30	23

*All valves and cocks to be fully open.
**Check valves require 0.50 psi pressure loss to open fully.
Welding elbow data from Midwest Piping Catalog 61 (1961).
L/D values from Crane Co. Technical Paper No. 410 (1957). Both L and D in feet.

LENGTH OF STRAIGHT PIPE IN FEET HAVING THE SAME PRESSURE LOSS AS THE TABULATED FITTING

EXAMPLE

What is the pressure drop for a compressed air pressure of 100 psig at a temperature of 100°F with a flowrate of 1500 cubic feet per minute through a pipe with an internal diameter of 3 inches and a length of 100 feet?

$$\Delta P_{air} = \frac{6.504\,E\text{-}6}{(4.39 + Log_{10} d)^2} \; \frac{T_{air} + 460}{P_{air} + 14.7} \; \frac{Q^2}{d^5}$$

$$\Delta P_{air} = \frac{6.504\,E\text{-}6}{(4.39 + Log_{10} 3)^2} \; \frac{560 + 460}{100 + 14.7} \; \frac{1500^2}{3^5} \qquad \mathbf{= 1.24\ psi/100ft}$$

AIR TEMPERATURE INCREASES ON COMPRESSION

The first law of thermodynamics is

$$u = q - w$$

If adiabatic, q = 0, then

$$u = - w$$

Ideal Gas Law is

$$PV = nRT$$

and work of compressing an ideal gas is

$$w = p\ dV = znRT \frac{dV}{V}$$

Heat Capacity in terms of internal energy is

$$du = n\ C_v\ dT$$

Substituting for internal energy and work gives

$$C_v\ dT = - RT \frac{dV}{V}$$

For ideal gas the following relationships hold

$$\frac{R}{C_v} = \frac{C_p - C_v}{C_v} = \frac{C_p}{C_v} - 1 = \gamma - 1$$

Substituting gives

$$d\ln T = - (\gamma - 1)\ d\ln V$$

Integration gives

$$T\ V^{\gamma-1}\ \text{constant} \qquad \text{and} \qquad \frac{T_2}{T_1} = \left[\frac{V_1}{V_2}\right]^{\gamma-1}$$

EXAMPLE

If air is compressed to half its volume and its initial temperature was 60°F, what will its final temperature be?

$$T_1 = 520\ R \quad \text{or} \quad 60°F$$

$$T = 520 \left[\frac{10}{5}\right]^{1.37 - 1} \quad \mathbf{= 672\ R\ or\ 212^\circ F}$$

Air Pressure Requirements

Pressure requirements are rather unpredictable. Hole problems, primarily associated with formation water production, have the major effect. If water is a distinct possibility, then pressure capability to 1200 psig may be required. However, many air drilling operations are conducted below 150 psig and a few require more than 750 psig. Most operations require multi-stage compressors or boosters. A "rule-of-thumb" is that if water is encountered air volumes must be increased by 20%. Smith's table summarizes air pressure requirements.

Compressor Design Problem

The output horsepower of a compressor is given by the formula:

$$HP = 0.015\, N_s\, Q_1\, P_1 \left[(P_2/P_1)^{\frac{.283}{N_s}} - 1.\right]$$

Q_1	=	Flow rate of free air, ft^3/min.
P_1	=	Intake free air pressure, psia
P_2	=	Discharge pressure, psia
N_s	=	Number of compression stages

The equation assumes equal division of work and intercooling back to intake temperature for all stages. Free air is the volume of space occupied by the air at the conditions of the intake with no corrections made for temperature or pressure. The value of Q_1 in the above equation may be measured with a meter, estimated with piston displacement data, and/or estimated with compressor performance charts. A typical performance chart is depicted for the Gardner-Denver WEN earlier in this chapter.

If the air flow rate is metered at a pressure and temperature different than those at the intake, then Q_1 may be calculated with the following formula:

$$Q_1 = Q_m \left[\frac{P_m * T_1}{T_m * P_1}\right]$$

T_1	=	Intake air temperature, °R
T_m	=	Metered air temperature, °R
P_m	=	Metered air pressure, psia
Q_m	=	Metered air flow rate, ft^3/min.

P_1 = Intake air pressure, psia

If the air flow rate is to be estimated from piston displacement data, then for single-acting cylinders:

$$PD = \frac{A * S * N * RPM}{1728}$$

and for double-acting cylinders (rod is accounted):

$$PD = \frac{A * S * N * RPM}{874}$$

$$Q_1 = E_v * \frac{PD}{100}$$

$$E_v = 100 - \left(\frac{P_2}{P_1}\right) + C - C \left(\frac{P_2}{P_1}\right)^{\frac{1}{k}}$$

per stage and C is volumetric clearance.

The factor P_2/P_1 in the horsepower equation is called the compression ratio. Modern compressors permit a discharge temperature of 300°F or less and a compression ratio of 4 or less per stage. Higher temperatures critically shorten valve and cylinder life. The temperature criterion, in turn, establishes a maximum compression ratio. The applicable equation is

$$\frac{P_2}{P_1} = \left[\frac{T_2}{T_1}\right]^{3.5\ N_S}$$

These ideas are clarified in the following example problem.

EXAMPLE

A three-stage, 300-hp compressor is to be operated at an altitude of 5000 feet. The ambient temperature is 100°F. What are the maximum flow rate and maximum pressure available?

Table 3 shows the normal absolute pressure (P_1) at 5000 feet to be 12.22 psia. T_2 is set at 300°F or 760°R by the temperature criterion. T_1 is the air temperature and is 100°F or 560°R. Then, P_2 is calculated to be

$$\frac{P_2}{P_1} = \left[\frac{T_2}{T_1}\right]^{3.5\ N_s} = \left[\frac{760}{560}\right]^{3.5 * 3} = \mathbf{24.7}$$

The compression ratio per stage is

$$\sqrt[3]{24.7} = \mathbf{2.91}$$

which is less than the limiting maximum value of 4.0.

$$P_2 = 24.7 * 12.22 \qquad = \mathbf{302\ psia}$$

Q_1 is calculated to be

$$300 = 0.0153 * 3 * Q_1 * 12.22 \left[25.8^{\frac{2.83}{3}} - 1.02 \right]$$

$$300 = 0.0153 * 3 * Q_1 * 12.22 [1.355 - 1.0]$$

$$300 = 0.0153 * 3 * Q_1 * 12.22 * .355$$

$$Q_1 \qquad = \mathbf{1500\ ft^3/min.}$$

Next, a correction back to standard air must be completed prior to comparison with either Angel's and/or Gray's published air volume requirement tables.

$$Q_s = 1500 * \left[\frac{12.22 * (460 + 60)}{14.7 * (460 + 100)}\right] \qquad = 1500 * 0.772$$

$$= \mathbf{1160\ ft^3/min\ of\ standard\ air.}$$

Angel's tables are used to determine the depth to which this compressor could drill a 7-7/8" hole with 4-1/2" drillpipe at a drilling rate of 60 feet/hour.

$Q_s = Q_o + N * H$ (Angel's equation from his table)

$$1160 = 670 + 82.7 * H$$

$H = 490/82.7$ = **5950 feet** (Maximum expected drilling depth)

Basic guidelines for compressor operation are these: Do not operate above the recommended horsepower rating, higher than recommended RPM, in excess of pressures above the maximum calculated discharge pressure (P_2), or in excess of compression ratios of four for any stage.

Compression Boosters

Boosters are compressors. They are called boosters only because their intakes are charged with air above ambient pressures. All equations for compressors apply equally to boosters. The following is a continuation of the foregoing example problem.

EXAMPLE BOOSTER HORSEPOWER

What is the maximum attainable working pressure that may be achieved with a single stage booster? What horsepower booster is required?

Assume that after-cooling reduces the intake air at the booster to 130°F or 590°R. Intake pressure is 311 psia if 5 psi is lost between the compressors and the booster. T_2 is again fixed at 300°F or 760°R.

$$R = \frac{P_2}{P_1} = \left[\frac{T_2}{T_1}\right]^{3.5 * 1} = \left[\frac{760}{590}\right]^{3.5}$$

$$\frac{P_2}{P_1} = 2.44$$ compression ration is less than the critical value of 4

$$P_2 = 311 * 244 = \mathbf{760\ psia}$$ (maximum discharge pressure)

Q_1 must be determined for P_1 = 311 psia and T_1 = 130°F

$$Q_1 = Q_s\left[\frac{P_s * T_1}{P_1 * T_s}\right] = 1160\frac{14.7 * (460 + 130)}{311 * (460 + 60)} = \mathbf{62.2\ ft^3/min}$$

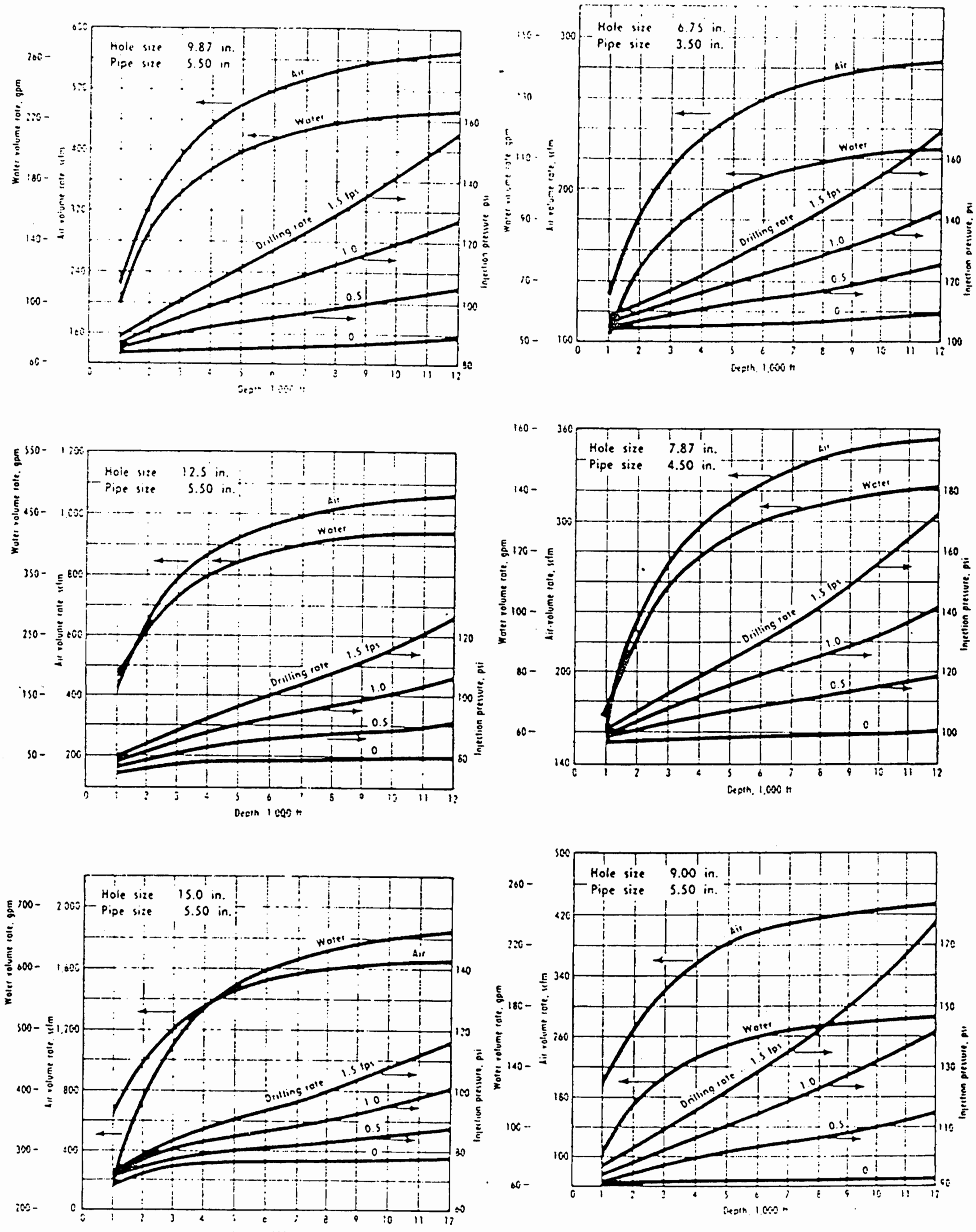
Required booster horsepower is

$$HP = 0.0153\, N\, Q_1\, P_1 \left[(P_2/P_1)^{\frac{.283}{N_s}} - 1.0\right]$$

$$HP = 0.0153 * 1 * 62.2 * 311 \left[\left[\frac{760}{311}\right]^{\frac{.283}{1}} - 1.0\right]$$

$$HP = 0.0153 * 1 * 62.2 * 311 * [1.287 - 1.0]$$

HP = **85** (output horsepower required)



Air & Water Volume and Pressure Requirements for Straight Foam Drilling (after Krug & Mitchell)

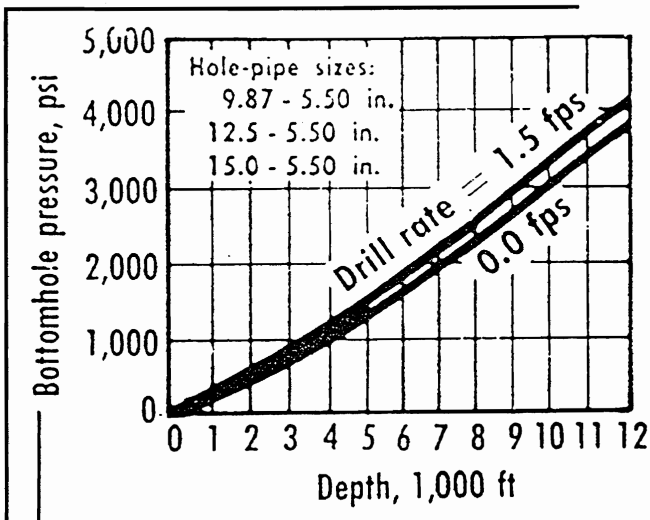
Mist Drilling Volumes and Pressure Requirements

Experience has shown that mist drilling requires up to 35% more air volume than straight air; however, no formulation of the problem has been published. Mist drilling pressures in deeper holes will normally increase above those of straight air drilling to a value that will require boosters.

Foam Drilling Volumes and Pressure Requirements

The pneumatics and hydraulics for straight air, water, and surfactant foam have been formulated and are presented in the figures of the previous page (Krug and Mitchell).

The stiff foam problem has not been formulated; however, field experience prescribe these guidelines. Air volumes will not exceed 400 scfm and most likely will range between 100 and 250 scfm. Mud volume requirements will be 10-20 gpm. Most work has been done with a conventional straight air arrangement; i.e., compressors capable of 1200 scfm and 1200 psig.



Expected Downhole Pressure in Straight Air - Water Foam Drilling (after J. A. Krug & B. J. Mitchell)

Pressure requirements may vary between 50 to 500 psig, but will generally range between 200 to 300 psig. One "rule-of-thumb" requires an air injection rate be such that the annular velocity of standard air be 200 ft/min with no allowances for compressibility of the air or the volume occupied by the mud. Figure 9 may also be used as an estimator of volume requirements for stiff foam drilling in large holes.

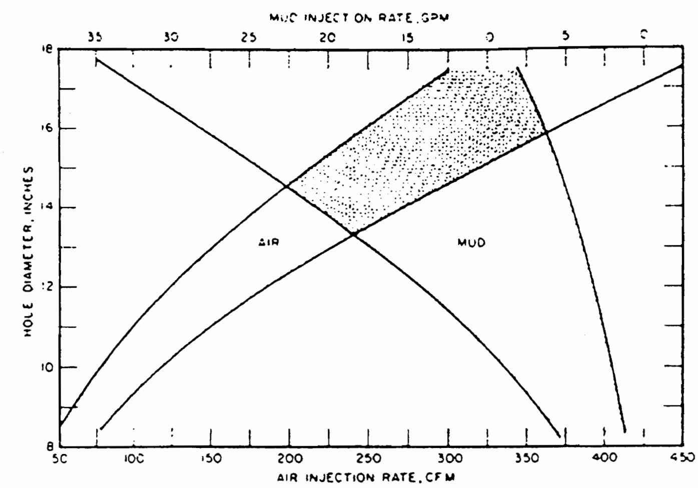


Figure 9 - Air and Mud Injection Rates for Stiff Foam (after O. Garavini, G. Radenti, and A. Sala)

WOODS AERATED MUD EQUATIONS

In the following equation **X** is the depth in the annulus at which pressure **P** is generated. In practice the air/mud ratio is adjusted until the desire pressure a depth is attained. The equation is also helpful in selecting setting depths for parasite strings.

$$\mathbf{X} = \frac{(\mathbf{P} - P_o)}{A + C} + \frac{B\ (A - C)}{2\,(A + C)^2}\ \log_e\left[\frac{(A + C)\,\mathbf{P}^2 + 2\,C\,B\,\mathbf{P} + C\,B^2}{(A + C)\,P_o^{\,2} + 2\,C\,B\,P_o + C\,B^2}\right]$$

$$- \frac{2\,B\,(A\,C)^{1/2}}{(A + C)^2}\left[\tan^{-1}\frac{(A + C)\,\mathbf{P} + C\,B}{B\,(A\,C)^{1/2}} - \tan^{-1}\frac{(A + C)\,P_o + C\,B}{B\,(A\,C)^{1/2}}\right]$$

$$A = 0.00708\left[(Q_a/Q_m)\ \rho_a + \rho_m\right]$$

$$B = 15\left[(Q_a/Q_m) + 0.02\right]$$

$$C = \frac{0.00063\ Q_m^{\,2}\left[(Q_a/Q_m)\,\rho_a + \rho_m\right]}{(\,H - D\,)\,(\,H^2 - D^2\,)^2}$$

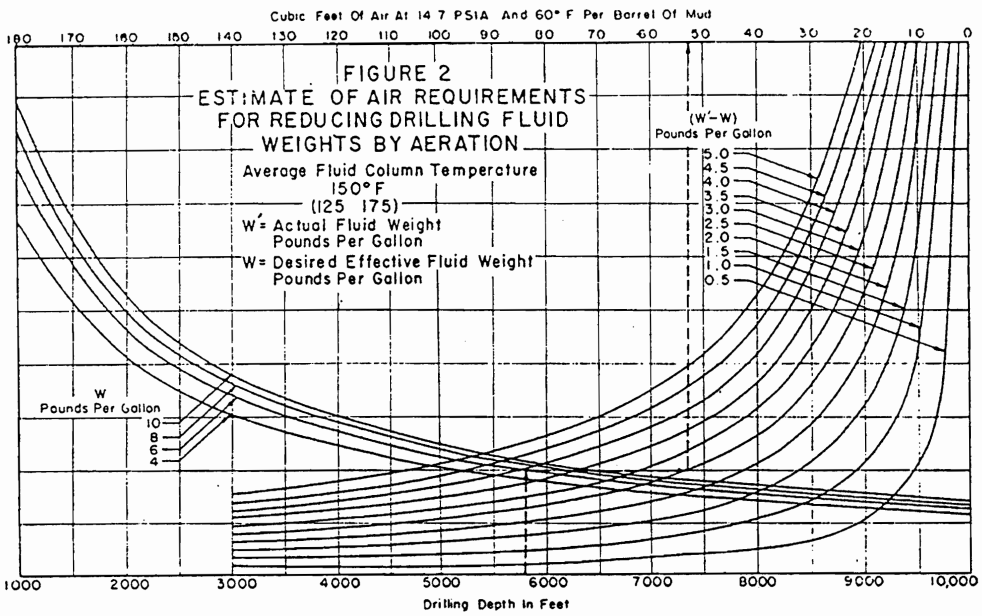
DEFINITIONS			EXAMPLE VALUES
Q_a	=	cu. ft. of free air per minute	339.1
Q_m	=	cu. ft. of mud per minute	16.957
ρ_a	=	pounds per cu. ft. of free air	0.07642
ρ_m	=	pounds per cu. ft. of mud	65.824
H	=	hole diameter, in.	8.5
D	=	drill pipe o.d., in.	4.5
P_o	=	annular pressure at the surface, psia.	14.7
X	=	depth at which pressure (**P**) occurs, ft	**10,000.**
P	=	bottom-hole pressure at (**X**), psia.	**3,307.**
Air/mud ratio =			**20 cf/cf**
AV of the annular mud =			**60 fpm**
Equivalent mud density at 10,000 ft =			**6.63 ppg**
A =			**477**
B =			**300.3**
C =			**0.00113**

Aerated Mud Volume and Pressure Requirements

H.B. Woods equation is recommended for the prediction of volumes of air required in aerated mud drilling. Jet bit hydraulics are calculated as if only mud is circulated. Initial annular velocity of mud should be 100 fpm without air volume considerations. Common air/water ratios are in the range of 5:1 through 20:1.

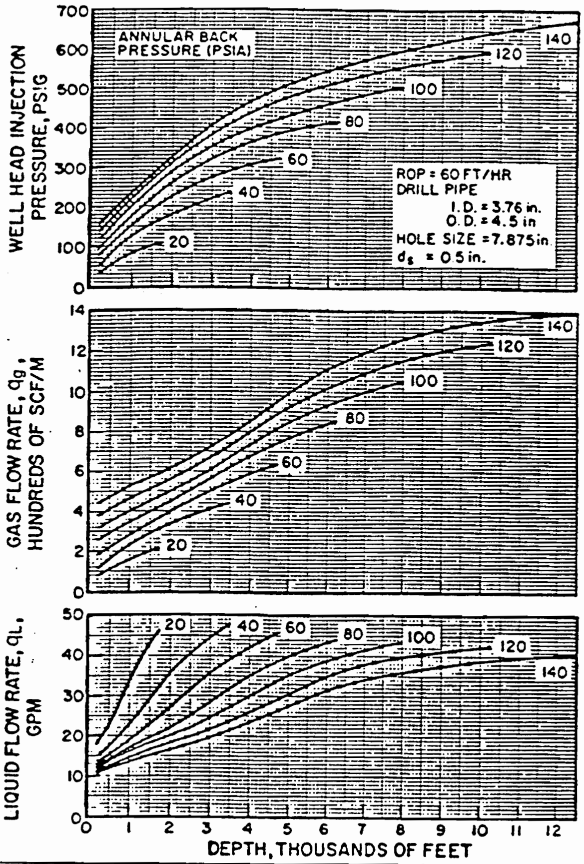
POETTMANN

F. H. Poettmann et.al. developed charts for the purpose of controlling bottom hole pressure by injecting air into the circulating mud. His chart is used to ascertain the cubic feet of standard air required per barrel of mud circulated with the mud pump. In their paper but not presented here, they publish the equations from which this was developed.

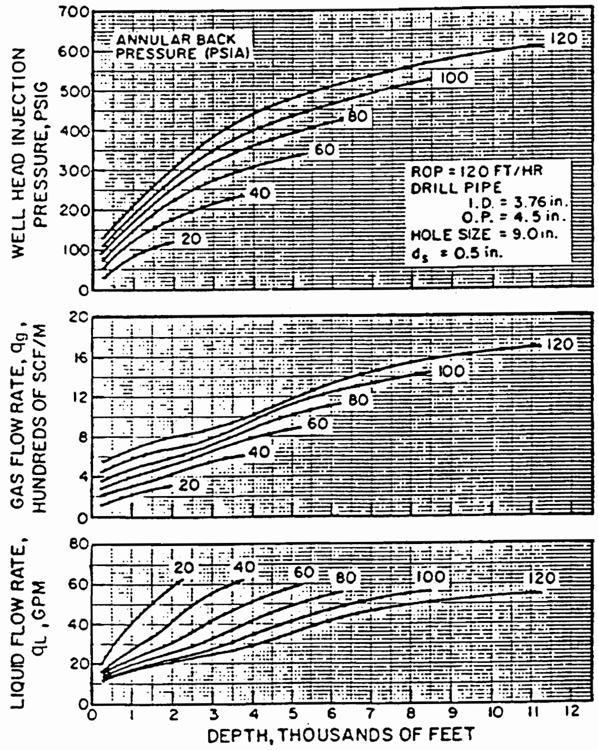


IKOKU

Okpobiri and Ikoku published charts which shows the requirements of liquid, air, and annular back pressure to lift 0.5 inch cuttings at 60, 90, and 120 feet\minute drilling rates versus depth. In their paper but not presented here, they present a computer program for making the requisite calculations.



·Volumetric requirements for 4.5-in. drillpipe in 7.875-in. hole (penetration rate = 60 ft/hr; particle size = 0.5 in.).



·Volumetric requirements for 4.5-in. drillpipe in 9.0-in. hole (penetration rate = 120 ft/hr; particle size = 0.5 in.).

OPERATIONAL PROCEDURES

Straight Air Drilling

Five operational procedures occur regularly in straight air drilling.

1. Monitoring air volumes is a must and measuring them with a meter run (see Meter Run) is recommended if water or any type of hole problem occurs. Guidelines are these: if excessive fill occurs after connections, then increase air volume; if very little fill occurs, then perhaps air volumes may be decreased. Excessive air may cause serious hole erosion.
2. During connections the backflow or annular air and cuttings through the bit and into the string may be prevented by a check valve above the bit. These cuttings often settle within the bit and plug the jets.

3. Also during connections stand pipe pressure must be bled off. To prevent the repressuring of the entire string, a check valve is placed in the string just below the kelly.

4. Deviation survey tools cannot be dropped without severely damaging them. Baker Oil Tools, Inc. manufacturers a full opening check valve which will pass survey tools on wire line.

5. Unloading the hole of water or mud in preparation for drilling with air is done with Randall's Air Unloading Curves. This circumvents staging or the use of high pressures compressors.

The charts contain three curves.

1. "time for air to reach bottom" which is of little interest.
2. "air rate" in CFM which is only of interest in preplanning but has no operational purpose.
3. "pump rate" in gpm which must be followed.

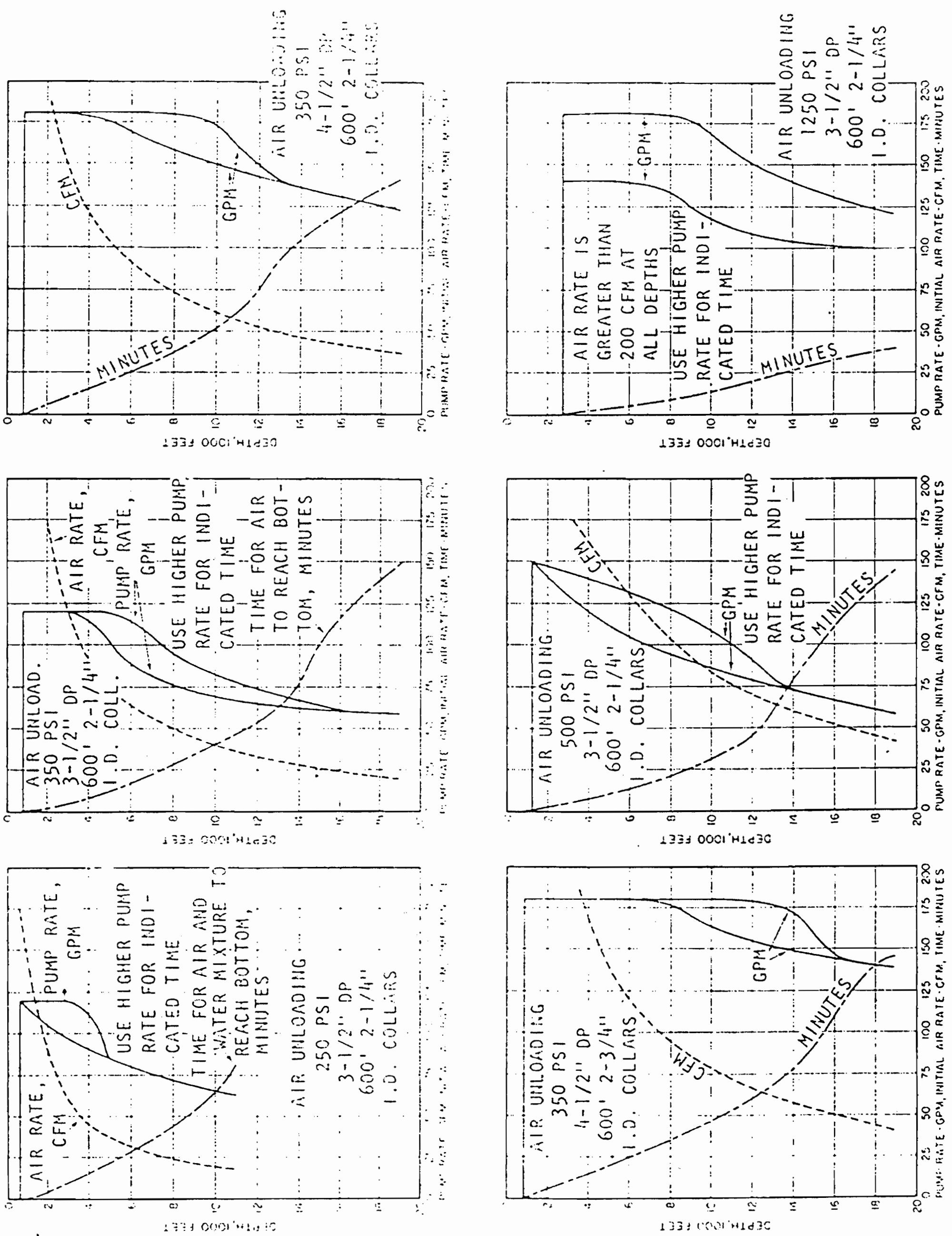
Randall's procedure requires these steps.

1. Compress air at the stand pipe to its maximum pressure and maintain that pressure.
2. Pump water or mud in accordance with Randall's charts.
3. Wait for air to reach flow line.
4. Then, stop pumping water or mud and continue circulating air. Expect heading of water or mud at the flow line.
5. The last operational procedure is the handling of small water flows (3 to 4 barrels per hour).

This problem may be solved by adding drying agents to the air stream through a solids injector or changing the system to mist. The quantity of drying agent is found by trial and error in accordance with the desired end result of re-establishing dusting.

Small water flows are most often detected by observing a decrease in cuttings return and a gradually increasing rise in circulating pressure. In some cases, produced free water may not reach the surface. If "dusting" cannot be regained, a change to mist is mandatory to prevent stuck pipe.

The mechanism that sticks the pipe is that water seeping zone(s) collect cuttings on exposed faces and deposit a part of these on the drillpipe. These are called "mud rings" and may seal the annulus while sticking the string.



AIR UNLOADING CURVES (AFTER RANDALL)

Mist Drilling

Mist drilling is mandatory once mud rings become a problem. Two operational problems will occur. The first is that a surfactant (soap or foaming agent) and perhaps additional water above that being produced by the formation will be required to combat the mud rings. The quantity of surfactant and any additional water is found by trial and error. The desired end result is the dissolution of the mud rings and an accompanying decrease in pressure. The surfactant may be injected with a pump or poured into the drillpipe during connections. A guideline quantity is one gallon of surfactant during each connection.

The second problem is that additional air volume will be needed. This quantity is also found by trial and error; however, experience dictates that 20% more air volume will be required. Water shut-off which is a possible solution is discussed in the special topics section. If the problems associated with water cannot be solved, the circulating system is generally changed to mud. See Smith's table on for a summary of air volumes required.

Foam Drilling

Operational procedures for stiff foam drilling require that the mud have satisfactory rheological properties and that the correct air and mud volumes be injected. The following mud composition for a stiff foam is recommended:

Bentonite	12	#/bbl
CMC HVd	1/2	#/bbl
Soda Ash	1	#/bbl
Foaming Agent	1	% by volume (3.5 gal/bbl)

If small quantities of salt water and/or oil are encountered, the following mud composition is recommended:

Polyanionic Cellulose (Drispac)	1/2	#/bbl
Sodium Polarcylonitrile (Cypan)	1	#/bbl
Foaming Agent (3.5 gal/bbl)	1	% by volume

Guidelines for air and mud volumes call for 60 to 90 feet/minute of annular velocities with air to mud volume ratios between 100:1 and 300:1. Excess air causes air to channel through the mud and insufficient air increases back pressure on the bottom of hole. Refer to the figures for recommended air-mud injection volumes.

Straight air-water foam also requires correct air-water injection rates. It is recommended that the water contain 1 to 3 gallons of foaming agent per barrel.

The charts of Krug and Mitchell may be used for selecting injection rates.

In both stiff foam and straight air-water foam, the factor limiting their use is the flow of excessive formation water (30 to 40 bph).

Aerated Mud or Water

The major factor in aerated mud drilling is the control of the gel strength of the mud. Control must be such that break-out of air within the annulus does not occur and that separation does occur in the pits. If break-out occurs in the annulus, heading (slugging) of the mud will erode the hole. Lack of separation on the surface causes pumping problems.

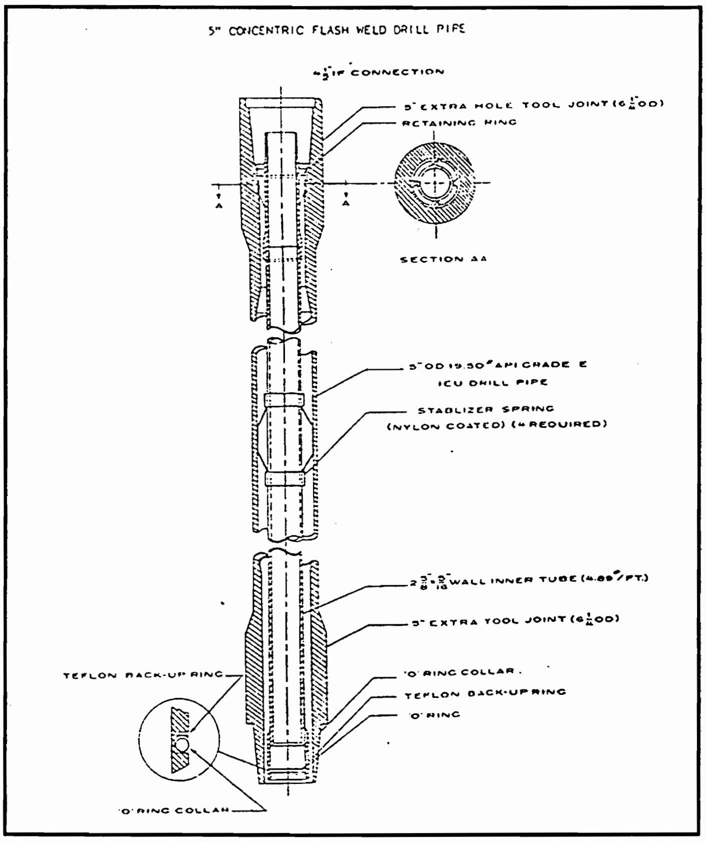
At the initiation of aerated mud or water drilling, straight mud is circulated followed by increasing volumes of air; however, violent unloading of the hole by initial excessive volumes of air are to be avoided. Maximum air volumes are usually attained by the second pass of the mud or water. A surge tank with back pressure control is recommended. Prior to trips, the hole is filled with mud. A corrosion inhibitor of polyphosphate-sodium chromate is commonly used. Refer to the tables for volumes required.

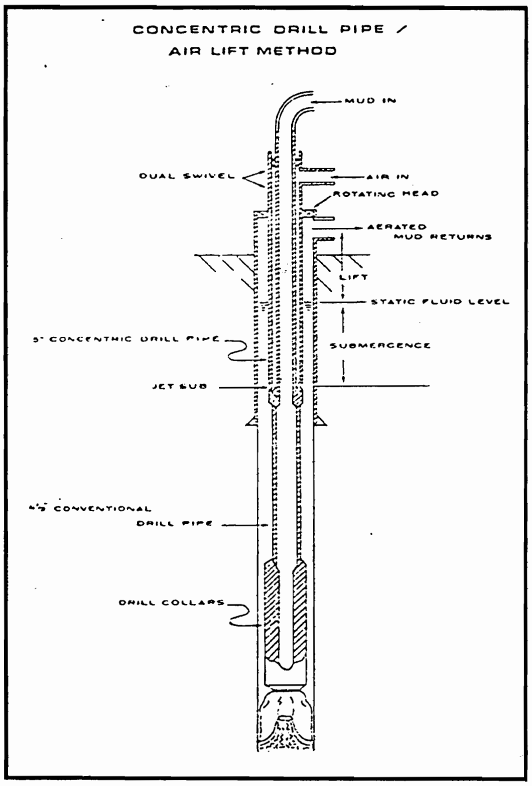
A successful aerated mud is a X-P20 lime mud with 10 % oil. Ideal properties are these:

(1)	Weight:	8.8	ppg
(2)	Viscosity:	38	sec
(3)	Apparent Viscosity:	20	cps
(4)	Yield:	5	lb/100 sq. ft.
(5)	Gels:	0	(initial)/2(3-minute) lb/100 sq. ft.
(6)	Water Loss:	6.0	cc
(7)	Cake Thickness:	2/32	inch
(8)	pH:	12.5	
(9)	Excess Lime:	4	lb/bbl

CONCENTRIC DRILLPIPE AND THE JET SUB

J. F. Binkley published a description of the concentric drillpipe system and the operational problems associated with drilling a hole from 12,630 feet to 14,866 feet. The concentric drillpipe and equipment are shown in the sketches.

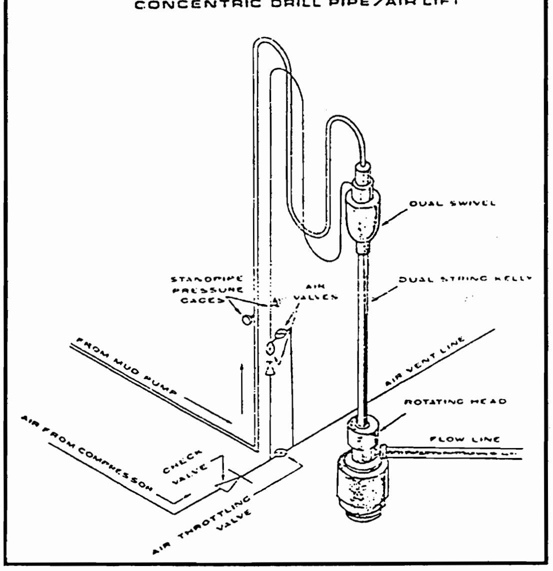




CONCENTRIC DRILL PIPE / AIR LIFT METHOD

PARASITE STRING

J. W. Murray published sketches, charts, operational procedures, and solutions of problems which arose in the drilling of a hole with a parasite string.



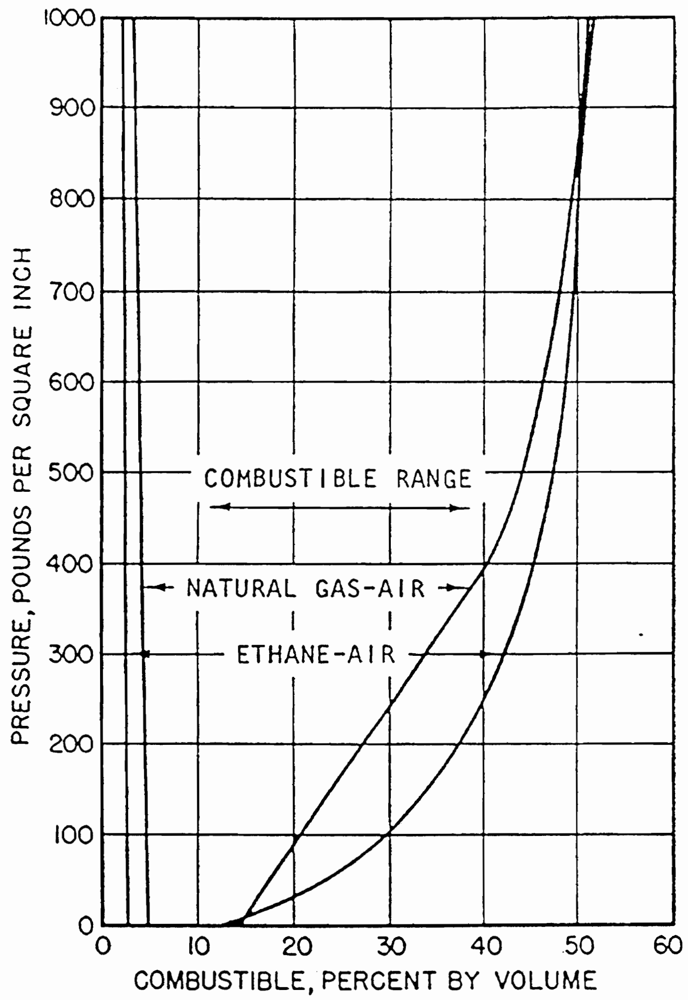
CONCENTRIC DRILL PIPE/AIR LIFT

SPECIAL TOPICS

1. Water Shut-offs

   A. Halliburton's low viscosity plastic is injected with a catalyst to seal water-bearing zones. The technique requires that the hole be full of water and high surface pressures for displacement.

   B. Technical Drilling Services, Inc., Midland, Texas, offers silicon tetrafluoride which is a heavy colorless gas with a sharp but harmless odor. The gas reacts with formation water to form a precipitate within the rock pores.



Combustible Concentrations of Natural Grades at Drilling Pressures (from Bureau of Mines R.I. 4751, 1950

   C. Gulf Oil Corporation's aluminum sulfate solution and gaseous ammonia form a solid precipitate within the pores to inhibit water flow.

2. API RP-46 - Testing Foaming Agents

   National API standard tests and reporting of results are presented.

3. Surge-Separator

Surge-Separator Tank in conjunction with back pressure valves for controlling water-air surges is described in the Oil & Gas Journal, April 19, 1965.

4. Venturi

Venturi tube for the injection of low pressure air into medium pressure mud line is described in Drilling, August, 1962.

5. Air Hammer

Mission Manufacturing Company rents pneumatic downhole percussion tools (Air Hammer). They are used primarily in crooked hole areas.

6. Explosions - Downhole Fires

Downhole fires are the result of combustion of circulated air with formation natural gas. Rare in drilling because requisite flame temperatures are not approached and combustible volume ratio criteria are not met. However, drillpipe and drill collars will be ruined and fishing is most likely if any combustion occurs.

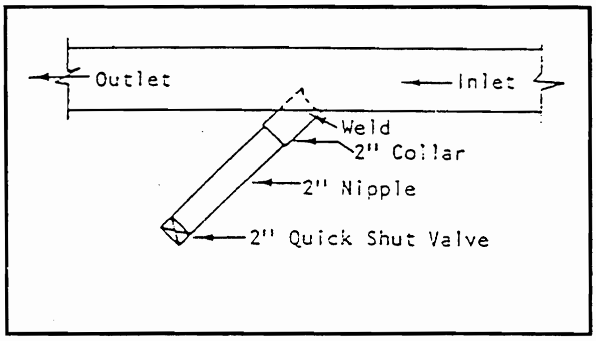
Fire is not likely to burn to the surface because of the always changing pressures and temperatures to the surface in the annulus. The figure on the preceding page depicts the flammable area in regard to volume percent and pressure. Required action to kill a fire is that air injection be discontinued.

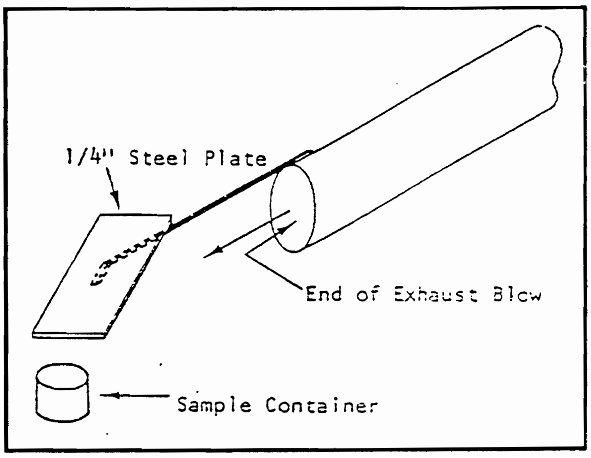
Nitrogen is safe and fire free.

7. Sample Catchers

Two types of sample catchers are popular. One is a deflection plate at the end of the blooey line where cuttings are deflected down into a container.

The other is a small diameter pipe (2") affixed to the bottom of the blooey line where cuttings will be caught. A valve on the end of this pipe is convenient for removal of cuttings.





8. Equations and Charts for Air or Gas Volume Measurements

Most satisfactory size meter runs are of diameters of 3" or 4". The pertinent equation is:

$$Q = F_b F_g \sqrt{h_w P_f}$$

Q	=	Volume flow rate, scf/M
F_b	=	Orifice flow factor - selected from Table 3
F_g	=	Specific gravity factor = $1/\sqrt{\text{Specific gravity}}$ (F_g = 1.0 for air)
G	=	Specific gravity of gas being measured
h_w	=	Differential pressure in inches of water - inches and is read from meter chart record.
P_f	=	Flowing static gas pressure, psia

EXAMPLE

Find the quantity of air if the meter pressure chart shows an average differential of 25.0" of water. Compressor elevation is 5000 feet. Meter run diameter is 4 inches. Orifice diameter is 2.000" with flange taps (taps for differential pressure lines are within the flange connectors 1" on each side of orifice plate). Static pressure is 210 psig.

F_b = 14.357 (from orifice table)

$F_g = 1/\sqrt{1.0}$ **= 1.0**

h_w = 25.0 (average pressure differential value from meter record)

P_f = 12.22 + 210.0 **= 222.22**

(The following table gives 12.22 and 210.0 is from the meter record)

$Q = 14.357 * 1.0 * \sqrt{25.0 * 222.22}$ $= 14.357 * 74.2$ **= 1062 scf/M**

Normal Atmospheric Pressure At Various Altitudes

Alt.,Ft.	Pres.,psia	Alt.,Ft.	Pres.,psia
0	14.70	6,000	11.77
500	14.42	6,500	11.55
1,000	14.16	7,000	11.33
1,500	13.91	7,500	11.12
2,000	13.66	8,000	10.91
2,500	13.41	8,500	10.70
3,000	13.16	9,000	10.50
3,500	12.92	9,500	10.30
4,000	12.68	10,000	10.10
4,500	12.45	10,500	9.90
5,000	12.22	11,000	9.71
5,500	11.99		

Orifice Factors (F_b) for Meter Runs

Orifice Diameter Inches	Meter Run Diameters (Nominal & Actual) 2" (2.067) Taps Flange	Pipe	3" (3.068) Taps Flange	Pipe	4" (4.026) Taps Flange	Pipe
0.250	0.217	0.218	0.217*	0.217	0.216*	0.217*
0.375	0.485	0.495	0.484	0.489	0.483*	0.487*
0.500	0.861	0.895	0.857	0.874	0.856	0.868
0.625	1.352	1.434	1.340	1.378	1.337	1.361
0.750	1.963	2.131	1.936	2.012	1.928	1.973
0.875	2.702	3.019	2.645	2.784	2.630	2.710
1.000	3.584	4.148	3.470	3.709	3.444	3.579
1.125	4.632	5.592	4.416	4.802	4.370	4.588
1.250	5.884	7.467	5.490	6.089	5.412	5.746
1.375	7.391	9.956	6.702	7.584	6.573	7.067
1.500	9.245	-	8.063	9.376	7.856	8.565
1.625	-	-	9.591	11.473	9.268	10.260
1.750	-	-	11.310	13.964	10.816	12.176
1.875	-	-	13.250	16.947	12.509	14.341
2.000	-	-	15.446	20.554	14.357	16.794
2.125	-	-	17.944	24.979	16.375	19.579
2.250	-	-	20.854	-	18.582	22.750
2.375	-	-	-	-	20.999	26.381
2.500	-	-	-	-	23.650	30.557
2.625	-	-	-	-	26.566	35.392
2.750	-	-	-	-	29.779	41.037
2.875	-	-	-	-	33.339	-
3.000	-	-	-	-	37.422	-

* Orifice diameter too small for accurate measurements

9. Water Content of Compressed Air

The figure depicts water content of air at various pressures and temperatures and may be used to predict the pressure and temperature at which free water will be formed.

10. Safety Practices

1. Smoking area should be designated. No cigarettes, matches ,or any spark or flame producing mechanism should be permitted on the rig floor.

2. There should be no open wires within 150' of the rig.

3. No welding should be performed around the rig when gas is being circulated.

4. All engine exhaust should be equipped with water injection system.

5. Rig ignition system should be spark proof.

6. When diesel engines are used, a device for closing air intake should be used. This should be located at an easily accessible location off the rig floor.

7. Entire rig light and wiring system, including switches, should be vapor proof.

8. Spark proof tools should be used. Consideration should be given to the use of a spinning rope instead of a chain. Kelly bushing should be well greased. All chain guards should be solidly built and oil bathed.

9. Flood lights should be required for illumination under the rig floor with no electrical connection or auxiliary gasoline engines under rig floor. The flood lights should be located at least 25 feet from rig floor.

10. A maximum amount of fire fighting equipment should be kept on hand. One operator suggests the following: three 30 pound dry chemical extinguishers near the light plant; one 350 pound dry chemical extinguisher on ground near substructure; one 150 pound dry chemical (nitrogen charge) "Fire Boss" manifold under the rig floor with 150 foot hand operated hoses.

11. Drilling crews should have regularly scheduled fire drills and be taught to use the fire fighting equipment that is available.

12. A pilot light should burn at the end of the blooey line at all times while circulating. When gas drilling, air should be cleared from the blooey line after tripping before the pilot is lit.

13. Maintain rotating head and blowout preventers in good working condition and watch for leaks of any kind under rig floor. Operate BOPs each trip. Consideration should be given to testing BOPs and connections every five days using a casing head housing test plug. The stripper rubber in the circulating head should be inspected frequently and should be replaced immediately if any leaks are detected. Grease and pack rotary

head regularly. All connections, unions, etc., should be inspected frequently with an explosimeter.

14. Remote blow out preventer controls should be rigged up a minimum 100' from rig and should be on windward side if possible.

15. Blow-out drills should be held regularly.

16. Gas should not be allowed to accumulate on the rig floor or under the substructure. Wind breaks should not be used around substructure or engines. Gas should be dissipated with bug fans operated by air or explosion proof electric motors.

17. Doghouse, trailers, light plants and etc., should be a minimum of 100' from the derrick floor.

18. Blow line should be in same direction from rig as prevailing winds if possible.

19. Bottom hole float should be used and should be inspected each trip.

20. Radio phones should be located no closer than 100' from rig floor.

21. Keep all personnel clear of well head during unloading operations; and, they should stay clear of the discharge of the blow line at all time.

22. Automobiles should be parked in a designated area 300 feet from the derrick floor. Only vehicles handling heavy equipment will be permitted beyond this point. These vehicles must obtain clearance from the driller or other authorized persons before proceeding into the danger zone.

23. In case of fire, all personnel should be instructed to protect their eyes and attempt to hold their breath until out of the burning area.

24. Visitors should not be allowed within 300 feet of the rig floor unless accompanied by authorized personnel.

25. Signs should be posted on the road leading into the location stating: "Visitors Are Not Allowed Beyond This Point." These signs shall be placed 300 feet from the rig floor.

26. There should be sufficient passageways from the derrick floor so that all employees can escape in an emergency. These passageways shall be kept open and free of all objects at all times.

27. All unnecessary equipment should be removed from the rig floor.

28. Good housekeeping practices should be followed at all times.

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1. R. R. Angel, "Volume Requirements for Air or Gas Drilling, " PETROLEUM TRANSACTIONS, AIME, Vol. 210, 1957, p. 325.

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CHAPTER X

CEMENT

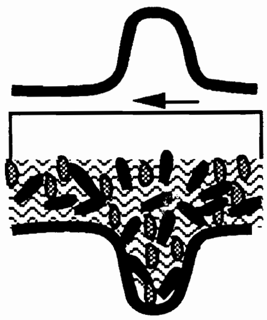
ONE DOZEN CEMENTATION PROBLEMS

There are a dozen major problems which may occur during primary cementations. The following is a list of those problems.

1. Poor displacement of the drilling mud, solids, and cuttings beds over the length of the hole that is being cemented.
2. Lost circulation during or after the cementation.
3. Bridges composed of cement filter cake.
4. Swapping out of drilling mud left below the pipe and cement circulated around the pipe (particularly bad in the setting of open plugs).
5. Flash setting of cement.
6. Shrinkage of cement.
7. Permeability after setting of the cement.
8. Gas migration (percolating gas) during the setting of the cement.
9. Micro-annulus from pressure and temperature within and of the pipe.
10. Temperature strength retrogration of the cement.
11. Perforation of cement.
12. Cement settling in high angle holes.
13. Equipment, planning, and execution failures (people errors) and the quality of cement and additives.

SOLUTIONS TO A DOZEN PROBLEMS

#1 PROBLEM: POOR DISPLACEMENT OF MUD



MECHANISM: The volumetric fraction of the mud removed from the wellbore annulus by the cement slurry is called displacement efficiency. High displacement efficiencies increase the probability that the set cement will not contain channels of mud or that the cement will not have channeled through the mud. Satisfactory displacement efficiencies depend on many factors; however, the type of flow regime in which the cement slurry and the mud being displaced is flowing during displacement is dominant.

SOLUTION: The recognized flow regimes are (1) plug, (2) laminar and, (3) turbulent. The dominant solution to poor mud displacement is cement hydraulics. Other aids are pipe rotation and reciprocation. Mobil Oil showed rotation speeds of 35 rpm are sufficient. Exxon showed 80% standoff with centralizers is sufficient with proper cement and hydraulics.

PLUG FLOW CEMENTATION

The plug flow technique is very efficient; but it is limited to cementations of small volumes and where the mud in the hole is of low density. The technique calls for Reynold's numbers of 100 or less, no cement slurry retarders or thinners, and for the cement to be in contact with the mud (no spacer). It can be demonstrated in laboratory experiments that a thickened plug or ball of flocculated cement and mud is formed in the annulus which when aided by buoyancy pushes the mud from the hole.

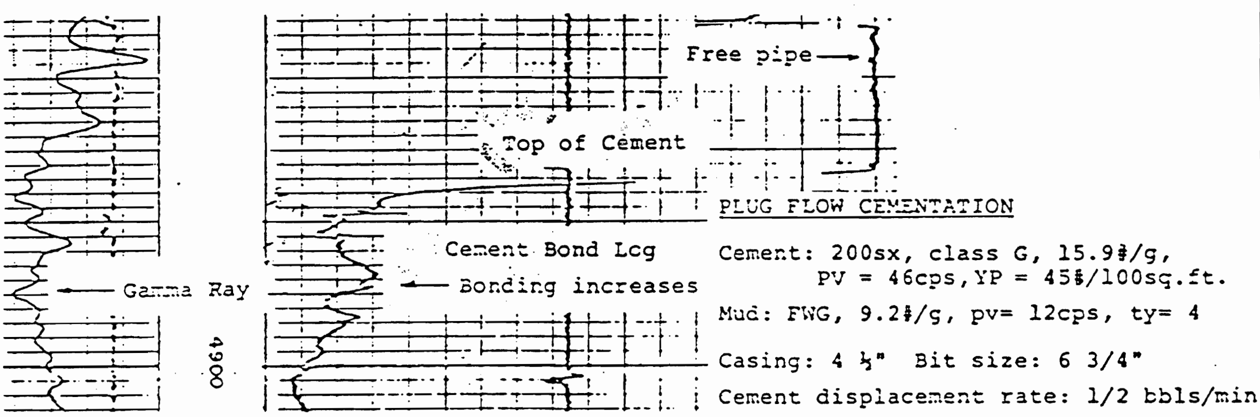
PLUG CEMENTATION VARIABLES

1) $Nr \leq 100$	1) Low hydraulic horsepower
2) No spacer	2) Low Pressures
3) No retarder	3) Low Displacement rates
4) No thinner	4) Long cementation times
5) No friction reducer	5) Low circulating bottom pressure

EXAMPLE

The cement bond log which was taken from a plug flow cementation indicates an excellent cement pipe bond. Other points are that the cement to mud transition is sharp, free pipe is clearly shown, and bonded pipe to free pipe signal strength is excellent. The following analysis points out that the cement was in plug flow.

Data is taken from the well log and the power law sheet. The cement properties are those for the class "G" cement.



The flow indices for the cement slurry are

$$n = \mathbf{.436} \qquad J = \mathbf{7.5}$$

$$K = \frac{.01066 * 7.5}{1.703^{.436}} \qquad = \mathbf{0.0634}$$

$$\frac{K_{an}}{K} = \qquad = \mathbf{1.125}$$

$$K_{an} = \qquad = \mathbf{0.0713}$$

$$v = \frac{1/2 * 42}{2.45\,(6.75^2 - 4.5^2)} \qquad = \mathbf{0.339\ ft/sec}$$

The Reynold's number is

$$N_R = \frac{2.79 \left[\frac{6.75 - 4.5}{144}\right]^{.436} .339^{2-0.436}\ 15.9}{.0713} \qquad = \mathbf{18}$$

$$f \qquad = \mathbf{.0889}$$

$$\Delta P = \frac{.889 * 15.9 * .339^2}{25.8 * (6.75 - 4.5)} * 6000 \qquad = \mathbf{167\ psid}$$

Because the Reynold's number is about 1/5 of the upper value of 100 for plug flow, an excellent plug should form in the annulus and a satisfactory cementation should result.

Turbulent Flow Cementation

The turbulent flow technique gives high displacement efficiencies and is applicable to large volume cementations or where the mud and cement slurry weight are similar. Two critical factors limit its use (1) excessive bottom hole circulating pressures and (2) insufficient surface pump horsepower or pressure. The technique calls for pumping at volumetric rates which places the mud, the spacer which should be used, and the cement slurry which is usually thinned (PV and YP are reduced) all into turbulent flow. Fully developed turbulent flow calls for Reynold's number of 3000 or more.

In all cases the Reynold's number should be calculated with the power law equation.

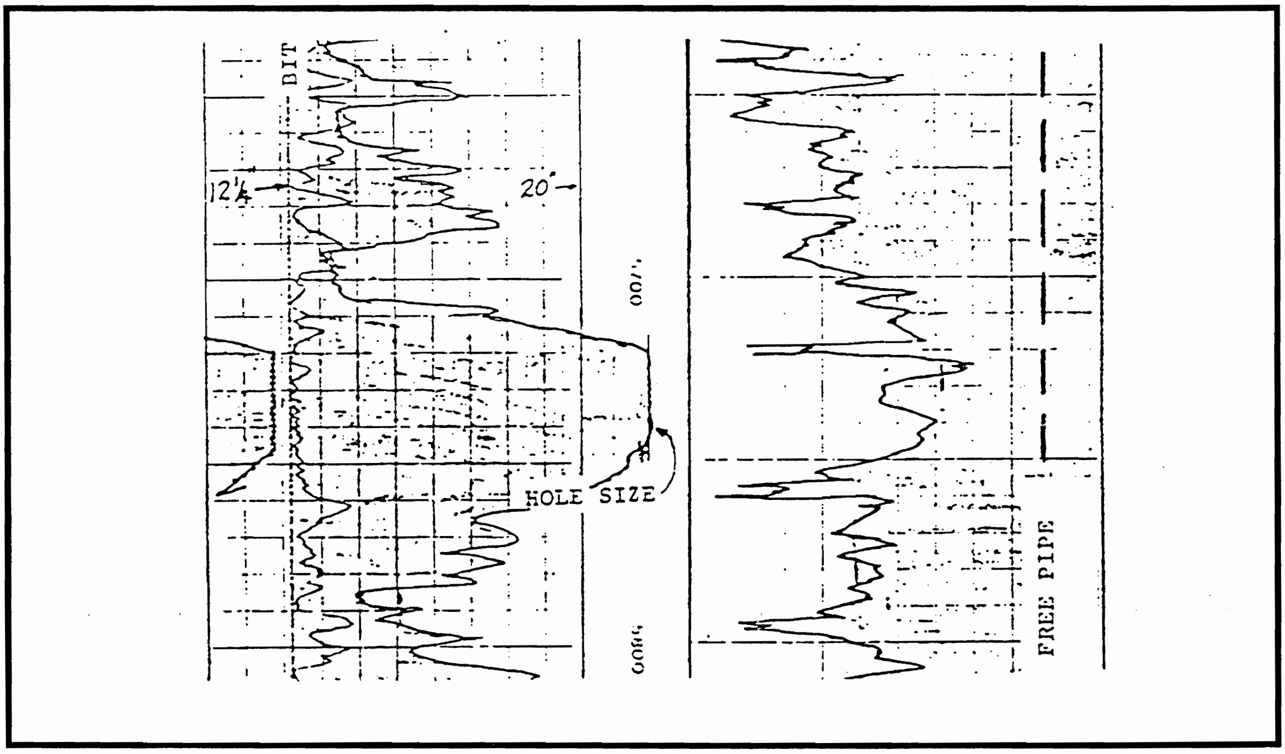
Turbulent flow cementations are primarily chosen because they require the shortest of all cementation times; and not chosen because of high circulating bottom hole pressures.

TURBULENT CEMENTATION VARIABLES

1) $N_r \geq$ Turbulent	1) High hydraulic horsepower
2) Spacers	2) High Pressures
3) Thinners	3) High rates
4) Friction reducers	4) Short cementation times
5) Retarders	5) High circulating bottom hole pressure

EXAMPLE

The cement bond log shown in the turbulent flow cementation figure illustrates better bonding where the smaller hole diameters exist. At depths of 5720' - 5750' the hole diameter is 24" or greater and less bonding has taken place than at the depth of 5695' where the hole is 13 1/4" diameter. The data for the calculations of the Reynold's numbers were taken from the figure. Casing TD is 6482'.



Well, Cement, and Pumping data:

Cement: Class 'G', 15.9 ppg, PV = 8, YP = 12

Mud: Fresh water gel with polymer, 12.2 ppg
Drill bit and casing: 12 1/4" by 9 5/8"
Maximum pumping rate: 922 gpm (22 bpm) with rig pump

The flow indices for the cement slurry were

$$n = .485 \qquad \gamma_{600} = 28 \qquad \gamma_{300} = 20$$

$$K = \frac{.01066 * 28}{(1.7 * 600)^{.485}} \qquad = \mathbf{.0104}$$

$$\frac{K_{an}}{K} = 1.12 \qquad K_{an} = .0116$$

The average flow velocity in the 24" hole is

$$v = \frac{922}{2.45\,(24^2 - 9.625^2)} \qquad = \mathbf{.779\ ft/sec}$$

The Reynold's number is

$$N_R = 2.79 \frac{(\frac{24 - 9.625}{144})^{.485}\ .779^{2-.485}\ 15.9}{.0116} \qquad = \mathbf{856\ (laminar)}$$

The average flow velocity in the $13\frac{1}{4}$ inches hole is

$$v = \frac{922}{2.45\,(13.25^2 - 9.625^2)} \qquad = \mathbf{4.54\ ft/sec}$$

The excellent Reynold's number for the same annulus is

$$N_R = 2.79 \frac{(\frac{13.25 - 9.625}{144})^{.485}\ 4.54^{2-.485}\ 15.9}{.0116} \qquad = \mathbf{6337\ (turb.)}$$

If the cement had been circulated to the surface as was planned, then the circulating bottom hole pressure would have been as calculated below.

The friction factor as read from Moody's diagram is

$$N_R = 9462 \text{ (for 12 1/4"hole)}$$

$$f = .0048$$

The pressure loss would have been

$$p_f = \frac{.0048 * 15.9 * 6.55^2 * 6482}{25.8 * (12.25 - 9.625)} \qquad = \mathbf{313.7\ psid}$$

The pressure head would have been

$$P_h \quad = \quad 0.052 * 15.9 * 6482 \qquad = \mathbf{5359.3\ psi}$$

The circulating bottom hole pressure would have been

$$P \quad = \quad 313.7 + 5359.3 \qquad = \mathbf{5673\ psi}$$

The equivalent circulating density would have been

$$ECD \quad = \quad \frac{5673}{.052 * 6482} \qquad = \mathbf{16.83\ ppg}$$

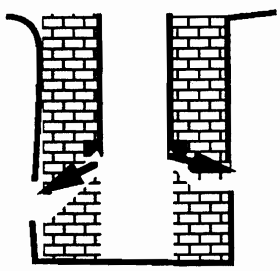
NOTE that this ECD must be less than the fracture extension gradient if circulation is not to be lost.

LAMINAR FLOW CEMENTATION

The laminar flow technique has the smallest displacement efficiencies (75% and less) of the three flow regimes and often gives inadequate cementations. It should be avoided when possible.

However, because of equipment limitations and poor planning, the technique is often inadvertently used.

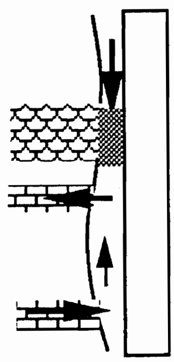
#2 PROBLEM: LOST CIRCULATION



MECHANISM: ECD is greater than the fracture strength of the hole.

SOLUTION: Reduce the density of the cement column and the circulating friction pressure loss. See the ECD calculation for the turbulent flow cementation section above for details.

#3 PROBLEM: Bridges composed of cement filter cake



MECHANISM: Bridges from cement filter cake are formed on thick permeable beds as the cement is circulated and thereafter until the cement sets. Once the cake grows to a point that it contacts the casing, it begins to support the weight of the cement and mud above it.

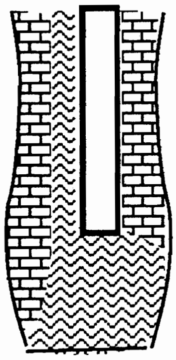
This in turn releases part of the pressure below the bridge which permits gas to percolate through the cement.

SOLUTION: The solution is use a filtrate control additive in the cement.

#4 PROBLEM: Swapping out of mud and cement below pipe.

MECHANISM: Casing or a plug back pipe is not run to the total depth of the well and the mud below the pipe buoyantly rises into the annulus where the cement was circulated.

While the mud is rising the cement must be falling along side it; thus, mud and cement are "swapping out."



SOLUTION: If pipe is close to the bottom of the hole (50 feet or so; as is the usual case of floaters), a pill of a density equal or slightly greater than that of the cement may be circulated to the bottom of the hole prior to the cementation of the pipe.

If an up hole cement plug is to be set, the pipe is run one stand below the desired location of the bottom of the plug and a very viscous bentonite pill is circulated.

Thereafter, the pipe is raised one stand and the cement is circulated into place.

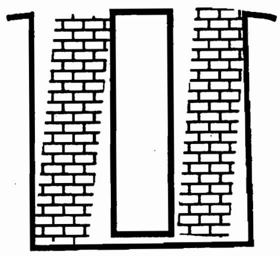
Of course, the viscous bentonite pill can be used if the casing is close to the bottom of the hole.

#5 PROBLEM: Flash setting of cement.

MECHANISM: Careful studies of flash setting of cement indicates that the mix water was much harder (Mg, Ca, and Fe) than anticipated.

SOLUTION: Check the mix water and retard the cement as needed.

#6 PROBLEM: Cement can shrink and may fail to isolate zones.



MECHANISM: Laboratory measurements have shown that cement can shrink as much as 5%.

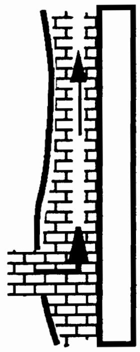
SOLUTION: Use non-shrinking cement. Non-shrinking cement has about 15% sodium chloride added to the base cement.

#7 PROBLEM: Permeability of cement may cause an interzonal flow.

MECHANISM: Laboratory measurements have shown that cement permeability increases to values of 5 millidarcys after 30 days. This may fail to isolate zones.

SOLUTION: Use an additive which blocks the growth of permeability in cement. Bentonite is good at low temperatures.

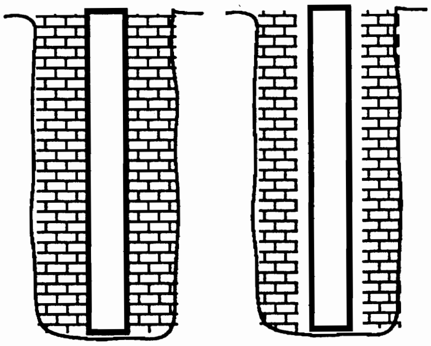
#8 PROBLEM: Gas migration may fail to isolate zones.



MECHANISM: There are two mechanisms of gas migration. The first is that during the setting of the cement, pencil size (1/8") channels form vertically in the setting cement column allowing gas to flow either to the surface or to another zone. The second is that in inclined drill holes, the cement particles settle to the low side allowing a water (or nearly water) channel to form on the high side of the drill hole.

SOLUTION: The gas migration resulting from the first mechanism has not been totally solved; however right angle setting cements and appropriate additives have been helpful. Sometimes successful additives are latex flakes and alumina particles. The second mechanism requires careful planning of cementation setting times and circulating times to eliminate long periods of quiescent unset cement in the drill hole.

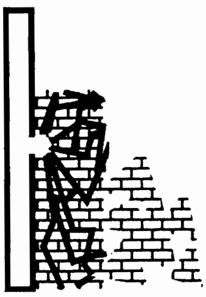
#9 PROBLEM: A micro-annulus can occur during and after pressures or temperatures are reduced in a casing or pipe. The pipe shrinks away from the cement. Isolation of zones may be lost.



MECHANISM: Steel shrinks as temperatures are reduced. Pipe diameters decrease as pressures are reduced. See the tubular chapter for an analytical example.

SOLUTION: One solution is to reduce the pressures within casing while the cement is setting. This may be done by bumping the cement plugs with a light fluid and not holding a surface pressure while the cement sets. Another is plan not to lower the pressure in the pipe at critical times.

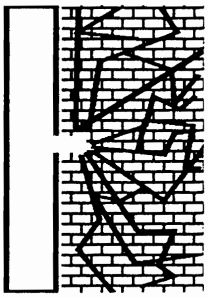
#10 PROBLEM: Temperature retrograde of cement.



MECHANISM: As temperatures approach 247°F, calcium silicate hydrate converts to crystalline forms which are weak and permeable. If at least 35% fine silica is added then tobermolite is formed (at 247°F). Xonotlite crystals which are strong and moderately impermeable form at temperatures of 302°F if at least 35% fine silica has been added. Tobermorite can be stabilized with alumina to 485°F. Truscottite crystals which are stable between 420°F and 600°F and are weaker and less permeable than xonotlite and can be formed in 50% silica flour (325 mesh or less) and portland cement.

SOLUTION: Use 35 to 50% by weight of silica flour to prevent the formation of weak and permeable cement.

#11 PROBLEM: Perforation of cement.



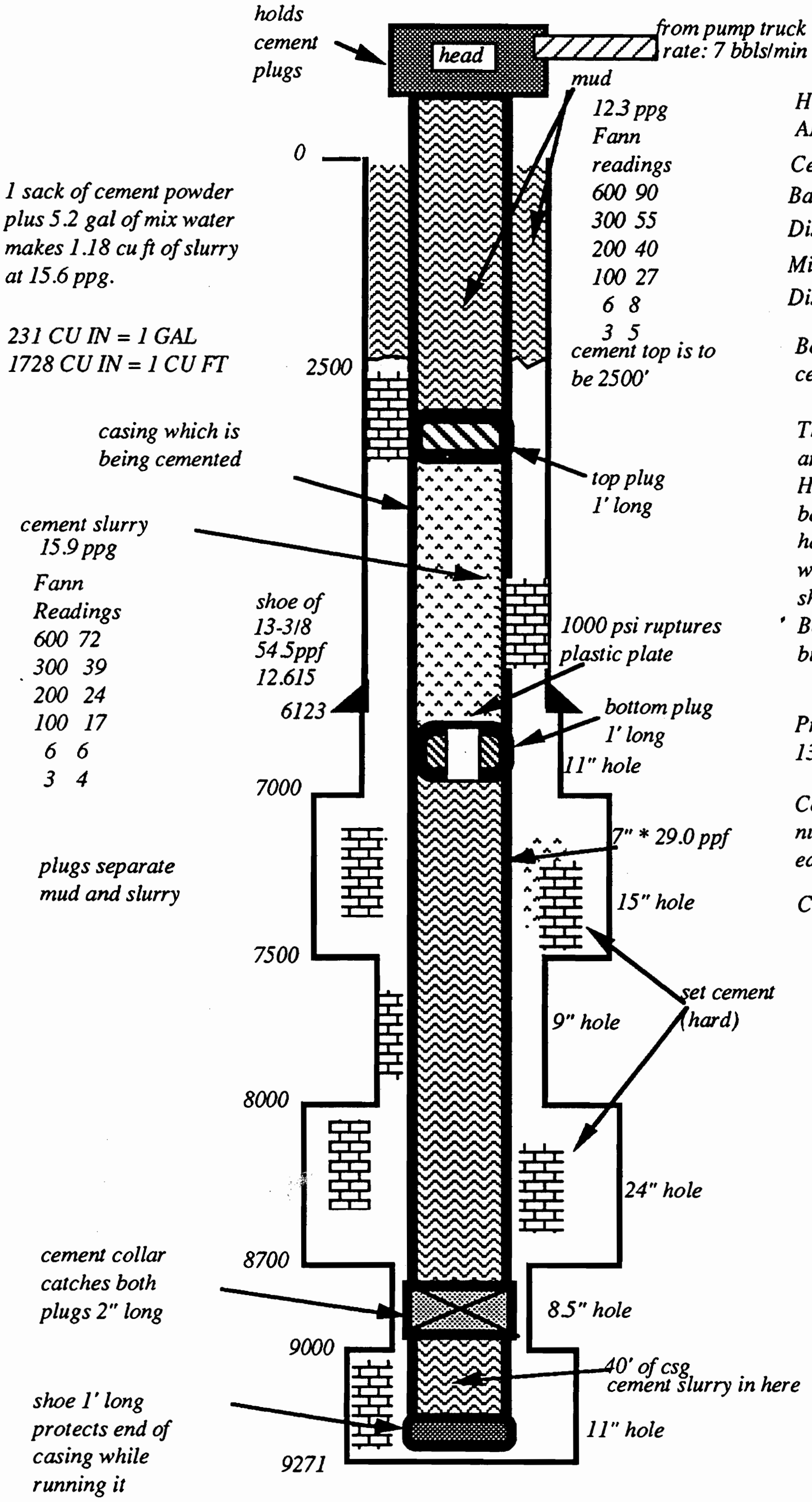
MECHANISM: Cement may shatter behind casing during perforation of the pipe leaving the cement weak and permeable.

SOLUTION: It has been found in the laboratory that cements with higher moduli of elasticity give less shattering (neat cement).

#12 PROBLEM: Equipment, material, planning, and execution failures (people errors).

SOLUTION: Hire good people.

PRIMARY CEMENTATION



HOW MANY SACKS OF CEMENT ARE REQUIRED?

Cementation time?

Barrels of mix water?

Displacement volume?

Mix time (min)?

Displacement time (min)?

Bottom hole pressure after all of cement has been mixed?

The elapsed time when the cement arrives at the shoe of 7" casing? How many sacks of cement or barrels of displacement will have been mixed or pumped when the cement arrives at the shoe?

Bottom hole pressure when plug is bumped?

Pressure at the shoe of the 13-38" when plug is bumped?

Compute power law Reynold's number for mud and cement for each hole size.

Circulating BHP when plug bumps?

Pumping rate, pump pressure and pump hhp to put both cement and mud into turbulent flow within 15" hole at the time the plug bumps?

Balanced Plug Cementation Formula

1. Capacity of hole $C_{an} = \frac{H^2}{1029.5} = \frac{17.5^2}{1029.5}$ **= .2975 bbl/ft**

2. Set Length of plug $L = B - T = 1715 - 1300$ **= 415 ft**

3. Cement Slurry Volume $C = \frac{H^2\,L}{1029.5} = \frac{17.5^2 * 415}{1029.5}$ **= 123.5 bbl**

4. Displacement depth (pipe is 5" * 21ppf)

$$D = B - \frac{C}{\frac{H^2}{1029.5} - \frac{W}{2747}} = 1715 - \frac{123.5}{\frac{17.5^2}{1029.5} - \frac{21}{2747}}$$ **= 1289 ft**

5. Displacement volume

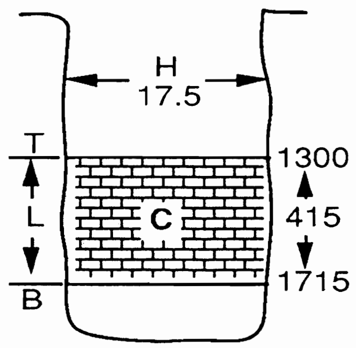
$$V = \frac{D\,d^2}{1029.5} = \frac{1289 * 4.276^2}{1029.5} - \#IUV$$ **= 22.9 bbl**

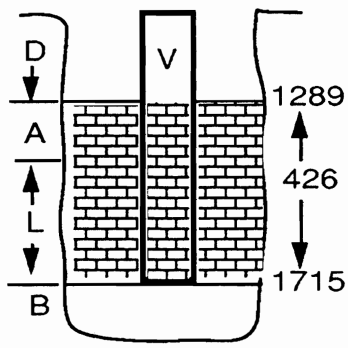
(IUV = Internal Upset Volume of tooljoints must be subtracted)

6. Sacks of cement $= \frac{5.615\ C}{Y\ E} = \frac{5.615*123.5}{1.18 * 1.335}$ **= 785 sk**

7. Volume of water $= \frac{785 * 5.27}{42}$ **= 98.5 bbl**

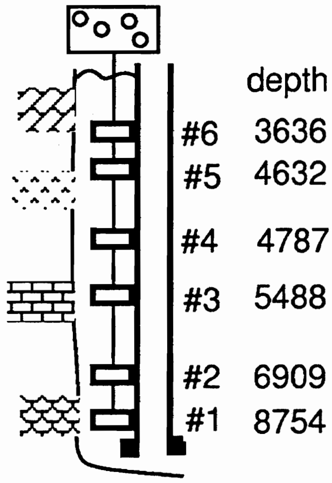
H	=	Hole diameter; inch
L	=	Length of plug; ft
C	=	Cement Slurry volume; bbl
W	=	Linear weight of string; lb/ft
d	=	inside diameter of pipe; inch
D	=	Displacing liquid depth; ft
V	=	Displacement volume; bbl
Y	=	Yield of Cement; cf/sk
T	=	Top of Plug; feet
E	=	Excess cement slurry factor





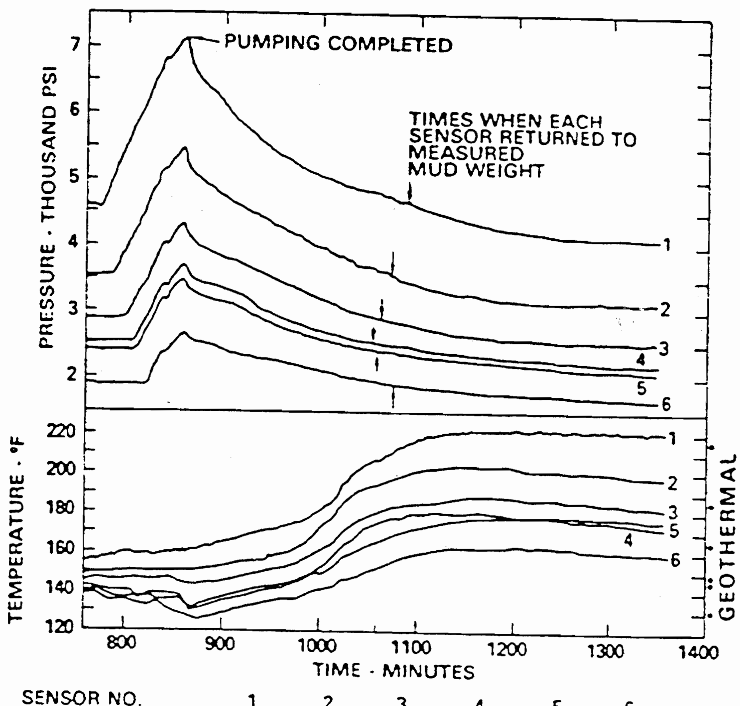
CEMENTATION TEMPERATURES

Cooke (EXXON) measured downhole temperatures and pressures during a primary cementation. Six temperatures and pressure sensors were attached to the outside of 2 7/8" casing at the depths shown on the chart.



Bit diameter was 7 7/8". A slurry volume of 390 barrels of 16.6 ppg Class H cement plus sand was circulated to a depth of 1,200' in the annulus. The three axes of the chart are sensor temperatures in °F, geothermal temperatures at each sensor in °F, sensor pressures in 1000's of psi, and time in minutes. Each line in the chart represents temperatures and pressures detected by a sensor whose number is at the end of that line. Corresponding geothermal temperatures are shown as black dots. Cement mixing and pumping began at the left axis at a time of 750 minutes. Displacement ended at 860 minutes. Prior pumping had lowered the temperatures at each sensor to the values shown on the left axis. The lines show that heating at the sensors was gradual until the time of 1010 minutes (4.3 hours). For example at sensor #1, located near the bottom of the hole where the geothermal temperature is 218°F, the gradual increase raised the temperature from 155°F to 190°F in 4.3 hrs. Thereafter, heat of hydration of the setting cement combined with conduction from the formations to quickly raise the temperature of the cement above the geothermal temperature at that sensor.

Their study shows that large bodies of cold fluid heats slowly in the annulus between formation and casing and that pressures within set cement drop substantially below the pressures developed by the unset cement slurry.



SENSOR NO.	1	2	3	4	5	6
DEPTH (FT) (RKB)	8754	6909	5488	4787	4632	3636

PORTLAND CEMENT — CLASS A OR B WITH BENTONITE

SLURRY PROPERTIES

Per Cent Bentonite	Maximum Water Requirements Gal./Sk.	Cu. Ft./Sk.	Slurry Weight Lbs./Gal.	Lbs./Cu. Ft.	Slurry Volume Cu. Ft./Sk.
0	5.2	0.70	15.6	117	1.18
2	6.5	0.87	14.7	110	1.36
4	7.8	1.04	14.1	105	1.55
6	9.1	1.22	13.5	101	1.73
8*	10.4	1.39	13.1	98	1.92

THICKENING TIME — HOURS:MINUTES

(Pressure-Temperature Thickening-Time Test)

Per Cent Bentonite	API CASING TESTS 4,000'	6,000'	8,000'	API SQUEEZE TESTS 2,000'	4,000'	6,000'
0	3:00+	2:25	1:40	2:14	1:32	1:01
2	2:25	1:48	1:34	2:25	1:29	0:56
4	2:34	1:57	1:32	2:26	1:18	0:58
6	2:35	1:45	1:22	2:16	1:26	0:56
8*	2:44	1:50	1:24	2:31	1:28	0:58

COMPRESSIVE STRENGTHS — PSI

Atmospheric Pressure

Per Cent Bentonite	60°F.	80°F.	100°F.	120°F.
		12 HOURS		
0	80	580	1035	1905
2	55	455	635	1280
4	20	220	375	780
6	15	85	245	500
8*	15	50	155	310
		24 HOURS		
0	615	1905	2610	3595
2	365	1090	1520	2040
4	225	750	1015	1380
6	85	360	730	925
8*	60	265	510	610
		72 HOURS		
0	2050	4125	6150	6650
2	1185	2840	3350	4110
4	960	1775	2430	2800
6	615	1170	1610	1710
8*	425	720	1045	1215

*For bentonite concentrations greater than 8 per cent, see modified cement data.

CALCIUM CHLORIDE ACCELERATOR

CLASS A CEMENT

SLURRY PROPERTIES

Per Cent Bentonite	Water Requirement Gals./Sk.	Slurry Weight Lbs./Gal.	Slurry Volume Cu. Ft./Sk.
0	5.2	15.6	1.18
2	6.5	14.7	1.36
4	7.8	14.1	1.55

THICKENING TIME — HOURS:MINUTES

(Pressure-Temperature Thickening-Time Test)

API CASING TESTS

Per Cent Calcium Chloride	0 PER CENT BENTONITE 2,000'	4,000'	2 PER CENT BENTONITE 2,000'	4,000'	4 PER CENT BENTONITE 2,000'	4,000'
0	3:36	2:25	3:20	2:25	3:45	2:34
2	1:30	1:04	2:00	1:30	2:41	2:03
4	0:47	0:41	0:56	1:10	1:52	2:00

CLASS A CEMENT WITH BENTONITE AND SALT

SLURRY PROPERTIES

Per Cent Bentonite	Water Gals./Sk.	Cu. Ft./Sk.	Per Cent Salt by Wt. of Water	Weight of Dry Salt, Lbs./Sk. Cement	Slurry Weight Lbs./Gal.	Lbs./Cu. Ft.	Slurry Volume Cu. Ft./Sk.
2	6.5	0.87	0	0	14.7	110	1.36
			5	2.17	14.8	111	1.37
			10	5.42	14.9	111	1.39
			18	9.75	15.1	113	1.41
			Sat. (140°F.)	20.15	15.3	114	1.48
4	7.8	1.04	0	0	14.1	105	1.55
			5	3.25	14.2	106	1.56
			10	6.50	14.3	107	1.58
			18	11.70	14.5	108	1.61
			Sat.	24.18	14.8	111	1.69
6	9.1	1.22	0	0	13.5	101	1.73
			5	3.79	13.7	102	1.75
			10	7.58	13.8	103	1.77
			18	13.64	14.0	105	1.80
			Sat.	28.21	14.3	107	1.90
8	10.4	1.39	0	0	13.1	98	1.92
			5	4.33	13.2	99	1.94
			10	8.66	13.4	100	1.96
			18	15.60	13.6	102	2.00
			Sat.	32.24	14.0	105	2.11
12	13	1.74	0	0	12.5	93	2.29
			5	5.41	12.6	95	2.31
			10	10.83	12.8	96	2.34
			15	16.24	12.9	97	2.38
			Sat.	40.30	13.4	100	2.53

GILSONITE CEMENT

Gilsonite is a special type of solid hydrocarbon (asphaltite) used with cements to reduce slurry weight and minimize lost circulation. It is a black granular material having a specific gravity of 1.07 and a relatively low water requirement (2 Gals. per Cu. Ft.). It has a bulk density of 50 Lbs. per Cu. Ft., and is supplied in 100 pound paper bags.

The following recommendations should be noted:

1. Bottom plugs should not be used ahead of Gilsonite cement.
2. A viscous slurry should precede the Gilsonite cement to help prevent separation of the light material.
3. The use of centralizers and scratchers should be minimal with this material.
4. Twenty-five pounds of Gilsonite per sack of cement should be the normally recommended amount although higher concentrations may be used for extremely severe lost circulation.

SLURRY PROPERTIES

Portland Cement

Class A Cement

Gilsonite Lbs./Sk.	Per Cent Bentonite	Water Per Sk. Gals.	Cu. Ft.	Slurry Weight Lbs./Gal.	Lbs./Cu. Ft	Slurry Volume Cu. Ft./Sk.
0	0	5.20	0.70	15.6	117	1.18
	4	7.80	1.04	14.1	105	1.55
12.5	0	5.60	0.75	14.4	108	1.42
	4	8.30	1.11	13.3	99	1.80
25	0	6.00	0.80	13.6	102	1.66
	4	8.80	1.18	12.7	95	2.05
50	0	7.00	0.94	12.5	94	2.17
	4	9.80	1.31	11.9	89	2.56

CLASS A CEMENT WITH SALT

Salt Cement slurries are recommended for use in cementing through salt sections, shale formations, bentonitic sands as well as other types of formations that are susceptible to fresh water contamination. When salt is preferentially dry blended with cement, ¼ lb. of D-AIR 1™ (Part No. 70.15764) per sack is recommended to reduce foaming of the slurry. Where blending facilities are not available and the salt must be placed in the mixing water, 1 quart of NF-1 (Part No. 70.15492) per 1,000 gallons of water will aid in reducing foaming during mixing.

SLURRY PROPERTIES

Water Requirements Gals./Sk.	Cu. Ft./Sk.	Per Cent Salt by Weight of Water	Weight of Dry Salt Lbs./Sk. Cement	Slurry Weight Lbs./Gal.	Lbs./Cu. Ft.	Slurry Volume Cu. Ft./Sk.
5.2	0.70	0	0	15.6	117	1.18
		5	2.17	15.7	117	1.19
		10	4.33	15.8	118	1.20
		15	6.50	15.9	119	1.21
		20	8.66	16.0	120	1.22
		Sat.(140°F.)	16.12	16.1	120	1.27

PHYSICAL PROPERTIES OF CEMENTING MATERIALS AND ADMIXTURES

MATERIAL	Bulk Weight Kg/m³	Specific Gravity	Weight 13.6 Liters*	Absolute Volume L/Kg
API Cements	1506	3.14	42.6	0.3185
Attapulgite	641	2.89	39.3	0.3460
Ciment Fondu	1442	3.23	44.0	0.3096
Lumnite Cement	1442	3.20	43.5	0.3125
Trinity Lite-Wate	1201	2.80	34.0‡	0.3571
Barite	2162	4.23	57.6	0.2364
Bentonite (gel)	961	2.65	36.1	0.3776
Calcium Carbonate Powder	357	2.71	36.7	0.3690
Calcium Chloride, flake**	903	1.96	26.7	—
Calcium Chloride powder**	809	1.96	26.7	—
Cal-Seal	1201	2.70	36.7	0.3704
CFR-1**	646	1.63	22.2	—
CFR-2**	689	1.30	17.7	—
CFR-2 (Liquid)	—	1.18	—	—
D-AIR 1**	404	1.35	18.4	—
D-AIR 2**	—	1.005	—	—
Diacel A**	966	2.62	35.7	—
Diacel D	268	2.10	28.6	0.4762
Diacel LWL**	465	1.36	18.5	—
Diesel Oil No. 1 (liquid)	819	0.82	11.2	1.2195
Diesel Oil No. 2 (liquid)	849	0.85	11.6	1.1765
Econolite**	1201	2.40	32.6	—
Econolite (Liquid)	—	1.40	—	—
Flocele	240	1.42	—	—
GAS-CHEK® (Solid)**	1153	2.70	36.7	—
GAS-CHEK® (Liquid)	1121	1.07	14.6	—
Gilsonite	801	1.07	14.5	0.9346
HALAD®-9**	596	1.22	16.6	—
HALAD®-14**	633	1.31	17.8	—
HALAD®-22A**	376	1.32	16.3	—
Hi-Dense® No. 3	2995	5.02	68.3	0.1992
HR-4**	561	1.56	21.2	—
HR-5**	615	1.41	18.6	—
HR-6L (Liquid)	—	1.21	—	—
HR-7**	481	1.30	17.7	—
HR-12**	372	1.22	16.6	—
HR-15**	711	1.57	21.4	—
HR-L (liquid)**	—	1.23	—	—
Hydrated Lime	497	2.20	29.9	0.4545
Hydromite	1089	2.15	29.3	0.4651

MATERIAL	Bulk Weight Lbs./Cu. Ft.	Specific Gravity	Weight 3.6* Absolute Gallons	Absolute Volume Gals./Lb.	Absolute Volume Cu. Ft./Lb.
Iron Carbonate	114.5	3.70	110.9	0.0324	0.0043
KCl (in solution @ 68°F. with fresh water)					
3%	—	1.019	—	0.0443	0.0059
5%	—	1.031	—	0.0450	0.0060
LA-2 Latex (liquid)	68.5	1.10	33	0.1087	0.0145
NF-1 (liquid)**	61.1	0.98	29.4	—	—
Perlite Regular	8***	2.20	66.0	0.0546	0.0073
Perlite Six	38†	—	—	0.0499	0.0067
Pozmix® A	74	2.46	74	0.0487	0.0065
Sea Water	—	1.025	—	—	—
Salt (dry NaCl)	71	2.17	65.1	0.0553	0.0074
Salt (in solution @ 68°F. with fresh water)					
6%—0.5 lb./gal.	—	—	—	0.0372	0.0050
12%—1.0 lb./gal.	—	—	—	0.0391	0.0052
18%—1.5 lb./gal.	—	—	—	0.0405	0.0054
24%—2.0 lb./gal.	—	—	—	0.0417	0.0056
Salt (in solution @ 140°F. with fresh water)					
Sat.—3.1 lb./gal.	—	—	—	0.0458	0.0061
Sand (Ottawa)	100	2.63	78.9	0.0456	0.0061
Silica Flour (SSA-1)	70	2.63	78.9	0.0456	0.0061
Spacer-Sperse™	40.0	1.32	39.6	—	—
Spacer Mix (Liquid)	—	.932	—	—	—
SPHERELITE††	25	.685	20.5	—	—
Coarse Silica (SSA-2)	100	2.63	78.9	0.0456	0.0061
Tuf Additive No. 1	—	1.23	36.9	0.0976	0.0130
Tuf Additive No. 2	15.38	.88	26.4	0.1364	0.0182
Tuf-Plug	48	1.28	38.4	0.0938	0.0125
THIX-SET A**	68.5	1.97	59.1	—	—
THIX-SET B**	36.5	1.37	41.1	—	—
Water	62.4	1.00	30.0	0.1200	0.0160

* Equivalent to one 94 lb. sack of cement in volume.

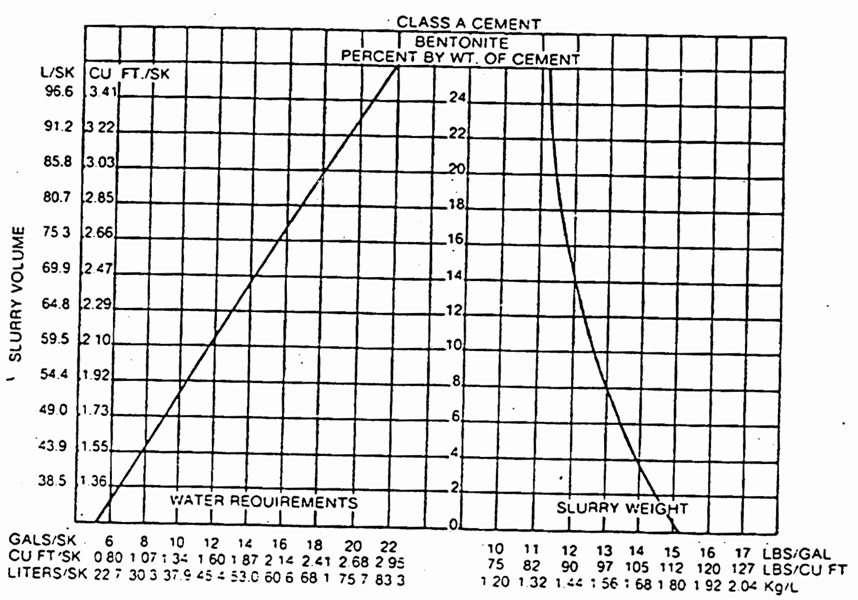
** When less than 5% is used these chemicals may be omitted from calculations without significant error.

*** For 8 lbs. of Perlite Regular use a volume of 1.43 gallons at 0 pressure.

† For 38 pounds of Perlite 6 use a volume of 2.89 gallons at 0 pressure.

†† Varies with pressure.

‡ 75 lbs. = 3.22 absolute gallons.



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CHAPTER XI

DRILL BIT SELECTION

INTRODUCTION

One of the more confusing aspects of oilwell drilling to the young and old a like is the numerous types of drill bits. The largest manufacturer of drill bits makes 34 types and 33 sizes (diameters) and many more on request.

The International Association of Drilling Contractors (IADC) organization has a system of assigning a designation code to each bit type. The meaning of the codes are shown in the table. The primary short coming of the API coding system is caused by the fact that manufacturers' bits which on the surface may appear similar are very different in substance.

The IADC category of bits and types (called bit codes) are

1. roller cone

IADC BIT CLASSIFICATION FORM				FEATURES								
	SERIES	FORMATIONS	TYPES	Standard Roller Bearing 1	Roller Bearing Air Cooled 2	Roller Bearing Gauge Protected 3	Sealed Roller Bearing 4	Sealed Roller Bearing Gauge Protection 5	Sealed Friction Bearing 6	Sealed Friction Bearing Gauge Protection 7	Directional 8	Other 9
Milled -Tooth Bits	1	Soft formations with low compressive strength and high drillability	1 2 3 4									
Milled -Tooth Bits	2	Medium to med-hard formations with high compressive strength	1 2 3 4									
Milled -Tooth Bits	3	Hard semi-abrasive and abrasive formations	1 2 3 4									
Insert Bits	4	Soft formations with low compressive strength and high drillability	1 2 3 4									
Insert Bits	5	Soft formations with low compressive strength	1 2 3 4									
Insert Bits	6	Medium hard formations with high compressive strength	1 2 3 4									
Insert Bits	7	Hard semi-abrasive and abrasive formations	1 2 3 4									
Insert Bits	8	Extremely hard and abrasive formations	1 2 3 4									

a. identified by four codes; (1) series, (2) type, (3) feature #1, and (4) feature #2
b. the following chart present the IADC codes
c. an example code is 5 2 2 R.

"5" is the code for "insert bits"

"2" is the code for "soft to medium formation"

"2" is the code for "roller bearing air cooled"

"R" is the code for "reinforced welds"

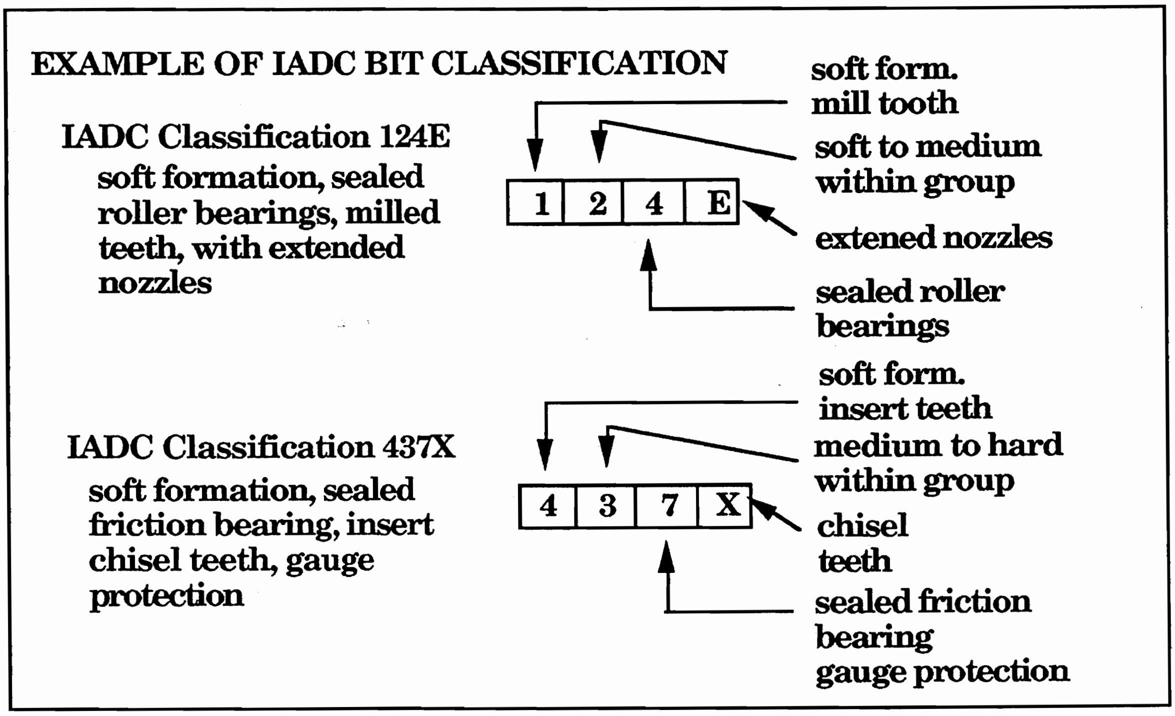
2. fixed cutters
   a. identified by four codes; (1) cutter type and body material, (2) cross-section profile, (3) hydraulic design, and (4) cutter size and density
   b. the following charts presented the IADC codes
   c. an example code is M 5 8 4

"M" = code for "polycrystalline diamond cutters/matrix body"

"5" = code for "medium taper/medium cone"

"8" = code for "fixed ports/open face"

"4" = code for "medium cutter size/light cutter density"



ADDITIONAL DESIGN FEATURES

CODE	FEATURE	CODE	FEATURE
A	Air application	N	
B		O	
C	Center jet	P	
D	Deviation control	O	
E	Extended jets	R	Reinforced welds
F		S	Standard steel teeth
G	Extra gauge/body protection	T	
H		U	
I		V	
J	Jet deflection	W	
K		X	Chisel insert
L		Y	Conical insert
M		Z	Other insert

Fixed Cutter Codes

First code designations are

1. D - natural diamond/matrix body
2. M - polycrystalline diamond cutters/matrix body
3. S - polycrystalline diamond cutters/steel body
4. T - thermally stable polycrystalline diamond cutters/matrix body
5. O - other

Second code designations are

1. long taper/deep cone
2. long taper/medium cone
3. long taper/shallow or no cone (parabolic)
4. medium taper/deep cone
5. medium taper/medium cone
6. medium taper/shallow or no cone(rounded)
7. short taper/deep cone (inverted)
8. short taper/medium cone
9. short taper/shallow or no cone (flat)

Third code designations are

1. bladed/changeable jets
2. bladed/fixed ports
3. bladed/open throat
4. ribbed/changeable jets
5. ribbed/fixed ports
6. ribbed/open throat

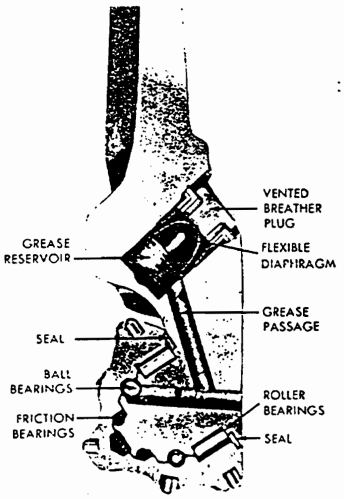
7. open face/changeable jets
8. open face/fixed ports
9. open face/open throat
10. alternative codes
    a. radial flow
    b. cross flow
    c. other

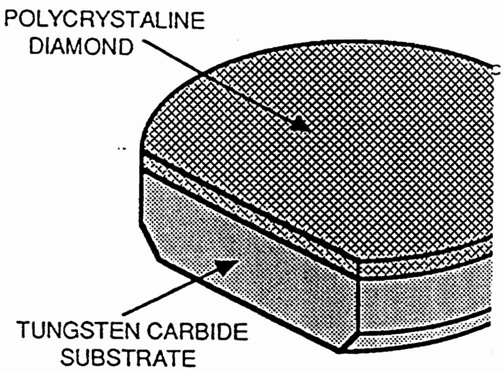
Fourth code designations are
1. cutter size; large, medium, and small
2. cutter density; light, medium, heavy

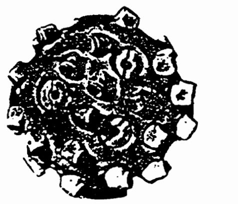
Drill Bit Characteristics

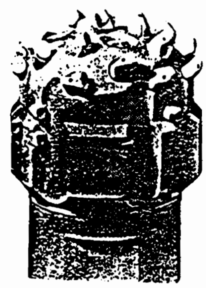
Roller cone drill bit characteristics of importance in regard to selection are:
1. Teeth length and size
2. Number of teeth (indexing)
3. Cone offset
4. Cone bearing type
5. Heel row teeth configuration
6. Gauge configuration
7. Journal shaft angle (how flat or round the bottom hole pattern)

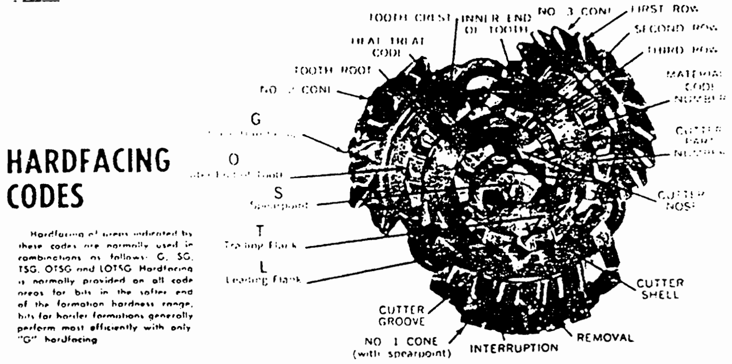
The teeth length and size and cone offset affect the drill bits penetration rate. The next three items on the list have a large affect on the bit life. The last item affects hole straightness.

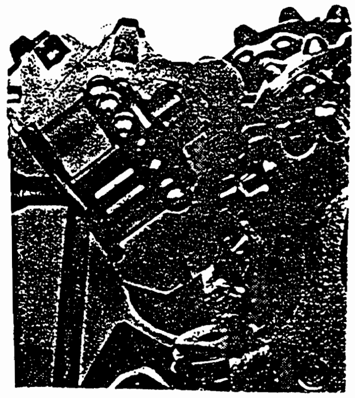


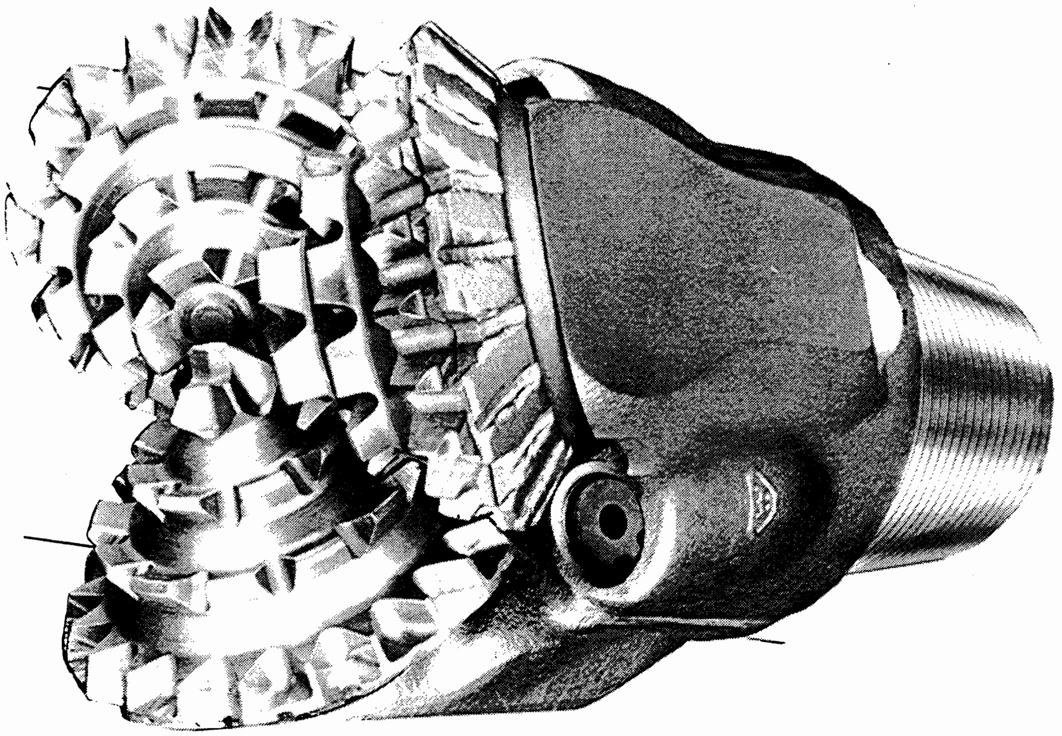


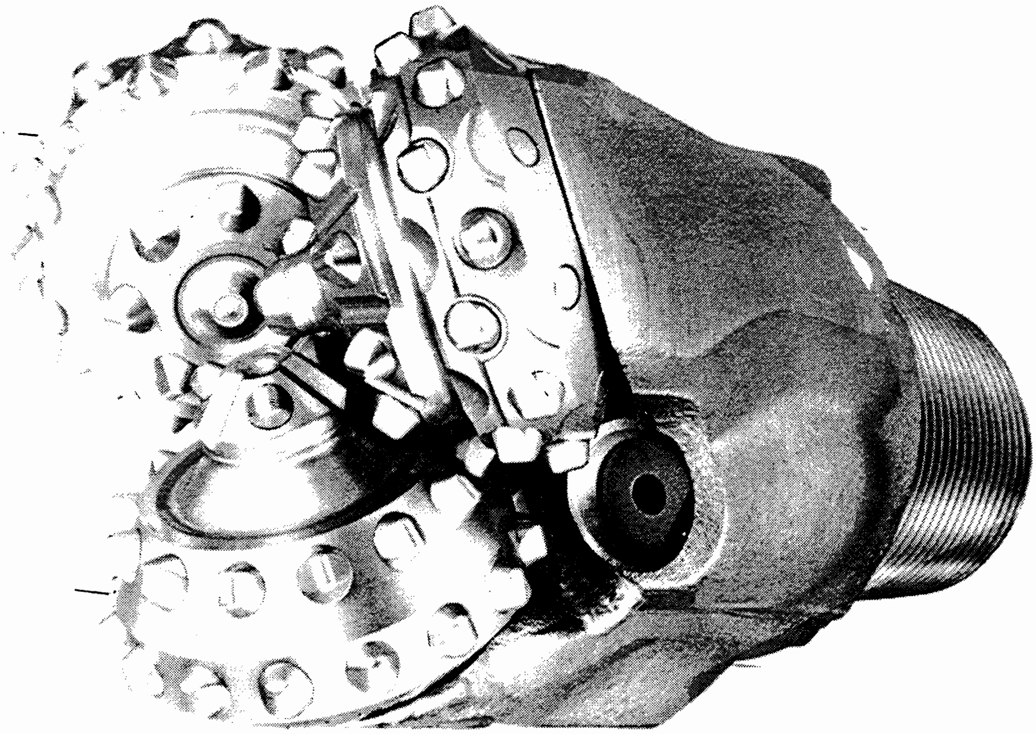




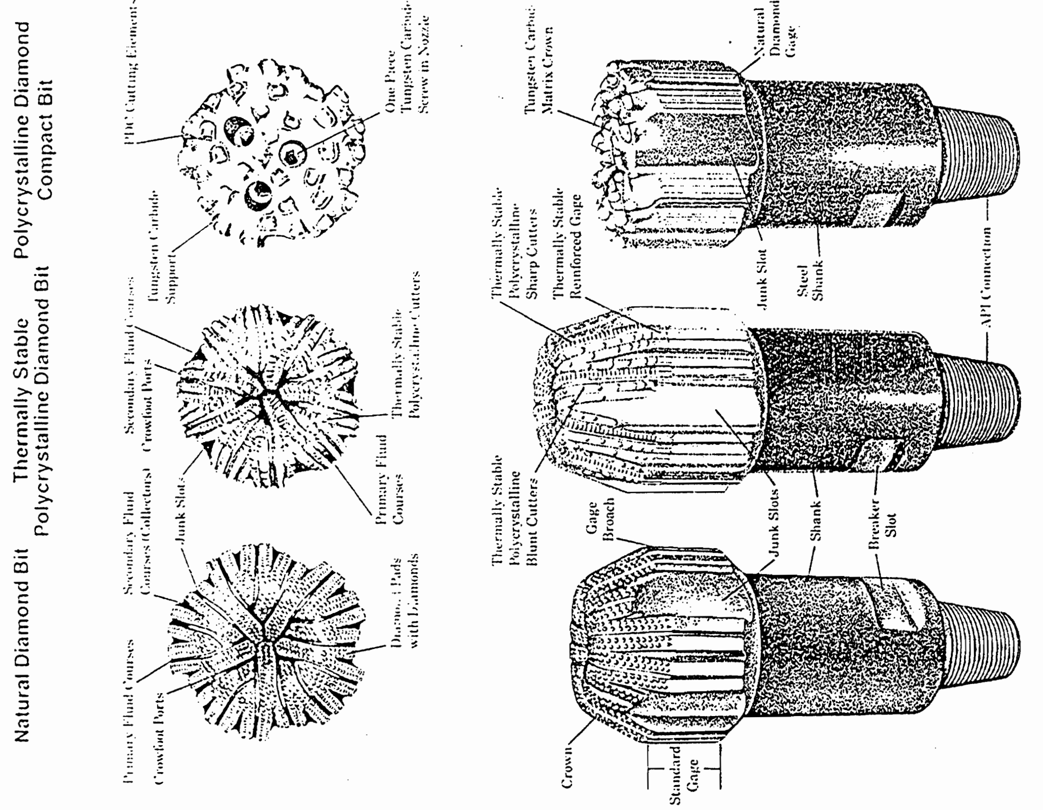


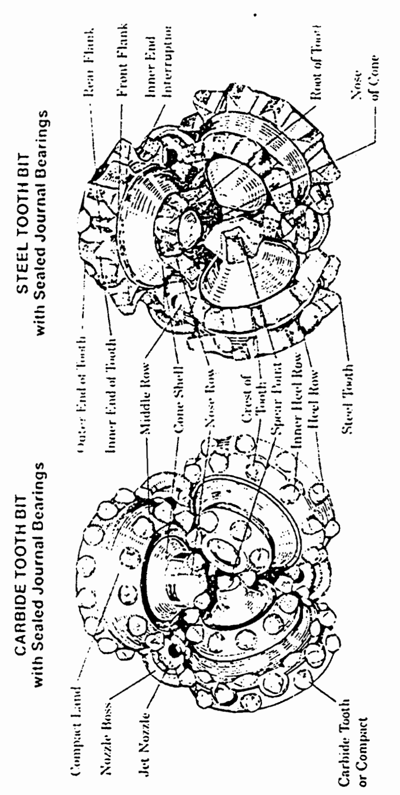


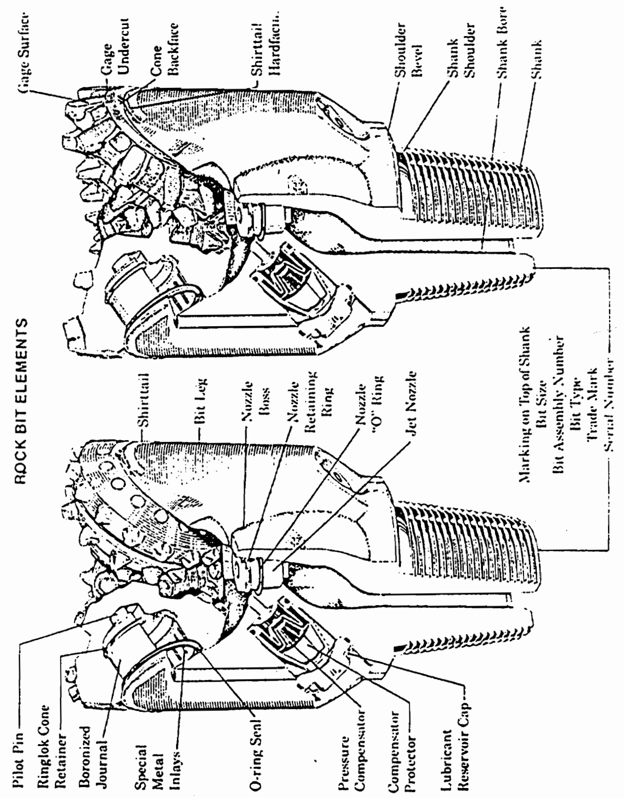




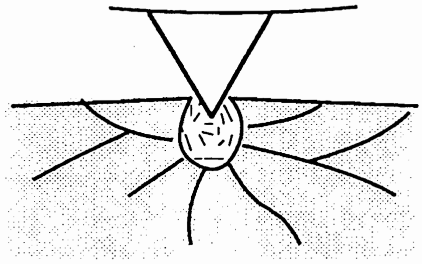
ROCK BIT TERMINOLOGY



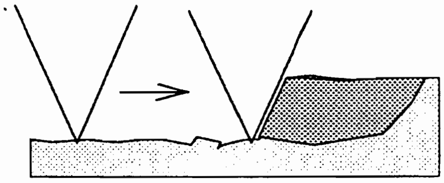




Rock Failure Models



Two rock failure models are recognized. These are the tension and shear models and are illustrated in the following sketches. The tension model may be demonstrated by hitting a rock with a hammer. The events are (1) the tooth hits the rock and makes a powdered sphere below it. The tooth enters the powder and increases its pressure which pushes the sides of the rock apart causing it to crack in tension.



The shear model (also called the 'slip-stick') has the tooth scraping. Chips of rock are formed along shear planes. It has been argued that the shear type action is more efficient than the tension model because the chips are larger; ie, very little powder size chips are made.

DRILL BIT SELECTION CRITERIA

A common criterion for drill bit selection is **'BALANCED WEAR'**. Balanced wear is the method of bit selection in which the teeth, bearings, and gauge, wear-out at exactly the same time. It is easy, but does not necessarily give minimum cost.

Some of the problems in selecting drill bits with the **'MINIMUM COST'** method are:

1. bit selection can only be based on drilling experience;
2. lowest cost is usually interpreted as lowest cost per foot which is not necessarily true;
3. cost comparisons between bits must be made with each bit operating at its optimal weight and speed and optimal life;
4. all other factors must be reasonably similar.

A popular equation for cost comparisons is the total cost per foot equation. The costs used in the equation are dictated by drilling rig operations and fall into four categories: bit, trip time, rotating time, and down time accountable to the drill bit. Rotation time is the time consumed while the bit is rotating on bottom with full weight on the formation. Down time is the time consumed other than rotating or trip times. For example, suppose a bit has such a short life that adequate mud treatment can not be accomplished while drilling and that the mud must be treated after the bit is worn out and before a trip is made. Then the time spent treating the mud must be charged to down time. Trip time is the time consumed while running and pulling the drill string for the purpose of replacing the bit. Time for connections is down time.

The total cost of operating and replacing a bit is the sum of the costs: bit, trip, rotation, and down time. If the hourly cost rate of a rig is comparable for all bits being compared; then, the average operating cost for a particular bit per foot is

$$TCF = \frac{\text{Bit + Opr * Trip T + Opr * Rot T + Opr * Down T}}{\text{Bit Footage}}$$

If the bit were operated to its optimal life at its optimum weight and speed, then the cost would be the lowest cost per foot for the bit. Optimal life occurs if the bit is terminated by bearing failure, total tooth wear, or the reduction of its drilling rate to an uneconomical one. Formation changes often mask the latter optimal life and care must be exercised.

The bit performance analyzed in the following example illustrates these ideas. The bit's optimal life corresponds with the minimum value of cost per foot on the plot. Continued drilling will result in increased cost although the teeth and bearings are not totally worn. Contrary to popular beliefs, the simultaneous wear out of bit teeth and bearings is not optimal, nor are the maximizations of rotation time or bit footage optimal.

EXAMPLE

A long steel tooth bit with unsealed roller bearings and a tungsten carbide insert bit with sealed and lubricated bearings have produced the following performance data.

ITEM	STEEL TOOTH	INSERT TOOTH
Bit Cost	$277.88	$1521.32
Trip	6:18 hrs/min	6:32 hrs/min
Rotation Time	10:01 hrs/min	58:42 hrs/min
Down Time	None	None
Bit Footage	180 ft	880.5 ft
Rig Cost Rate	375 $/hr	Same
Cost Per Foot	$35.54/ft	$35.28/ft
Fraction Tooth Wear	1/2	Not Applicable
Fraction Bearing Wear	6/8	8/8
Average Drilling Rate	18 ft/hr	15 ft/hr
Bit Size	7-7/8 in.	7-7/8 in.
Bit Weight	30,000 lbs	40,000 lbs
Rotary Speed	85 rpm	60 rpm

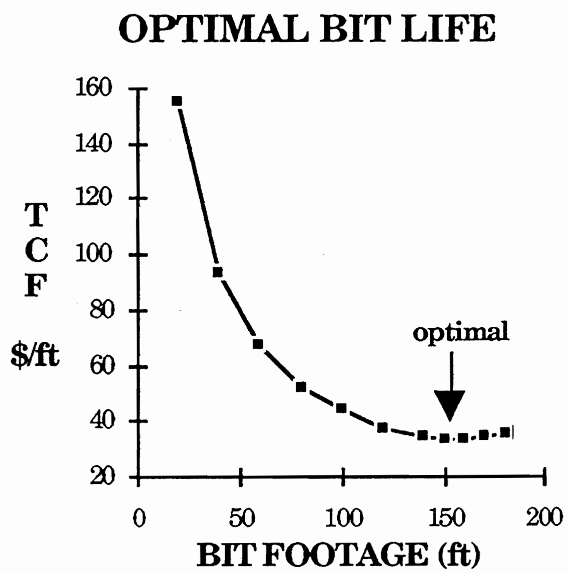
$$TCF_{steel} = \frac{277.88 + 375*6.3 + 375*10.02 + 375*0}{10.02*18} = \mathbf{\$35.47/ft}$$

$$TCF_{insert} = \frac{1521.32 + 375*6.5 + 375*58.7 + 375*0}{58.7 * 15} = \mathbf{\$29.50/ft}$$

Values in the table indicate that the insert tooth bit (1) was on bottom for a longer period, (2) drilled more footage, and (3) produced a lower cost per foot. The plot shows that had the steel tooth bit been pulled at its optimal life (6 hours and 34 minutes) it would have had a lower cost per foot: \$34.05/ft versus \$35.47/ft. The section of the wellbore (880.5 feet) would have cost \$369.81 less at optimal life.

Either bit may have drilled at a lower cost had they been operated at their optimal weights and speeds.

No direct method for determining the optimal weight and speed is available for the insert bit with journal bearings; however, analyses of several insert bit performances at various weights and speeds near the manufacturer's recommended values usually delineates the optimal values.

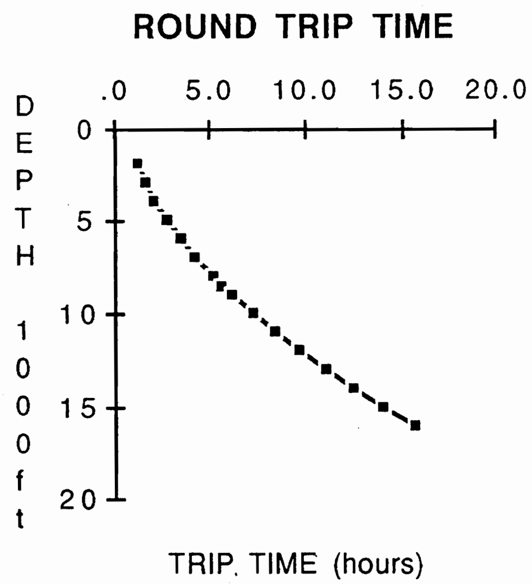


TRIP TIME

Trip times may be taken from the IADC tour sheet or if greater precision is desired they may be taken from the Geolograph unit or some other device.

The tour sheet will usually be sufficient and much faster.

A chart versus depth must be constructed for each rig for ultimate analyses. Such a chart is shown in the sketch.



EXAMPLE

A bit is run at a depth of 8,000 feet and pulled at a depth of 8,500 feet. How many hours is the trip time for this bit?

From the chart a round trips at 8,000 feet and 8,500 feet require 5.2 hours and 5.7 hours, respectively.

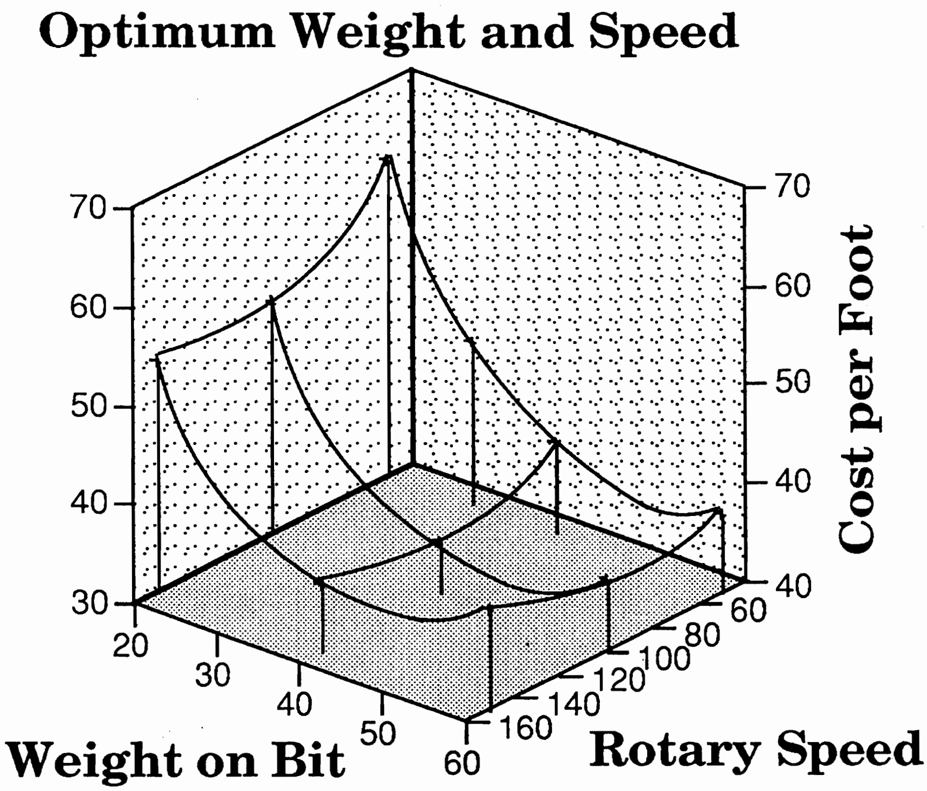
Sufficient accuracy for field operations will be achieved if the average of the two trip times is selected.

$$\text{Trip times} = \frac{5.2 + 5.7}{2} \qquad \mathbf{= 5.45\ hr}$$

OPTIMAL WEIGHT ON BIT AND ROTARY SPEEDS

The purpose of selecting bit weight and rotary speeds are to

1. produce the lowest drilling cost per foot
2. control the direction of the drill hole
3. find overpressured zones
4. satisfy equipment limitations (solids removal, bha, etc.)
5. develop a satisfactory cuttings description log



CONTOUR METHOD

The following example illustrates the contour method of selecting optimal weight on bit and optimal rotary speed. The method consists of constructing the above figure in a two dimensional presentation. This is accomplished by plotting drilling cost per foot contours with known drill bit data on a set of axes. The weight of bit is set as the abscissa and the rotary speed as the ordinate. Most likely no two bits will have the same drilling cost. Thus, one draws the contour lines to either include a value or exclude it within the contour. Those bits which are nearest the optimal weight on bit and rotary speed will be the most cost effective bits.

EXAMPLE

Six insert bits have been run and their performance and total cost per foot data are listed in the table.

An analysis of the figure showing bit weights and rotary speeds versus cost per foot for each of the bits indicates that the optimal weight and speed is 40,000 pounds and 50 rpm.

The resulting cost per foot is about **$35.00/foot**.

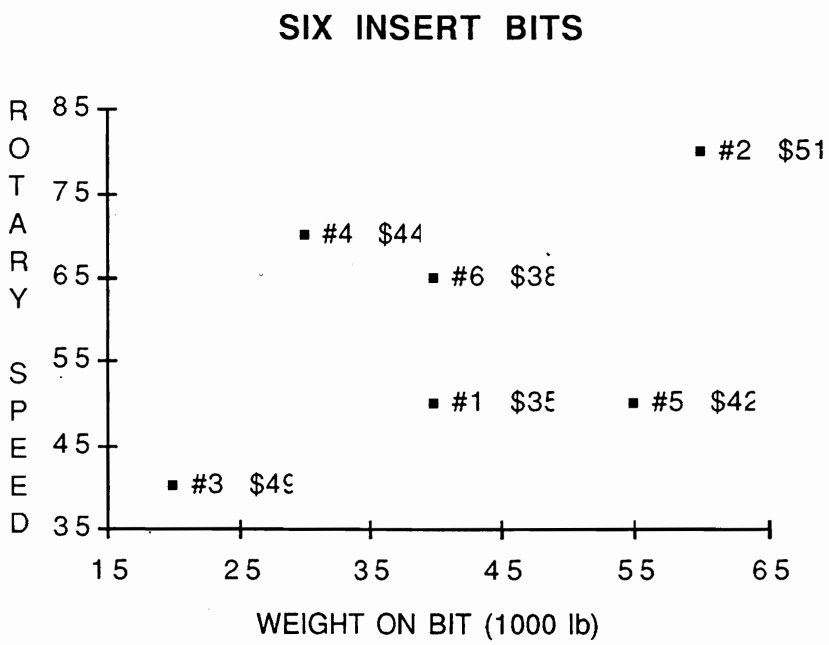
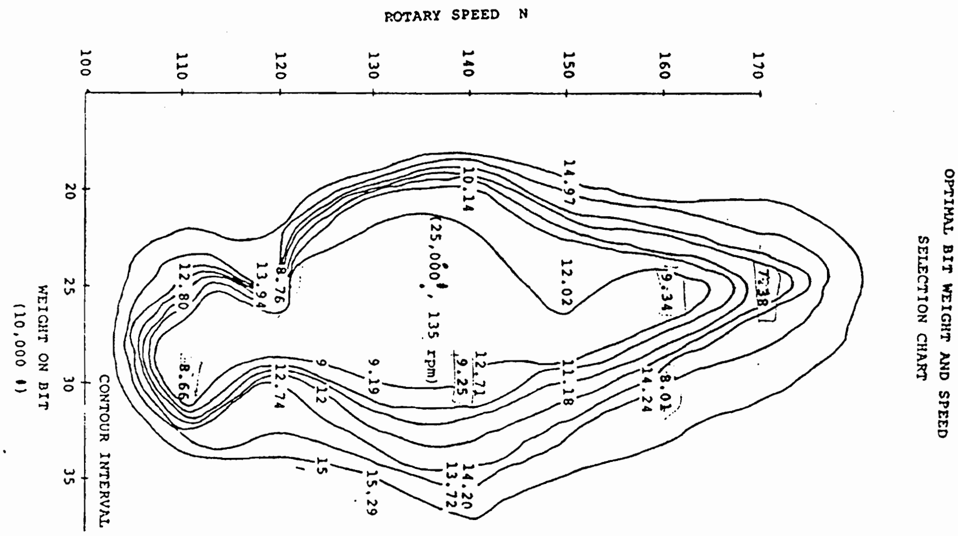


TABLE OF BIT PERFORMANCE

BIT #	ROT. TIME(HR:MIN)	FOOTAGE	TCF($/FT)
1	58:42	734	35.28
2	39:05	360	51.51
3	101:21	855	48.99
4	62:07	614	44.24
5	47:16	517	41.27
6	56:33	658	38.14

EXAMPLE

The following contour chart was extracted from a study of drill holes in the Gulf Coast.



ANALYTICAL METHOD

The same procedure could be used for determining optimal weights, and speeds for the steel tooth, roller bearing bit; however, an analytical method is available for these bits. The equations of Galle and Woods are the following.

DEFINITION OF TERMS

BS	=	Bit diameter; inch
W'	=	Actual weight on bit; 1000 pounds
W	=	Equivalent weight on a 7-7/8 inch bit; 1000 pounds
TT	=	Trip time; hrs
R	=	Cost rate of the drilling operation; $/hrs
T	=	Time; hrs
G	=	Bit Cost + Trip Cost; $
TC	=	Trip cost; R * TT; $
D	=	Fraction of tooth height worn away; 8ths
B	=	Fraction of bearing life consumed; 8ths

DT	=	Down time; hrs
RT	=	Rotation time; hrs
F	=	Bit footage; ft
N	=	Rotary speed; rpm

Subscripts

i	=	initial
f	=	final

Galle and Woods published the following relationships for the rate of tooth wear.

TOOTH WEAR

$$\frac{dD}{dT} = \frac{1}{A}\frac{i}{a\,m}$$

$$i = N + 4.348 * 10^{-5} N^3$$

$$a = 1 + 6D + .928\,D^2$$

$$\int a\,dD = D + 3\,D^2 + .309\,D^3$$

$$\int a^{1-p}\,dD = -.0067 + 1.15\,D + 0.871\,D^2$$ (least squares approximation; has less than 2% error).

$$m = 1359.1 - 714.19 \log_{10}(W)$$

$$W = W_{actual} * \frac{7.875}{BS}$$ and is called the equivalent weight

A = correlation coefficient for tooth wear and is called the formation abrasiveness factor

Others published the following relationships for the rate of bearing wear and penetration rate for bits.

BEARING WEAR

$$\frac{dB}{dT} = \frac{1}{S}\frac{N}{L} \qquad B = \frac{N}{S\,L} * RT$$

$$L = e^{8.9609 + 0.90393\,LnW - 0.36297\,(LnW)^2}$$ (least squares eqn; less than 3% error)

S = correlation coefficient called the drilling fluid factor

RATE OF PENETRATION

$$\frac{dF}{dT} = C\frac{W^{\alpha}N^{\beta}}{a^{p}}$$

C = correlation coefficient called the formation drillability factor

The values of α and β range between

$.6 \leq \alpha \leq 2.0$
$.4 \leq \beta \leq 1.0$

p = 1.0 if flat tooth wear (non-sharpening)
p = 0.5 if self sharpening

Integration of the tooth wear equation gives bit rotation life (time)

$$RT = A\frac{m}{i}\int a\,dD$$

Substitution into the bearing wear equation for differential time (dT) gives the bearing wear in terms of tooth dullness.

$$B = \frac{N}{SL}\frac{A\ m}{i}\int a\,dD$$

Bit footage is

$$F = \frac{dF}{dT}\,dT = C\,W^{\alpha}N^{\beta}A\frac{m}{i}\int a^{1-p}\,dD$$

The total cost per foot equation is

$$TCF = \frac{G + R\,A\frac{m}{i}\int a\,dD}{C\,W^{\alpha}\,N^{\beta}\,A\frac{m}{i}\int a^{1-p}dD}$$

EQUATIONS FOR THE PARAMETERS, G, A, C, & S

These parameters are usually computed with data gathered from other similar drill bits which were run in similar formations, at similar depths, with similar bottom hole assemblies, with similar hydraulics, with similar muds, and similar rigs.

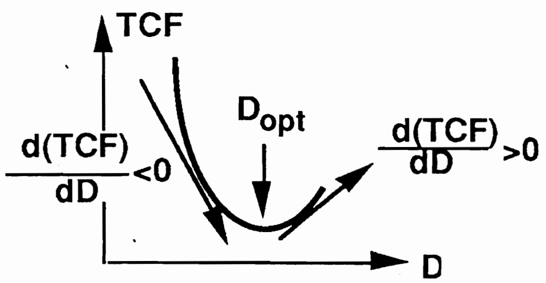
G = Drill bit cost \$ + Round trip cost \$

$$A = \frac{RT}{\frac{m}{i}\int a dD}$$

$$S = \frac{NT}{BL}$$

$$C = \frac{F}{W^{\alpha} N^{\beta} A \frac{m}{i} \int a^{1-p} dD}$$

MATHEMATICAL CONSTRAINTS



and the constraint is the bearing wear equation. The restrictions are that the life of the bit is ended when the teeth or bearings are totally worn or where the slope of the plot of cost per foot versus tooth wear for the operation of the bit becomes positive.

$$B \leq 1.0 \qquad D_f \leq 1.0 \qquad \frac{d(TCF)}{dD} \leq 0.0$$

OPTIMIZING EQUATIONS

If p = 1.0 the two equations become

$$B = \frac{N}{SL} A \frac{1359.1 - 714.19 \log_{10}(W)}{N + 4.348 * 10^{-5} N^3} [(D + 3D^2 + .309 D^3)_{i+1} - (D + 3D^2 + .309 D^3)_i]$$

$$TCF = \frac{G + R A \frac{1359.1 - 714.19 \log_{10}(W)}{N + 4.348 * 10^{-5} N^3} [(D + 3D^2 + .309 D^3)_{i+1} - (D + 3D^2 + .309 D^3)_i]}{c W^a N^b A \frac{1359.1 - 714.19 \log_{10}(W)}{N + 4.348 * 10^{-5} N^3} [D_{i+1} - D_i]}$$

If p = 0.5 then

$$TCF = \frac{G + R A \frac{1359.1 - 714.19 \log_{10}(W)}{N + 4.348*10^{-5} N^3} [(D + 3D^2 + .309 D^3)_{i+1} - (D + 3D^2 + .309 D^3)_i]}{c W^a N^b A \frac{1359.1 - 714.19 \log_{10}(W)}{N + 4.348 * 10^{-5} N^3} [f_{i+1} - f_i]}$$

$$f_i = \frac{D_i + 3.233}{2.076}\sqrt{(D_i + 3.233)^2 - 9.373} - 4.515 * Ln_e[D_i + 3.233 + \sqrt{(D_i + 3.233)^2 - 9.373}\,]$$

OPTIMIZING STEPS

The following steps may be followed to arrive at a set of optimal weights and speeds for dulling the drill bit's teeth. This is called a minimum path for total cost per foot.

1. Choose practical ranges on weight of bit and of rotary speed for a 7.875 inches drill bit.

2. Set $D_i = 0$ and $D_{i+1} = D + \Delta D$ and solve the equations for TCF and B. A satisfactory value for ΔD is 0.1.

3. Select the W and N for dulling this fraction of the drill bit such that the following two constraints are met.

   a. $TCF_i = TCF_{minimum}$

4. Terminate incrementing D when either one of the following occurs.

   a. $D = 1.0$
   b. $B = 1.0$
   c. $TCF_{i+1} \geq TCF_i$

EXAMPLE

Calculate an optimal bit weight and rotary speed for the steel tooth bit in the first problem if from a previous bit the following data were noted.

W = 40,000 lb N = 80 rpm p = 1.0 $\alpha = 1.0$

$\beta = 0.7$ RT = 10 hr F = 180 ft

and all other data remain as in the first problem.

$$A = \frac{10 * (80 + 4.348 * 10^{-5} * 80^3)}{(1359 - 714 \log_{10} 40)(.5 + 3 * .5 + .309 * .5^3)} \qquad \mathbf{= 3.693}$$

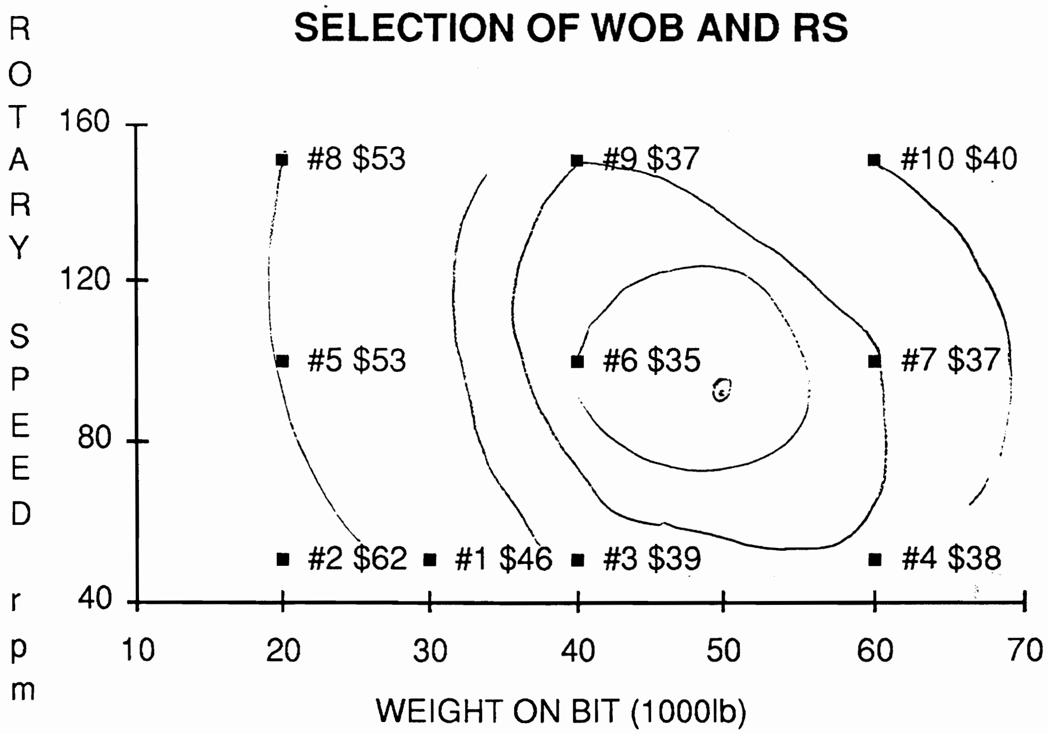
$$C = \frac{180 * (80 + 4.348 * 10^{-5} * 80^3) * \int a^{1-P} dD}{40^1 * 80^{.7} * 3.693 * (1359 - 714\, LOG_{10}(40))} \qquad \mathbf{= 21.4}$$

$$S = \frac{80 * 10}{.75 * \frac{8755}{10^{.01782*40}}} = .629$$

$$G = 277.8 + 6.3 * 375 = 2640.4$$

TABLE OF WEIGHT AND SPEED

W	N	D	B	TCF
30	50	.3	.364	46.23
20	50	.3	.342	61.73
40	50	.4	.608	39.09
60	50	.5	.815	37.70
20	100	.4	.411	53.12
40	100	.6	.906	34.81
60	100	.6	.853	37.02
20	150	.6	.578	53.17
40	150	.7	.857	36.52
60	150	.7	.807	40.05



The weight and speed versus cost per foot figure shows that the example steel tooth drill bit should have been operated at 50,000 pounds and 90 rpm rather than 45,000 pounds and 80 rpm in order to save about $.90/ft or an amount of $660.60 over the 734 foot interval.

The steel tooth bit would have been $ 2.13/ft less costly than the insert tooth bit. A savings of $1563.42 over the 734 foot interval would have been realized.

OPTIMAL WOB AND ROTARY SPEED CHARTS

Galle and Woods published charts for the selection of optimal weight on bit and rotary speed. The following steps outline and illustrate their procedure.

1. Get a record of the bit's performance and wear data.
2. Record bit size (d) and type 8 - 3/4" OSC
3. Record hourly operational costs (C_{opr}) $ 50.00/hr
4. Record bit cost (C_b) $ 200.00
5. Record the bit depth in 10,000 feet and depth out 10,180 feet.
6. Record the round trip time (t_t) for the bit 6 hrs
7. Record the bit footage (F) 180 feet
8. Record the rotating time (T) of the bit 12 hours
9. Record the weight on bit (WOB) 35 1000's lb
10. Record the rotary speed (N) 100 RPM
11. Record the tooth dullness (D) 6/8
12. Record the bearing wear (B) 4/8
13. Calculate the equivalent weight on bit (W).

    $$W = \frac{7.875}{\text{step } 2} * \text{step } 9 = \frac{7.875}{8.75} * 35 \qquad = \mathbf{31.5}$$

14. Using the result from step 13, obtain the weight on bit parameter (m) from Table Bit-1. The bit parameter (m) can also be calculated from the equation if you so choose.

    M **= .404**

15. Using the result from step 13, obtain the bearing life parameter (L) from Table Bit-1.

    L **= 2,316**

16. Using the result from step 10, obtain the rotary speed parameter (i) from Table Bit-1. The rotary speed parameter (i) can also be calculated from the equation if you so choose.

i = 143

17. Using the result from step 10, obtain the rotary speed parameter (r) from Table Bit-1.

r = 31.8

18. Using the result from step 11, obtain the tooth dullness parameter (U) from Table Bit-1.

U = 1,834

19. Using the result from step 11, obtain the tooth dullness parameter (V) from Table Bit-1.

V = 967

20. Calculate the formation abrasiveness coefficient (A).

$$A = \frac{T * i}{m * U} = \frac{\text{step 8} * \text{step 16}}{\text{step 14} * \text{step 18}} = \frac{12 * 143}{.404 * 1834} = 2.32$$

21. Calculate the formation drillability coefficient (C).

$$C = \frac{F * i}{A * r * W * m * V}$$

$$= \frac{\text{step 7} * \text{step 16}}{\text{step 20} * \text{step 17} * \text{step 13} * \text{step 14} * \text{step 19}}$$

$$= \frac{180 * 143}{2.32 * 31.8 * 31.5 * .404 * 967} = 0.0284$$

22. Calculate the drilling fluid coefficient (S).

$$S = \frac{T * N}{B * L} = \frac{\text{step 8} * \text{step 10}}{\text{step 12} * \text{step 15}} = \frac{12 * 100}{4/8 * 2316} = 1.04$$

23. Calculate the cost parameter (G).

$$G = \frac{C_b}{C_{opr}} + t_t = \frac{\text{step 4}}{\text{step 3}} + \text{step 6} = \frac{200}{50} + 6 = 10$$

24. Calculate the normalized chart coefficient (A_n).

$$A_n = \frac{G}{A} = \frac{\text{step 23}}{\text{step 20}} = \frac{10}{2.32} = 4.31$$

25. Calculate the normalized chart coefficient (S_n).

$$S_n = \frac{S}{A} = \frac{\text{step 22}}{\text{step 20}} = \frac{1.04}{2.32} \qquad \mathbf{= 0.45}$$

26. Go to the "tornado" chart.

27. Draw a small "dot" with a pencil where the values on the two axes intersect. Note the value on the abscissa, S_n, is 0.45 and the value on the ordinate, A_n, is 4.31. The "dot" on the "tornado" chart is shown for the example in these steps. The location of the "dot" will change for another example.

28. If, as in this example, the "dot" is drawn in the "Teeth Limit Bit Life" region, then and only then a horizontal line is drawn to the left until it intersects the first vertically curved line. A new "dot" is drawn at that point and it is the "dot" to be used. The above has been illustrated in the "tornado" chart.

29. Record the new value of S_n which is the value on the abscissa directly below the new "dot" and call the new value S_n'__**.425**__. If the dot is drawn in the envelope, then it is the "dot" to be used. In this instance, S_n' is changed.

30. With the "dot" drawn in step 28, record the cost per foot parameter (K)__**.00267**__.

31. With the "dot" drawn in step 28, record the optimal equivalent weight on bit (W')__**49.5 1,000's lb**__.

32. With the "dot" drawn in step 28, record the optimal rotary speed (N')__**96**__RPM.

33. With the "dot" drawn in step 28, record Df'__**8/8**__.

34. Compute optimal weight on bit (WOB') __**55**__1'000's lbs $(49.5 * \frac{8.75}{7.875})$.

35. Using the result from step 31, obtain L' from Table Bit-1.

L' **= 1,084**

36. Compute the expected rotation time (t')__**11.13**__hours.

$$t' = \frac{S'_n * L' * A}{N'} = \frac{\text{step 29} * \text{step 35} * \text{step 20}}{\text{step 32}}$$

$$t' = \frac{.425 * 1084 * 2.32}{96} \qquad = \mathbf{11.13\ hours}$$

37. Compute the expected bit footage (F')<u> 224 </u>feet.

$$F' = \frac{C * (A * A_n + t')}{K}$$

$$= \frac{\text{step 21} * (\text{step 20} * \text{step 24} + \text{step 36})}{\text{step 30}}$$

$$= \frac{.0284 * (2.32 * 4.31 + 11.13)}{.00267} \qquad = \mathbf{224\ feet}$$

38. Compute expected cost per foot (C/F)'<u> 4.70 </u>$/ft.

$$(C/F)' = \frac{K * C_{opr}}{C} = \frac{\text{step 30} * \text{step 3}}{\text{step 21}} = \frac{.00267 * 50}{.0284} \qquad = \mathbf{4.70\ \$/ft.}$$

39. Compute actual drilling cost per foot (C/F)<u> 6.11 </u>$/ft.

$$C/F = \frac{\text{step 4} + (\text{step 3} * \text{step 6}) + (\text{step 3} * \text{step 8})}{\text{step 7}}$$

$$= \frac{200 * (50 * 6) + (50 * 12)}{180}$$

$$= \mathbf{6.11\ \$/ft}$$

40. Compare the savings and percent savings.

Savings = [C/F - (C/F)'] * F'

= [step 39 - step 38] * step 37

= (6.11 - 4.70) * 224 **= $ 315.84**

$$\% \text{ Savings} = \frac{C/F - (C/F)'}{(C/F)'} * 100$$

$$= \frac{6.11 - 4.70}{4.70} * 100 \qquad = \mathbf{30.0\ \%}$$

Thus, a savings in drilling costs of about 30% is expected for this example.

Table Bit - 1

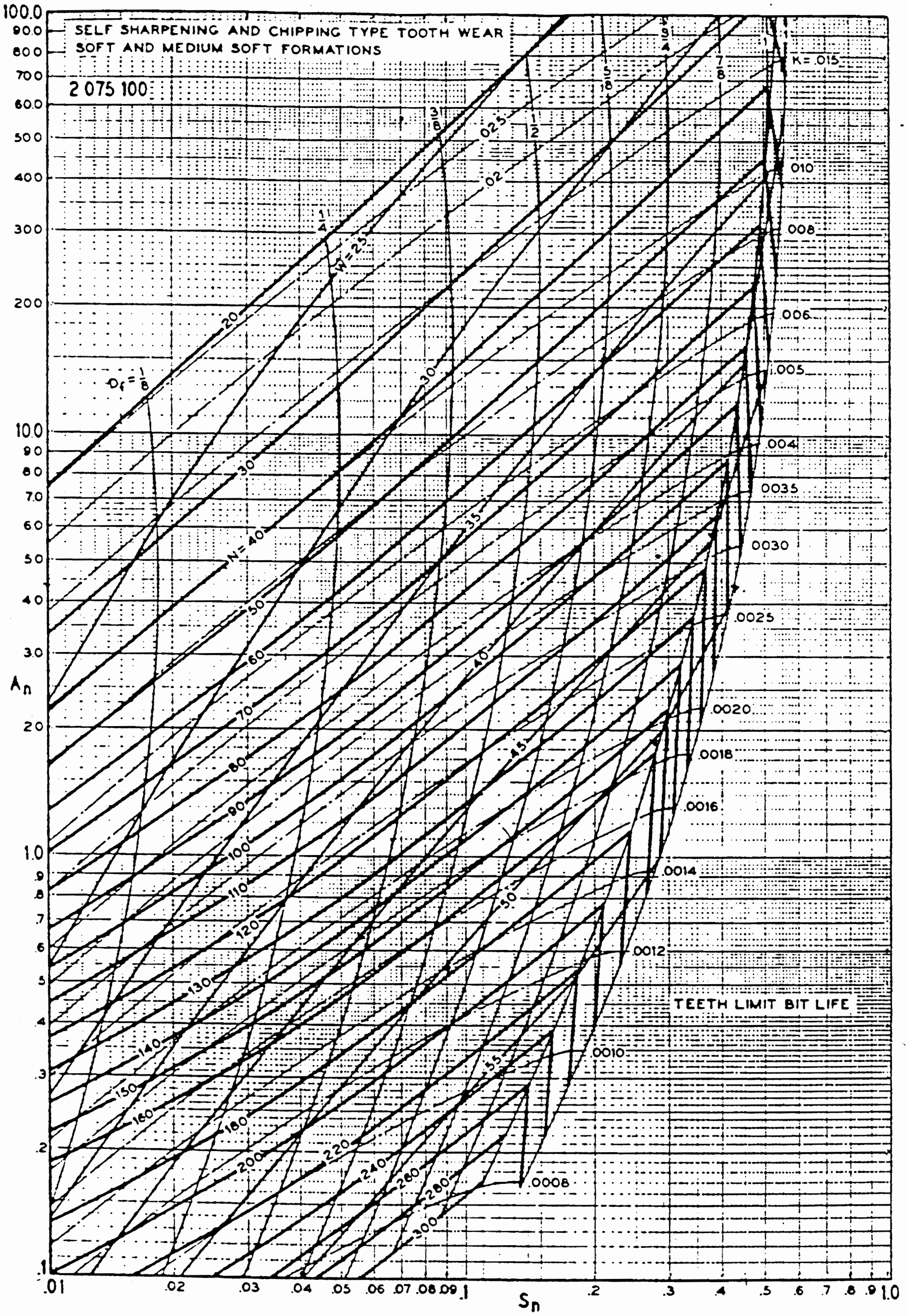
GALLE & WOODS WEIGHT AND SPEED TABLE

N	i	r
10	10	5.2
15	15	7.6
20	20	9.6
25	26	11.4
30	31	13.0
35	37	14.6
40	43	16.2
45	49	17.6
50	55	19.0
55	62	20.4
60	69	21.8
65	77	23.1
70	85	24.4
75	93	25.7
80	102	27.0
85	112	28.2
90	122	29.4
95	132	30.6
100	143	31.8
105	155	33.0
110	168	34.1
115	181	35.3
120	195	36.4
125	210	37.5
130	226	38.7
135	242	39.8
140	259	40.8
145	278	41.9
150	297	43.0
160	338	45.1
170	384	47.2
180	434	49.3
190	488	51.3
200	548	53.3
225	720	58.2
250	929	63.0
275	1179	67.6
300	1474	72.2
325	1818	76.6
350	2214	81.0
375	2668	85.3
400	3183	89.5

W	m	L
5	1.204	13239
6	1.124	12279
7	1.057	11376
8	.999	10532
9	.948	9745
10	.903	9016
11	.861	8360
12	.823	7758
13	.789	7207
14	.756	6702
15	.726	6240
16	.698	5813
17	.672	5422
18	.647	5064
19	.624	4738
20	.601	4439
21	.580	4162
22	.560	3909
23	.541	3677
24	.522	3465
25	.505	3270
26	.488	3089
27	.471	2922
28	.445	2768
29	.440	2627
30	.425	2496
31	.411	2373
32	.397	2259
33	.384	2153
34	.371	2055
35	.358	1963
36	.346	1876
37	.334	1794
38	.323	1718
39	.311	1646
40	.300	1578
41	.290	1513
42	.279	1452
43	.269	1395
44	.259	1340
45	.249	1288
46	.240	1238
47	.230	1191
48	.221	1146
49	.212	1104

W	m	L
50	.204	1063
51	.195	1024
52	.186	987
53	.178	951
54	.170	917
55	.162	884
56	.154	853
57	.147	822
58	.139	793
59	.132	766
60	.124	739
61	.117	713
62	.110	689
63	.103	665
64	.096	642
65	.090	620
66	.083	599
67	.076	578
68	.070	558
69	.064	539
70	.057	520
71	.051	502
72	.045	484
73	.039	467
74	.033	450
75	.027	434
76	.022	418
77	.016	403
78	.010	388
79	.005	373

D	U	V
0	0	0
1/8	123	105
1/4	316	236
3/8	581	389
1/2	920	563
5/8	1337	756
3/4	1834	967
7/8	2413	1194
4/4	3078	1437



BEST COMBINATION OF WT AND RPM FOR GIVEN $\bar{A}_n$ AND S_n

DIAMOND BIT HYDRAULIC LIFT OFF

Winters and Warren of Amoco teach diamond bit hydraulic lift off and two procedures for ascertaining the numerical value of bit lift off while circulating and making hole. They suggest that lift off occurs because of the "tight fit" of the body of the bit in the drill hole and the circulation of fluid. They compare the bit with a piston being pushed on by fluid pressure acting on its surface.

The BHA weight slacked-off on the diamond bit and which is reflected on the weight indicator is consumed by the contact force of the diamonds on the rock at the bottom of the drill hole and the lift off force.

$$\text{Slacked off weight} = F_{contact} + F_{lift}$$

The wear of the diamonds and its penetration rate will depend only on the contact force and not on the lift off force. The problem is that the lift off force masks the contact force.

If a bottom hole motor is in use two other problems occur; ascertaining and then applying the correct pressure differential across the motor. Without a specific procedure it is not possible to separate pressure drops of the bit, the motor, and the drill string. The other problem is that the life of the bearings within the motor is dependent on the pressure drop across the bit.

The weight on bit (actually this axis should have been named slack off) versus pump pressure chart was derived from a drill off test. For this test 12,000 pounds were slacked off at the surface, the drawworks brake was set, and while continuing to drill weight on bit and pump pressure were recorded. During the test the circulation rate was held constant. It must be noted that the pressure differential across a mud motor is proportional to the torque imposed on it by the bit; ie, no drilling, no torque, and no pressure drop across the motor.

The test begins at the upper right on the chart with a surface pump pressure above 3,300 psi and a slack off weight of 12,000 pounds. As the bit drills new hole the contact force between the diamonds and the rock decreases, the torque decreases as it should, and the decreasing pressure differential across the motor is reflected in the pump pressure. The data from point 1 to 2 evolves.

At point number 2 it is observed that the bit is no longer drilling and that the diamonds are not contacting the rock. The pump off point has been reached. It must be that the motor is no longer contributing to the pump pressure and the bit's jets, the lift off force pressure, and drill string are still contributing the same quantities; thus the drop in pump pressure from point 1 to 2 is the motor pressure differential. In this example its value is 563 psi (3,440 - 2,877).

Further, as the bit is picked up off the bottom of the hole, it is noted that the pump pressure drops to point 3 where the pump pressure is 2,445 psi. It must be that the lift off force is the only pressure drop which is no longer active. The lift off pressure drop is then 432 psi (2,877 - 2,445).

After ascertaining these values and having knowledge of the operating parameters of the mud motor, a correct hydraulic balance and slack off weight can be chosen.

Winters and Warren Drilloff Test Procedure

Winters and Warren recommend the following drilloff test procedure.

1. With the pump on and the bit rotating, position the bit about 1 foot off bottom.

2. Record the standpipe pressure and zero the weight indicator.

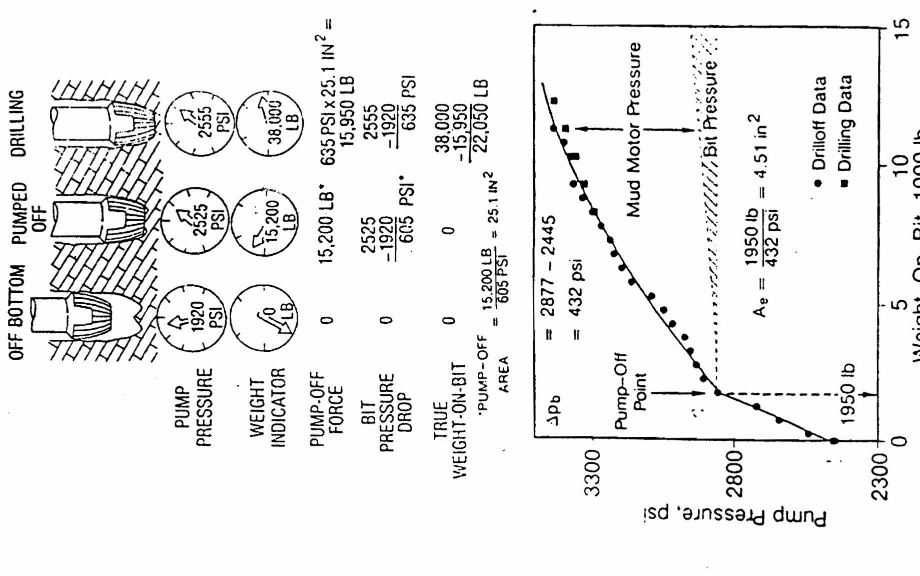
3. Lower the bit to the bottom and cut a fresh bottom hole pattern by drilling about 1 foot with normal weight on bit.

4. Lock the drawworks brake and let the bit drill off.

5. Record standpipe pressure, indicated weight, and time as the bit drills off. The data are usually recorded with each 1,000 lbs.

6. End the test when the indicated weight has clearly stopped decreasing. The weight indicator gives the pump off force at this point. Record the values and standpipe pressure.

7. Raise the bit about 1 foot off bottom to check that the standpipe pressure and indicated weight on bit have returned to their initial values at the beginning of the test.



DULL BIT GRADING

In 1987 the Internal Association of Drilling Contractors published a new dull bit grading procedure. The essence of the new procedure is presented in the figure and charts.

T				B	G	REMARKS	
CUTTING STRUCTURE				B	G	REMARKS	
Inner Rows (I)	Outer Rows (O)	Dull Char. (D)	Location (L)	Brng. Seal (B)	Gage 1/16 (G)	Other Dull (O)	Reason Pulled (R)

(I)- CUTTING STRUCTURE- Inner 2/3 of bit

(O)-CUTTING STRUCTURE- Outer 1/3 of bit

In columns (I) and (O) a linear scale from 0-8 is used to describe the condition of the cutting structure as follows:

STEEL TOOTH BITS-a measure of lost tooth height

0-no loss of tooth height
8-total loss of tooth height

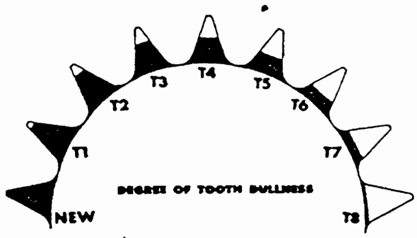
INSERT BITS-measure of total cutting structure reduction due to lost, worn and/or broken inserts.

0-no reduction of cutting structure
8-total reduction of cutting structure

(D) DULL/(O) OTHER DULL CHARACTERISTICS

*BC–Broken Cone
BT–Broken Teeth/Cutters
BU–Balled Up
*CC–Cracked Cone
*CD–Cone Dragged
CI–Cone Interference
CR–Cored
CT–Chipped Teeth
ER–Erosion
FC–Flat Crested Wear
HC–Heat Checking
JD–Junk Damage
*LC–Lost Cone
LN–Lost Nozzle
LT–Lost Teeth/Cutters
OC–Off-Center Wear
PB–Pinched Bit
PN–Plugged Nozzle
RG–Rounded Gauge
RO–Ring Out
SD–Shirttail Damage
SS–Self-Sharpening Wear
TR–Tracking
WO–Wash Out on Bit
WT–Worn Teeth/Cutters
NO–No Dull/No Other Wear

*show cone # or #'s under Location (L), Column 4



(L)-LOCATION

N–Nose Rows	Cone # or #'s
M–Middle Rows	1
H–Heel Rows	2
A–All Rows	3

(B) BEARING/SEALS

NON-SEALED BEARINGS-a linear scale from 0-8 is used to indicate the amount bearing life that has been used.
0-no bearing life used
8-all bearing life used

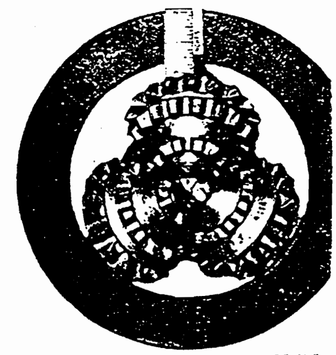
SEALED BEARINGS
E-seals effective
F-seals failed

(G) GAUGE
I-in gauge
1/16-1/16" out of gauge
2/16-1/8" out of gauge
10/16-10/16" out of gauge

(R)-REASONS PULLED

BHA–Change Bottom Hole A
DMF–Down Hole Motor Failure
DSF–Drill String Failure
DST–Drill Stem Test
DTF–Down Hole Tool Failure
LOG–Run Logs
CM–Condition Mud
CP–Core Point
DP–Drill Plug
FM–Formation Change
HP–Hole Problems
HR–Hours on Bit
PP–Pump Pressure
PR–Penetration Rate
RIG–Rig Repairs
TD–Total Depth/Casing Depth
TQ–Torque
TW–Twist Off
WC–Weather Conditions

MEASURING AMOUNT OF UNDERGAUGE



CONDITION: 0 ⅜—BIT OUT OF GAUGE ⅜"

BIT RECORD												
BIT NO.				1				2				3
SIZE				7 7/8"				7 7/8"				12 1/4"
IADC CODE				5	1	7	X	8	3	2	A	5	1	7	X
DULL CUT. STRUC. I	O	D	L	6	2	BT	M	5	8	WT	A	0	0	NO	A
B	G	O	R	E	I	NO	PR	3	3/16	FC	HR	E	I	LN	PP

TABLE 4–EXAMPLE DULL BIT GRADINGS

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CHAPTER XII

FISHING

DEFINITIONS

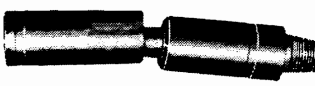
A fish is any undesirable object; such as, pipe, tools, wireline, which can not be removed from a borehole with ordinary practices. Fishing is any action taken which attempts to remove a fish from a borehole. For example, lowering a magnet into the bore hole for the purpose of removing a cone lost from a drill bit is fishing. Removing a broken wireline and a directional survey tool is fishing.

Indirect fishing time is the elapsed time which is required to regain the borehole depth and be in the drilling mode from the time the fish evolved. For example, if on day number 5 with the borehole at a depth of 5,000 feet and while drilling a drill bit cone is lost and thereafter 500 feet of hole are lost, and if on day 20 the borehole is once again drilled to 5,000 feet, then the fishing time is 15 days. Direct fishing time is the time which was spent attempting to remove the fish. In the previous example the time to regain the 500 feet of hole would not be direct fishing time.

Direct and indirect fishing cost parallels the thought of fishing time except that dollars replace time. For example daily rig cost and an overshot rental fee would be direct fishing costs; however if a bottom hole assembly were lost then its cost would be an indirect fishing cost.

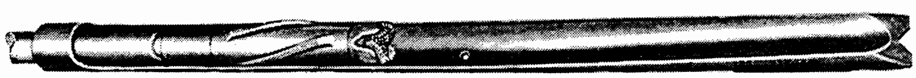
TO FISH OR NOT TO FISH

There are at least three popular methods for deciding whether to fish or not to fish. The most popular method is to fish for two days and if no progress is made other alternatives are chosen. One of the others is to construct break-even charts. These charts weigh the value of the borehole and equipment against the cost of fishing. The last method is based on the expected value method of decision making. The problem with it is that a data base is required.



Universal Joint Assembly for Standard or Circulating Whipstock

Standard Spiral Stabilizer

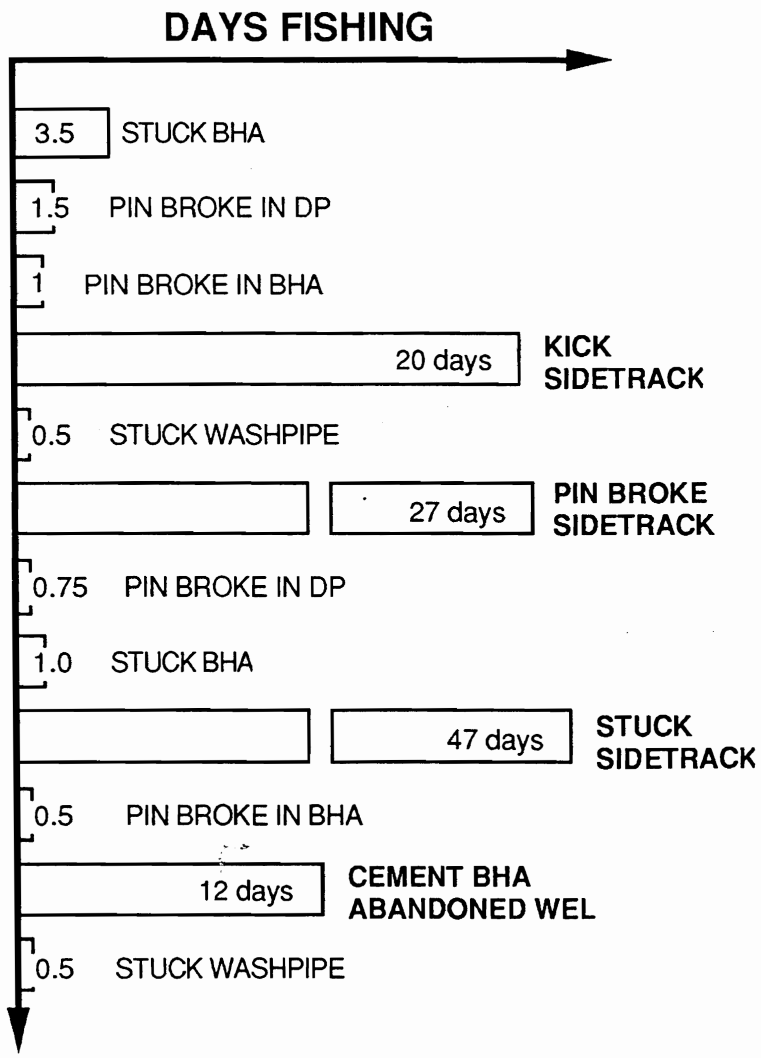


Standard Whipstock

FISH FOR NO MORE THAN TWO DAYS WITHOUT PROGRESS METHOD

The "fish for no more than two days without progress" rule is a decision method based on drilling experience. It is a rule of thumb. The following fishing analysis, which is extracted from the morning reports of a real well and shown in the chart, substantiates this rule.

In the chart if a sidetrack was not required or the well was not abandoned, then the fish was successfully extracted. Note that those fishing operations in which the fish were not extracted within two days ended in unsuccessful fish extraction and sidetracks were required with the exception of the first fishing operation (3.5 days).



Break-even Charts as a Decision Tool

"Break-even fishing cost" is a comparison of the direct costs of fishing with the potential of the salvage value of the fish and the drill hole. For example let the direct fishing cost be $6,000 per day, the fish is a $50,000 bottom hole assembly, $12,000 of hole will be lost, and a sidetrack at a cost $28,000 will be required if the fish is not removed. The break-even number of days of fishing provided that all the fish can be removed by the last day is

Direct   Indirect

$$6{,}000\ T = 50{,}000 + 12{,}000 + 28{,}000$$

$$T = 15 \text{ days}$$

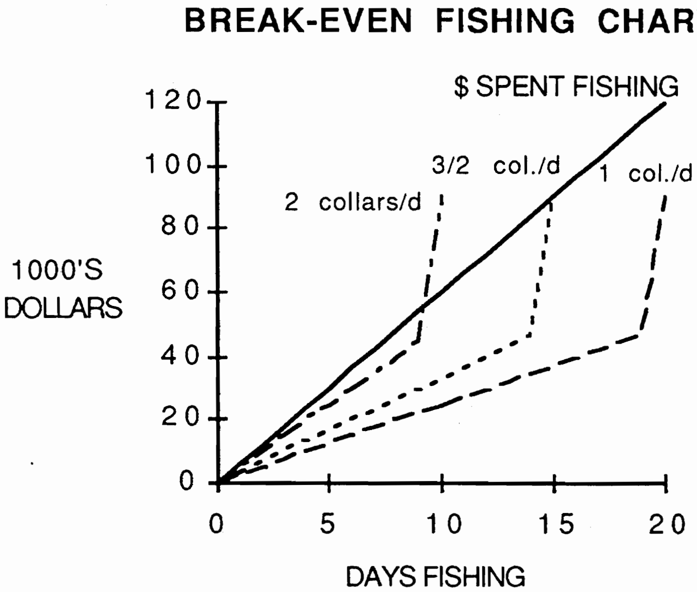
A popular variation of the break-even chart is to anticipate the partial recovery of the fish on a daily basis. For example let the fish be 20 drillcollars (at a cost of $2,500 each) and it is desired to ascertain the anticipated recovery rate of drillcollars to break even.

BREAK-EVEN TABLE OF FISHING COST

DAYS	DIRECT	RECOVERY		RECOVERY
RECOVERY				
FISHING	FISHING	@1 COL/d	@3/2 COL/d	@ 2 COL/d
	$1000	$1000	$1000	$1000
=======	=======	========	==========	=========
0	0	0	0	0
4	24	10	13	20
8	48	20	27	40
10	60	25	33	90
12	72	30	40	
15	90 <==	38	90 <== BREAK-EVEN	
18	108	45		
20	120	90		

Formula for direct fishing costs is $ = 6,000 * days fishing
Formula for recovery @ 1 collar/day is $ = 1 * 2,500 * days fishing
Formula for recovery @ 3/2 collar/day is $ = 3/2 * 2,500 * days fishing
Formula for recovery @ 2 collar/day is $ = 2 * 2,500 * days fishing

Note that on the last day when all of the collars have been removed that the hole will have been regained and the sidetrack eliminated; thus $40,000 will have been recovered over the alternative of sidetracking.



The chart shows the money spent and money recovered for each day of fishing. Recovering 3/2 collars per day on the average exactly breaks evens with sidetracking while recovering 1 collar per day produces a net loss. Recovering 2 collars per day gives a net gain. Thus if it is anticipated that more than 3/2 collars can be recovered per day and all of the collars can be successfully fished, then fishing should be attempted.

Expected Value Method as a Decision Tool

The expected value method is based on averages and probabilities. The expected value of an event is equal to the probability of it succeeding times the cost of succeeding, added to the probability of it failing times the cost of it failing. The sum of the probabilities of success and failure must have a value of one. The formulas for computing the expected value of fishing is

$$EV = P_s * C_s + P_f * C_f \qquad P_s + P_f = 1$$

For example it may be reasoned that the expected value of winning on betting that a tossed coin lands on heads, is to break-even. The expected value method solves the problem; thus, the cost of succeeding is +$1.00 and cost of failing is -$1.00 and the probability of success is .5 and the probability of failing is .5. A check of the probabilities show that the probability equation is satisfied

$$.5 + .5 = 1$$

and the EV equation gives

$$EV = .5 * (+1.00) + .5 * (-1.00)$$

$$= \mathbf{0} \text{ (no money will be won or lost)}$$

For example, in the drillcollar problem above, let a data base show that stuck collars were fished 27 times in 2 days out of 34 times that collars were stuck and cost \$15,000. The other 7 times that the collars were stuck, sidetracking was required at a cost \$40,000; however one of the sidetracks failed and the borehole was finally abandoned at a cost of \$250,000. In those cases in which fishing was not attempted prior to sidetracking, no boreholes were lost and the sidetracks cost \$35,000.

The probability involved with fishing are

$$P_s = \frac{27}{34} = 0.794 \qquad \text{and} \qquad P_f = \frac{7}{34} = 0.206 \qquad \text{Sum} = 1 \text{ as is required}$$

The probabilities involved with sidetracking after fishing are

$$P_s = \frac{6}{7} = 0.857 \qquad \text{and} \qquad P_f = \frac{1}{7} = 0.143 \qquad \text{Sum} = 1 \text{ as is required}$$

The expected value for the fishing and sidetracking is

$$EV = \frac{27}{34} * 15{,}000 + \frac{7}{34} * (\frac{6}{7} * 40{,}000 + \frac{1}{7} * 250{,}000)$$

$$= 11{,}912 \quad + \quad 14{,}412$$

$$= \mathbf{\$26{,}324} \text{ (fishing then sidetrack)}$$

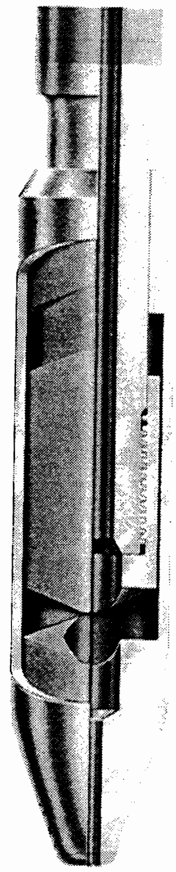
The expected value for sidetracking without fishing is

$$EV = 1 * 35{,}000 + 0 * 0$$

$$= \mathbf{\$35{,}000} \text{ (sidetrack without fishing)}$$

A comparison of the expected values shows that \$8,676 will be saved on the average if fishing is attempted before sidetracking in this example.

CASING SWEDGE



ESTIMATE OF COST: Confidence Lines and Least Squares

Confidence lines are computed and drawn to provide the following type of answer: 'if 7" casing is selected, 95% of the pipe will have an expected maximum diameter of 7.012" and a minimum expected diameter of 6.889" and an expected diameter of 7.001"'. These are often called the maximum, minimum, and best values.

Draw two confidence lines and the least squares line of the equation

$$Y = a + b\,X$$

Begin by computing a, b, and c

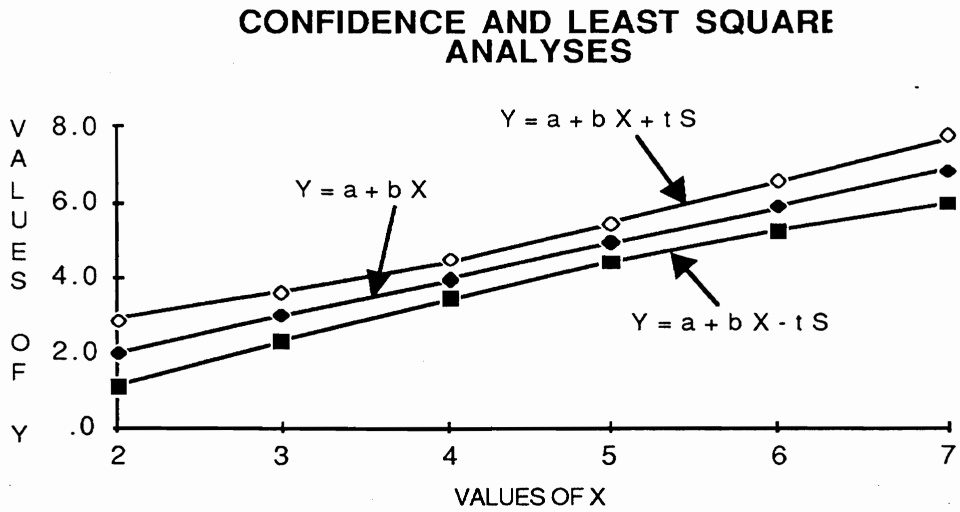
$$b = \frac{N\,\Sigma XY - \Sigma X\,\Sigma Y}{N\,\Sigma X^2 - (\Sigma X)^2} \qquad a = \frac{\Sigma Y}{N} - b\frac{\Sigma X}{N} \qquad c = \frac{\Sigma X}{N}$$

X, Y are data point pairs
N is number of data point pairs

The least squares line is given by the equation

$$Y = a + b\,X$$

To obtain confidence intervals, ascertain the value of the student's 't' distribution.



For example:

if 95% confidence is desired and N = 4

then G = 1 - 0.95 and

then $t_{n-2,\,1-G/2} = t_{2,\,.975} = 4.30$ (value depends on number of data points)

The two confidence lines as shown in the sketch are

$Y = a + b\,X + t\,S$ upper line

$Y = a + b\,X - t\,S$ lower line

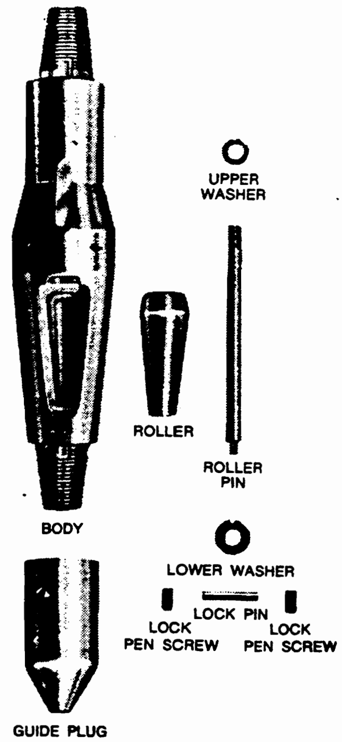
Where S is given by the equation below

$$S = \left[\frac{1}{N} + \frac{(X - c)^2}{\Sigma(X - c)^2}\right]^{\frac{1}{2}} * \left[\frac{\Sigma(Y - a - bX)^2}{N - 2}\right]^{\frac{1}{2}}$$

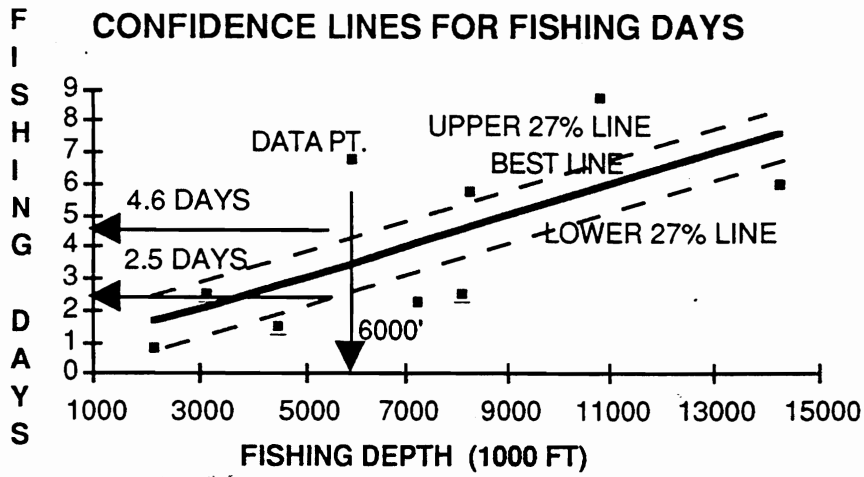
For example it is believed that fishing days increase with deeper depths. A data base contains the following information.

CONFIDENCE TABLE

Depth 1000 ft	Fishing Days	Y Lower	Y Best	Y Upper
=======	=======	======	=====	=====
2203	0.75	1.17	1.61	2.04
3163	2.5	1.70	2.09	2.47
4512	1.5	2.44	2.76	3.08
5949	6.75	3.30	3.47	3.74
7238	2.25	3.86	4.11	4.37
8129	2.5	4.29	4.56	4.82
8273	5.75	4.36	4.63	4.90
10834	8.75	5.54	5.90	6.27
14285	6.0	7.05	7.62	8.19



CASING ROLLER

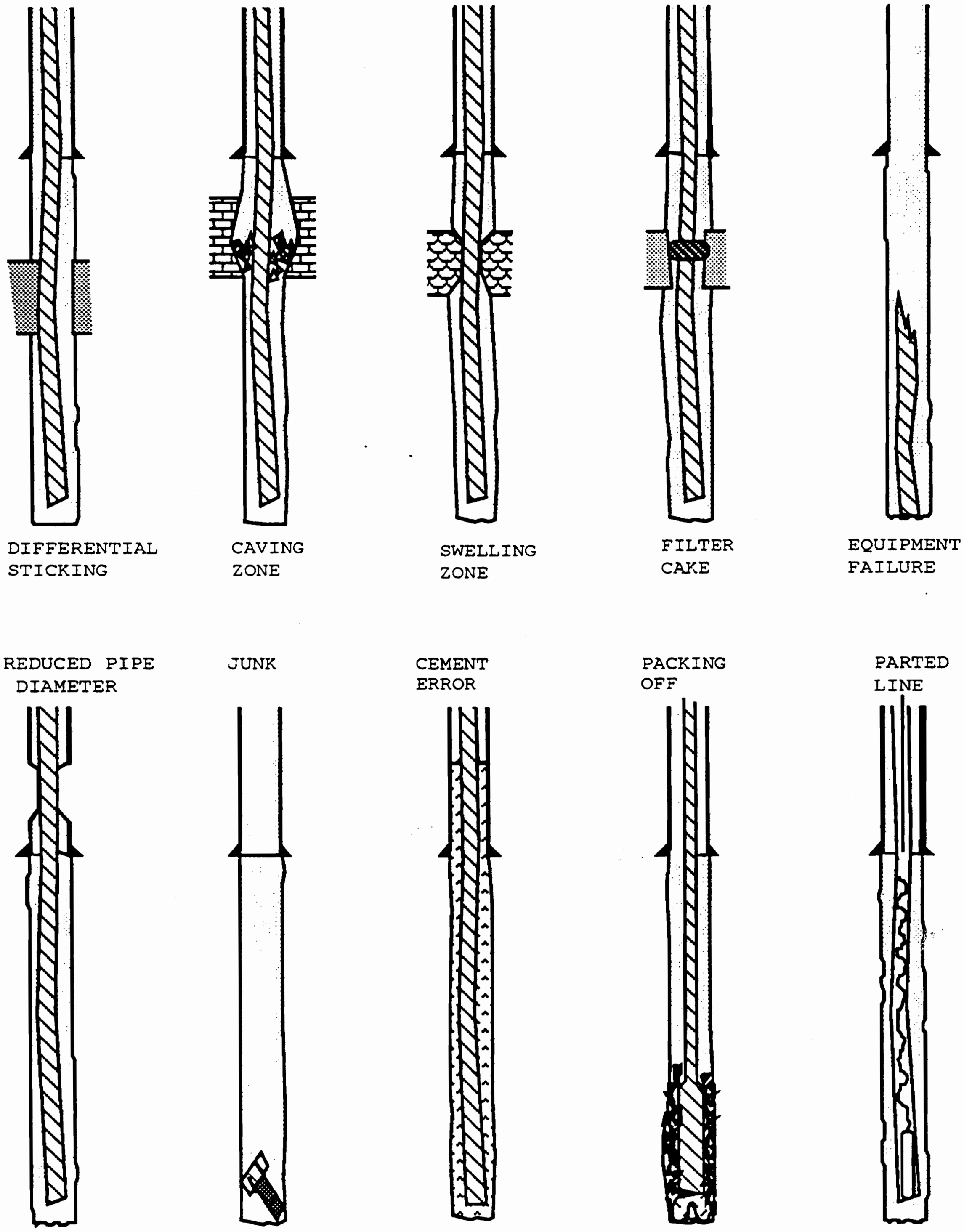


On the chart the data points are the solid squares. The chart should be read in this manner. At the fishing depth of 6,000 feet, the expected number of days for 27 out of 100 fishes will be no more than 4.6 days and no less than 2.5 days. The best estimate of the number of days will be 3.5 days. The equation of the best estimate line is

$$\text{Days} = 0.5 + 0.00049 * \text{Depth}$$

CAUSES OF FISH

Perhaps the most important aspect of fishing is to ascertain the events which caused the fish to occur. Without knowing the causation correct tool selection and procedures will be impossible.



DIFFERENTIAL STICKING: Usually occurs after the drill string is idle. Mud cake thickness is a factor as well as excessive overbalance. Circulation at normal rates is possible. Jarring is not effective.

CAVING ZONE: Caused by low density mud. Big problem in horizontal wells and high angle wells. Reactive filtrate may be a factor in shales. Can't circulate but may be able to move drill string a little. Jarring may help. Think about washing over. Sidetrack may be required.

SWELLING ZONES: Could be salt or diaperic shale. Run string reamers to bring hole back to gauge. Make short trips. May need to washover. Jarring usually helps.

FILTER CAKE: Thick filter cake can reduce the hole diameter. Usually occurs in consort with differential sticking. Reduce filter cake thickness. Short trips help. Can be washed-over.

EQUIPMENT FAILURE: May need to polish with mill. Retrieve with overshot.

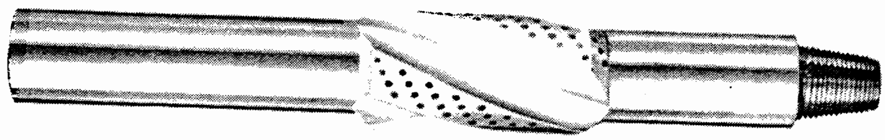
REDUCED PIPE DIAMETER: Try rolling back diameter. May sidetrack if a mill is used. Probably abandon well.

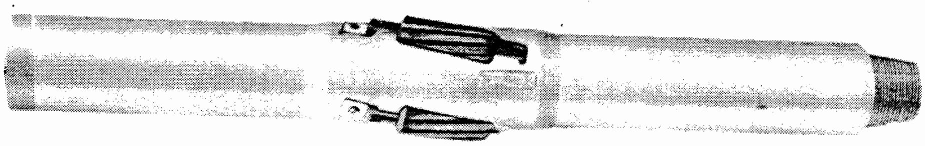
JUNK: Try magnet or junk basket. Try mill. Try jets. Try acid.

CEMENT ERROR: Try washpipe if outside. Try mill. If inside use drillbit.

PACKED OFF: If can circulate raise circulation rate. Raise viscosity if in near vertical hole. May be able to washover. Probably sidetrack borehole.

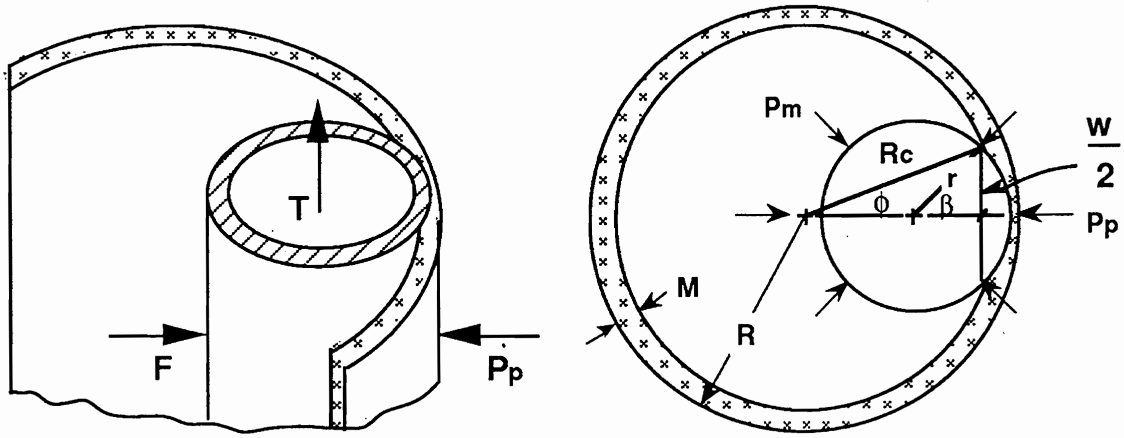
PARTED LINE: Retrieve with spear.





DIFFERENTIAL STICKING

Differential sticking of pipe, collars, or casing requires at least (1) a mud cake and (2) that the pressure within the wellbore exceeds the pore pressure. The sketch shows the mechanics of differential sticking.



MECHANICS OF DIFFERENTIAL STICKING

The vertical force required to pull the pipe is

$$T = C * F \qquad P_d = P_m - P_p$$

$$T = C * A * P_d \qquad A = W * L$$

The equations describing the parameters are

$$M = R - R_c$$

$$R_c \sin\phi = r \sin\beta$$

$$R_c \cos\phi = r \cos\beta + (R - r)$$

$$W = 2\, r \sin\beta$$

$$P_d = \frac{T}{C * L * W}$$

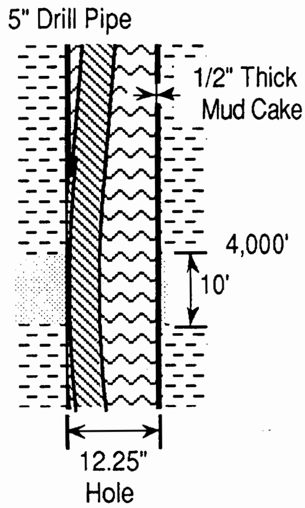
$$W = d \sin\left[\operatorname{acos} \frac{(D - 2\,M)^2 - d^2 - (D - d)^2}{2 * (D - d) * d}\right]$$

D_c = diameter of the hole with the mud cake
L = contact length of pipe with the cake
D = diameter of hole
d = diameter of pipe

M	=	mud cake thickness
W	=	projected width of contact area
C	=	coefficient of friction
T	=	tension
P_d	=	differential pressure

EXAMPLE

What is the allowable mud weight at 4,000 feet if the hole is 12 1/4" in diameter, the cake is 1/2" thick, the drill pipe is 5", the zone in which the pipe is stuck is 10 feet. The coefficient of friction is 0.3, the margin of overpull for the pipe is 100,000 pounds.



$$w = 5 \sin [\text{acos} \frac{(12.25 - 2 * .5)^2 - 5^2 - (12.25 - 5)^2}{2 * (12.25 - 5) * 5}]$$

$$= \mathbf{3.68\ inches}$$

$$P_d = \frac{100000}{.3 * 3.68 * 10 * 12} \qquad = \mathbf{755\ psi}$$

Then the allowable pressure of the mud at 4,000 feet is 2878 psi if the pore pressure is 9.0 ppg.

$$P_m = .052 * 9.0 * 4000 + 755 \qquad = \mathbf{2627\ psi}$$

$$MWE = \frac{2627}{.052 * 4000} \qquad = \mathbf{12.62\ ppg}$$

FREEING DIFFERENTIALLY STUCK PIPE

There are five methods of freeing differentially stuck pipe. One requires that the mud cake be destroyed. Two are based on the sound principle that the pressure in the annulus must be reduced. The circulation of oil into the annulus is the more popular of the two. The procedure is to circulate a quantity of oil into the annulus above the stuck point sufficient to lower the differential pressure between the mud and the pore pressure.

First Method

The first method involves circulating a volume of oil into the stuck pipe and displacing it into the annulus to a location where the oil is above the stuck point. This reduces the effective pressure at the stuck zone in the drill hole. If the hole were to gauge, then the quantity of oil and the mud to displace the oil is

$$v_o = c_{an} \frac{19.25\,P_p - D\,W_m}{W_o - W_m} \qquad V_d = C_p * (L - l_h) + C_{an} * H$$

EXAMPLE

Suppose 5" pipe within a 9" hole is stuck at a depth of 8100 feet. The hole depth is 10,000 feet. The pore pressure of the zone where the pipe is stuck is 4100 psig and the mud weight is 11.3 ppg. The weight of the displacing oil is 7.5 ppg.

The required volume of the oil (if the hole is to gauge) is

$$V_o = .0544 \frac{\frac{4100}{.052} - 8100 * 11.3}{7.5 - 11.3} \qquad = \mathbf{181.7\ bbls}$$

The volume of mud to pump behind the oil is

$$V_d = \frac{4.276^2}{1029} * 10{,}000 + 1900 * .0544 \qquad = \mathbf{281\ bbl\ mud}$$

Caution must be used with this procedure. The mud weight equivalent at 8100 feet is

$$MWE = \frac{4100}{.052 * 8100} \qquad = \mathbf{9.73\ ppg}$$

However, when the oil is circulated above 5,000 feet, then the mud weight equivalent at 5000 feet is only

$$MWE = \frac{1}{5000}\left(\frac{181.7}{.0544} * 7.5 + \left(5000 - \frac{181.7}{.0544}\right) 11.3\right) \qquad = \mathbf{8.76\ ppg}$$

Second Method

The second method of freeing differentially stuck pipe is to pump oil or water into the pipe and then bleed the oil or water from the pipe. This procedure drops the mud level in the annulus which in turn reduces the pressure of the mud at the stuck point. The procedure is very precise because the drop of the fluid level in practical cases occurs in the last set casing string. The minimum volume of oil or water to be pumped into the pipe and the maximum pipe gauge pressures are:

$$V_o = \frac{DC_p}{W_m - W_o} \left[W_m - \frac{P_p}{.052D}\right] \left[1 + \frac{C_{an}}{C_p} \left(1 - \frac{W_o}{W_m}\right)\right]$$

$$P_{gm} = .052 \frac{V_o}{C_p} (W_m - W_o)$$

and the fluid level drop when the pressures of mud and the formation are equal is

$$F.\,L.\,D. = D - \frac{P_p}{.052\ W_m}$$

The mud weight equivalent at depth during the bleed-off of the oil or water is

$$MWE = W_m - \frac{P_{gm} - P_g}{.052\ D\left[1 + \frac{C_{an}}{C_p}\left(1 - \frac{W_o}{W_m}\right)\right]}$$

When the pipe is released, the remaining oil in the pipe should be removed by filling the annulus with mud.

EXAMPLE

Free the pipe in the previous problem with the second method.

$$V_o = \frac{8100 * .0178}{11.3 - 7.5}\left[[11.3 - \frac{4100}{.052 * 8100}]\,[1 + \frac{.0544}{.0178}(1 - \frac{7.5}{11.3})]\right] \qquad \mathbf{= 120.5\ bbls}$$

$$P_{gm} = .052 \frac{120.5}{.0178}(11.3 - 7.5) \qquad \mathbf{= 1337.7\ psig}$$

$$F.\,L.\,D. = 8100 - \frac{4100}{.052 * 11.3} \qquad \mathbf{= 1122.5\ feet}$$

The mud weight equivalent at 5000 feet after a drop in the annulus of 1122.5 feet is

$$MWE = 11.3 - \frac{1337.7 - 0}{.052 * 5000\left[1 + \frac{C_{an}}{C_p}\left(1 - \frac{W_o}{W_m}\right)\right]} \qquad \mathbf{= 8.76\ ppg\ at\ 5{,}000\ ft}$$

Thus caution must be used in order to prevent zones high in the hole from kicking.

SYMBOLS

V_o	=	volume of oil pumped; bbls
V_d	=	volume of mud pumped behind oil; bbls
C_{an}	=	annular capacity; bbls/foot
C_p	=	pipe capacity; bbls/foot
W_m	=	weight of mud; ppg
W_o	=	weight of oil; ppg
P_p	=	pore pressure; psi
MWE	=	mud weight equivalent at a particular depth; ppg
D	=	depth to stuck zone or total depth; feet
P_g	=	pressure within the pipe at the surface; psi
P_{gm}	=	maximum P_g; psi

THIRD METHOD

The method is called the 'soak' method. A chemical wash which is designed to break the filter cake is circulated across the stuck zone and is then allowed time to react with the filter cake. Thereafter the chemical wash which remained below the zone is periodically circulated across the zone. During the soaking, the drillstring is stretched and relaxed in tension. Jarring is thought to be of little practical value. A significant variation is to slack-off and then pull on the drillstring.

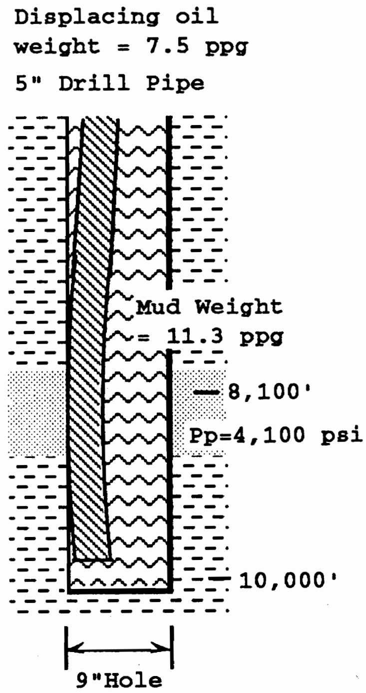
The buckling of the string during slack-off is thought to "un-zipper" the stuck portion from the wall of the hole. If a section of the hole is enlarged, it is possible to permanently cork-screw the drillpipe during the slack-off. A volume of chemical wash of 25 to 50 barrels is usually sufficient.

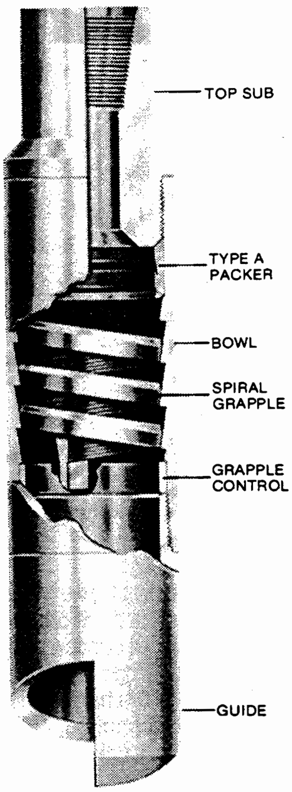
FOURTH METHOD

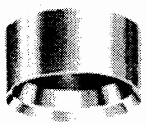
A popular method of freeing stuck pipe in which pressure control is critical is the drill stem test method. The essentials of the method is that the drillstring is backed off just above the stuck zone; a drill stem test tool is run, set, and opened above the zone. The test tool is pulled out of the hole and the once stuck string is retrieved with a screw-in sub or overshot. A variation is to attach an overshot to the test tool and eliminate the intermediate trip.

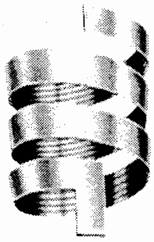
FIFTH METHOD

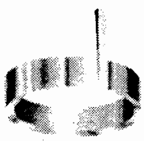
This is the quickest method of all. Slack-off a portion of or all of the weight of the drillstring onto the stuck zone. Increase the pump pressure until the safety pin in the pop-off value shears. Pull the once stuck string up the hole. The mechanism is that the drill string is buckled by the slack-off and extended by the pump pressure. The shearing of the pin allows the drillstring to contract and jar the pipe in the stuck zone. If a section of the hole is enlarged, it is possible to permanently cork-screw the drillpipe during the slack-off.

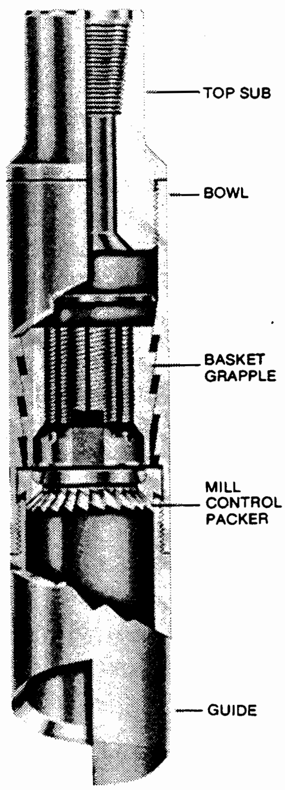


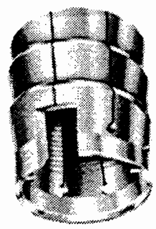


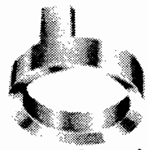

Packer

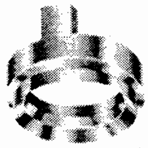

Spiral Grapple

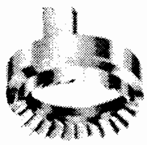

Spiral Grapple Control




Basket Grapple

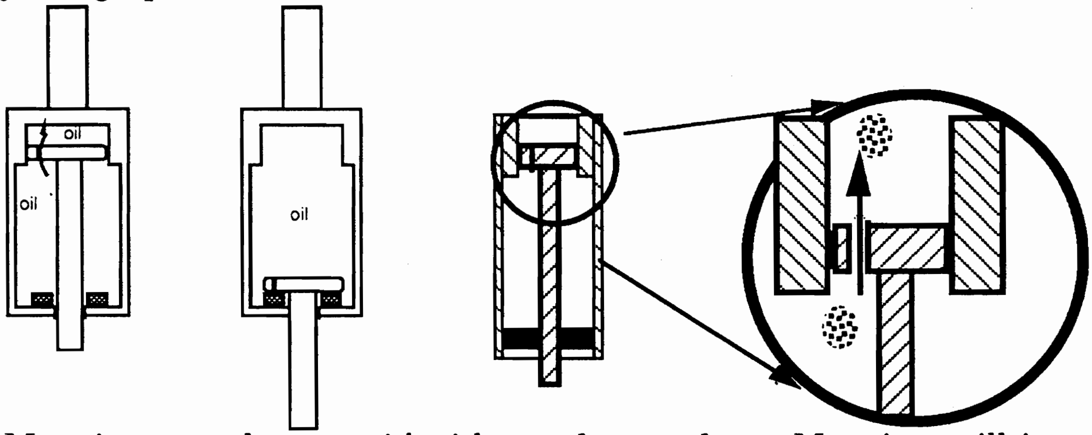

Basket Grapple Control


Basket Grapple Control Packer


Basket Grapple Mill Control Packer

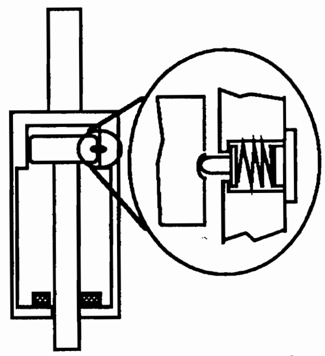
JARS AND ACCELERATORS (Boosters & Intensifiers)

There are two popular types of jars: (1) oil triggered and (2) mechanically triggered. The oil jars are common in fishing operations because the triggering load placed on the jar is only set by pulling on the drillpipe; i.e., if 50,000 pounds are pulled on the drillstring, then the jar triggers at 50,000 pounds. Some mechanical jars require pre-setting of the triggering load at the surface prior to running the jars, while others require the torquing of the drill string during the jarring operation.



Most jars may be run with either end up or down. Most jars will jar either up or down.

Mechanical jars have proven to be more popular than oil jars while conducting normal drilling, testing, and coring operations primarily because they are always the same length in the drillstring. Also, oil jars may trigger under small tension loads at in-opportune times. In general jars are run in drillstrings to reduce the risk of permanent sticking of the string.

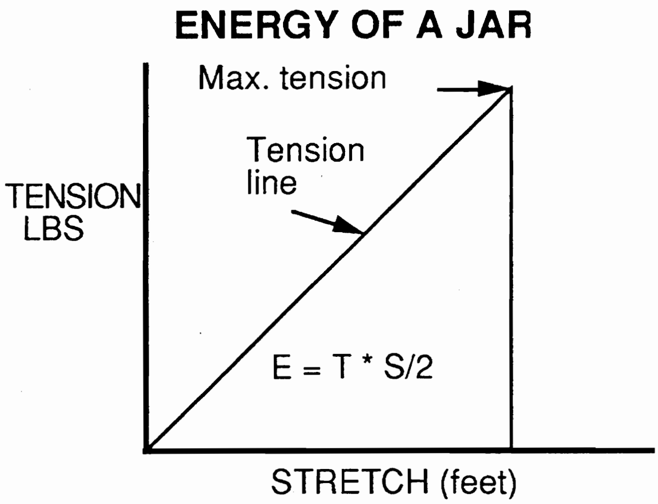


Normally jars are run below several drill collars which act to concentrate the blow of the jar. The popular numbers of collars are three to nine collars. Nine collars acts as a big hammer while three collars act as a small hammer. The momentum of the collars created by the contraction of the stretched drillpipe gives the blow to the fish. A hammer blow equal to three to four times the tension put into the drillstring at the surface (triggering tension) is possible.

The blow is reduced by drag of the jarring string on the wall of the hole and the mud. The absolute maximum amount of energy which can be placed into a jar without an accelerator is one-half the product of the stretch of the drillpipe and the strength of the drillpipe. Note in the figure that at the beginning of the pull on the drillstring, the tension and stretch of the string are zero; thereafter, the tension stretches the drillstring linearly, conforming to Hooke's law, to a triggering value. At that tension the energy is the area below the tension line. The equation is

$$\text{Energy} = \frac{\text{Tension} * \text{Stretch}}{2}$$

It is advantageous to use an accelerator for three situations. One is while jarring near the surface where it is not possible to stretch the drillpipe sufficiently to give a good jar blow. The second is while jarring in a crooked hole or any hole which has a lot of drag. In this case the drillpipe will not be capable of contracting quickly and giving a hard blow. The third is any time that a harder blow is desired than can be given with only a drillstring.



Care must be exercised in setting a triggering load. The triggering load must not exceed the difference between the strength of the pipe and loads created by the weight of the string and the internal fluid pressure-area force acting at the top of the drillpipe.

In directional holes and horizontal holes it is popular to place the jar at the top of the regular collars and below the heavy weight drillpipe. If there are no regular collars in the drill string and only heavy weight, the jar is usually placed between the heavy weight and the drillpipe.

Most manufacturers think that there is little difference in the wear of their jars in regard to whether they are run cocked or extended.

It is thought that placing jars in the drill string above the neutral point of bending will prolong their life. (The neutral point of bending is often incorrectly believed to be the neutral point of tension and compression.)

EXAMPLE

Compute the maximum jar triggering load if 90 feet of 8" * 3" * 147 ppf drill collars and 8,000 feet of 35% worn 5" * 19.5 ppf * E grade drillpipe (actual weight is 21 ppf) is run above the jar in 11.0 ppg mud.

The maximum triggering load without hydraulic pressure assistance is the difference between the tensile strength of the drillpipe and the buoyed weight of the string above the jars. If the hydraulic pressure is assisting the triggering of the jars then the pressure-area force created by the hydraulic pressure must be added to the buoyed weight of the string.

The working tensile strength of the grade E drillpipe is computed as follows:

$$S_t = (1-.35) * \frac{\Pi}{4} * (5^2 - 4.276^2) * 75{,}000 * 0.9 * 0.875 \qquad = \mathbf{202{,}496\ lb}$$

The buoyed weight of the string above the jars is the buoyed weights of the drill collars and drillpipe

$$W = (1 - \frac{11}{65.45}) * (90 * 147 + 8{,}000 * 21) \qquad = \mathbf{150{,}771\ lb}$$

Thus, the maximum triggering load is the difference in strength and buoyed load

Max. Triggering Load = 202,496 - 150,771 = **51,725 lb**

If the hydraulic pressure within the jar can be raised to 1,000 psi above the exterior pressure by pumping at the surface into the drillpipe and the jar has a seal area of 15 sq. in., then the hydraulic assistance (called pump extending force) will be

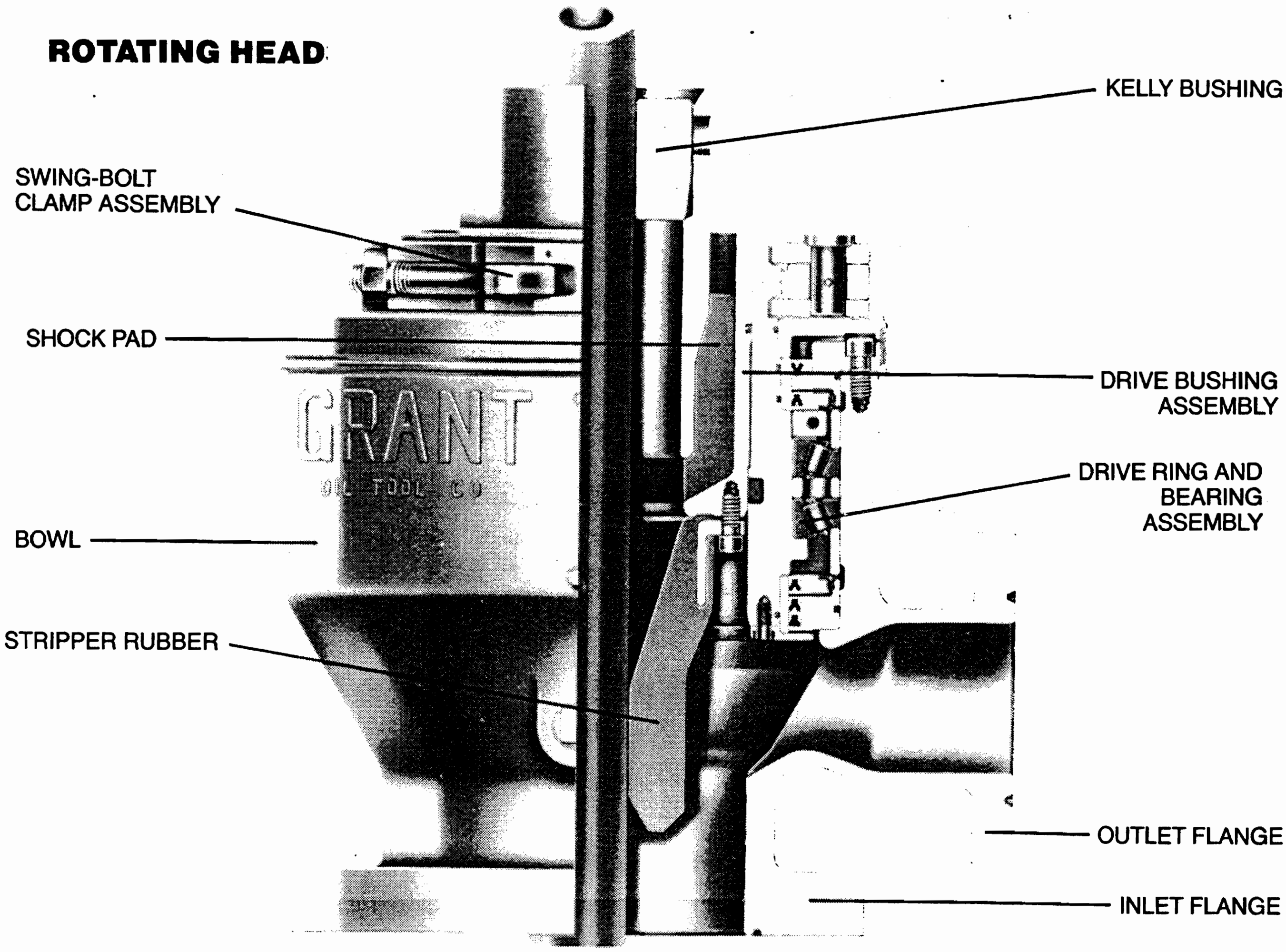
Pump extending force = jar pressure * seal area

= 1,000 * 15 = **15,000 lb**

The maximum loads which can be placed on the drillpipe at the rotary table are

Max. Load = **202,496 lb** without hydraulic assistance

Max. Load = **187,496 lb** with hydraulic assistance



BACK-OFF AND FREE POINT TOOL PROCEDURES

Using the Free Point Tool

Selecting a location to back-off a stuck string of pipe is assisted with a free point tool. In most cases the desirable location to back-off will be at a collar just above the stuck point. Situations which may dictate other locations are to back-off inside casing, above a dogleg, above a ledge, or within a gauge hole. Backing-off in an over gauge hole may allow the string to fall to one side prohibiting the latching onto it with an overshot or other tool.

The free point tool identifies the depth at which a stuck string can be moved and not moved (strained). It does this by detecting the stretch (tension) and twist (torque) of the string with a inductance coil frequency generator and sensor. It is known that when steel is strained, it will change the frequency of the voltage in a nearby coil. Free point tools are run on an electrical cable within the stuck string. The free point (and stuck point) is found by trial and error by lowering and raising the tool. Because of drag of the stuck string on the wall of the borehole, the tool will show zones where the string is partially free and, of course, partially stuck. The free point is usually selected as that location above which the string is 80% or more free all the way to the surface.

Using Pipe Stretch to Locate the Free Point

Two factors adversely affect the location of the free point with pipe stretch. One factor is that pipe buckles in a hole. The other factor is friction between the wall of the hole and pipe. The lower portion of a pipe will be buckled if the buoyed weight of the pipe above the stuck point is not pulled at the hook.

This means higher tension on the pipe, and higher tension means higher friction. Thus, a reasonable procedure for finding a free point is to slack off until buckles form in the pipe, pull the buckles out, and continue pulling to 80% of the pipe strength. During the pulling, make a graph of the stretch versus pull data. The following equations and an ideal graph will assist in the interpretation of the graph.

Helical Buckling Equation (Lubinski Eqn.)

$$\Delta L_b = \frac{r^2\,\Delta F_b^{\,2}}{8\,E\,I\,w_b} \qquad \text{if } \Delta F_b > 0$$

$$\Delta L_b = 0 \qquad \text{if } \Delta F_b <= 0$$

ΔF_b = Bouyed wt. of pipe - Pull Load (ΔF)

ΔL_b = apparent stretch caused by buckling; in

r = radial clearance of pipe and hole; in

ΔF_b	=	pulling load; lb
E	=	Young's modulus for steel; 30,000,000 psi
I	=	cross-sectional moment of inertia; in^4
w_b	=	buoyed weight of pipe per foot; lb/ft

Stretch Equation (Hooke's Law)

$$\Delta L_s = \frac{L\ 12\ \Delta F\ Dens\ 12}{E\ w_s} \quad \text{or} \quad L = \frac{\Delta L\ w_s\ 735389}{\Delta F}$$

ΔL_s	=	real stretch caused by pulling; in
L	=	length of pipe to the free point; ft
ΔF	=	pulling load; lb
Dens	=	density of steel; .2833 lb/in^3
E	=	Young's modulus of steel; 30,000,000 psi
w_s	=	weight of the steel in the pipe per foot; lb/ft

Total movement at the point of pulling is

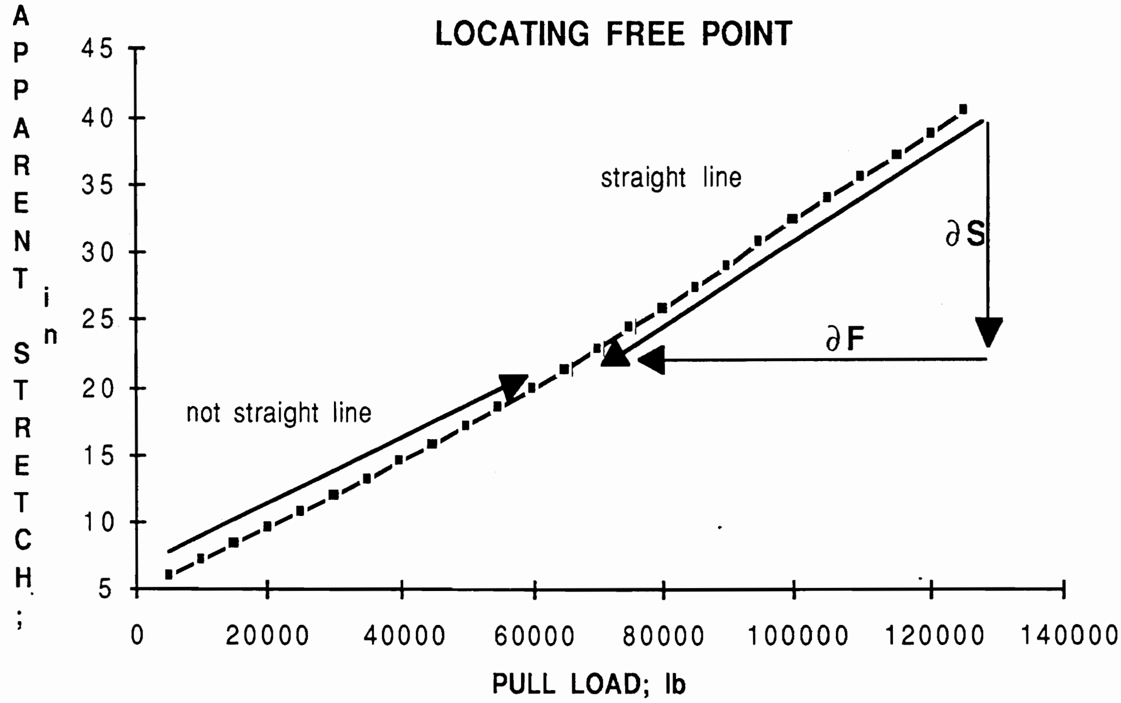
$$\Delta L_t = \Delta L_b + \Delta L_s$$

STRETCH OF PIPE

Pipe: OD = 5" ID = 4.276 w_s = 21 lb/ft I = 14.27 in^4
Hole: 17.5" bit dia. r = 6.25 in
Mud: 9.5 lb/gal

Pull (F) lb	ΔL_s in	F_b lb	ΔL_b in
5000.0	1.6	84759.4	4.6
10000.0	3.2	79759.4	4.0
15000.0	4.9	74759.4	3.6
20000.0	6.5	69759.4	3.1
25000.0	8.1	64759.4	2.7
30000.0	9.7	59759.4	2.3
35000.0	11.3	54759.4	1.9
40000.0	13.0	49759.4	1.6
45000.0	14.6	44759.4	1.3
50000.0	16.2	39759.4	1.0
55000.0	17.8	34759.4	.8
60000.0	19.4	29759.4	.6
65000.0	21.0	24759.4	.4
70000.0	22.7	19759.4	.2
75000.0	24.3	14759.4	.1
80000.0	25.9	9759.4	.1
85000.0	27.5	4759.4	.0
90000.0	29.1	-240.6	.0

95000.0	30.8	-5240.6	.0
100000.0	32.4	-10240.6	.0
105000.0	34.0	-15240.6	.0
110000.0	35.6	-20240.6	.0
115000.0	37.2	-25240.6	.0
120000.0	38.9	-30240.6	.0
125000.0	40.5	-35240.6	.0



Depth to the free point is $L = \frac{(40.5 - 32.4) * 21 * 735389}{25000} = 5003 \text{ ft}$

Back-off Procedure

The back-off tool is a steel bar wrapped with primer cord hanging below a collar locator tool. A collar locator tool measures the change in the magnetic flux put into the stuck string by the tool. Essentially, it locates the air gap or line of demarcation which exists between two tubes. With the collar locator tool hanging within the collar to be backed-off, the tool is raised the requisite distance (not more than ten feet) which places the primer cord within the collar. After twisting the stuck string the primer cord will be detonated.

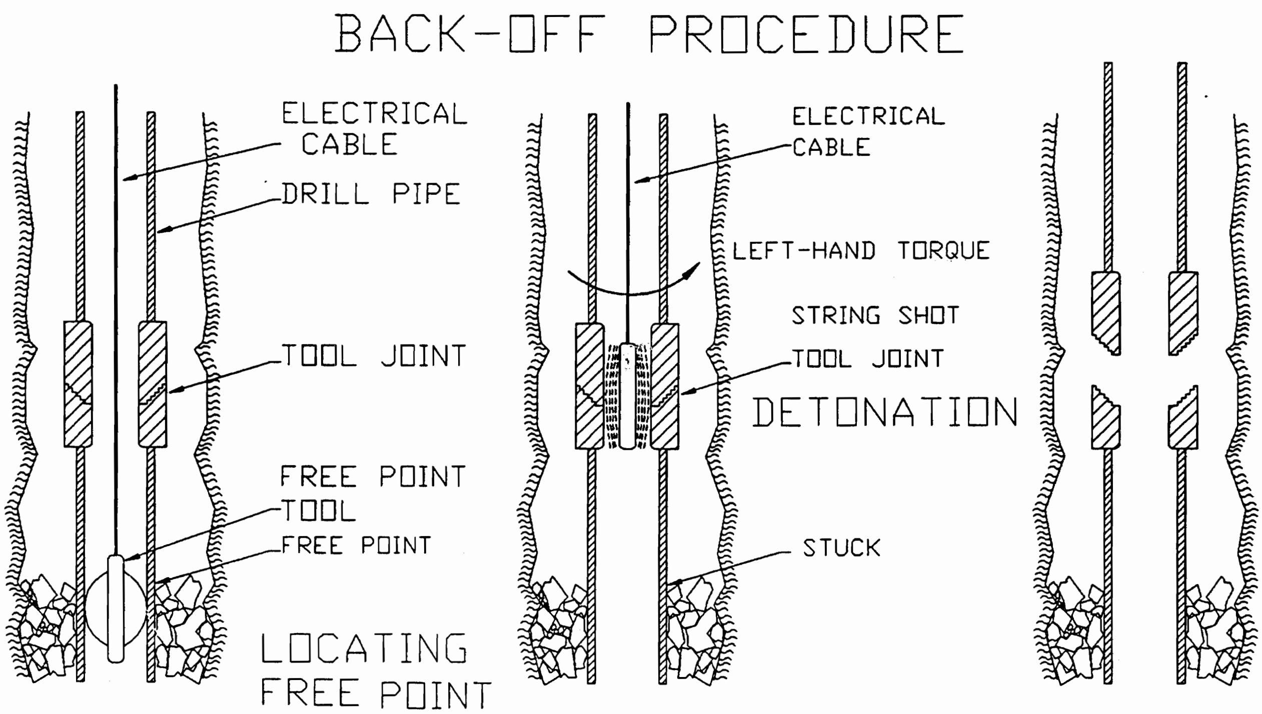
The detonation of the primer cord only momentarily relieves the stress between the shoulders within a tooljoint and the threads. Prior to detonation lefthand torque is put into the stuck string by twisting at the surface. To enhance the possibility that the torque is also put into the tooljoint to be backed-off the stuck string is alternated stretched and relax. The rule of thumb for the amount of twist to put into the stuck string is one turn per 1,000 feet of free string.

The amount of tension pulled at the surface will have an affect on the back-off. Two points of view are popular. One, the one to which the author subscribes, is that the buoyed weight of the stuck string above the stuck point must be pulled. The other view is that the air weight of the stuck string above the stuck point must be pulled.

Pulling the buoyed weight above the stuck point will pull all of the buckles out of the string placing a minimum amount of load and therefore friction between the wall of the borehole and the stuck string. Thus, the twist will have the maximum affect. Pulling the air weight will also pull the buckles out of the stuck string but will at the same time pull the stuck string more tightly into the wall of the hole which will add to the friction and negating a portion of the twist put into the stuck string.

BACK-OFF RULES

1. **PULL THE BUOYED WEIGHT OF THE STRING ABOVE THE STUCK POINT**
2. **TWIST THE STRING ONE TURN PER 1,000 FEET OF FREE STRING**
3. **FIRE THE SHOT AFTER WORKING THE TORQUE DOWN TO THE TOOLJOINT TO BE BACKED-OFF**



LATCHING ONTO THE FISH

The most popular tool for latching onto fish is the overshot. The overshot allows near full circulation rates, has high tensile strength for pulling heavy loads, and permits long periods of jarring. However, less popular latching tools; such as, the inside tap, the outside tap, and the spear, may be functional. The significant limitations of the overshot are that it requires a significant diametrical clearance between the inside wall of the hole and the fish. In order to latch over the fish, a single size overshot has a narrow diametrical range over which it will latch, and it will not securely latch onto a mildly tapered fish.

LATCHING AND UNLATCHING PROCEDURE

Exact measurements of the depth of the top of the fish and the length of the fishing string are best. Lowering the overshot onto the fish must be done with precision in order to avoid flaring or otherwise damaging the top of the fish. The overshot should slide over the top of the fish with one point of weight or less. If higher weight is needed to force the fish into the overshot, then the overshot may not release the fish at a later time.

To correctly latch onto a fish, the overshot should slowly be rotated to the right (5 rpm or less) as the overshot is slid over the fish. If the overshot does not slide freely over the fish, the most likely reason is that the top of the fish has been flared or it has ears. These must be removed with a mill. Mills attach to overshots; therefore, the milling and latching can be accomplished in one run.

Most overshots release by torquing the fishing string to the right and mildly jarring down onto the overshot and fish. This action loosens the grapple within the overshot so that it may spring back into it recess.

Some overshots are designed such that wireline tools may be run through them and near full circulation is allowed while others are simple devices.

SELECTING THE SIZE OF THE OVERSHOT

The OD of an overshot can be no larger than the drift diameter of the casing into which it is to be run or run through. The maximum catch size is the OD of the fish onto which the overshot can be latched. The minimum size which can be latched onto is about 1/2 inch less than the maximum latch size. The maximum and therefore minimum latch size for any one OD size overshot depends on the type of installed grapple. The spiral grapple gives a larger latch size but has less tolerance for the OD of the fish than does the basket type of grapple.

It should be noted that spiral grapples require less diametrical clearance of the fish and the hole than do basket grapples. If the OD of an overshot is reduced by machining away the steel of its body, the overshot will be weakened. When ordering an overshot, the top connection must be specified, the OD of the fish to be latched, and the OD of the overshot. If the overshot is to be run above wash pipe or other tools, then a cross-over sub may be needed.

The following table shows a typical specifications for overshots.

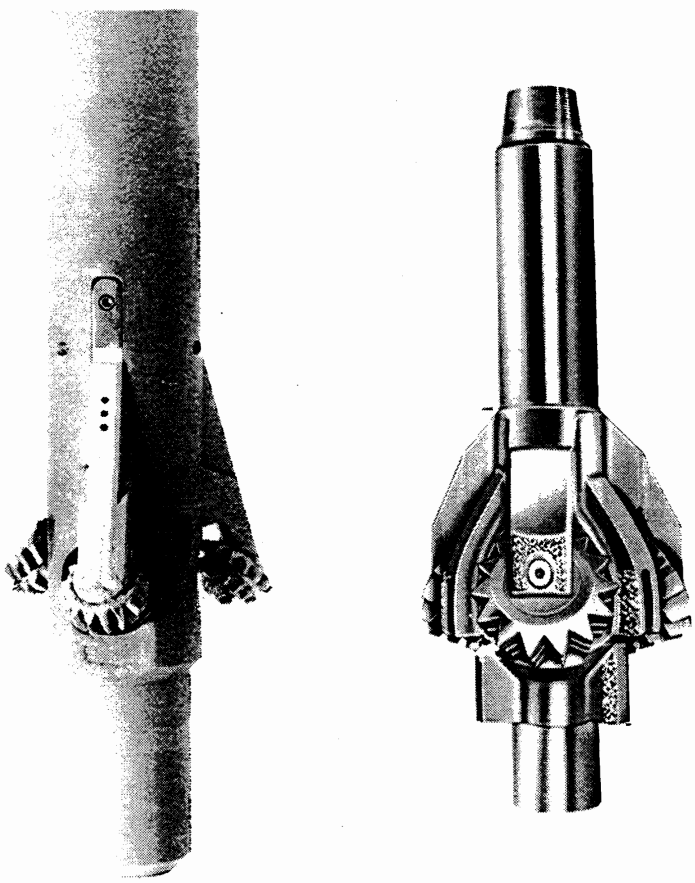
OVERSHOT SPECIFICATION TABLE

	1	2	3	4	5
MAX. LATCH SPIRAL	5	5 1/2	5 3/4	5 3/4	6
MAX. LATCH BASKET	4 1/4	4 3/4	5 1/8	5	5 1/2
OVERSHOT OD	6 5/8	7 1/8	6 7/8	7 3/8	7 5/8

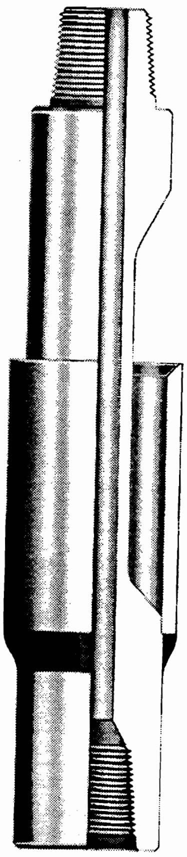
EXAMPLE OVERSHOT

Five inch drillpipe which is residing within 8 5/8" by 44 ppf casing is to be latched with an overshot. The casing has a drift diameter of 7.5 inches. Select number 3 from the above table. It has the 6 7/8" OD overshot with a basket latch which has a maximum latch size of 5 1/8 inches.

UNDER REAMER AND HOLE OPENER



JUNK BASKET

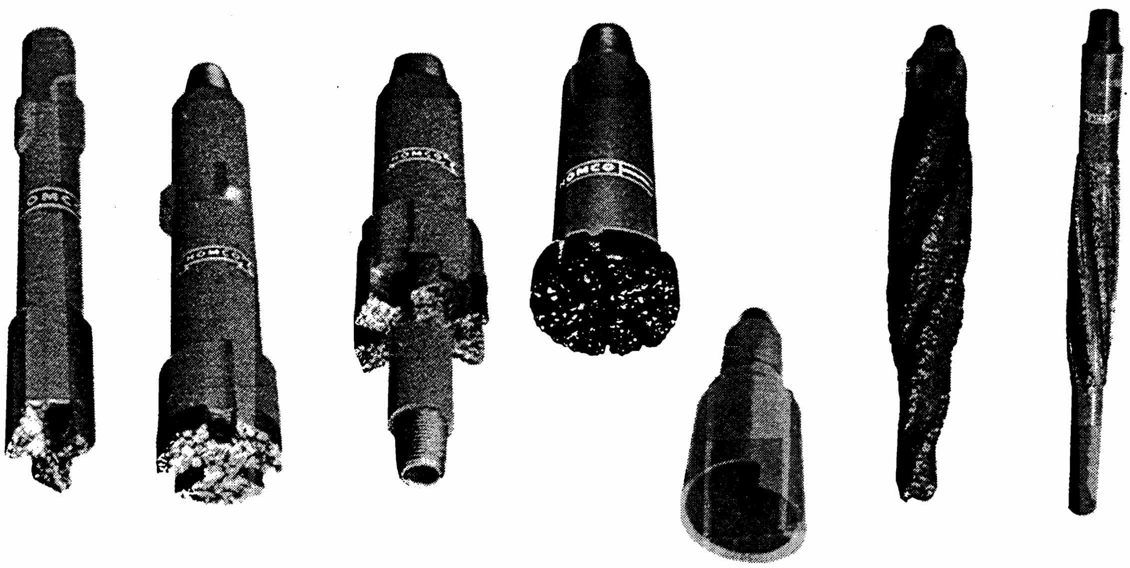


MILLING

Milling is the cutting of steel. The result is the making of steel "filings" and the removal of steel which is blocking the hole or a hole to be drilled. Milling is designed to accomplished the following goals:

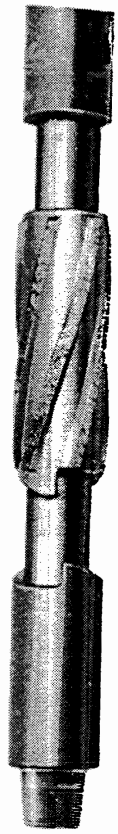
1. Cut up small tools or other objects in the hole; such as, bits, hand wrenches, clamps, etc.
2. Cut up tools purposefully placed in the hole; such as, packers, cement shoes, bridge plugs, port collars, etc.
3. Cut up parts of a drillstring lost in the hole; such as, drilling subs, stabilizers, stabilizer blades, drillpipe, casing, collars, etc.
4. Cut a window in casing (usually done for the purpose of sidetracking out of casing).
5. Cut a hole through collapsed pipe
6. Cut packer and bridge plug slips so as to release the tool
7. Cut the lips off overly torqued casing pins
8. Cut tops of liners and broken pipe to "true" the tops up

CHOOSING THE CORRECT MILL



The cutting surface of mills are manufactured of either tungsten carbide, diamond, or a combination of both. Tungsten carbide is the prevalent material. Diamonds are thought to be more fragile than tungsten carbide. Also, tungsten carbide particles can be sweated onto a mill on the rig floor with a welders torch.

Mills may be designed to cut only on bottom, cut on the inside, cut on the outside, or any combination. If for example it were desired to mill the slip of a packer without damaging the casing, one would choose a mill with tungsten carbide on the inside and the bottom and without tungsten carbide on the outside of its cutting surfaces.



Key Seat Wiper

The configuration of a mill refers to its shape. For example if a tooljoint is to be removed from drillpipe, the mill would be configured with blades of sufficient length to cut the outside diameter of the tooljoint and a stinger in its center to guide and stabilize the mill. The stinger could have a mill attached to its end to cut and remove objects or set mud or set cement. Mills are convex and concave on bottom. Some have blades, or are solid donuts.

The size of the tungsten carbide dictate the size of the filings and in a major way affects the speed of the milling. Big tungsten carbide particles cut bigger filings and may be better suited for milling a liner for example.

MUD, WEIGHT, AND ROTARY SPEED FOR MILLING

Mill filings are not easily circulated from the hole or suspended in the hole. Raising circulation rates to facilitate the removal of filings is usually not possible because of close clearance tools and pump limitations. Increasing the API yield point of the mud to a value of 35 lbf/100 sq.ft. may be required in order to remove filings from the cutting face of the mill, remove them from the hole, and to suspend them.

These same filings are not easily removed from the mud at the surface with normal mud cleaning equipment if the yield point of the mud is raised. If left in the mud, the circulating system is in jeopardy of rapid wear. Rubber pump parts are especially at jeopardy. One solution is to place several magnets in the suction pit. These must be cleaned frequently if a great deal of milling is done.

Selecting the best combination of weight and rotary speed has five important facets other than fast milling. These are pump off force, slack off weight, vibration, buckling, and tungsten carbide cooling.

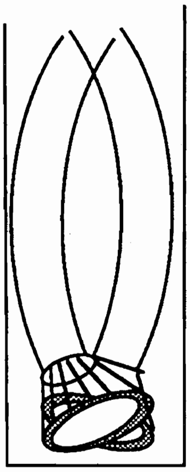
Because most mills have a natural close clearance with the bottom of the hole and the steel being milled, the circulation of fluid lifts the bit partially off bottom. Thus the weight slacked off at the surface is not placed on the bottom of the hole. Hole drag may, of course, also be a factor. A series of hydraulic tests can identify the pump off force and the weight being applied to the bit. A more practical method is to observe the rotary table torque and its changes as weight is slacked off. Drilling with a combination of slack off weight and torque is the practice.

The upper limit of rotary speed is set by either vibration or the cooling of the mill. Two types have been identified. One is torsional vibration and the other is longitudinal. If either is induced the other is sure to occur. Torsional is often called bit hang-up while the other is called bit bounce. The solution to bit hang-up is to add more collars if reasonable. The solution to bit bounce is to install a shock sub, add more collars, and/or stabilize the milling string.

The additional collars act to smooth the vibration as does a fly wheel on an engine. The bigger the fly wheel the smoother the engine runs.

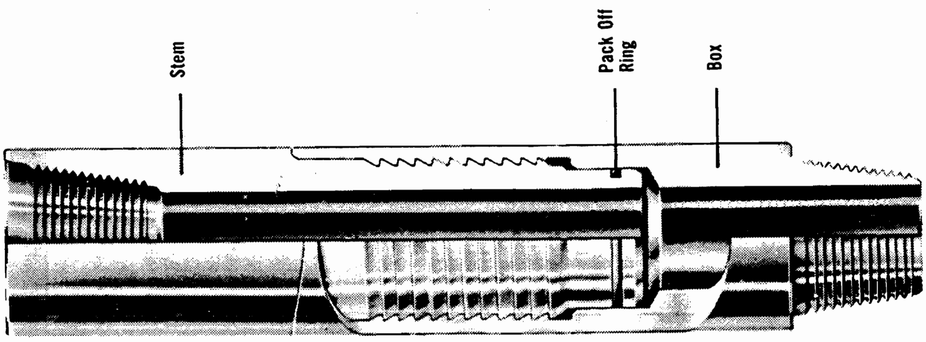
The shock sub acts as a damper which consumes vibrational energy.

Stabilizers better align the milling string with the hole and reduce the affect that buckling has on canting the mill. The idea is captured in the drawing. Regardless, of these facets, milling rotary speeds will fall into a 35 rpm to 100 rpm range.



RULES FOR SUCCESSFUL MILLING

- MUD API YIELD POINT TO 35
- HIGHEST CIRCULATION RATES
- STABILIZE THE FISHING BHA OR STRING
- SELECT WEIGHT ON THE MILL WITH TORQUE
- ROTATE AS NOT TO BURN THE TUNGSTEN CARBIDE
- ROTATE AS NOT TO VIBRATE THE MILL



Super Safety Joint – Drill Pipe

WASHOVER PIPE and ROTARY SHOES

Rotary shoes are a piece of pipe with tungsten carbide or diamonds applied to a smooth or jagged end. Washover pipe must fit within the borehole or the drift diameter of the last casing with a clearance of about 1/4 inch. It must fit over the pipe being fished with a minimum clearance of 1/8 inch. If washing over a string of pipe is a real possibility, the sizes of washover pipe, rotary shoes, and overshots best be considered prior to the drilling of the hole. A popular washover pipe connection is the Hydril Flush Joint Washover Pipe connection. It is truly flush and is a two step, three shoulder, square thread. It is cut directly into the wall of the pipe with no upsets. This connection in 7 inches washover pipe permits torsion loads to 5,000 ft-lbs.

Washover pipe is manufactured in standard API grades and proprietary grades on request. A cross-over sub is usually required to make-up with tools.

OPERATING HINTS

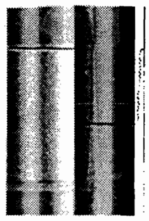
Washover pipes are run in conjunction with rotary shoes. Two problems are always prevalent while washing over. One is that the weight on the rotary shoe is difficult to ascertain because of hydraulic lift off caused by the close clearances of the rotary shoe and the washover pipe. The second, which is critical because of the low torsional strength of the connections on the washover pipe, is masking of the torque at the rotary shoe and on the washpipe by the drillpipe and any other tools in the fishing string.

A third but lesser problem to resolve is the selection of a rotary speed. Two factors governs the selection: one is the damage which can be done to the washover pipe by a high inertial load if the rotary shoe stops while rotary at a high rpm, the other is vibration. Rotary speeds between 35 and 65 rpm seem to reduce both of these effects.

The following table of washover pipe was published by HOMCO.

Hydril FJ-WP*

A special wash-pipe connection designed for higher torque strength. Develops about twice the torque load of ordinary threads. Available for most sizes and weights of pipe.



HOMCO WASHOVER PIPE TABLE

NOM. OD	NOM. ID	WEIGHT lb/ft	THREAD	MIN. HOLE ID inch	MAX. FISH OD inch
=======	========	=======	=========	=========	=========
3 1/2	2.992	9.2	Hyd FJ-WP	3 3/4	2 11/16
3 3/4	3.250	9.5	Hyd FJ-WP	4 1/8	2 7/8
4	3.240	15.7	Hyd FJ-WP	4 1/2	2 7/8
4	3.340	14	Hyd FJ-WP	4 1/2	3
4	3.428	11.6	Hyd FJ-WP	4 1/2	3 1/16
4 1/2	3.920	13.5	Hyd FJ-WP	4 3/4	3 13/16
4 1/2	3.826	16.6	Hyd FJ-WP	4 3/4	3 3/4
4 3/4	4.082	16	Hyd FJ-WP	5 3/4	3 15/16
5	4.408	15	Hyd FJ-WP	5 3/4	4 1/4
5	4.276	18	Hyd FJ-WP	5 3/4	4 1/8
5 3/8	4.938	19.5	Hyd FJ-WP	6 1/8	4 3/4
5 1/2	4.892	17	Hyd FJ-WP	6 1/8	4 3/4
5 1/2	4.778	20	Hyd FJ-WP	6 1/8	4 9/16
5 1/2	4.548	26	Hyd FJ-WP	6 1/8	4 3/8
5 3/4	5.125	18	Hyd FJ-WP	6 3/8	5
5 3/4	4.990	22.5	Hyd FJ-WP	6 3/8	4 7/8
6	5.352	20	Hyd FJ-WP	7	5 1/8
6	5.240	23	Hyd FJ-WP	7	5 1/16
6 3/8	5.625	24	Hyd FJ-WP	7	5 1/2
6 3/8	5.499	28	Hyd FJ-WP	7	4 5/16
6 5/8	5.921	24	Hyd FJ-WP	7 3/8	5 3/4
7	6.275	26	Hyd FJ-WP	7 7/8	6 1/8
7	6.184	29	Hyd FJ-WP	7 7/8	6
7 3/8	6.625	29	Hyd FJ-WP	7 7/8	6 1/2
7 1/2	6.765	28	Hyd FJ-WP	8 1/8	6 9/16
7 5/8	6.875	29.7	Hyd FJ-WP	8 1/8	6 3/4
8	7.250	31	Hyd FJ-WP	8 1/2	7 1/16
8 1/8	7.385	32	Hyd FJ-WP	8 5/8	7 1/8
8 1/8	7.185	39.5	Hyd FJ-WP	8 5/8	7
8 3/8	7.625	35.5	Hyd FJ-WP	8 7/8	7 3/8
8 5/8	7.825	36	Hyd FJ-WP	9 5/8	7 3/4
8 5/8	7.625	44	Hyd FJ-WP	9 5/8	7 3/4
9	8.150	40	Hyd FJ-WP	9 7/8	8
9 5/8	8.835	40	Hyd FJ-WP	10 5/8	8 11/16
9 5/5	8.755	43.5	Hyd FJ-WP	10 5/8	8 5/8
9 5/8	8.681	47	Hyd FJ-WP	10 5/8	8 1/2
10 3/4	9.950	45.5	Hyd FJ-WP	11 3/4	9 3/4
10 3/4	9.850	51	Hyd FJ-WP	11 3/4	9 1/2
11 3/4	10.880	54	Hyd FJ-WP	12 3/4	10 3/4
11 3/4	10.772	60	Hyd FJ-WP	12 3/4	10 9/16

PERFORATION OF PIPE

PURPOSES

Pipe is perforated for one or more of the following purposes:

- To drain mud out of the string to prevent a wet trip and to reduce the weight of the string.
- To equalize pressures and/or fluid heads.
- To balance fluid columns above a pack-off; such as, cement, mud, oil, etc.
- To circulate above a plugged point in a string.
- To squeeze cement or mud.

CASING STRENGTH

Most fishing operations will require one to four perforation holes. The holes weaken the wall of the perforated pipe; fewer holes raise internal pressures during circulation or cause the string to drain or to equalize more slowly.

In deep holes the loss of pipe strength because of perforations is critical and as few holes (often no more than one hole) should be made.

PRESSURE LOSS ACROSS PERFORATIONS

The size and number of perforations will have an affect on the pressure drop across them while injecting fluids. However, in many drilling situations part of the perforations may become plugged.

The jet nozzle equation which is used for drill bit jets often predicts the pressure drop through perforations sufficiently well for fishing.

GAS

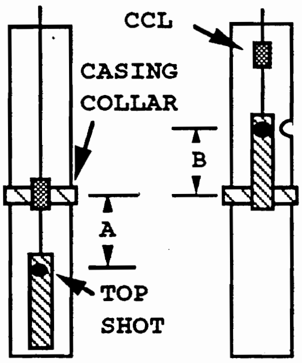
While perforating a high permeability gas zone from within an empty or partially empty wellbore, the gun can be blown up the hole. If the logging cable becomes entangled with itself or the gun, a fishing job will result. This situation can be circumvented by "shooting on the run."

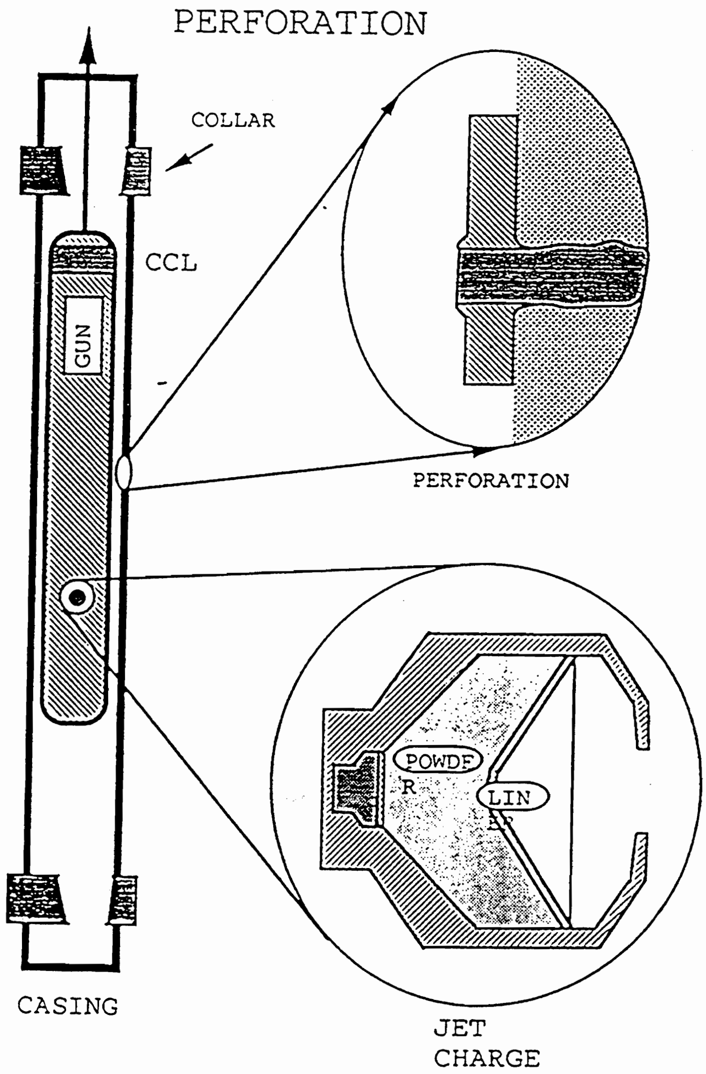
Shooting on the run is similar normal perforation with the one exception that the tool is not stopped when the top shot is at it shooting depth.

PERFORATING PROCEDURE

A perforating gun consists of the gun which contains the jet charges and a collar locator tool. The collar locator tool detects the air gap in either casing connectors or tooljoints.

After it is decided where the hole or holes are to be shot from logs, a nearby casing collar, which is easily identified by its proximity to collars on either side of it, the top shot in the perforating gun is raised (never lowered) to the pre-selected depth and fired. The distance to raise the CCL is equal to the distance the top shot is to be above the selected casing collar and the distance between the CCL and the top shot; i.e., the distance A +B in the figure. Setup is shown in the following figure.





FISHING LOGGING TOOLS AND WIRE LINE TOOLS

Wire line logging tools and other wireline tools are usually less than forty feet in length and three inches in diameter. Three prevalent causes for fishing these tools are differential wall sticking, cavings which wedge them in the hole, or junk falling into the hole on top of them. In most cases if the cable were stronger the tools could be pulled with the cable on which they were run. The usual fishing operation is running into the hole with pipe, latching onto the tool, pulling it free, and pulling it out of the hole.

PROCEDURE

If the cable is not broken and is attached to the tool, the tool can be retrieved directly with pipe, an overshot, and the cable guide method. If the line is broken and has fallen down the hole, then several runs with wireline spears may be required to fish the wire line. After the wire line is fished to the top of the stuck tool, a stout pull will pull the wire line out of the rope socket at the top of the stuck tool. Getting over the stuck tool with an overshot will most likely require a funnel type guide shoe. If the tool is cocked in an enlarged hole, it may require milling.

The cable guide method is outlined below. The cable guide method should be used when possible because the cable guides the overshot to and over the stuck tool.

STEPS in the CABLE GUIDE METHOD

1. Pull the slack out of the cable, and cut it about 3 to 4 feet above the rotary table.

2. Attach a line clamp to the cable and let the clamp rest in a guide base which has been placed on the rotary table.

3. Attach a fishing spear rope socket to the end of the line which is in the hole. Attach a rope socket with threads to the end of the cable which is in the mast.

4. Make-up a sinker bar and an female spear overshot to the threaded rope socket. The sinker bar will be used to drag the free cable in the mast down through the many stands of the fishing string. The female spear overshot can easily be latched and unlatched to the spear rope socket.

5. Pick-up the fist stand of the fishing string and screw-on an overshot which will latch onto the stuck tool when it gets to the bottom of the hole.

6. Lower the sinker bar and female spear rope socket down through the first stand of the fishing string.

7. Connect the female overshot to the male spear. This connects the two ends of the line back together; pull up on the cable, pull the

rope guide base to one side, and lower the overshot and first stand of the fishing string into the hole.

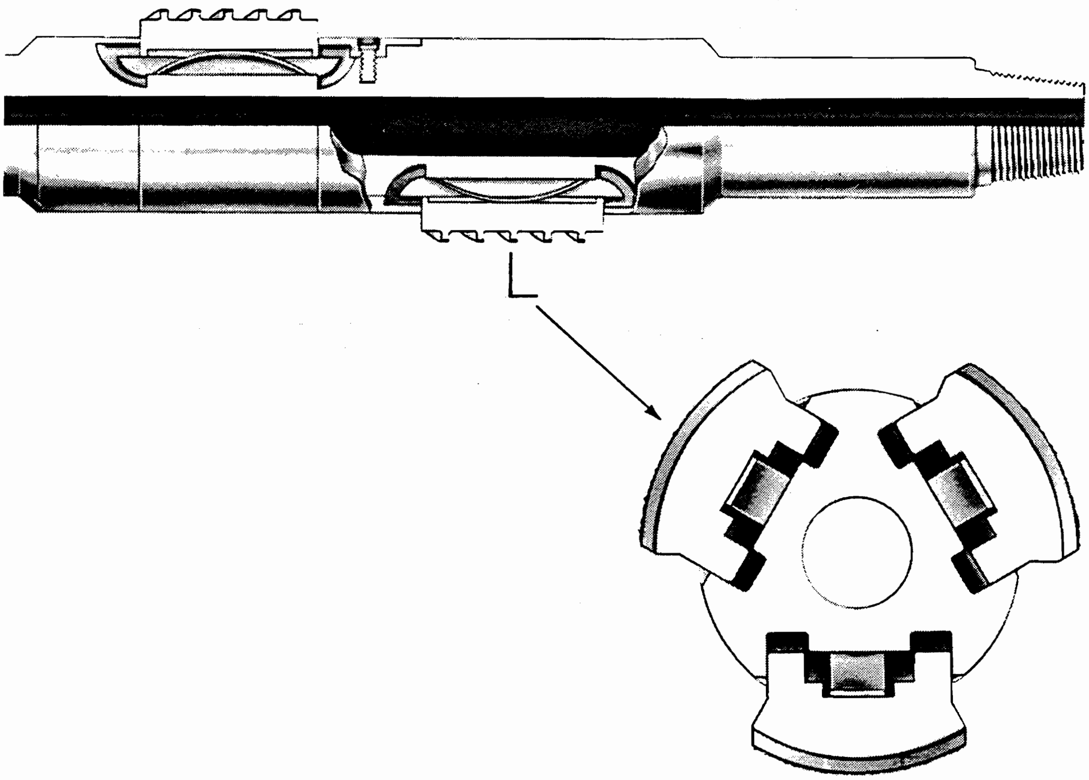
8. Pull the male spear until it is above the tooljoint of the fishing string and slide the guide base on the cable. After slacking off on the cable, disconnect the female overshot and spear.

9. Lowering of the female overshot and sinker bar is a repeat of step #6. Repeat steps #6 though #8 until the overshot screwed to the fishing string is just above the fish.

10. With the overshot just above the fish, pull any slack out of the cable, lower and turn the fishing string slowly to right to engage and allow the stuck tool to enter the grapple section of the overshot. If this operation is not done carefully, the stuck tool could be damaged.

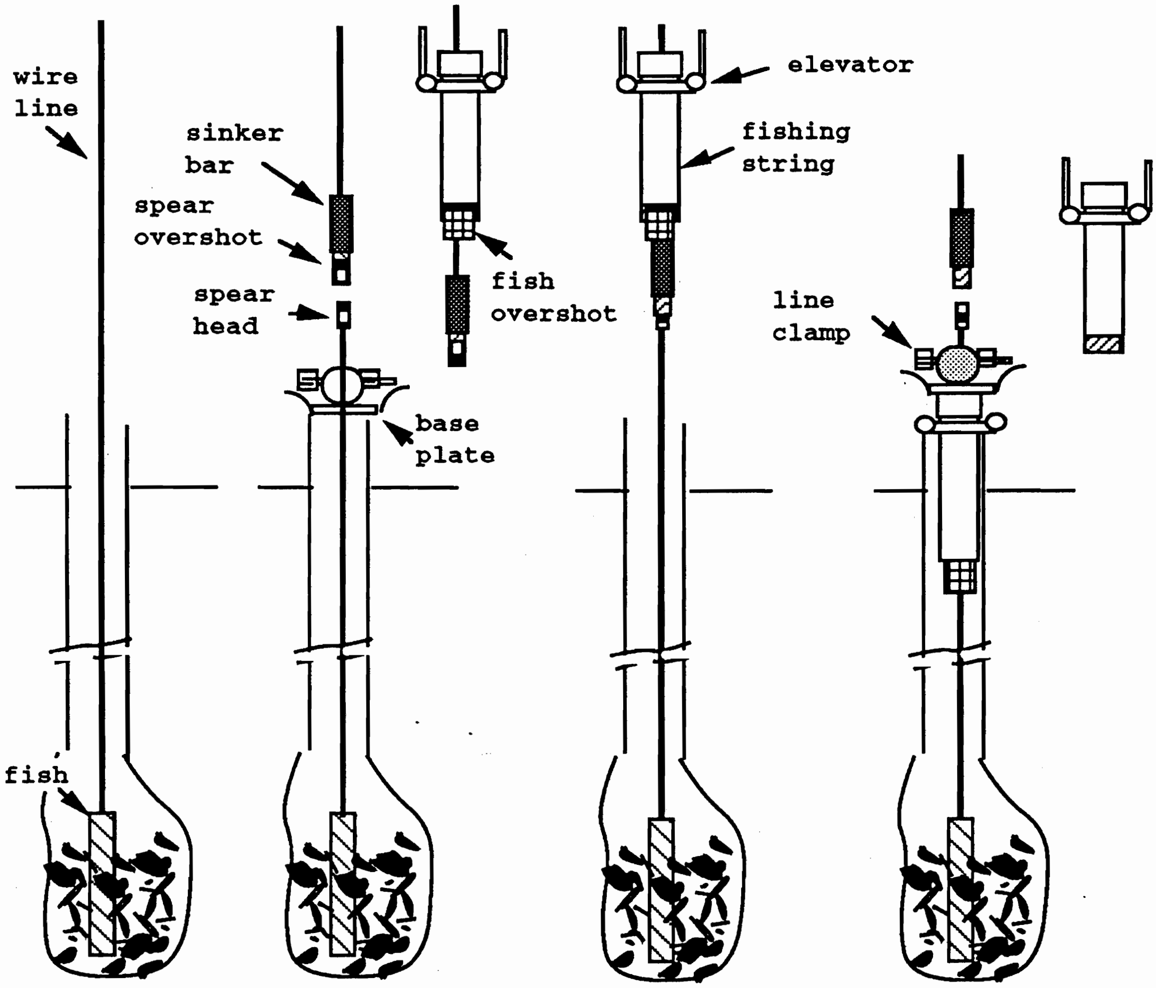
11. Pull the cable out of the rope socket attached to the stuck tool and remove the cable from the hole.

12. Free the stuck tool and pull it from the hole. The fishing string should not be turned to the left at any time. Turning to the right is not recommended because overshots release fish by turning to the right while bumping down on the fish. Chaining out of the hole is advised.

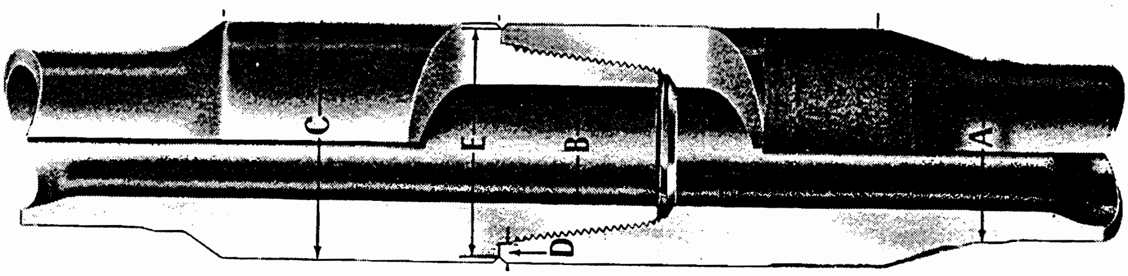


Casing Scraper

Blade Cross Section


wire line
sinker bar
spear overshot
spear head
fish overshot
elevator
fishing string
line clamp
base plate
fish

Tool Joint

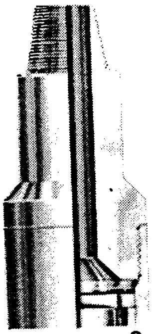

A
B
C
D
E

FISHING SMALL OBJECTS

Hand wrenches, bolts, drill bit cones, clamps, hammers, pieces of steel are examples of small objects. These may be successfully fished in open hole with the following.

- Magnets
- Jet charged cutters
- Junk baskets
- Mills

MAGNETS



If the object is lying on the bottom of the hole and not covered with cavings or can not be removed by circulation, magnets may be successful. Magnets may be run in conjunction with rotary shoes and junk baskets. Magnets may be run on wire line or on pipe. Wire line runs are faster while pipe runs allow circulation.

JET CHARGED CUTTERS

Jet charges cut steel with a high velocity hot jet gas stream. The purpose of the jet charge is to fragmentize larger objects which will not fit in a junk basket or can not be held by a magnet. The operation of the jet charged cutter is to run it to the top of the object and fire into it. Several runs may be required to cut up larger objects. To be effective the jets must be within one foot or so of the object.

JUNK BASKETS

Many types of junk baskets exist. The three categories are the plain, the reverse circulating, and the pump junk baskets. The operation of the plain junk basket is to run into the hole, circulate, cut a short core to push the object into the barrel of the junk basket, grasp the core and/or object with the fingers or grapple of the junk basket. The pipe is usually chained out of the hole (not rotated) to prevent the dropping of a partially grapped object.

Reverse circulating junk baskets are the most popular; however, they sometimes plug preventing circulation. The operation is similar to the plain junk basket with the exception that a jet fluid cross above the barrel diverts the mud stream out of the pipe, down the annulus, and then up through the barrel of the junk basket. In practice the objects are swept into the barrel of the basket by the mud. Larger objects may be cut up with a rotary shoe mill.

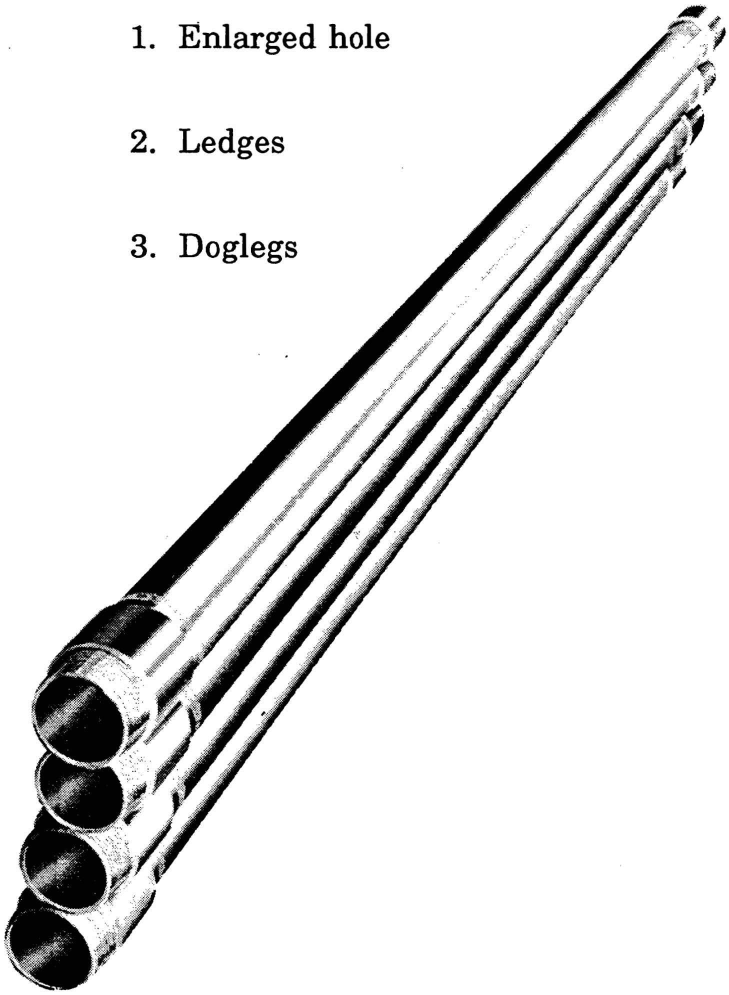
Mills are usually run below either a junk basket or magnet. Their purpose is to dislodge or fragmentize an object. Full gauge complete bottom hole coverage mills are used to grind the object into shavings and small pieces. A junk sub is usually run above the mill to catch the small pieces which are lifted off bottom by mud circulation.

FISHING DRILL COLLARS and DRILLPIPE

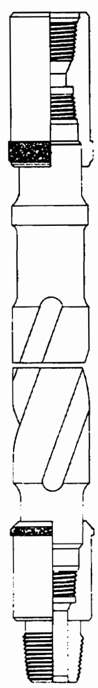
Two important problems can surface in the fishing of drill collars and drillpipe. One is that drill collars which are off bottom can fall to the bottom of the hole after they are washed over. The other is that the top of the collars, if free, can lean over in an enlarged hole or under a ledge and make it impossible to get over them with tools. The first potential problem of letting washed over collars or pipe fall to the bottom of the hole may be circumvented with a "washover back-off tool." These tools connect to the fish prior to washing over. If the fish is not free after the first washover, a string shot back-off may be performed to remove all free fish. Also, these tools permit faster recovery because collars or pipe may be washed over and pulled in one trip.

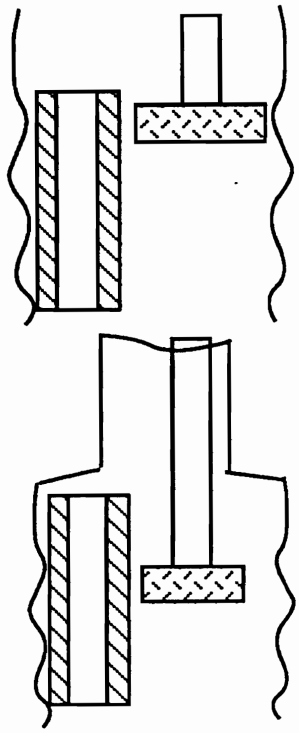
The second problem in which the collars or pipe lean over in an enlarged hole or under a restricted diameter hole may be alleviated by backing off in carefully chosen points. The factors and situations which promote trouble are the following: enlarged hole, ledges, and crooked hole (doglegs).

1. Enlarged hole
2. Ledges
3. Doglegs



DRILL COLLAR





These three potential problems may be identified by coordinating the proposed back-off depths with the well site geologist and the rig drillers. The geologist will be aware of enlarged zones and the drillers will have noted the locations of doglegs while tripping the drill string.

It is possible that the collars or pipe will be stuck in more than one zone.

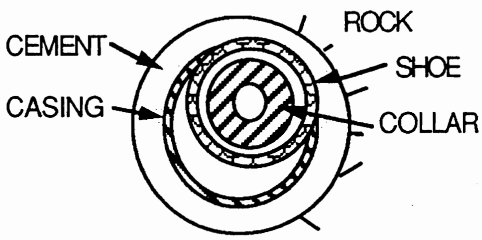
If wall sticking (differential sticking) has occurred circulation is usually possible. If cavings have stuck the string then circulation is usually not possible.

Cutting off of blades on stabilizers or other centralizing tools may allow the fish to lean to the side of the hole and make getting over the fish impossible.

A change in mud properties is almost always required to successfully fish stuck collars or drillpipe. The mud properties which may have the more significant affect are filter cake thickness and density. Higher mud densities usually better stabilize the wall of the hole. Thinner filter cakes reduce the risk of differential sticking.

Washpipe is very susceptible to wall sticking because of the small diametrical clearance between the washpipe and the hole.

Washing over drill collars which are frozen to the wall of casing is almost impossible. It is thought that the washover shoe wears or breaks up because of the vibration of cutting across the steel in casing and the cavings or cement freezing the collars.



Backing off inside casing is the safest location, but may not be the most practical.

Rules for selecting back off locations:

- At or above the free point
- Inside casing if practical
- Within gauge hole
- Within straight hole

A string shot back off may not be possible because of plugged collars or drillpipe. The plug may consist of barite, viscous mud, cement, collapsed pipe, or small pieces of equipment (rubber, steel). This situation calls for an outside cut of the pipe or collars.

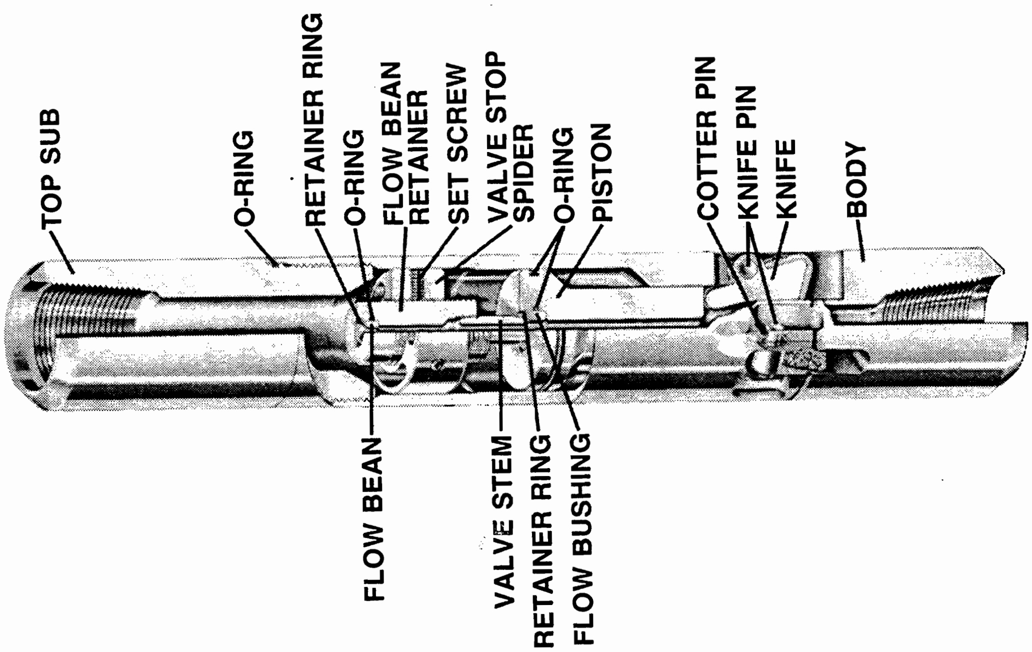
CUTTING OF TUBULARS

There are three classes of pipe cutters: jet shaped charge, acid, and mechanical. The jet shape charge is used to cut through several concentric casings with one charge. It cuts steel with a high velocity hot jet gas stream and is directed with the shape of the container of the charge. It makes a neat clean cut.

The acid cut is made with a high velocity stream of acid which impinges on the wall of the pipe. It leaves a smooth cut end.

There are two popular mechanical cutters: the inside cutter and the outside cutter. Knives are forced out or in against the wall of the pipe with either hydraulic force or set down or pull up force and rotation of the string produces the cutting action. The knives function well for single pipe wall cuts; however, in holes which have considerable drag, the knives may be damaged because of excessive but undetectable strains on the fishing string. Knives are not likely to cut or damage other pipe strings in the hole.

Holes can be cut into pipe with either of the three cutters; however, the mechanical punch may be best for thin wall pipe such as tubing. It is not likely to damage outer pipe. These are usually spring actuated.



Sidetracking

Sidetracking starts with the setting of a cement plug and the selection of the method to drill off of the cement plug. Also, there are the open hole and cased hole sidetrack.

SELECTING THE SIDETRACK METHOD

There are four popular methods of sidetracking. They named by the equipment used: the gilligan, the whipstock, the bottom hole motor with angle building device, and jetting.

THE GILLIGAN

The gilligan is the easiest and fastest sidetrack method; however, it can not be oriented in azimuth and the angle of build is not particularly predictable. The method calls for the placement of one joint of drillpipe below the BHA, a cross-over sub, and then the drill bit. One thousand to five thousand pounds are put on the drill bit for the purpose of buckling the drillpipe and making the bit drill. The canting of the bit by the buckling of the drillpipe causes the bit to build angle away from the cement hole and the old plug. Normal rotary speeds are used. After drilling off the plug about ten to fifteen feet, a regular drilling BHA is run to drill the hole.

The equation derived by Lubinski gives the minimum weight on bit which will buckle the drillpipe and cant the bit.

$$F = 80 * \left[B_f(D^2 + d^2) * (D^2 - d^2)^3 \right]^{1/3}$$

F	=	minimum force to bend the gilligan; lb
B_f	=	buoyancy factor
D	=	OD of gilligan; in
d	=	ID of gilligan; in

BUCKLING OF A GILLIGAN

A gilligan is 5" by 4.276" drillpipe. How much weight must be placed on the bit to buckle the gilligan in 11.3 ppg mud?

The buoyancy factor is

$$B_f = (1 - 11.3/65.45) \qquad = \mathbf{0.827}$$

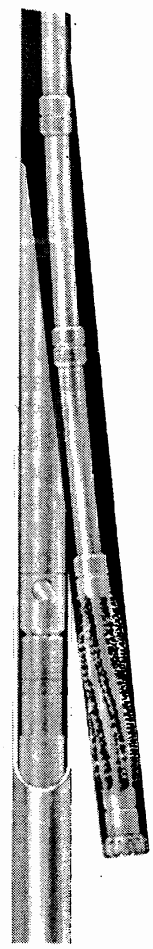
Lubinski's equation predicts the following:

$$F = 80 * \left[0.827\,(5^2 + 4.276^2) * (5^2 - 4.276^2)^3 \right]^{1/3}$$

$$F \qquad = \mathbf{1{,}771\ lb}$$

THE WHIPSTOCK

The whipstock is a steel ramp shaped device which deflects the bit off the cement plug. Two types are popular. One is called a permanent whipstock because it is cemented and left in the hole. The other is called a retrievable because after deflecting the bit off the cement plug it is pulled from the hole. The basic problems which often occur are similar for both types. The first problem is that the whipstock is not set on bottom which allows the bit to drill around the end of the whipstock and back into the old hole. This problem is usually caused by bottom fill which was not removed before setting the whipstock. The whipstock is set on the bottom of the hole with exacting precision measurements of the setting string and drilling string to which the setting string must be matched. After pulling out of the pilot hole, the whipstock may turn which prevents the re-entry into the pilot hole. The rock may be of sufficient strength that the drill bit will cut through the steel of the whipstock rather than the surrounding rock.



The procedure for running an open hole cemented whipstock for the purpose of sidetracking without azimuth orientation is the following.

1. Set a cement plug
2. Run the whipstock in the hole
3. With precision measurements set the whipstock on bottom
4. Cement the whipstock in place
5. Drill off the whipstock

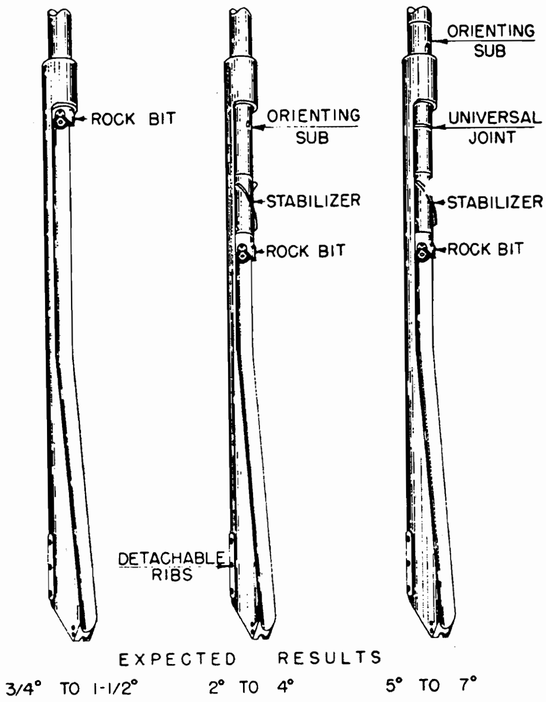
The procedure for setting a retrievable whipstock is

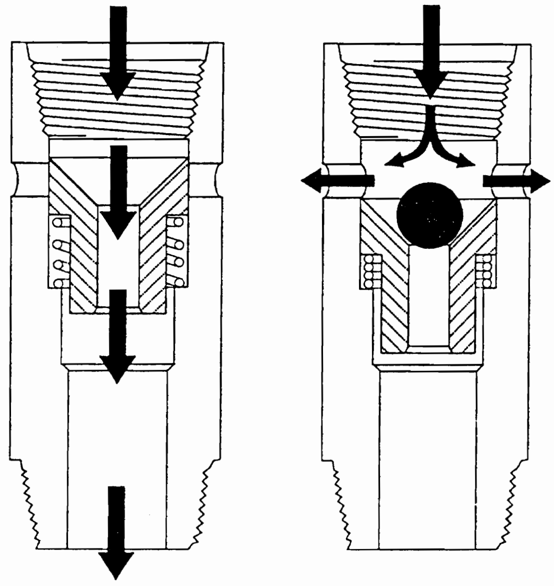
1. Set a cement plug
2. Run the whipstock into the hole with a small diameter bit
3. Set the whipstock on bottom
4. Shear the pin which connects the whipstock to the drilling string
5. Drill 10 feet of pilot hole with the bit
6. Pull the small bit and the whipstock
7. Run in the hole with a full size bit and a knuckle joint
8. Drill to about 30 feet
9. Pull the knuckle joint and bit
10. Run a full size bit and a drilling BHA

Sidetracking from a window within casing with a whipstock may be less expensive and faster in some cases. After the window is cut, a whipstock fastened to a hanger is run into the hole and the hanger is set within the casing below the window. The hanger locks the whipstock in place with the assurance of the casing. A cement plug is not required.

WHIPSTOCK

SKETCH SHOWING THE VARIOUS BOTTOM ASSEMBLIES FOR THE EASTMAN REMOVABLE WHIPSTOCK AND THE EXPECTED RESULTS. DRAG BITS, ROCK BITS OR DIAMOND BITS MAY BE USED.





Circulating Sub

The bottom hole motor and bent sub is both fast and effective. The procedure is to set a good solid cement plug, pick up a bottom hole motor with a bent sub, a full size bit, and run into the hole and begin drilling. If the rock is moderately strong, the bit may drill the cement rather than sidetracking into the adjacent rock. This problem may be alleviated by underreaming a notch into the hole such that the drill bit will have a ledge to biteinto.

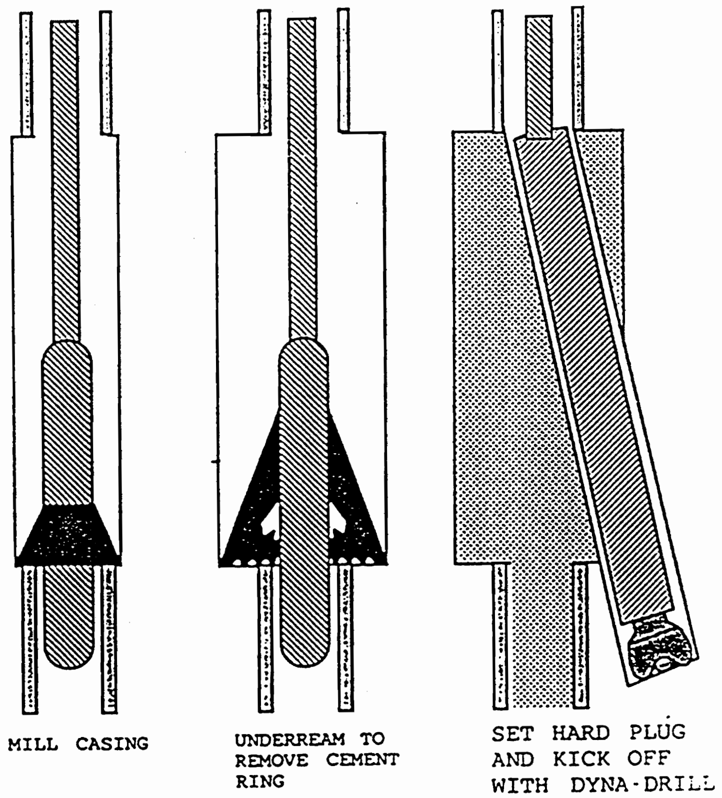
A second problem is that the binding of the bottom hole motor and the bit with the dogleg of the hole may overcome the torque output of the motor and prohibit the turning of the bit. The use of a bent housing motor or a shorter motor may ease this problem.

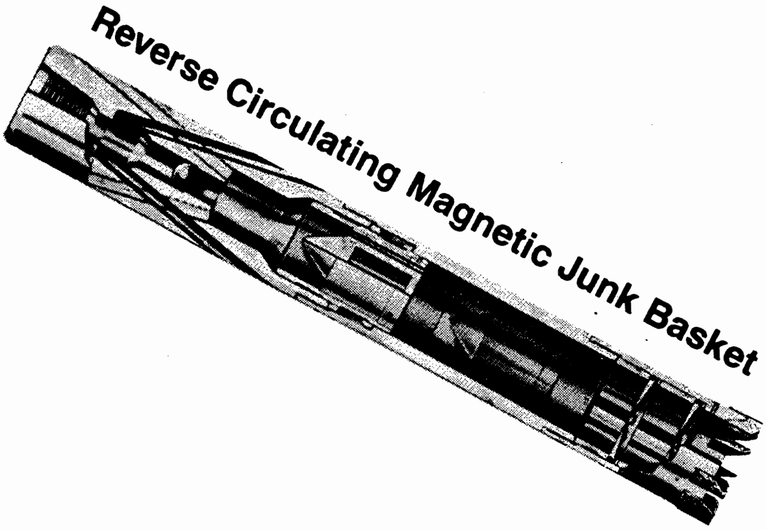
CEMENT PLUGS FOR SIDETRACKING

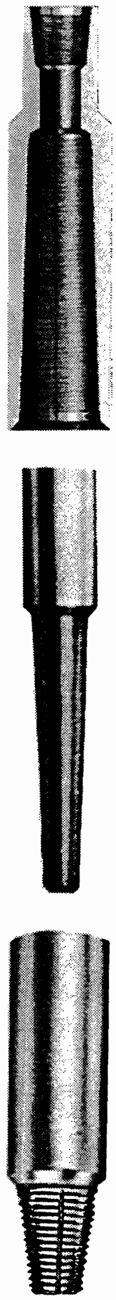
The strongest cement plug is a plug composed entirely of neat cement. However, setting a plug in deep hot wells may require silica flour to prevent retrograde of its strength. The problem in setting good cement plugs is the swapping out of the cement and the mud. Swapping out is a process in which lighter mud below the un-set heavier cement buoys a passage way up and through the cement.

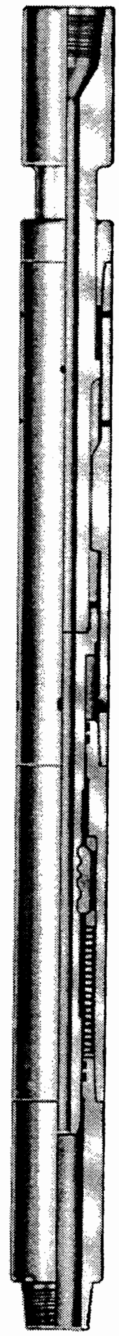
Swapping out is routinely prevented by setting a viscous bentonite pill in the hole below the bottom of the cement. The practice is to run an extra stand of drillpipe into the hole and spot 90 feet of the viscous pill. The pill can be drill mud (if water base) and 5 to 10 sacks of bentonite poured into it as it is pumped. The bentonite should hydrate and become truly viscous after it is spotted. The next steps are to pull one stand of drillpipe and then while spotting the cement plug, rotate the drillpipe. The rotation of the drillpipe will assist the cement in the displacement of the mud out of the plug zone. The mud not displaced will mix into the ~~mud~~ cement and form concrete.

CASING SIDETRACKING









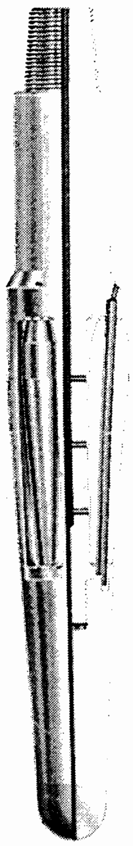
DRILLING JAR

OTHER COMMON FISHING TOOLS

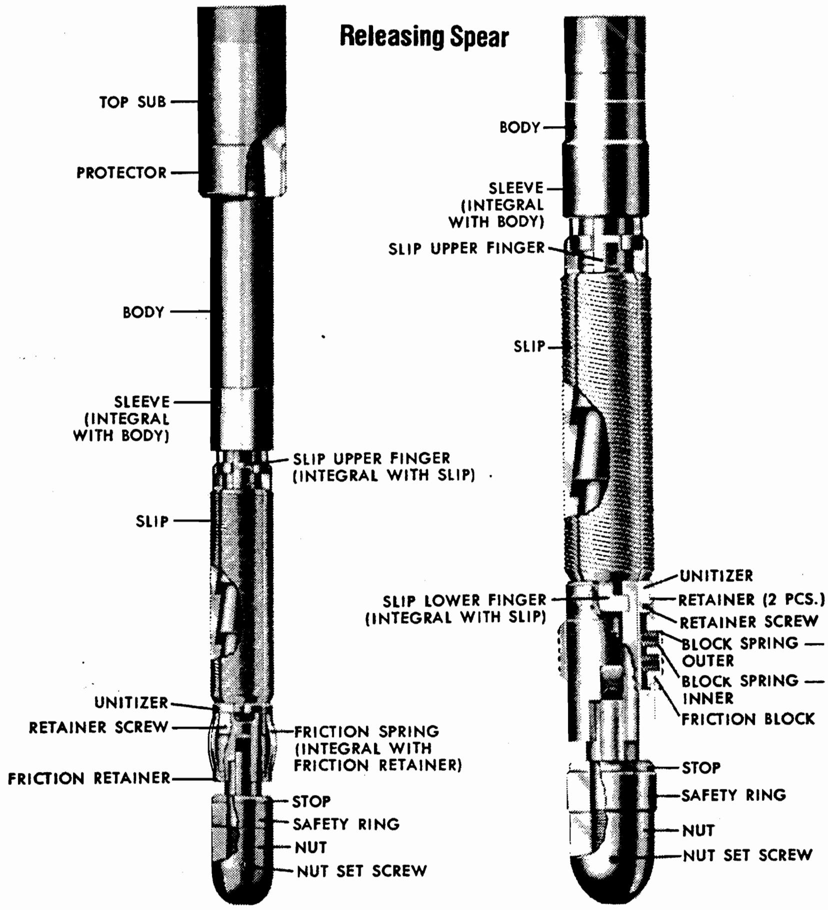
There are a wide variety of fishing tools and equipment. The following is a short description of common tools.

BAILER: A pipe with a compression actuated valve which is used to place cement in a hole or clean cavings out of the hole.

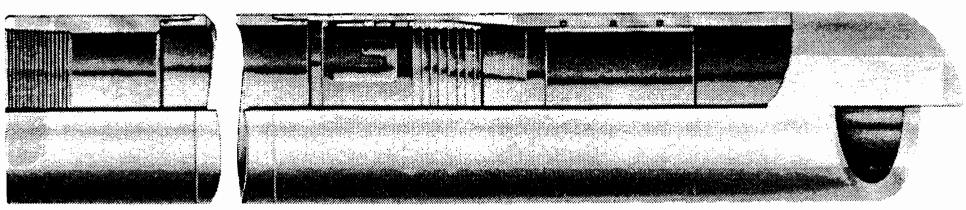
BUMPER SUB: A drill string sub with a mandrel and housing which may be extended with tension or closed with compression. Used in combination with jars.



casing roller



CASING PATCHES: Casing which is run inside a damaged casing.



DIE COLLAR: A collar which has internal die threads for screwing over and onto a fish.

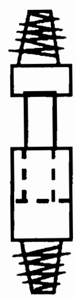
GUIDE SHOE: A pipe like tool attached to other tools to guide the tool over the fish.

IMPRESSION BLOCK: Usually made of lead. The block is pushed down on a fish to ascertain the fish's size dimensions or state of disarray.

JAR INTENSIFIER: Also called an accelerator and jar booster. The intensifier gains energy from the pull of the drillpipe and released to the jar.

JUNK BASKET: Used to collect small pieces of junk off the bottom of the borehole.

JUNK SUB: It is placed near the bit to collect small pieces of junk. It has an exterior basket.



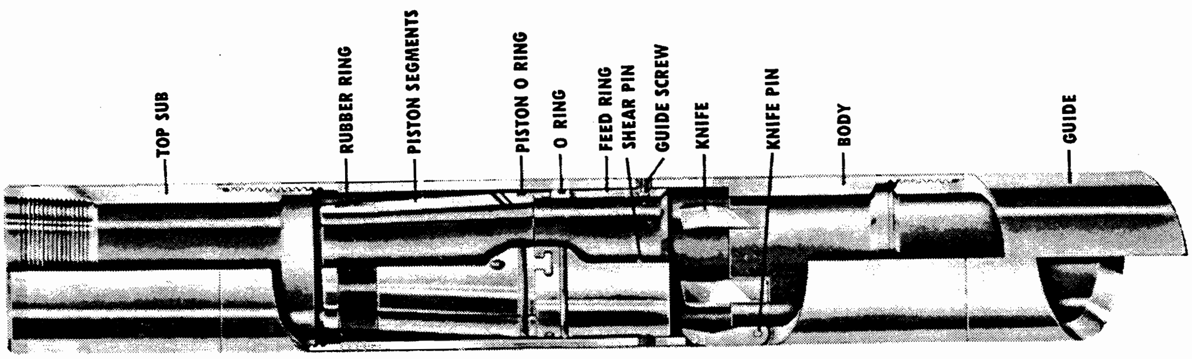
KEY SEAT REAMER: Also called a key seat wiper. It is blade tool used to cut key seats out of the wall of the borehole.

KNUCKLE JOINT: A device which permits sharp angles in a fishing string.

MAGNET: It is used to pick small pieces of steel off the bottom of the hole.

MECHANICAL JAR: Jars which have a mechanical detente for releasing the housing or mandrel within a jar.

MILL: A bit with a diamond or tungsten carbide surface for the purpose of cutting steel.



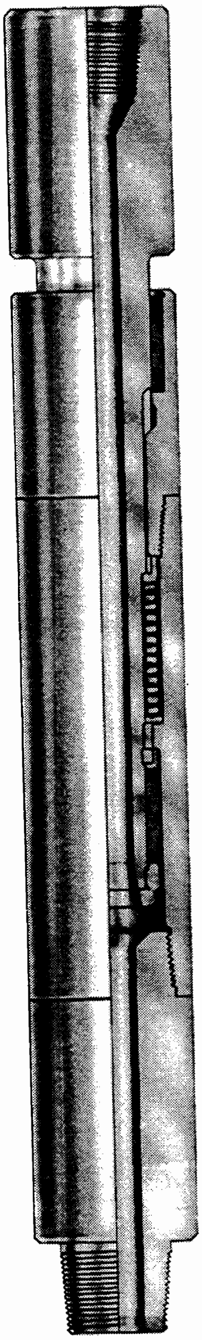
External Cutter

OIL JAR: Often called fishing jars. The detente is the passing of oil from one reservoir to another through a very small passage.

OUTSIDE CUTTER: Use to sever pipe or tools.

OVERSHOT: A hollow tool with a grapple which is used to attach a fishing string to fish and pull or push on the fish.

PACKER RETRIEVER: Use to retrieve a packer.



SHOCK TOOL

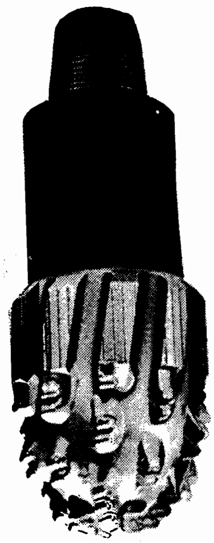
PIPE CUTTER: Any device, mechanical blades, chemical, jets, etc. which severs pipe.

PIPE ROLLER: Used to expand damaged pipe back to its original diameter.

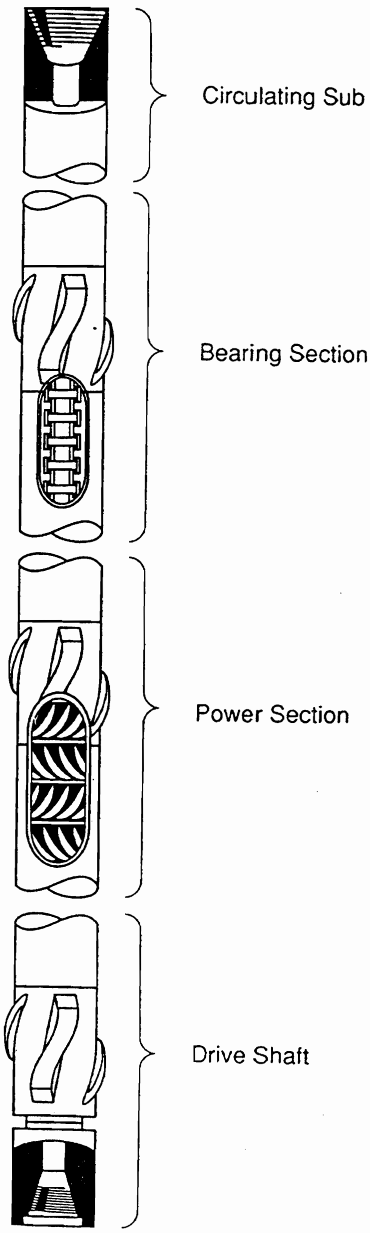
PLUGS: Placed inside pipe to block the passage of liquids or tools.

SAFETY JOINTS: Placed within fishing strings to assure that the string can be disconnected from the fish at a desired location in the string.

SINKER BARS: Sinker bars are solid steel bars used to pushed tools down the borehole.



TURBINE



SLIM LINE DRILLPIPE: HOMCO has slim line drillpipe in the sizes of 1 13/16 inches and 1 9/16 inches OD. The pipe is handled with lift plugs and elevators. It has a small external and internal upset much as large drillpipe

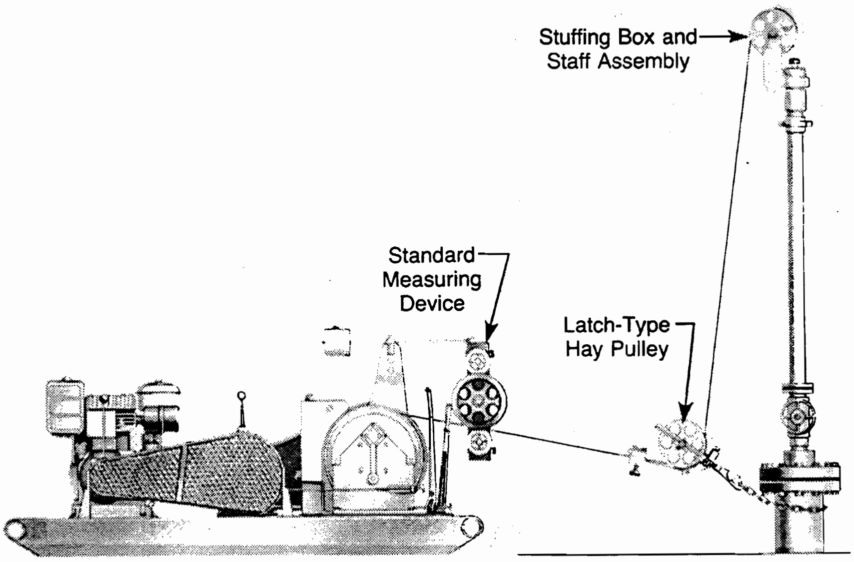
SPEAR: A tool which latches within and onto a fish for the purpose of pulling and pushing on the fish.

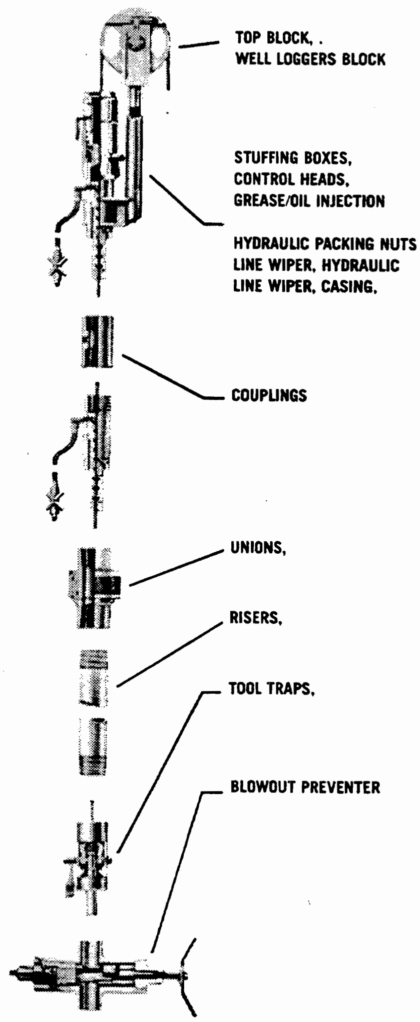
STUFFING BOX: Used to contain pressure with a pipe while running and pulling tools into and out of the pipe.

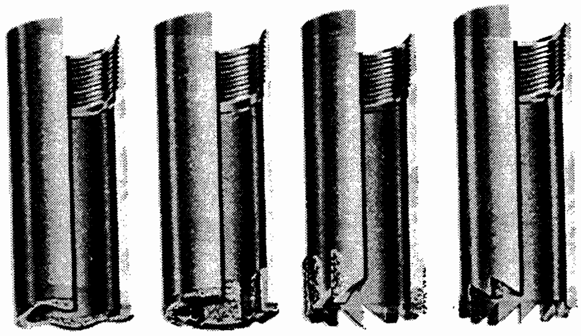
TAPER TAP: It is a threaded tapered mandrel which is screwed into a fish.

LUBRICATOR: A lubricator is composed of a stuffing box at the top of a length of pipe with a valve at its lower end and a flange or threaded connection below the valve. It is used in a three step mode. The first step is to lower a tool into the pipe of the lubricator. The second step is to attach the stuffing box to the top of the pipe. The third step is to tighten the stuffing box and open the lower valve. At this point the tool may be lowered into the hole.

WASHPIPE: A pipe which is ~~which is~~ sufficiently small to go into the borehole and large enough to go over a fish. It will have a shoe at it lower end.

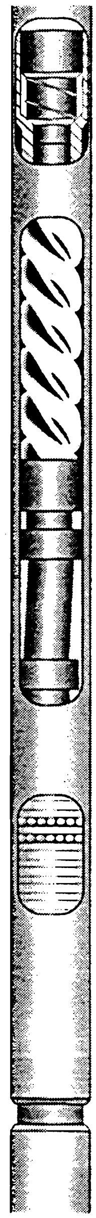






ROTARY SHOES

THE BOTTOM HOLE MOTOR (PDM)



The popular positive displacement rotary Moineau motor (PDM) is popular because it has the following operational advantages over other motors which is the turbine.

1. Its lower rotary speeds significantly lengthens bit life with both on bottom time and footage increased. Trip time is reduced.

2. Available torque is proportional to pressure drops across the motor which gives precise monitoring of torque on tools, bits, etc. by observing the standpipe pressure.

3. RPM is proportional to circulation rate allowing for precise control of bits, reamers, cutting blades, etc.

4. Thrust bearing sections allow for high set down loads without axially over loading the motor.

5. Its operation is independent of pump pressure.

6. A wide range of circulation rates are operationally possible with any one motor.

7. At maximum recommended operation torque, the volumetric efficiency (theoretical mud volume per revolution/actual mud volume per revolution) is constant near 70%.

The sections of the typical PDM are the thrust bearing, universal joint, rotor, stator, bypass valve, and connections.

1. The thrust bearing keeps weight on bit or other axial loads off the rotor. This allows for high weight on bit operation.

2. The rotor obits (wobble) within the stator and the universal joint permits this movement.

3. The rotor is a multi-lobed gear operating within a corresponding stator with one tooth difference. Both the rotor and stator are helical down their axes.
4. The number of lobes and the pitch of the spirals affect the torque and RPM parameters of the motor.
5. The stator is molded elastomer and the rotor is metal.
6. The bypass valve permits the draining and filling of the drillstring during trips. The motor will drain and fill slowly with rotation. The mud drains and fills the string through the hollow rotor in the motor.

OPERATION

The operations with a bottom hole motor are of six categories:

1. Motor check prior to tripping into the hole.
   a. drill bit jet size check
   b. operation of the bypass valve
2. While tripping into the hole, circulation should be broken every 20 to 30 stands to keep the motor running free.
3. At total depth and before drilling, circulation is initiated and set to close the automatic bypass valve and to start the motor and bit turning. This will establish the off bottom (no load) standpipe pressure. The off bottom pressure drop through the motor may be 50 to 100 psi. Agreement of the expected standpipe pressure and the actual standpipe pressure will assure that the motor is functioning correctly. This will also warm-up the oil in the bearings.
4. If orientation of the PDM and bent sub are desired, it must be recalled that the PDM imparts a left hand torque to the drill string and the amount of torque is generated by the bit, not the motor. Once drilling is established and the orientation is set, any change in the standpipe pressure will indicate a change in the torque generated at the bit and a corresponding turn of the bent sub and PDM.
5. If the bit generates excessive torque (want turn), the motor will be stalled and the standpipe gauge will be unresponsive to additional weight on bit. Pickup to restart the PDM to turning. At the stalled condition all of the mud is slipping between the rotor and stator. Re-orientation may be required because of positioning after a high torque has been applied.
6. While tripping out, the bypass valve allows the string to drain. With a bent sub, rotating out with the rotary table may be hazardous.

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