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Petroleum and natural gas industries — Drilling and production equipment — Part 1: Drill stem design and operating limits

Élément introductif — Élément central — Partie 1: Titre de la partie

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Foreword

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ISO 10407-1 was prepared by Technical Committee ISO/TC 67, *Materials, equipment and offshore structures for petroleum, petrochemical and natural gas industries*, Subcommittee SC 4, *Drilling and production equipment*.

ISO 10407 consists of the following parts, under the general title *Petroleum and natural gas industries — Drilling and production equipment*:

— *Part 1: Drill stem design and operating limits*

— *Part 2: Inspection of drilling equipment*

Introduction

The function of this part of ISO 10407 is to recommend design criteria for the drill stem and to define the operating limits of the equipment used to make up the drill stem. It contains formulas and figures to aid in the design and selection of equipment to meet a specific drilling condition.

A major portion of this part of ISO 10407 is based upon API Recommended Practice 7G, *Recommended practice for drill stem design and operating limits,* Sixteenth edition, August 1998.

In this International Standard, data are expressed in the International System (SI) of units in tables in Annex A and in the United States Customary (USC) system of units in tables in Annex B. Data within the text of this International Standard is expressed in SI units followed by USC units in brackets.

Users of this part of ISO 10407 should be aware that further or differing requirements may be needed for individual applications. This part of ISO 10407 is not intended to inhibit a vendor from offering, or from the purchaser from accepting, alternative equipment or engineering solutions for the individual application. This may be particularly applicable where there is innovative or developing technology. If an alternative is offered, the vendor should identify any variations from this part of ISO 10407 and provide details.

Petroleum and natural gas industries — Drilling and production equipment — Part 1: Drill stem design and operating limits

1 Scope

1.1 Coverage

This part of ISO 10407 provides design guidelines and operating limits for steel drill stem components including kelly valves, kellys, drill pipe, heavy weight drill pipe, drill collars, rotary subs, bit connections and rotary shouldered connections.

1.2 Section coverage

Clauses 4, 5, 6, and 7 provide procedures for use in the selection of drill string members. Clauses 8, 9, 10, 11, 12, and 15 are related to operating limitations that can reduce the normal capability of the drill string. Clause 13 contains a classification system for used drill pipe and used tubing work strings, and identification and inspection procedures for other drill string members. Clause 14 contains statements regarding welding on down hole tools. Clause 16 contains a classification system for rock bits.

2 Conformance

2.1 Units of measurement

In this International Standard, data are expressed in both the International System (SI) of units and the United States Customary (USC) system of units. For a specific order item, it is intended that only one system of units be used, without combining data expressed in the other system.

Product manufactured to specifications expressed in either of these unit systems shall be considered equivalent and totally interchangeable. Consequently, compliance with the requirements of this International Standard as expressed in one system provides compliance with requirements in the other system.

For data expressed in the SI, a comma is used as the decimal separator and a space as the thousands separator. For data expressed in the USC system, a dot is used as the decimal separator and a space as the thousands separator.

Data within the test of this International Standard are expressed in SI units followed by data in USC units in parenthesis.

2.2 Tables and figures

Separate tables for data expressed in SI units and in USC units are given. The tables containing SI units are given in Annex A and the tables containing data in USC units are given in Annex B. For a specific order item, only one unit system shall be used.

Figures containing SI units are provided in Annex C and figures containing USC units are provided in Annex D.

3 Normative references

The following referenced documents are indispensable for the application of this document. For dated references, only the edition cited applies. For undated references, the latest edition of the referenced document (including any amendments) applies.

3.1.1 ISO

ISO 6506-1, *Metallic materials —Brinell Hardness test — Part 1:Test method*

ISO 9303, *Seamless and welded (except submerged arc-welded) steel tubes for pressure purposes — Full peripheral ultrasonic testing for the detection of longitudinal imperfections*

ISO 9304, *Seamless and welded (except submerged arc-welded) steel tubes for pressure purposes — Eddy current testing for the detection of imperfections*

ISO 9402, *Seamless and welded (except submerged arc-welded) steel tubes for pressure purposes — Full peripheral magnetic transducer/flux leakage testing of ferromagnetic steel tubes for the detection of longitudinal imperfections*

ISO 10407-2, *Petroleum and natural gas industries – Drilling and production equipment – Part 2: Inspection of drilling equipment*

ISO 10424-1, *Petroleum and natural gas industries – Drilling and production equipment – Part 1 - Specification for rotary drill stem elements*

ISO 10424-2, *Petroleum and natural gas industries – Drilling and production equipment - Threading, gauging and testing of rotary shouldered thread connections*

ISO 13665, *Seamless and welded steel tubes for pressure purposes — Magnetic particle inspection of the tube body for the detection of surface imperfections*

ISO/TR 9769, *Steel and iron — Review of available methods of analysis*

3.1.2 API

API Spec 7, *Rotary drilling equipment*

API RP 7A1, *Recommended practice for testing of thread compound for Rotary Shouldered Connections*

API RP 7G, *Recommended practice for drill stem design and operating limits*

API Bul 5T1, *Imperfection terminology*

3.1.3 ASTM

ASTM A370, *Standard test methods and definitions for mechanical testing of steel products, Annex II, Steel tubular products*

ASTM A751, *Methods, practices and definitions for chemical analysis of steel products*

ASTM A941, *Terminology relating to steel, stainless steel, related alloys and ferro-alloys*

ASTM E10, *Standard method of test for Brinell hardness of metallic materials*

ASTM E23, *Methods for notched bar impact testing of metallic materials*

ASTM E165, *Standard practice for liquid penetrant inspection method*

ASTM E213, *Standard practice for ultrasonic examination of metal pipe and tubing*

ASTM E309, *Standard practice for eddy-current examination of steel tubular products using magnetic saturation*

ASTM E570, *Standard practice for flux leakage examination of ferromagnetic steel tubular products*

ASTM E709, *Standard practice for magnetic particle examination*

3.1.4 NACE

NACE MR 01 75, *Petroleum and natural gas industries-materials for use in H2S-containing environments in oil and gas production* – Part 1, Part 2 and Part 3

4 Terms, definitions, and abbreviations

4.1 Terms and definitions

For the purposes of this document, the following terms and definitions apply.

4.1.1
addendum
distance between the radius of the pitch circle and the crest of a thread

4.1.2
alpha-numeric code
system of 3 numbers and 1 letter used to identify the design characteristics of a drill bit

4.1.3
API sizes
tool joint size and styles and drill pipe size, weight and strength grades defined by API Spec 7 or API Spec 5D

4.1.4
axial stress
stress produced by a load applied along the central axis of a straight bar of uniform cross section of homogeneous material

4.1.5
axisymmetric
exhibiting similarity in all planes about a central axis

4.1.6
beachmark
progression marks on a fatigue fracture surface that indicate successive positions of the advancing crack front

4.1.7
belled box
increase in swelled box OD adjacent to the make-up shoulder caused by exceeding box torsional strength

4.1.8
bending strength ratio
ratio of the section modulus of the box at its last engaged thread to the pin at its last engaged thread

4.1.9
bevel
conical surface machined 45° from the axis at the juncture of the tool joint make-up shoulder and OD

4.1.10
bevel diameter
outer diameter of the contact face of the rotary shouldered thread connection

4.1.11
bit sub
sub with two box connections used to connect the bit to the drill string

4.1.12
bore back
box stress relief feature in which the unengaged threads at the back of the box vanish into a cylindrical section

4.1.13
bottleneck
reduction in diameter for transition from one size drill stem component to another

4.1.14
bottom hole assembly
section of the drilling assembly from the bit to, but not including, the drill pipe

4.1.15
break-out
connection break-out
loosening a rotary shouldered connection

4.1.16
brinell hardness
hardness determined by a test in which a steel ball 1 cm in diameter is pressed into the material with a 3000 kg force

NOTE See BHN.

4.1.17
brinell hardness number
number computed by dividing the applied load by the surface area of the indention in Brinell hardness tests

4.1.18
brittle
separation of a solid accompanied by little or no macroscopic plastic deformation

4.1.19
bushing
kelly bushing
device that transfers rotary motion from the rotary table to the kelly

4.1.20
casing-friendly
non-damaging to casing

4.1.21
chemical reaction
change in which a substance (or substances) is changed into one or more new substances

4.1.22
chromium carbide hard-banding
hard-banding consisting primarily of chromium carbide Cr_3C_2

4.1.23
circulation
flow of drilling fluid through the drill pipe

4.1.24
Class 2
third in a hierarchy of three drill pipe wear classifications, preceded by new and premium, defined in ISO 10407-2 and API RP7G

4.1.25
coefficient of friction
ratio of the frictional force between two bodies in contact, parallel to the surface of contact, to the force, normal to the surface of contact with which the bodies press against each other

4.1.26
cold rolling
process in which a shaped roller is used to achieve cold working of a material

4.1.27
cold working
plastic deformation, to induce strain hardening, of the thread roots of a rotary shouldered connection

4.1.28
corrosion
deterioration of a material by chemical or electrochemical reaction with its environment

4.1.29
critical hole angle
angle at which gravity alone will no longer cause drill pipe to move toward the bottom of the hole

4.1.30
critical rotary speed
rotary speed at which harmonic vibrations occur

4.1.31
decarburization
loss of carbon from the surface of a ferrous alloy as a result of heating in a medium that reacts with the carbon at the surface

4.1.32
dedendum (thread)
difference between the radius of the pitch circle and the root of a thread

4.1.33
diamond drill bits
bits that used industrial grade synthetic diamonds as the cutting structure

4.1.34
directional wellbore
wellbore that is not vertical

4.1.35
dogleg
sharp change of direction in a wellbore

NOTE Applied also to the permanent bending of wire rope or pipe.

4.1.36
double-shouldered connections
connections with torque shoulders at each end of the threaded section

4.1.37
drift
gauge used to check functional internal diameter

4.1.38
drift diameter
maximum diameter tool that can pass through the bore of a drill string member

4.1.39
drill collar
thick-walled cylindrical member with rotary shouldered connections used to provide stiffness to the drill string and supply weight to the bit

thick-walled pipe to provide stiffness and concentration of weight at or near the bit

4.1.40
drill pipe
upset seamless steel pipe with weld-on tool joints

4.1.41
drill pipe body
seamless steel pipe with upset ends used to weld to tool joints

4.1.42
drill stem
drill string
drilling assembly from the swivel to the bit

4.1.43
drill string element
drill pipe with tool joints attached

4.1.44
electrochemical corrosion
corrosion that is accompanied by a flow of electrons between cathodic and anodic areas on metallic surfaces

4.1.45
electrochemical reaction
flow of electrons between cathodic and anodic areas on metallic surfaces

4.1.46
endurance limit
maximum stress below which a material can endure an infinite number of stress cycles without fatigue failure

4.1.47
equivalent stress
uniaxial stress that is equivalent to the triaxial stress condition

4.1.48
erosion
destruction of metals or other materials by the abrasive action of moving fluids, usually accelerated by the presence of solid particles or matter in suspension

4.1.49
failure
improper performance of a device or equipment that prevents completion of its design function

4.1.50
Farr's modified screw jack formula
formula used to calculate the make-up torque and the yield torque of a rotary shouldered connection

4.1.51
fatigue
phenomenon leading to fracture under repeated or fluctuating stresses having a maximum value less that the tensile strength of the material

4.1.52
fatigue crack
crack resulting from fatigue

4.1.53
fillet
radius imparted to inside meeting surfaces

4.1.54
finite element analysis
stress analysis method (as it applies to this part of ISO 10407) where the part or system is broken into elements interconnected at discrete nodal points

4.1.55
fishing
removing a twisted drill string from the hole

4.1.56
fishing neck
region with a reduced diameter at or near the upper end of a drill string member which fishing tools can grab

4.1.57
forging, verb
plastically deforming metal, usually hot, into desired shapes with compressive force

4.1.58
fracture
act, process or state of being broken

4.1.59
friction factor
make-up torque multiplier used when the thread compound coefficient of friction is not 0,08

4.1.60
gall
damage on contacting surfaces caused by localized friction welding of high spots

4.1.61
galled threads
thread damage caused by galling

4.1.62
galling
localized welding between two surfaces caused by excessive friction and high spots

4.1.63
gouge
elongated grooves or cavities caused by mechanical removal of metal

4.1.64
hard banding
hard facing
sacrificial or wear resistance material applied to component's surface to prevent wear of the component

4.1.65
harmonic vibrations
periodic motion that is a sinusoidal function of time

4.1.66
heat checking
formation of surface cracks formed by the rapid heating and cooling of the component

4.1.67
heat-treat
heating and cooling a solid metal or alloy in such a way as to obtain desired conditions or properties

4.1.68
heavy weight drill pipe
non-API design pipe with thick wall used in the transition zone to minimize fatigue and as bit weight in directional wells

4.1.69
helically buckle
buckle in three dimensions

4.1.70
hook load
maximum axial tension load on the drill string at the uppermost joint of pipe

NOTE See tensile capacity.

4.1.71
inclusion
foreign material or non-metallic particles entrapped within the metal during solidification

4.1.72
insert bit
bits that use sintered tungsten carbide shapes as the cutting structure

4.1.73
inspection
process of measuring, examining, testing, gauging, or otherwise comparing the unit of product with the applicable requirements

4.1.74
jack-screw
jack operated by a screw mechanism

4.1.75
jarring
applying axial impact loads with a jar to a stuck drill string

4.1.76
jars
mechanical or hydraulic devise used in the drill stem to deliver an impact load to another component of the drill stem, especially when that component is stuck

4.1.77
kelly
square- or hexagonal-shaped steel pipe connecting the swivel to the drill string

NOTE The kelly moves through the rotary table and transmits torque to the drill string.

4.1.78
kelly saver sub
sub positioned between the kelly and drill pipe to limit wear on kelly threads from adding and removing drill pipe from drill string

4.1.79
keyseat wiper
mechanical devise used in the drill stem to ream and enlarge a restriction in the well bore that is often referred to as a keyseat

4.1.80
last engaged thread
last thread on the pin engaged with the box or the box engaged with the pin

4.1.81
lead
distance the pin will advance in the box in one complete turn

4.1.82
low torque feature
design feature where the box counterbore is increased in size thus reducing the cross sectional area of the sealing face to increase the compressive stress in the sealing face that is induced by make-up torque

4.1.83
lower kelly valve
full-opening valve installed immediately below the kelly, with outside diameter equal to the tool joint outside diameter

4.1.84
macroscopic
large enough to be observed by the naked eye

4.1.85
make-up shoulder
torque shoulder or shoulder
sealing shoulder on a rotary shouldered connection

4.1.86
make-up torque
torque applied to tighten rotary-shouldered connections

4.1.87
mean stress
algebraic mean of the maximum and minimum values of a periodically varying stress

4.1.88
mechanical property
property that involves a relationship between stress and strain or reaction to an applied force

4.1.89
microscopic
of extremely small size

4.1.90
normalize
heating a ferrous alloy to a suitable temperature above the transformation range and then cooling in air to a temperature substantially below the transformation range

4.1.91
oil muds
type of drilling fluid where oil is the continuous phase and water is the dispersed phase

4.1.92
overshot
fishing tool attached to drill pipe or tubing and lowered into the wellbore over the outside wall of the pipe or sucker rods lost or stuck in the wellbore

NOTE A friction device in the overshot firmly grips the pipe or sucker rods allowing them to be pulled from the hole.

4.1.93
pitch circle (thread)
circle concentric with the thread axis with a diameter in which the width of the thread is equal to the distance between two adjacent threads

4.1.94
pitch diameter
diameter of the pitch circle

4.1.95
plain end
pipe with no upset

4.1.96
premium class
second in a hierarchy of three drill pipe wear classifications, preceded by "new" and followed by "Class 2", defined in ISO 10407-2 and API RP7G

4.1.97
prima facie
evidence adequate to establish a fact or raise a presumption of fact without further examination

4.1.98
quench
rapid cooling during heat treatment

4.1.99
range (drill pipe length)
drill pipe length specification defined in ISO11961(?) and API Spec 5D

4.1.100
reamer
mechanical device used in the bottom hole assembly to maintain hole gauge if the drill bit becomes undersized due to wear

4.1.101
relief groove
stress relief feature for rotary shouldered connection that consists of the use of a groove to remove the unengaged threads of the pin or box to make the connection more flexible and reduce the likelihood of fatigue cracking

4.1.102
roller cone bit
bits that use cones rotating around shafts to reduce torque as the bit turns

4.1.103
rotary shouldered connection
two member threaded connection with sealing shoulders

4.1.104
rotary subs
short drill stem members with different rotary shouldered connections at each end for the purpose of joining unlike members of the drill stem

4.1.105
section modulus (Z)
measure of the capacity of a section to resist any bending moment to which it is subjected

4.1.106
shear strength
stress required to produce fracture in the plane of cross section, calculated by dividing the force required to produce shearing by original cross-sectional area of a section

4.1.107
shot peening
cold working the surface of a metal by metal-shot impingement

4.1.108
slip area
distance of 122 cm (48 in) along the pipe body from the juncture of the tool joint OD and the elevator shoulder

4.1.109
spall
spalling
cracking or flaking of particles out of a surface

4.1.110
split box
box with axial crack caused by belling

NOTE See belled box.

4.1.111
stabilizer
member of the drill stem assembly used to centralize or control the direction of the bottom hole assembly

4.1.112
steel tooth bit
milled tooth bit
bits that use milled and hardened teeth as the cutting structure

4.1.113
strain hardening
increase in hardness and strength caused by plastic deformation

4.1.114
stress concentrator
feature on a part, usually abrupt changes in shape, at which a stress distribution has high localized stresses

4.1.115
stress corrosion cracking
failure by cracking under combined action of corrosion and stress, either external (applied) stress or internal (residual) stress

4.1.116
stress-relief feature
modification performed on rotary shouldered connections which removes the unengaged threads of the pin or box to make the connection more flexible and reduce the likelihood of fatigue cracking

NOTE This process makes the joint more flexible and reduces the likelihood of fatigue cracking in this highly stressed area.

4.1.117
stretched pin
threaded portion of a tool joint pin which has been plastically deformed by high stresses in the axial direction

4.1.118
swivel
device at the top of the drill stem which permits simultaneous circulation and rotation

4.1.119
sulphide stress cracking
brittle failure by cracking under the combined action of tensile stress and corrosion in the presence of water and hydrogen sulphide.

4.1.120
taper
change in diameter over a given length

4.1.121
temper
reheating a quench-hardened or normalized ferrous alloy to a temperature below the transformation range and then cooling to soften and remove stress

4.1.122
tempered
reheating to some temperature below the critical temperature followed by any desired rate of cooling

4.1.123
tensile capacity
axial load a drill string member can withstand without permanent deformation

4.1.124
tensile yield strength
stress at which a material exhibits a specified deviation from proportionality of stress and strain

4.1.125
thread compound
lubricant used on rotary shouldered connections to add lubricity and protect the mating surfaces from galls during make-up

4.1.126
thread form
family of threads defined by the dimensions of its features

4.1.127
thread protector
cap (for pins) or plug (for boxes) placed on rotary shouldered connections to protect the threads and shoulders while moving or during pick-up and lay-down operations

4.1.128
tolerance
permissible variation

4.1.129
tong space
cylindrical, outside, surface of a tool joint or other threaded drill string member

4.1.130
tool joint box
box
tool joint with internal threads

4.1.131
tool joint pin
pin
tool joint with external threads

4.1.132
tool joint
threaded connection, welded to the drill pipe body, for coupling lengths of drill pipe

4.1.133
top drive
electrical or hydraulic device used at the top of the drill string to provide rotation to the drill stem

4.1.134
torque capacity
torsional load a string member can withstand without permanent deformation

4.1.135
torque measuring device
gauge used to monitor the amount force used to make-up a rotary shouldered connection

4.1.136
torsional balance
point where the torsional area of the box is equal to the torsional area of the pin

4.1.137
tortuosity
spiralling of a drilled hole

4.1.138
transformation temperature (transformation range)
during the heating of steel, the temperature at which a change in phase occurs

4.1.139
tungsten-carbide
hard, wear resistant compound of tungsten and carbon and other bonding materials and used in hardfacing and drilling tools

4.1.140
twist-off
parting of the drill stem, from fatigue or other events, that results in a fishing job

4.1.141
ultimate tensile strength
tensile stress at which a body will fracture, or continue to deform under a decreasing load

4.1.142
upper kelly cock
valve immediately above the kelly that can be closed to confine pressures inside the drill string

4.1.143
upset
forged end of a drill pipe tube used to increase wall thickness

4.1.144
washover
fishing operation to go over the outside of drill pipe or tubing that is stuck in the borehole

NOTE This (washover) is accomplished by externally removing debris, formation, cement, junk or anything else that causes the string to become stuck in the wellbore.

4.1.145
washover pipe
washpipe
specialty casings used in washing over a stuck downhole assemble.

4.1.146
working gauges
gauges used by manufacturers or machine shops for measuring threads or other close tolerance features

4.1.147
working pressure
maximum pressure to which a part (drill stem member) will be subjected

4.1.148
working temperature
maximum temperature to which a part (drill stem member) will be subjected

4.2 Abbreviations

For the purposes of this document, the following abbreviations apply.

7,842 specific gravity of steel

65,44 density of steel, lbs/gal

489,5 density of steel, lbs/cu ft

API American Petroleum Institute

BHA bottom hole assembly

BHN Brinell hardness number

BSR bending strength ratio

DF design factor

DR design ratio, DF or K_{SCF}, which ever is greater

F_{AT} maximum allowable tension load in the drill pipe, lbs

F_{ABT} maximum allowable buoyed tension load in the drill pipe, lbs

F_{ABT1} maximum allowable buoyed tension load in the first section of drill pipe above the drill collars, lbs

F_{ABT2} maximum allowable buoyed tension load in the second section of drill pipe above the drill collars, lbs

F_{ABT3} maximum allowable buoyed tension load in the third section of drill pipe above the drill collars, lbs

F_{ABT4} maximum allowable buoyed tension load in the fourth section of drill pipe above the drill collars, lbs

F_{DR} maximum allowable buoyed tension load based on the design factor or the slip crushing constant, whichever is applicable, lbs

F_{MOP} maximum allowable buoyed tension load based on the Margin of Overpull, lbs

F_Y pipe body yield strength

NOTE See Table A.2 (Table B.2) for new drill pipe, Table A.4 (Table B.4) for premium class drill pipe, and Table A.6 (Table B.6) for Class 2 drill pipe, lbs.

H-90 H-90 style of thread design

IADC International Association of Drilling Contractors

ID inside diameter

K_{BF} buoyancy factor

K_{SCF} slip crushing factor, the ratio of S_H/S_T

L_{DP1} maximum length of the first section of drill pipe above the drill collars, feet

L_{DP2} maximum length of the second section of drill pipe above the drill collars, feet

L_{DP3} maximum length of the third section of drill pipe above the drill collars, feet

L_{DP4} maximum length of the fourth section of drill pipe above the drill collars, feet

L_S length of the slips, inches

MOP margin of overpull, lbs

NC API number style of thread design

OD outside diameter

PAC Phil A. Cornell style of thread design

S_H hoop compressive stress in the drill pipe that is surrounded by the slips, psi

S_T axial tension stress in the pipe hanging below the slips, psi

$W_{C\&B}$ combined weight of the drill collars and bottom hole assembly, lbs

W_{DP1} approximate weight per foot of the first section of drill pipe above the drill collars, adjusted for the weight of the upsets and tool joints, lb/ft

NOTE See column 3 of Tables 1.10 and 1.11 of API Bulletin 7G.

W_{DP2} approximate weight per foot of the second section of drill pipe above the drill collars, adjusted for the weight of the upsets and tool joints, lb/ft

W_{DP3} approximate weight per foot of the third section of drill pipe above the drill collars, adjusted for the weight of the upsets and tool joints, lb/ft

W_{DP4} approximate weight per foot of the fourth section of drill pipe above the drill collars, adjusted for the weight of the upsets and tool joints, lb/ft

$W_{\#F}$ weight of the drilling fluid, lbs/cubic foot

$W_{\#G}$ weight of the drilling fluid, lbs/gal

W_{SG} specific gravity of the drilling fluid

5 Symbols

For the purposes of this document, the following symbols apply.

ρ density

m mass

ϑ Inclination at lower point of curved section

σ Inclination at upper point of curved section

ϑ is the wellbore inclination

a A constant that depends on the drill pipe size, grade and mean stress

A_{DP} Drill pipe cross sectional area

A' Formula factor

A_{TOR} AB or AP whichever is smaller

A cross sectional area

a Distance across flats

AMMT abbreviation for the American macaroni tubing style of thread design

AMT alternate abbreviation for the American macaroni tubing thread style of thread design

b Constant that depends on the drill pipe size and grade.

B' Formula factor;

B is the hole curvature or build rate

b_K Kelly bore

b Thread root diameter of box threads at small end of pin

B_{BUC} curvature of buckled pipe

BC Buoyancy Factor

B_C Critical hole curvature

B_L Lateral curvature rate

BR Build rate (deg/100 ft)

BSR bending strength ratio

B_T Total curvature rate

BV_{MAX} Maximum vertical curvature rate for buckling

BV_{MIN} Minimum vertical curvature rate for bucklingBW Critical curvature that defines the transition from point contact to wrap contact

$\angle C$ angle of run-out of elevator recess

C′ Formula factor.

C Pitch diameter at gauge point

c is the maximum permissible dogleg severity

$\angle D$ angle of run-out of slip recess

d is the inside diameter

D is the outside diameter

dB Decible

db is the maximum permissible bending stress

D_F bevel diameter

D_{FL} is the distance across drive section flats

D_{FR} diameter float valve recess

D_H hole diameter

D_L outside diameter lift shoulder

D_{LR} outside diameter kelly lower upset

D_P elevator recess diameter

D_R outside diameter reduced section

D_{RG} diameter of relief groove1

D_S diameter slip groove

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1) API Specification 7, Table 16, Column 5.

D_{TJ} tool joint outside diameter

D_U outside diameter upper kelly upper upset

E Young's modulus, psi

e differential stretch

f is the coefficient of friction of mating surfaces

F axial load

F_B critical buckling force

F_{CRIT} Critical buckling load from curve

$F_{CRIT-ADJ}$ Adjusted critical buckling load

FH abbreviation for the API full hole style of thread design

f_M is the root truncation

f_{MW} Buoyancy factor

f_{RN} is root truncation

H is the thread height not truncated

H_2S hydrogen sulphide

h_C radial clearance

Hmax height of the tool joint above the shoulder of the slips

I area moment of inertia

I/C section modulus

ID is inside diameter

J is the polar moment of inertia

k conversion constant

k1 conversion factor

k2 constant

k2 length from the shoulder to the plane of the gauge point

k3 is the constant

k4 is the constant

ka surface finish factor

K_B is the Buoyancy factor

kb size factor

kc load factor,

kd temperature factor,

ke miscellaneous effects factor.

L_C depth at which P_C acts

L_{TAN} tangency length

L_{TJ2} half the distance between tool joints

L_{WC} Length for wrap contact

L_{HC} length of high curvature hole

L_{PTC} is the length of one joint of drillpipe for point contact of pipe body

L1 is the length of free drill pipe, expressed in feet,

L_D length kelly drive section

l_E elevator groove recess depth

L_E effective span length for wrap contact and equals L for point contact

L_{Fpipe} lateral force on pipe body

L_{FTJ} lateral force on tool joint

L_{FV} length float valve assembly

L_G minimum length kelly sleeve gauge

LH left hand

L_L lower upset length kellys

L_{pc} length of the pin that mates with the box

L_R depth of float valve recess

l_S slip recess groove depth

L_t tong arm length

L_U upper upset length kellys

L_W length of pipe body touching hole

M_W mud density

N revolutions of exposure

N_1, N_2,…are the cycles to failure at Sf1, Sf2,…respectively.

NC abbreviation for the API number style of thread design

NDT non-destructive testing

OD outside diameter

P_{DIFF} differential pull

p lead of the thread

P_{PB} minimum tensile strength of pipe body

P_{TL} total load in tension

P line pull

P1 yield strength of the tool joint pin

P_{AC} allowable collapse pressure

P_C minimum collapse pressure

P_C collapse pressure

P_I internal pressure

P_O tension required to separate the tool joint shoulders

P_P theoretical collapse pressure from tables

P_Q yield of the drill pipe tube in the presence of torsion

PT liquid penetrant testing

P_{T3T2} Tension required to yield pin

P_{T4T2} Tension required to yield pin

Q minimum torsional yield strength

Q_c box counterbore

Q_T minimum torsional yield strength under tension

R_{TR} thread root diameter of pin threads

R radius

R_c corner radius forged kelly

R_{cc} corner radius machined kelly

REG API Regular style of thread design

R_H maximum fillet radius hexagonal kelly sleeve gauge

RH right hand

R_L lateral build radius

ROP penetration rate

RPM rotary speed

R_S average mean shoulder radius

R_T average mean thread radius

S stress

S_B bending stress

S_e endurance limit at zero mean stress of a Moore machine sample

S_{eo} drill pipe endurance limit

SF safety factor

S_f Stress amplitude

S_m Axial stress in drill pipe due to tension at the bottom of the dogleg

S_m Mean stress

S_{rs} truncation

S_s minimum shear strength

S_{ult} Tensile strength of drill pipe

T_{BW} buoyant weight

t wall thickness

T torque

T1 Torsional strength of box

T2 Torsional strength of pin of the tool joint

T3 Torsional load required to produce additional make-up of connection

T4 Make-up torque at which pin yield and shoulder separation occur simultaneously

T_A Torque that is applied to the tool joint before tension is applied

T_{DH} Applied downhole torque

TJOD OD tool joints

T_{MU} make-up torque

tpr taper

T_S tensile strength of pipe

T_Y torque required to yield

UT ultrasonic testing

W_A weight in air

W_{BUILD} is the Buoyed weight of pipe in the build section

W_c weight of collars in air

W_{DP} weight per footr of pipe

W_{DROP} Buoyed weight of pipe in the dropping section

W_{EQ} buoyant weight equivalent for pipe in curved borehole

$W_{F'}$ weight of drilling fluid inside pipe

W_F weight of drilling fluid

$W_{G'}$ weight of drilling fluid inside pipe

W_M buoyant weight of pipe

x fractional exponent

Y depth to fluid inside drill pipe

Y_M material minimum yield strength

Z_B box section modulus

Z_P pin section modulus

ΔD diameter difference tool joint minus pipe outside diameter

θ half the included angle of thread

Θ inclination angle

μ Poisson's ratio

6 Performance properties of drill stem components

6.1 Kelly valves

See ISO 10424-1 and API Specification 7 for information on kelly valves.

6.2 Kellys

6.2.1 Kellys are manufactured with one of two drive configurations, square or hexagonal. Dimensions are listed in Tables 2 and 3 entitled "Square kellys" and "Hexagonal kellys" of API Specification 7.

6.2.2 Square kellys are furnished as forged or machined in the drive section. On forged kellys, the drive sections are normalized and tempered and the ends are quenched and tempered.

6.2.3 Hexagonal or fully machined square kellys are machined from round bars. Heat treatment may be either:

a) Full length quenched and tempered before machining; or

b) The drive section normalized and tempered and the ends quenched and tempered.

NOTE Fully quenched drive sections have higher minimum tensile yield strength than normalized drive sections when tempered to the same hardness level. For the same hardness level, the ultimate tensile strength may be considered as the same in both cases.

6.2.4 The following criteria should be considered in selecting square or hexagonal kellys.

6.2.4.1 When the appropriate kelly is selected for a given casing size, the drive section of the hexagonal kelly is stronger than the drive section of the square kelly (see Table A.15 (Table B.15)).

EXAMPLE A 10,80 cm (4,25 in) square kelly or a 13,34 cm (5,25 in) hexagonal kelly would be selected for use in 21,91 cm (8,63 in) casing.

NOTE The connections on these two kellys are generally the same and unless the bores (inside diameters) are the same, the kelly with the smaller bore could be interpreted to have the greater pin tensile and torsional strength.

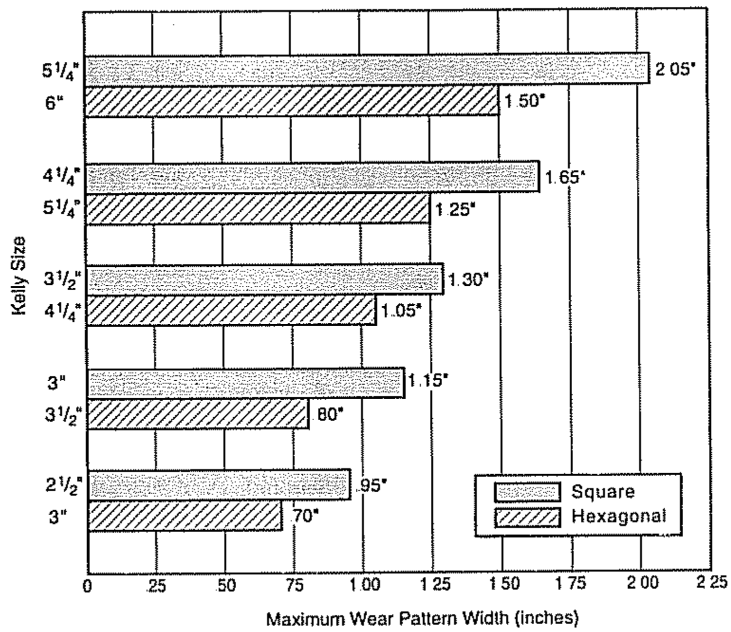
6.2.4.2 For a given tensile load, the stress level is less in the hexagonal section.

6.2.4.3 Due to the lower stress level, the endurance limit of the hexagonal drive section is greater in terms of cycles to failure for a given bending load.

6.2.4.4 Surface decarburization (decarb) is inherent in the as-forged square kelly which further reduces the endurance limit in terms of cycles to failure for a given bending load. Hexagonal kellys and fully machined squares have machined surfaces and are generally free of decarb in the drive section.

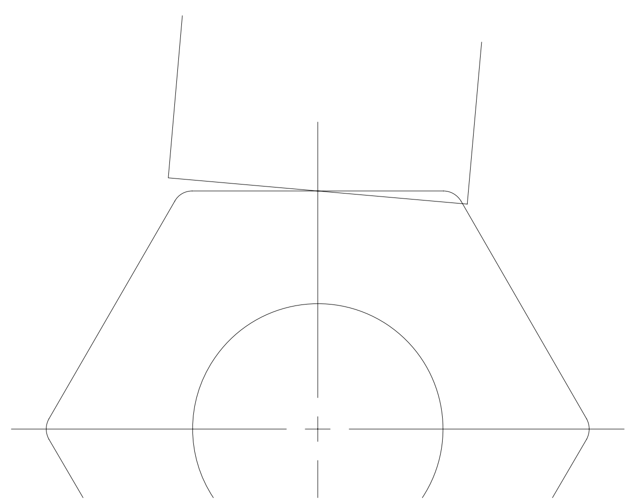
6.2.4.5 It is impractical to remove the decarb from the complete drive section of the forged square kelly; however, the decarb should be removed from the corners in the fillet between the drive section and the upset to aid in the prevention of fatigue cracks in this area. Machining of square kellys from round bars could eliminate this undesirable condition.

6.2.4.6 The life of the drive section is directly related to the kelly fit with the kelly drive. A square drive section normally will tolerate a greater clearance with acceptable life than will a hexagonal section. A diligent effort by the rig personnel to maintain minimum clearance between the kelly drive section and the bushing will minimize this consideration in kelly selection. New roller bushing assemblies working on new kellys will develop wear patterns that are essentially flat in shape on the driving edge of the kelly. Wear patterns begin as point contacts of zero width near the corner. The pattern widens as the kelly and bushing begin to wear until a maximum wear pattern is achieved. The wear rate will be the least when the maximum wear pattern width is achieved. Figure 1 illustrates the maximum width flat wear pattern that could be expected on the kelly drive flats if the new assembly has clearances as shown in Table A.16 (Table B.16). The information in Table A.16 (Table B.16) and Figures 1 and 2 may be used to evaluate the clearances between kelly and bushing. This evaluation should be made as soon as a wear pattern becomes apparent after a new assembly is put into service.



Note: The maximum Wear Pattern Width is the average of the Wear Pattern Widths based on calculations using minimum and maximum clearances and contact angles in Table 16 and is accurate within 5 percent.

Figure 1 — Maximum wear for New Kelly Drive Assembly



Note: Drive edge will have a wide flat pattern with small contact angle.

Figure 34-New kelly-New drive assembly

Figure 2 — New kelly – New drive assembly

EXAMPLE At the time of evaluation, the wear pattern width for a 13,34 cm (5,25 in) hexagonal kelly is 2,54 cm (1,00 in). This could mean one of the following two conditions exist:

a) If the contact angle is less than 8 ° 37 ', the original clearances were acceptable. The wear pattern is not fully developed.

b) If the contact angle is greater than 8 ° 37 ', the wear pattern is fully developed. The clearance is greater than is recommended and should be corrected.

6.2.5 Techniques for extending life of kellys include remachining drive sections to a smaller size and reversing ends.

a) Remachining: Before attempting to remachine a kelly, it should be fully inspected for fatigue cracks and also dimensionally checked to assure that it is suitable for remilling. The strength of a remachined kelly should be compared with the strength of the drill pipe with which the kelly is to be used. (See Table A.17 (Table B.17)) for drive section dimensions and strengths.)

b) Reversing Ends: Usually both ends of the kelly should be butt welded (stubbed) for this to be possible as the original top is too short and the old lower end is too small in diameter for the connections to be reversed. The welds should be made in the upset portions on each end to insure the tensile integrity and fatigue resistance capabilities of the sections. Proper heating and welding procedures should be used to prevent cracking and to recondition the sections where welding has been performed.

6.2.6 The internal pressure at minimum yield for the drive section may be calculated from Equation E.9.

6.3 Drill pipe and tool joints

6.3.1 This subclause contains a series of Tables A.1 through A.10 (Tables B.1 through .B.10) designed to present the dimensional, mechanical, and performance properties of new and used drill pipe. Tables are also included listing these properties for tool joints used with new and used drill pipe.

6.3.2 All drill pipe and tool joint properties tables are included in Annex A (Annex B).

6.3.3 Values listed in drill pipe tables are based on accepted standards of the industry and calculated from formulas in Annex E.

6.3.4 Recommended drift diameters for new drill string assemblies are shown in column 10 of Tables A.8 and A.9 (Tables B.8 and B.9). Drift bars shall be a minimum of four in long. The drift bar shall pass through the upset area but need not penetrate more than twelve inches beyond the base of the elevator shoulder.

6.3.5 The torsional strength of a tool joint is a function of several variables. These include the strength of the steel, connection size, thread form, lead, taper and coefficient of friction on mating surfaces, threads, or shoulders. The coefficient of friction for the purposes of this recommended practice is assumed a constant; it has been demonstrated, however, that new tool joints and service temperature often affect the coefficient of friction of the tool joint system. While new tool joints typically exhibit a low coefficient of friction, service temperatures greater than 300 °F can dramatically increase or decrease the coefficient of friction depending primarily on thread compound. See 13.13.2.5, Thread compound friction factor, for a more detailed discussion. The torque required to yield a rotary shouldered connection may be obtained from the equation in Section E.9.

6.3.6 The pin or box area, whichever controls, is the largest factor and is subject to the widest variation. The outside diameter of the tool joint box and the inside diameter of the tool joint pin largely determine the strength of the tool joint in tension. The OD of the tool joint box affects the cross sectional area of the box and the ID of the tool joint pin affects the cross sectional area of the pin. Choice of box OD and pin ID determines the cross sectional areas of the pin and box and establishes the theoretical torsional strength, assuming all other factors are constant.

6.3.7 The greatest reduction in theoretical torsional strength of a tool joint during its service life occurs with OD wear. At whatever point the tool joint box area becomes the smaller or controlling area, any further reduction in OD causes a direct reduction in torsional strength. If the box area controls when the tool joint is new, initial OD wear reduces torsional strength. If the pin controls when new, some OD wear may occur before the torsional strength is affected. Conversely, it is possible to increase torsional strength by making joints with oversize OD and reduced ID.

6.3.8 Minimum OD, box shoulder, and make-up torque values listed in Table A.10 (Table B.10) were determined using the following criteria:

6.3.8.1 Calculations for recommended tool joint make-up torque are based on the use of a thread compound containing 40 to 60 % by weight of finely powdered metallic zinc applied to all threads and shoulders, and containing not more than 0,3 % total active sulphur (reference the caution regarding the use of hazardous materials in API Specification 7, Appendix G). Calculations are also based on a tensile stress of 60 % of the minimum tensile yield for tool joints.

6.3.8.2 In calculation of torsional strengths of tool joints, both new and worn, the bevels of the tool joint shoulders are disregarded.

6.3.8.3 Premium Class: t = (0,8)(nominal wall), D = (nominal OD) – (0,4)(nominal wall). Premium Class Drill String is based on drill pipe having a minimum wall thickness of 80 %.

6.3.8.4 Class 2: t = (0,7)(nominal wall), D = (nominal OD) – (0,6)(nominal wall). Class 2 drill string allows drill pipe with a minimum wall thickness of 70 %.

6.3.8.5 The tool joint to pipe torsional ratios that are used here ($\geq$0,80) are recommendations only and it should be realized that other combinations of dimensions may be used. A given assembly that is suitable for certain service may be inadequate for some areas and over designed for others.

6.3.9 Many sizes and styles of connections are interchangeable with certain other sizes and styles of connections. These conditions differ only in name and in some cases thread form. If the thread forms are interchangeable, the connections are interchangeable. These interchangeable connections are listed in Table A.12 (Table B.12).

6.3.10 The curves of Figures C.1 through C.25 (Figures D.1 through D.25) depict the theoretical torsional yield strength of a number of commonly used tool joint connections over a wide range of inside and outside diameters. The coefficient of friction on mating surfaces, threads and shoulders, is assumed to be 0,08 (See Clause 3 of API RP 7A1, Recommended Practice for Testing of Thread Compound for Rotary Shouldered Connections). The make-up torque should be based on a tensile stress level of 60 % of the minimum yield for tool joints.

6.3.11 The curves may be used by taking the following steps:

6.3.11.1 Select the appropriately titled curve for the size and type tool joint connection being studied.

6.3.11.2 Extend a horizontal line from the OD under consideration to the curve and read the torsional strength representing the box.

6.3.11.3 Extend a vertical line from the ID to the curve and read the torsional strength representing the pin.

6.3.11.4 The smaller of the two torsional strengths thus obtained is the theoretical torsional strength of the tool joint.

6.3.11.5 It is emphasized that the values obtained from the curves are theoretical values of torsional strength. Tool joints in the field, subject to many factors not included in determination of points for the curves, may vary from these values.

6.3.11.6 The curves are most useful to show the relative torsional strengths of joints for variations in OD and ID, both new and after wear. In each case, the smaller value should be used.

6.3.12 The recommended make-up torque for a used tool joint is determined by taking the following steps:

6.3.12.1 Select the appropriately titled curve for the size and type tool joint connection being studied.

6.3.12.2 Extend a horizontal line from the OD under consideration to the curve and read the recommended make-up torque representing the box.

6.3.12.3 Extend a vertical line from the ID under consideration to the curve and read the recommended make-up torque representing the pin.

6.3.12.4 The smaller of the two recommended make-up torques thus obtained is the recommended make-up torque for the tool joint.

6.3.12.5 A make-up torque higher than recommended may be required under extreme conditions.

6.4 Proprietary rotary shouldered connections

Proprietary, or non-API, rotary shouldered connections are becoming increasingly common. Compared to API connections, proprietary connections often provide improved torsional strength for equivalent box OD and pin ID dimensions. The increased desire of oil companies to drill directional wellbores including extended-reach, horizontal and deep directional leads to increased side loading to the drill string and increased torque requirements. In these cases, proprietary connections often provide the torque capacity for common drill pipe sizes not always provided by API connections.

Another benefit of proprietary connections is the hydraulic advantage often provided. When torque requirements are not too high, proprietary drill pipe manufacturers are able to offer reduced OD and increased ID drill pipe products that are more “streamlined” than API drill pipe configurations (tool joint OD and ID dimensions more closely approximating that of the tube’s). In some cases, larger drill pipe sizes can now be run inside casing or liner sizes than that which was traditionally run before. An example of this is the increasing use of 4 in drill pipe with proprietary connections over traditional 3 ½ in NC38 drill pipe inside 7 in liners and casing strings. Use of products such as this reduce bore pressure losses through the drill string for equivalent flowrates and often result in reduced pressure requirements to the rig’s hydraulic pump system.

Though some performance properties may be improved through the use of proprietary connections, other performance may suffer when compared to API connections. Readers are encouraged to investigate all performance properties of each connection when selecting one for use.

There are a number of different manufacturers who produce and sell proprietary connections. Most of these different proprietary connections can be grouped into two categories: 1) double shoulder and 2) non-shouldering dovetail thread form connections. In addition, some proprietary rotary shouldered connections are offered with radial metal-to-metal seals to promote increased pressure integrity.

General performance data for various proprietary rotary shouldered connections is provided in Tables B.46 to B.50. This data has been obtained from the connection manufacturer and is intended for reference only. Readers are encouraged to contact the manufacturer directly for updated and/or more detailed information. Many of these connections are offered in sizes not presented in the tables. Table headings are provided below to indicate the connection types presented in Annex B.

Table B.46 – Grant Prideco GPDS™ Connection Information

Table B.47 – Grant Prideco HI TORQUE® Connection Information

Table B.48 – Grant Prideco eXtreme™ Torque (XT™) Connection Information

Table B.49 – Grant Prideco eXtreme™ Torque Metal Seal (XT-M™) Connection Information

Table B.50 – Omsco Tuff-Torque™ Connection Information

6.5 Alternative drill pipe sizes and weights

Alternative, non-API, drill pipe sizes and weights now frequent the industry. One such example of this is the predominant use of drill pipe products to land deepwater casing strings. Various casing load requirements have yielded a multitude of various drill pipe wall thicknesses in many API and non-API drill pipe OD sizes. In addition, intermediate OD size (between API OD sizes) drill pipe products have been manufactured to more optimally match casing designs and bit programs of the extended-reach and deepwater markets.

Table B.51 presents a summary of performance properties for many of these alternative drill pipe sizes and weights in a New Class condition. For reference, these products are also presented in alternative drill pipe grades. Table B.52 presents performance properties for these products in a Premium Class condition. These tables were developed by information received from various drill pipe manufacturers and are not intended to represent all available drill pipe sizes and weights. The reader should use this table for reference only and contact the drill pipe manufacturer for more information on all available drill pipe sizes and weights.

6.6 Heavy weight drill pipe

Heavy weight drill pipe (HWDP) is commonly used to provide a gradual transition from heavy drill collars to relatively lightweight drill pipe. By providing a gradual change in weight and rigidity, HWDP reduces connection fatigue problems within the bottom hole assembly (BHA) and increases drill stem service life. Additionally, HWDP bends more easily than drill collars and is often a preferred BHA component for directional control in high angle and horizontal drilling.

There are a variety of HWDP products available, ranging from integral products manufactured of drill collar bar stock to welded-on tool joint products manufactured similar to drill pipe. In both these types, a variety of configurations are available. These configurations range from conventional single wear section products to an array of spiralled configuration products for improved cuttings transport and reduction in differential sticking tendencies.

The most common HWDP product is comprised of a 55 000 psi minimum yield strength tube with welded on 100 000 or 110 000 psi minimum yield strength tool joints (depending on tool joint outside diameter size). The most common configuration in this product type is a single wear section in the middle of the tube, conventional heavy weight drill pipe. Figure 3 below illustrates this conventional heavy weight drill pipe product.

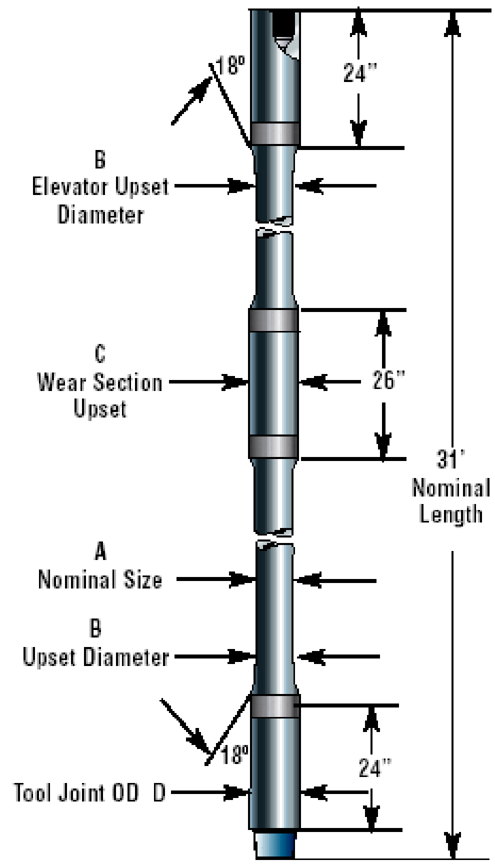


Figure 3 — Conventional Heavy Weight Drill Pipe

Table B.53 provides nominal dimensions and general performance properties for conventional HWDP. This table was developed based on comments from a number of HWDP manufacturers and does not represent all HWDP products offered by the various manufacturers. The reader is encouraged to use this information as general reference only. More specific and detailed, product-specific information should be obtained direct from its manufacturer.

6.7 Drill collars

6.7.1 Drill collar weights

6.7.1.1 Table A.13 (Table B.13) contains the weights of drill collar steel in kg/m (lbm/ft) for a wide range of OD and ID combinations, in both API and non-API sizes. Values in the table may be used to provide the basic information required to calculate the weights of drill collar strings that are made up of drill collars not being of uniform OD's and ID's.

6.7.1.2 The weight of drill collars not listed in Table A.13 (Table B.13) may be calculated from the following formula:

$$m = k\rho Al \qquad (1)$$

$$A = \frac{\pi}{4} \times \left(OD^2 - ID^2 \right) \qquad (2)$$

Where:

m is mass, expressed in kg (lbm)

ρ is density, expressed in kg/m^3 (lbm/ft^3)

l is length, expressed in m (ft)

A is cross sectional area of drill collar, expressed in mm^2 (in^2)

OD is outside diameter, expressed in mm (in)

ID is inside diameter, expressed in mm (in)

k is conversion constant, m^2/10^{6}mm^2 (ft^2/144 in^2)

For the purpose of this calculation, the density of drill collar steel is taken to be 7841,72 kg/m^3 (489,54 lbm/ft^3).

6.7.2 Make up torque

6.7.2.1 Recommended make-up torque values for rotary shouldered drill collar connections are listed in Table A.14 (Table B.14). These values are listed for various connection styles and for commonly used drill collar OD and ID sizes.

6.7.2.2 The values in Table A.14 (Table B.14) are considered to be minimum torque required in average conditions. The accepted range is the value in Table A.14 (Table B.14) plus 10%.

6.7.2.3 The table also includes a designation of the torsionally weak member (pin or box) for each connection size and style.

6.7.2.4 Make-up torques for OD and ID sizes not shown in Table A.14 (Table B.14) can be determined by extrapolating between the values listed. If one desires to be more precise, the exact value can be calculated by using Farr's Modified Screw Jack formula shown in subclause 5.8.2.3.

6.7.3 Bending strength ratio (BSR)

6.7.3.1 Figures C.26 through C.32 (Figures D.26 through D.32) are plots of bending strength ratios for many of the rotary-shouldered connections used in the drill stem.

6.7.3.2 These curves are very useful if a new string of drill collars are to be purchased as both ends of each new collar in the string will have the same OD and ID.

6.7.3.3 On used strings of collars it is difficult to match up OD's and ID's so as to have a consistent bending strength ratio throughout the string.

6.7.3.4 On used drill collar strings, it recommended that the larger drill collars be picked up first so as to be placed below the next smaller collar. In this manner, the box OD is larger than the OD of the mating pin, thus not adversely affecting the bending strength ratio of the combination.

6.7.3.5 Subclause 5.8.2.3 contains a more in depth discussion on the effects of the bending strength ratio on the fatigue life of the drill collar.

6.8 Rotary subs

6.8.1 Purpose of rotary subs

Rotary subs are used to connect members of the drill stem if different connections or different outside diameters exist between the two parts.

6.8.2 Mechanical properties of rotary subs

Straight OD subs below the kelly have the same mechanical properties as drill collars unless a special function has been identified and arrangements have been made to acquire special material. Refer to ISO 10424-1 or API Specification 7 for details on the mechanical properties of drill collars.

On top drive rigs, the drive sub can require higher strength material. The manufacturer of the top drive should be contacted to supply information on the material requirements.

Routinely, metallurgical test results are furnished that are indicative of the mechanical properties of the steel at a point 25,4 mm (1 in) below the original heat-treated surface.

The extent of the depth of acceptable mechanical properties below the above tested depth depends upon the chemistry of the steel, the OD of the material and the manner in which it was heat-treated, that is, quenched and tempered, normalized and tempered, etc.

If the material was originally 203 mm (8 in) or less, acceptable properties usually exist throughout the thickness of the material.

As material size increases above 203 mm (8 in), it becomes more difficult to obtain the desired properties throughout the section.

Caution should be taken using material larger than 203 mm (8 in) if the sub OD has been turned to a smaller size or if the sub has a reduced section (bottleneck) on either end as the mechanical properties in the reduced section may be less than those reported.

Reducing the diameter of one section of the sub by as much as 38,1 mm (1 ½ in) from the original “as heat-treated” OD (if heat-treatment was by quenching and tempering) will not materially reduce the mechanical properties.

However, as an example, if the original OD was 254 mm (10 in) and it is turned down to 178 mm (7 in) then the mechanical properties are questionable.

Another area of concern is if a sub is manufactured from material larger than 203 mm (8 in), is machined down on one end and has a pin connection on the smaller end. If the pin on the small end has a pitch diameter 76 mm (3 in) smaller than the connection on the larger end, concern is justified as to the mechanical properties of the pin on the small end. An example of this would be a reduced section sub with a 7 5/8 Regular on the larger end and a NC 38 pin on the smaller end. The mechanical properties in the NC 38 pin connection may not be as expected.

It is not practical to re-test the mechanical properties in the field. Performing a Brinell Hardness test on the reduced section will usually confirm the acceptability of the material. A minimum BHN of 285 for reduced diameter sections less than 175 mm (6,89 in) and a BHN of 277 for reduced subclauses greater than 175 mm (6,89 in) is considered prima facie evidence of satisfactory properties.

6.8.3 Reducing diameters between sections

If a sub has a reduced section, care should be taken so as not to concentrate stress at the point of diameter change. An area of transition should exist between the two diameters. To accomplish this, a taper of 45º maximum from the larger diameter should blend into a smooth radius at the junction of the taper to the reduced section. A smooth radius of 38 mm (1,5 in) or larger has proven adequate for this purpose.

6.8.4 Fishing necks on reduced diameter subs

Ensure the length of the fishing neck is long enough for its purpose. A minimum length of 254 mm (10 in) is required for a successful fishing job. This is the length after the twist-off. Consider if a NC 50 box twists-off, one will lose at least 114 mm (4,5 in) of length. This length should be added to the 254 mm (4,5 in) required for fishing. Thus, the fishing neck for a sub with a NC 50 box connection should never be less than 368 mm (14,5 in).

This simple logic can be applied to other connections.

EXAMPLE The twist-off of a 7 5/8 Regular box will require a minimum fishing neck of 387 mm (15,25 in).

To calculate the minimum fishing neck length:

$$\text{Length} = 254 \text{ mm} + L_{PC} \tag{3}$$

$$= (10 \text{ in} + L_{PC})$$

where

L_{PC} is the length of the pin that mates with the box (which can be found in ISO 10424-2 or API Specification 7), expressed in mm (in).

6.9 Bits

6.9.1 Classification of rock bits

6.9.1.1 A classification system and form for designating roller cone rock bits according to the type of bit (steel tooth or insert), the type of formation drilled, and mechanical features of the bit, was developed by a special subcommittee within the International Association of Drilling Contractors (IADC). The system as proposed by the IADC subcommittee was accepted by the IADC membership in March 1987.

In this part of ISO 10407, the classification form is designated as Table A.25 (Table B.25).

6.9.1.2 Bits may be classified and designated by an alpha-numeric code on the bit carton with a series of three numbers and one letter keyed to the classification system shown in Tables A.25 and A.26 (Tables B.25 and B.26). The three numbers originate from Table A.25 (Table B.25) and the letter from Table A.26 (Table B.26).

6.9.1.3 In using the form shown in Table A.25 (Table B.25), in conjunction with Table A.26 (Table B.26), the manufacturer assigns to each bit design three numbers and one letter that correspond with a specific block on the form. The sequence to be followed is:

— First number designates Series.

— Second number designates Type.

— Third number designates Feature.

— Letter designates additional features, as shown in Table A.26 (Table B.26).

6.9.1.4 Series numbers 1, 2, and 3 are reserved for milled tooth bits in the soft, medium, and hard formation categories. Series numbers 4, 5, 6, 7, and 8 are for insert bits in the soft, medium, hard, and extremely hard formations.

6.9.1.5 Type numbers 1 through 4 designate formation hardness sub-classification from softest to hardest within each series classification.

6.9.1.6 On Table A.25 (Table B.25), the seven column listings under the heading "Features" include seven features common to the milled tooth and insert bits of most manufacturers. The original chart consisted of nine columns, but Columns 8 and 9 have been removed and reserved for future bit development.

6.9.1.7 The form is designed to include only one manufacturer's listing on each sheet, and to allow each specific bit designation in only one classification position. It is recognized, however, that many bits will drill efficiently in a range of types and perhaps in more than one series. Particular attention is therefore invited to the note on the form which contains a statement of this principle. It is the responsibility of the manufacturer and user to determine the range of efficient use in specific instances.

6.9.1.8 Classification forms are available from: International Association of Drilling Contractors (IADC), P.O. Box 4287, Houston, Texas 77210.

6.9.2 Make-up torque for rock bits

6.9.2.1 Recommended torque for roller cone bits is shown in Table A.27 (Table B.27).

6.9.2.2 Recommended torque for diamond drill bits is shown in Table A.28 (Table B.28).

6.9.3 Bit sizes

6.9.3.1 Common sizes for roller bits are listed in Table A.29 (Table B.29). Sizes other than those shown may be available in limited cutting structure types.

6.9.3.2 Common sizes for fixed cutter bits are listed in Table A.30 (Table B.30). Sizes other than those shown may be available in limited cutting structure types.

6.10 Rotary shouldered connections

6.10.1 The purpose of a rotary shouldered connection

6.10.1.1 The threads on rotary-shouldered connection do not form a seal, but act as a jack-screw to force the shoulders together. By forcing the shoulders together under high load, the shoulders provide the necessary seal to prevent leakage.

6.10.1.2 The connection acts as a structural member in the drilling assembly. This means that the connection has to be strong in torsion, tension and in bending. Strong in torsion to take the rotational forces as the bit turns, strong enough in tension to lift the drill string from the hole and strong in bending strength to resist fatigue failures.

6.10.1.3 The rotary-shouldered connection is designed with the best overall combination of these functions so as to utilize the tensile and torsional strength of the steel, the maximum possible stiffness and shoulder sealing efficiency.

6.10.1.4 The ideal condition would be to have the connection as stiff and strong as the body of the drill stem member of which it is a part. Since this is impossible, design calculations are made to obtain the best balance between relative capacities of the box and pin to resist bending; the best torsional balance to give optimum shoulder load to prevent shoulder separation; and to provide ample tensile load requirements.

6.10.1.5 The shoulders acting under high compressive load form the only sealing surface. All threads on rotary shouldered connections are purposely truncated to prevent interference between the thread crests and thread roots, in addition to providing a space for excess thread compound and foreign particles in the drilling fluid. The space between the box counterbore and pin base also serves this purpose.

NOTE This space, as well as the space between the non-loaded thread flanks, forms a helical opening that follows the thread all the way to the bore of the connection.

6.10.1.6 The shoulders should always be treated with care and good rig practices to ensure proper sealing.

6.10.2 Make up torque

6.10.2.1 Recommended make-up torque values for rotary shouldered connections for drill pipe tool joint are listed in Table A.10 (Table B.10). For most connections used in the bottom hole assembly (drill collars, subs, reamers, stabilizers and other tools between the drill pipe and the bit) the minimum make-up torques are shown in Table A.14 (Table B.14). These values are listed for various connection styles and for commonly used OD and ID sizes.

6.10.2.2 The tables also includes a designation of the torsionally weak member (pin or box) for each connection size and style.

6.10.2.3 Make-up torques for OD and ID sizes not shown in Tables A.10 and A.14 (Tables B.10 and B.14) can be calculated by using Farr's Modified Screw Jack formula shown below:

$$T_{MU} = \frac{SA}{k_1} \times \left(\frac{p}{2\pi} + \frac{R_T f}{Cos\theta} + R_S f \right) \quad (4)$$

where:

T_{MU} is the minimum recommended make-up torque, expressed in N·m (ft-lbf);

S is the applied make-up stress, expressed in MPa (psi);

For drill pipe tool joints in Table A.10 (Table B.10), the make-up stress (S) is 496,42 MPa (72 000 psi);

For all rotary shouldered connections in the bottom hole assembly except the H-90 and PAC designs, the make-up stress (S) is 430,92 MPa (62 500 psi);

For the H-90 design, S is 387,49 MPa (56 200 psi);

For the PAC design, S is 603,29 MPa (87 500 psi);

A_{TOR} is the cross sectional area of the pin or box, A_B or A_P, whichever is smaller, expressed in mm^2 (in^2);

NOTE A_P should be based on pin connections without relief grooves.

P is the lead of the thread, expressed in mm (in);

R_T is the average mean thread radius, expressed in mm (in);

R_S is the average mean shoulder radius, expressed in mm (in);

NOTE The affect of the shoulder bevel is neglected;

θ is half the included angle of thread, expressed in degrees;

f is the coefficient of friction of mating surfaces, threads and shoulders, and is assumed to be 0,08 for thread compounds containing 40 to 60 % by weight of finely powdered metallic zinc;

k_1 is the conversion factor, 1000 mm/m (12 in/ft);

Where

$$R_T = \frac{C + \left[C - \left(L_{PC} - k_2\right) \times tpr\right]}{4} \quad (5)$$

$$R_S = \frac{OD + Qc}{4} \quad (6)$$

The maximum value of R_S is limited to the value obtained from the calculated OD where

$$A_P = A_B \tag{7}$$

$$A_P = \pi / 4\left[(C - B)^2 - ID^2\right] \tag{8}$$

$$B = 2\left(\frac{H}{2} - Srs\right) + tpr \times (k_3) \tag{9}$$

$$A_B = \pi / 4\left[OD^2 - (Qc - E)^2\right] \tag{10}$$

$$E = tpr\ (k_4) \tag{11}$$

Where

C is pitch diameter of thread at the gauge point, expressed in mm (in);

L_{PC} is length of pin threads, expressed in mm (in);

k_2 is the length from the shoulder to the plane of the gauge point, 15,875 mm (0,625 in);

tpr is taper, expressed in mm/mm (in/in);

OD is the outside diameter of the drill collar, expressed in mm (in);

Q_C is the box counterbore, expressed in mm (in);

H is thread height not truncated, expressed in mm (in);

Srs is root truncation, expressed in mm (in);

ID is inside diameter, expressed in mm (in);

k_3 is the constant 3,175 mm (0,125 in);

k_4 is the constant 9,525 mm (0,375 in).

6.10.3 Torque to yield a rotary shouldered connection

6.10.3.1 Torque required to yield a rotary shouldered connection can be calculated using Farr's Modified Screw Jack formula described in subclause 5.8.2.3. To calculate the yield torque, simply substitute the yield strength (Y_M) of the material for the applied make-up stress (S).

6.10.3.2 If the recommended make-up torque is known but not the yield torque, the yield torque can be determined by the formula below:

$$T_Y = T_{MU} \frac{Y_M}{S} \tag{12}$$

where:

T_Y is the torque to yield the connection, pin or box whichever is weaker, expressed in N • m (ft-lbf);

T_{MU} is the known recommended make-up torque of the connection, expressed in N • m (ft-lbf);

Y_M is the yield strength of the material, expressed in MPa (psi);

S is the applied make-up stress, expressed in MPa (psi).

6.10.4 Bending strength ratio (BSR)

6.10.4.1 Many rotary shouldered connection failures are a result of cyclic bending stresses rather than torsional and tensile loads. A recognized method to measure the capacity of a section to resist any bending moment to which it may be subjected is by comparing its Section Modulus (Z) to other sections of interest. In rotary shouldered connections, the areas of interest are located in the thread root of the box at the small end of the pin and in the thread root of the pin located 19,05 mm (0,750 in) from the sealing shoulder. By comparing the Section Modulus (Z) of the box to the Section Modulus of the pin at the points described above, one can determine the bending strength ratio (BSR) of the connection.

6.10.4.2 The curves in Figures C.26 through C.32 (D.26 through D.32) were determined from bending strength ratios calculated by using the Section Modulus as the measure of the capacity of the connection to exhibit a balanced resistance to fatigue. Figures C.26 through C.32 (D.26 through D.32) may be used for determining the most suitable connection to be used on new tools in the bottom hole assembly or for selecting a new connection to be used on tools which have been worn down on the outside diameter.

6.10.4.3 A connection that has a bending strength ratio of 2,50:1 is generally accepted as an average balanced connection. Since an absolute 2,50:1 ratio is not practical, a range of 2,25:1 to 2,75:1 is generally considered as an acceptable range for most drilling conditions. However, depending upon the drilling conditions, drill collar size and the connection selected, the acceptable range may vary from 1,90:1 to 3,20:1.

6.10.4.4 One should understand that even when the connections have a BSR that is considered "balanced", this will not prevent fatigue failures. "Balanced" simply means that one should experience an equal number of pin and box failures from fatigue but at a reduced rate.

6.10.4.5 To achieve the "balanced" rate of failures one also should torque the connections up to a torque value that will keep the shoulders from opening while rotating. Connections that are made up to insufficient torque will see the shoulders open and experience excessive pin fatigue failures.

6.10.4.6 As the outside diameter of the box will wear more rapidly than the pin inside diameter, the resulting bending strength ratio will be reduced accordingly. Therefore, it is recommended to select connection sizes where the initial (new) BSR is between 2,75:1 and 2,50:1.

6.10.4.7 Sometimes it will be necessary to use ratios larger than 2,75:1. Ratios greater than 2,75:1 usually result in torsionally box strong connections where it is difficult to develop sufficient compression in the shoulders to prevent the shoulders from opening. If the ratio is greater than 2,75:1, care should be exercised to ensure that proper make-up torque is applied.

6.10.4.8 For high rev/min, soft formations, and where drill collar OD is small compared to hole size (example: when body of the tool size is less than 75% of hole size), avoid ratios above 2,85:1 or below 2,25:1.

6.10.4.9 For low rev/min, hard formations, and when the tool OD is close to hole size (example: when tool OD is more than 80% of hole size), avoid ratios above 3,20:1 unless low torque features are used. When low torque features are used on large tools, ratios as large as 3,40:1 will perform satisfactorily.

6.10.4.10 The smaller a tool is, the more limber it is. Since small tools (less than 12,7 cm (5 in) OD) bend easier, ratios less than 2,25:1 have been found to perform satisfactorily in fatigue resistance. But when the bending strength ratio falls below 2,00:1, other connection troubles may begin. These troubles may consist of swollen or split boxes in addition to fatigue cracks in the boxes.

6.10.4.11 The minimum bending strength ratio acceptable in one operating area may not be acceptable in another. Local operating practices, experience based on recent predominance of failures and other conditions should be considered when determining the minimum acceptable bending strength ratio for a particular area and type of operation.

6.10.4.12 Minor differences between measured inside diameter and the inside diameters in Figures C.26 through C.32 (D.26 through D.32) are of little significance; therefore select the figure with the inside diameter closest to the measured inside diameter.

6.10.4.13 The effect of stress-relief features is disregarded when calculating the BSR.

6.10.4.14 The equation for calculating the BSR of a connection is as follows:

$$BSR = \frac{Z_B}{Z_P} \tag{13}$$

$$Z_B = \frac{0.098\left(OD^4 - b^4\right)}{OD} \tag{14}$$

$$Z_P = \frac{0.098\left(R^4 - ID^4\right)}{R} \tag{15}$$

Where

BSR is the bending strength ratio;

Z_B is box section modulus, expressed in mm³ (in³);

Z_P is pin section modulus, expressed in mm³ (in³);

OD is outside diameter of pin and box (Figure 4), expressed in mm (in);

ID is inside diameter or bore (Figure 4), expressed in mm (in);

b is thread root diameter of box threads at small end of pin (Figure 4) expressed in mm (in);

R is thread root diameter of pin threads 19,05 mm (0,750 in) from shoulder of pin (Figure 4), expressed in mm (in).

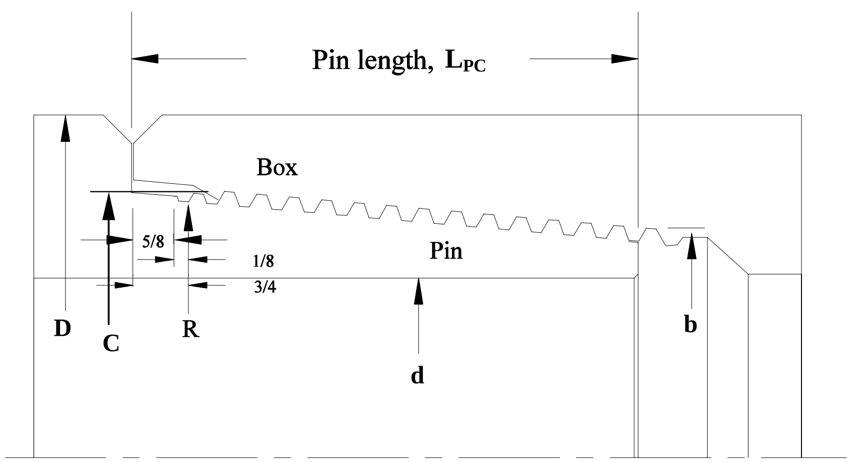


Figure 4 — Rotary shouldered connection location of dimensions for bending strength ratio calculations

To use the BSR equation, first calculate the *Dedendum, b and R*

$$Dedendum = \frac{H}{2} - f_M \tag{16}$$

$$b = C - tpr\left(L_{PC} - k_2\right) + \left(2 \times dedendum\right) \tag{17}$$

$$R = C - \left(2 \times dedendum\right) - tpr \times k_3 \tag{18}$$

Where

H is thread height not truncated, expressed in mm (in)

f_{RN} is root truncation, expressed in mm (in)

C is pitch diameter at gauge point, expressed in mm (in)

tpr is taper, expressed in mm/mm (in/in)

k_2 is the constant 15,875 mm (0,625 in)

k_3 is the constant 3,175 mm (0,125 in)

6.10.5 Stress relief features

6.10.5.1 Relief grooves

6.10.5.1.1 Stress is concentrated in the thread roots of rotary-shouldered connections. Relief grooves were designed to remove the thread roots and spread the bending over a larger area. Original laboratory fatigue tests performed by several drill collar manufacturers in the 1950’s demonstrated the beneficial effects of the stress-relief grooves at the pin shoulder and at the base of the box thread.

6.10.5.1.2 These laboratory tests indicated an improvement by raising the endurance limit of the connections by as much as 4 to 5 fold.

6.10.5.1.3 Subsequent tests made under actual service conditions verified the improvement in fatigue life gained if these features are provided.

6.10.5.1.4 Finite element analysis of pin stress relief grooves in the 1990’s confirmed that the most benefit of the groove is realized if the groove is 25,4 mm (1 in) in length. See Table A.231 (Table B.31).

6.10.5.1.5 These tests also indicated the groove length can vary from 19,0 mm (0,75 in) to 31,8 mm (1 ¼ in) and still provide identifiable benefits although not as much as the 25,4 mm (1 in) long groove.

6.10.5.1.6 Even with the obvious benefits, providers of short-term usage rental components have been reluctant to use the pin relief groove because of the rapid material loss in repairing shoulder and thread damage. Re-facing the pin shoulder and reconditioning the threads is restricted by the tolerances on the groove width per ISO 10424-2 and API Specification 7.

6.10.5.1.7 Rental components such as subs, drilling jars, vibration dampeners, stabilizers, reamers, etc., are usually employed for relatively short periods of time before being returned to service centers for inspection and repair. On rental equipment, connection repairs are made primarily because of galling, shoulder leaks and handling damages while repairs caused by fatigue failures are secondary in occurrence.

6.10.5.1.8 To encourage the use of stress relief grooves on pins of short term usage tools, the following is recommended. For tools returned to a service facility after each well, such as rental tools, a modified pin stress relief groove is recommended. It is recommended that the modified pin stress relief groove conform to Figure 5. Dimensions for DRG are found in ISO 10424-2 and API Specification. 7.

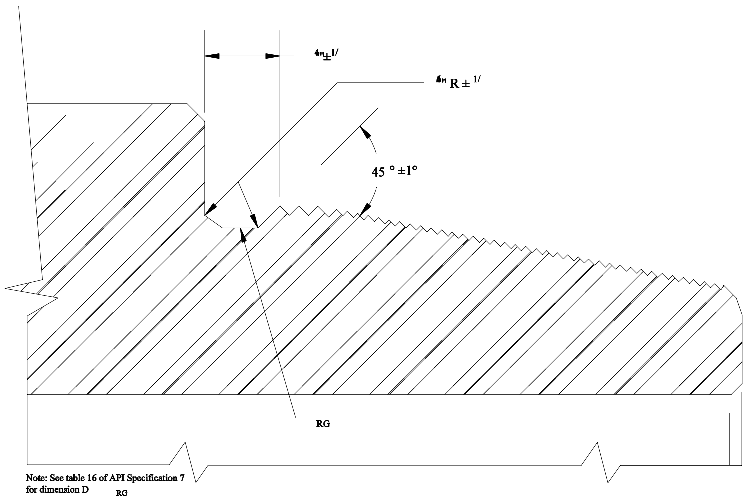


Figure 90-Modified pin stress-relief groove

Figure 5 — Modified pin stress – relief groove

6.10.5.1.9 Recommended initial width of the modified stress relief groove is 19,0 mm 0 to + 1,6 mm (0,75 in 0 to +1,06 in). After reworking for damaged threads and shoulders, the width of the modified stress relief groove should not exceed 31,8 mm (1,25 in).

6.10.5.1.10 Technical data-stress at the root of the last engaged thread of the pin depends on the width of the stress relief groove (SRG). Table A.31 (Table B.31) shows calculated relative stresses for an NC50 axisymmetric finite element model with 6,5 in box OD and 3 in pin ID. A pin with no stress relief groove is the basis for comparison.

6.10.5.1.11 It is recommended for tools that are acquired for long term usage at the rig and where the possibility of fatigue failures at points of high stress are a problem, stress relief grooves be provided and that relief groove conform to the 25,4 mm (1 in) length specified in ISO 10424-2 and API Specification 7.

6.10.5.2 Bore back relief features for the box

6.10.5.2.1 The relief groove on the pin is easy to machine because the machinist can see his work. But machining the relief groove in the box presents problems. One of the problems of machining a relief groove in the box is the ability of the machinist to see over the thread crest and thus assure the surfaces of the groove were smooth and free of tool marks. The surface of relief features should be free of machining marks for benefits to be seen.

6.10.5.2.2 The Bore Back relief feature was designed to eliminate these problems. Instead of a groove, the Bore Back is cut as a cylinder with the taper of the threads causing the thread roots to gradually vanish at the back of the box.

6.10.5.2.3 The Bore Back is the preferred relief feature for the box but the box relief groove is still available as an alternate.

6.10.6 Torsional balance of a rotary shouldered connection

6.10.6.1 If one will study the make-up torque equation in subclause 5.8.2.3, you will notice the term “SA”. The term “SA” is the load developed in the connection by the applied make-up torque. This load is a tensile load in the pin and a compressive load in the shoulders resulting from the applied make-up torque.

6.10.6.2 However, it is not the compressive “load” in the shoulders that keeps the shoulders together, it is the compressive “stress”.

6.10.6.3 For equilibrium to exist: $S_BA_B = S_PA_P$ or $A_B/A_P = S_P/S_B$

6.10.6.4 Also, $S_B = S_P$ if the torsional area of the box is equal to the torsional area of the pin.

6.10.6.5 So the ideal condition is developed if $A_B = A_P$ because at this point the compressive stress in the shoulders is equal to the tensile stress in the pin and the compressive stress is the maximum that can be developed by the applied make-up torque. It is desired to develop the maximum possible compressive stress in the shoulders. For drill collars, this stress is 430,9 MPa (62 500 psi) and for drill pipe it is 496,4 MPa (72 000 psi).

6.10.6.6 Another way to look at torsional balance is by the A_B/A_P ratio. Optimum conditions exist if $A_B/A_P = 1$.

6.10.6.7 For properly made-up drill collar connections, S_B is always 430,9 MPa (62 500 psi) if the $A_B/A_P = 1$ or less. This condition will always exist if the connection is torsionally box weak and the recommended make-up torque is applied.

6.10.6.8 If the drill collar connection is pin weak, the A_B/A_P ratio is greater than 1,00 and the compressive stress in the shoulders is less than 430,9 MPa (62 500 psi). And the compressive stress becomes smaller and smaller as the A_B/A_P ratio becomes greater.

6.10.6.9 As the A_B/A_P ratio increases greater than 1,00, it becomes increasingly more difficult to keep the shoulders from opening while rotating. For example, if the A_B/A_P ratio is 1,20, the average compressive stress in the shoulders is reduced to 359,1 MPa (52 083 psi). And if the ratio is 1,50, the average compressive stress in the shoulders will be further reduced to 287,3 MPa (41 667 psi). Often this is not enough compression to offset the bending caused by rotation and other down hole conditions, the result is the shoulders open on the outer periphery, the thread compound is lost and galling of the shoulders and hard-to-break connections occur.

6.10.6.10 It is extremely important to follow good make-up procedures and apply the correct amount of make-up torque if the A_B/A_P ratio is greater than 1,00. If the ratio is greater than 1,20, it may be necessary to apply more make-up torque than that which is recommended in Table A.14 (Table B.14). The exact amount depends upon the drilling conditions.

6.10.6.11 Remember the recommended range is the tabulated value to plus 10 %. If galling still persists, increase the torque in 5 % increments until the problem disappears.

6.10.6.12 In large diameter drill collars is very difficult is select a connection where the A_B/A_P ratio is near 1,00 because the number of connections to select from is limited to a very few. The solution to this problem is to use the Low Torque feature to reduce the torsional area of the box.

6.10.7 Low torque feature

6.10.7.1 The name “Low Torque” is misleading. It does not reduce the make-up required for the connection unless it is used on an OD smaller than the minimum OD for which it is recommended.

6.10.7.2 It does not change the bending strength ratio (BSR) discussed in 5.8.4 if the OD is the same. But it does permit the use of bending strength ratios as large as 3,40.

6.10.7.3 The torsional area of the box connections machined with the Low Torque feature is reduced because the counterbore is cut to a diameter larger than the counterbore on a standard “full faced” connection.

6.10.7.4 With the box torsional area reduced, the A_B/A_P ratio is reduced and the compressive stress in the shoulders is increased.

6.10.7.5 Caution should be noted in that connections with the Low Torque feature should not be mated with connections with the standard “full face” because both the outside bevels and the counterbores are different. To mate such a difference will reduce the contact face by as much as 50% causing the shoulders to yield in compression.

6.10.8 Cold working

6.10.8.1 As noted in subclause 5.8.5.1, stress is concentrated in the thread roots of a rotary shouldered connection and cyclic loading (bending while rotating) often results in fatigue.

6.10.8.2 At about the same time research work was being performed on stress relief features. Laboratory investigations were done as to the benefits of cold working the thread roots.

6.10.8.3 The conclusions of these tests indicated almost the same improvements could be achieved by cold working the thread roots of the thread.

6.10.8.4 Cold working the thread roots raises the endurance limit of the connection by as much as four times over the limit of a non-cold worked thread.

6.10.8.5 Cold working puts the thread root into a compressive state by yielding the surface of the thread root. With the surface in compression, the connection can withstand higher bending loads before fatigue begins.

6.10.8.6 Cold working can be applied by several methods: (1) cold rolling under pressure with a roller shaped like the thread form, (2) shot peening or (3) by working with a small special shaped air hammer. All methods achieve acceptable results.

6.10.8.7 It is recommended that, where fatigue failures in the thread roots are a problem, the thread roots be cold worked to improve the fatigue life.

6.10.8.8 The gauging of the connection should be done before cold working because the gauge stand-off will change. But the process will not interfere with the make-up of the connection.

6.10.9 Goodman diagram

6.10.9.1 Goodman diagram for tool joints

The Goodman diagram illustrates that the fatigue strength of a material is reduced when the static stress is increased. Generally, the higher the make-up torque for the connection the lower the allowable bending loads. The Goodman diagram can be used to evaluate the effect that the mean stress and fluctuating stresses has on pin fatigue resistance by plotting values on the diagram.

Fatigue failures in drill collar connections can occur in either the pin or box but fatigue failures in drill pipe tool joints almost always occur in the pin. These pin failures usually occur near the last engaged thread about (¾ in). from the make-up shoulder. Cyclic loading from down-hole vibrations and rotating in doglegs causes the failures.

Tool joint pins are preloaded by make-up torque that produces a stress at the last engaged thread of about 60% of its yield strength. The preload and other unvarying service loads result in a static or mean stress that has a significant influence on the potential for fatigue failures.

Tool joint boxes don’t generally experience fatigue failures because the cross sectional area and moment of inertia are great enough to prevent high stresses and because there is no preload to elevate the mean stress.

6.10.9.2 Fatigue failures in tool joints

Rotation and axial vibration of the drill string produces cyclic stresses in the tool joint. The endurance limit is the amplitude of the cyclic stresses below which a fatigue failure will not occur; that is, the stress could be repeated forever with no fatigue failure. The endurance limit of an API tool joint pin as reported by A. P. Farr in *Torque Requirements for Rotary Shouldered Connections and Selection of Connections for Drill* Collars is 13 000 psi.

$$S_A = \frac{M \cdot c_L}{I} \tag{19}$$

M is the bending moment induced on the pin;

C_L is the distance from the pin axis to the root of the thread ¾ in. from the make-up shoulder;

I is the moment of inertia of box and pin (root of the thread ¾ in. from the make-up shoulder).

The calculated stress value is not modified by a stress concentration factor. All API NC thread forms are the same regardless of the tool joint size, so the 13 000 psi number could vary some with size.

The results of the tests indicated that fatigue failures would not occur when the bending moment produced stresses less than about 13 000 psi.

When tool joints are in service, static stresses resulting from make-up torque, drill string weight and internal pressure are superimposed on the fluctuating stresses. The amplitude of the combination of these stresses is, in most cases, well below the yield strength of the tool joint pin but, under certain conditions of loading, can cause a failure in the pin.

6.10.9.3 Construction of a Goodman Diagram

The Modified Goodman diagram can be used to determine how the combination of static stresses and fluctuating stresses influence the fatigue life of tool joint. The diagram below illustrates the effect of imposing increasing static stresses on the cyclic stresses.

1) On the horizontal axis, mark the value of the material yield strength and ultimate strength. For API tool joints these values are 120 000 psi and 140 000 psi respectively.

2) Do the same on the vertical axis.

3) Mark the value of the endurance limit on the vertical axis. 13 000 psi.

4) Draw a line from the endurance limit value to the ultimate strength value on the horizontal axis. This line is the Goodman Line and represents the maximum stress amplitude at which a fatigue failure will not occur.

5) From the yield strength value on the horizontal axis, draw a line in the direction of yield strength on the vertical axis but stop the line at the Goodman line from step 5. The mean stresses are plotted on the horizontal axis. This line is the limiting mean stress in terms of material yield.

6) From the value of the mean stress on the horizontal axis, draw a vertical line to the Goodman line (step 5).

7) Finally, draw a horizontal line from the intersection of the mean stress line, step 6, and the Goodman line, step 5, to the vertical axis. The value at which this line crosses the axis is the maximum allowable stress amplitude at which a fatigue failure will not occur.

8) Alternatively, the maximum mean stress can be determined by projecting the stress amplitude value from the vertical axis to the Goodman line. This could be useful in selecting drill pipe high in the hole where the tensile loads are high.

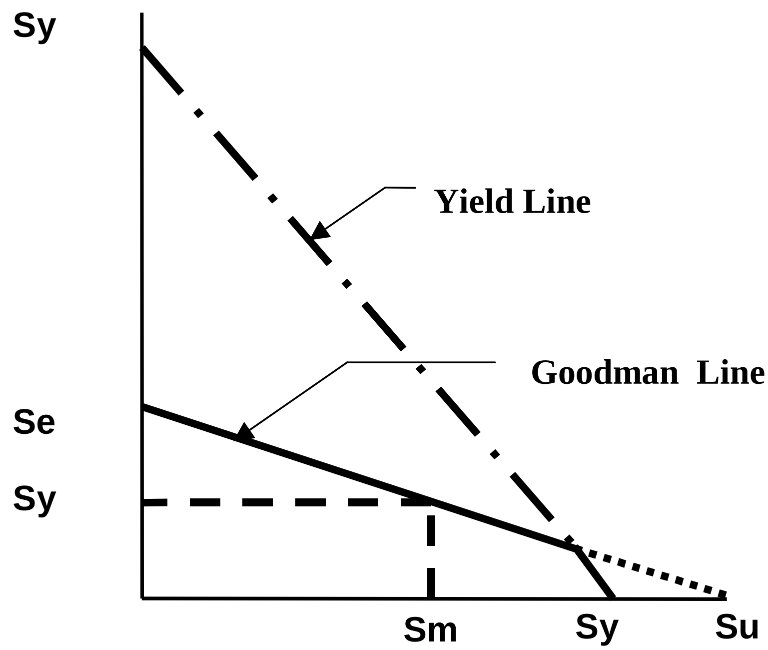


Figure 6 — Construction of Goodman diagram

6.10.9.4 Calculation of stresses for a Goodman diagram

6.10.9.4.1 General

The following formulas are used to calculate the values needed to construct a Goodman diagram. A sample calculation for the 5 X 2-11/16 NC38 is in the appendix.

Mean stress and static stress are used synonymously. Static stress is a stress that exists because of fixed loads applied to the tool joint. These fixed loads include preload from make-up torque, tensile load from supporting string weight and internal pressure.

$$S_S = S_{MU} + S_T + S_P \tag{20}$$

The mean stress is

$$S_M = \frac{S_{MAX} + S_{MIN}}{2} \tag{21}$$

For a properly made-up tool joint where the shoulders will not yield or separate during cyclic bending or variations in axial loads, the mean stress is equal to the static stress.

6.10.9.4.2 Tool joint make-up

The calculated make-up torque using the formulas in API RP 7G is 10,840 ft-lbs for a 5 X 2-11/16 NC38.

$$T_{MU} = \frac{S_{\mathrm{MU}} \cdot \mathrm{A_L}}{12} \cdot \left(\frac{\mathrm{p}}{2 \cdot \pi} + \frac{\mathrm{R_T} \cdot \mathrm{f}}{\cos\theta} + \mathrm{R_S} \cdot \mathrm{f} \right) \tag{22}$$

S_{MU} is generally equal to 72,000 psi for drill pipe and 62,500 psi for most other drilling tools. To solve for the mean stress in the pin, knowing the make-up torque, use equation

$$S_M = \frac{12 \cdot T_{MU}}{A_L \cdot \left(\frac{p}{2 \cdot \pi} + \frac{R_T \cdot f}{\cos\theta} + R_S \cdot f \right)} \tag{23}$$

The coefficient of friction is generally considered to be 0.08. The value could vary with different compounds and, if it is less, the likely hood of a fatigue failure is increased. Some drilling fluids, when not properly cleaned off the connections, could also affect the frictional properties.

6.10.9.4.3 Tensile load

The mean stress resulting from the tensile load is calculated using the following formula:

$$S_T = \frac{P}{A_L + A_B} \tag{24}$$

P is the total weight supported by the tool joint.

When P is a compressive load, S_L is negative and is subtracted from the mean stresses caused by make-up and internal pressure.

6.10.9.4.4 Internal pressure

The mean stress in the pin from internal pressure is calculated using the following formula:

$$S_P = \frac{P \cdot \pi \cdot \left(Q_C^2 - D_L^2\right)}{4 \cdot A_L} \tag{25}$$

6.10.9.4.5 Bending stress

The maximum allowable bending stress, S_a at the last engaged thread at which a fatigue failure will not occur is found in Step 7 of the Goodman diagram construction procedure. Equation 26 shows the relationship between S_A and the bending moment M induced on the tool joint.

$$S_a = \frac{M \cdot D_L}{2 \cdot (I_P + I_B)} \tag{26}$$

I_P+I_B is calculated collectively for the pin and box using

$$I_P + I_B = I_{TJ} = \frac{\pi \cdot \left(OD^4 - ID^4\right)}{64} \tag{27}$$

M is the bending moment caused by the pipe following the curvature in the hole. A more rigorous approach would be that described by Arthur Lubinski in *Fatigue of Range 3 Drill Pipe.* In Lubinski's paper, the difference in diameters of the tool joint and drill pipe and the tensile load in the pipe are considered in calculating M.

6.10.9.5 Example

Goodman Diagram for 5 X 2-11/16 NC38 tool joint made up to 10 843 ft-lb, with a tensile load of 200 000 lbs, an internal pressure of 3 500 lb and a bending load that induces 23 000 psi bending stress in 3.50 13.30 lb/ft pipe.

Figure 7 — Total steady state stress at LET of pin (psi)

This sample calculation shows that, with the tool joints made-up to 50 % of their yield torque, the maximum allowable bending stress is 5 600 psi. Assuming the pipe follows the wall of the hole, the tool joint can rotate in a 43°/100 ft dogleg without experiencing a fatigue failure. As described in reference 3 of this paper, three 5 X 2-11/16 NC38 tool joints were fatigue tested after being made-up to 50 % of their torsional yield. At the time the testing took place, API recommended new tool joints be made-up to 50 % of their yield torque therefore a make-up torque of 50% of yield was selected for the sample calculation. If the make-up torque was 60 %, or 10 800 ft-lbs as currently recommended by API, the maximum allowable dogleg or build rate would be about 35°/100 ft.

6.10.9.6 Calculated values for Goodman diagram

C is 3.808 in.

D_F is 4.578 in.

D_{LF} is 3.891 in.

f is 0.08

H is 0.216 005 in.

OD is 5 in.

OD_{PIPE} is 3.500 in.

ID is 2-11/16 in.

ID_{PIPE} is 2.764 in.

L_{PC} is 4.000 in.

P is 200 000 lbs

p is 0.25 in.

Q_C is 4.078 in.

S_{RS} is 0.038 in.

S_U is 140 000 psi

S_Y is 120 000 psi

S_E is 13 000 psi

T is 2 in/ft

T_{MU} is 9 000 ft-lbs

θ is 30°

$$R_T = \frac{(2)(3.808) - (4.0 - 0.625)\frac{2}{12}}{4} = 1.763 \quad (28)$$

$$R_S = \frac{4.578 + 4.078}{4} = 2.164 \quad (29)$$

$$A_B = \frac{\pi}{4}\left(5.0^2 - \left(4.078 - \frac{.375 \cdot 2}{12}\right)^2\right) = 6.971 \quad (30)$$

$$D_L = 3.808 - 2 \cdot \left(\frac{0.216005}{2} - 0.038\right) - \frac{2}{96} = 3.647 \quad (31)$$

$$A_L = \frac{\pi}{4} \cdot (3.647^2 - 2.688^2) = 4.772 \quad (32)$$

$$S_{MU} = \frac{(12)(9{,}000)}{4.772 \cdot \left(\frac{0.25}{2 \cdot \pi} + \frac{(1.763)(0.08)}{\cos(30)} + (2.164)(0.08)\right)} \quad (33)$$

$$S_{MU} = 60{,}000 \quad (34)$$

$$S_T = \frac{200{,}000}{4.772 + 6.971} = 17{,}031 \quad (35)$$

$$S_P = \frac{3500 \cdot \pi \cdot \left(4.078^2 - 3.647^2\right)}{(4) \cdot (4.772)} = 1{,}918 \quad (36)$$

$$S_s = 60{,}000 + 17{,}031 + 1{,}918 = 78{,}949 \quad (37)$$

$$I_{TJ} = \frac{\pi \cdot \left(5.0^4 - 2.688^4\right)}{64} = 28.12 \quad (38)$$

$$M = \frac{2 \cdot S_a \cdot I_{TJ}}{D_L} \quad (39)$$

$$M = \frac{(2)(5{,}600)(28.12)}{3.647} = 86{,}350 \quad (40)$$

$$I_{PIPE} = \frac{\pi \cdot \left(3.5^4 - 2.764^4\right)}{64} = 4.501 \quad (41)$$

$$R = \frac{E \cdot I_{Pipe}}{M} \quad (42)$$

$$R = \frac{30 \times 10^6 \cdot 4.501}{(12) \cdot (86{,}350)} = 130.3 \quad (43)$$

$$BR = \frac{180}{\pi} \cdot \frac{100}{R} \quad (44)$$

$$BR = \frac{180}{\pi} \cdot \frac{100}{130.3} = 43.30 \quad (45)$$

6.10.9.7 Conclusions

The fatigue life of pins in drill pipe and other rotary shouldered connections is reduced as the make-up torque is increased beyond the point required to prevent shoulder separation during bending. Shoulders should be tight enough to not open when they are on the outside of the bend in a dogleg and to seal the internal pressure of the drilling fluid. As the shoulder load increases due to increased make-up or string tension increases, the additional tension imposed on the pin causes a reduction in the fatigue life as is shown by examination of the Goodman diagram.

The optimum make-up torque varies for different loading conditions. The endurance limit of made-up tool joints is difficult to ascertain, especially since there will be tensile loads and internal pressure loads that add to the mean stress, but by using the technique presented here the causative factors can be evaluated.

7 Steady state drill string design considerations

7.1 Tension design of drill strings in vertical wellbores

7.1.1 Design assumptions

7.1.1.1 General

The design of drill strings for tension loads requires sufficient strength in the topmost length of each size, weight, grade, and classification of drill pipe to accommodate the three modes of loading due to tension that regularly occur during drilling operations.

This design procedure assumes minimal combined effects from hole angle, hole tortuosity, and hole depth. This design approach may be sufficient for wells with inclinations less than 10°, doglegs less than 1°/100 ft and total depths less than 12 000 ft to 15 000 ft. The combined effect of two or more of these conditions may cause significant inaccuracies in the vertical well drill string design.

A good indicator of the potential of a tortuous wellbore is correlation wells with pick-up and slack-off weights greater than 5 % of the total hook load.

7.1.1.2 Acceleration forces

Sufficient strength in excess of the buoyed weight of the drill stem should be provided to accommodate tension loads due to acceleration forces that occur when the drill pipe is picked-up or stopped. This should be accomplished by providing a minimum ratio of the maximum allowable tension load on the drill pipe and the buoyed weight of the drill stem. This ratio is called the design factor. Load measurements and experience has shown that a design factor of 1,30 is usually adequate to accommodate acceleration forces.

7.1.1.3 Slip crushing

In their analysis of slip crushing problems, Reinhold & Spiri and Vreeland demonstrated that the ratio of the hoop compressive stress (S_H) in the drill pipe that is surrounded by the slips is greater than the axial tension stress in the pipe (S_A) hanging below the slips

Sufficient strength in excess of the buoyed weight of the drill stem should be provided to assure that the hoop compressive stress in the drill pipe caused by the crushing action of the slips does not exceed the yield strength of the drill pipe. This should be accomplished by providing a minimum ratio of the hoop compressive stress (S_H) in the drill pipe that is surrounded by the slips and the axial tension stress in the pipe (S_A) hanging below the slips. This ratio is called the slip crushing constant and can be determined with the following equation:

$$K_{SCF} = \frac{S_H}{S_A} = 1 + \left[\frac{DK}{2L_S} + \left[\frac{DK}{2L_S} \right]^2 \right]^{0,5} \tag{46}$$

Slip crushing constants have been calculated for 2 3/8 through 6 5/8 drill pipe and are tabulated in Table A.32 (Table B.32). They range from 1,09 (1.09) to 1,91 (1.91) depending on the size of the drill pipe, the length of the slips, and the coefficient of friction between the backs of the slips and the master bushing. Note that the slip crushing constant increases as the pipe size increases, decreases as the slip length increases, and decreases as the coefficient of friction between the slip backs and the master bushing increases.

It is important to note that the above formula is only a guideline. Other variable that could result in crushing the pipe wall as well as catastrophic fracture of the toe section of the slip segments which are not accounted for in this formula.

The coefficient of friction between the surfaces of the slip dies and the slip segments into which they are fitted. This affects the amount of drill stem load that is supported by the toe section of the slip segment, independent of the friction between the slip segments and the drilling bowls.

Contamination of the slips and drilling bowls by certain oil-based drilling fluids may alter the friction between the slip segments and the drilling bowels, as well as between the slip dies and the segments such that crushing loads derived from the above formula may not accurately reflect the actual crushing load, as well as the load on the toe section of the slip segments.

When a portion of the toes of the slips are unsupported by the drilling bowls, an outward radial bending moment is exerted when drill sting hook load is transferred from the hook to the slips, with the fulcrum located at the last point of contact between the slips and the drilling bowls. The shear stress that results can be further exacerbated on floating drilling rigs where the plane of the rig floor is forced out of perpendicularity with the axis of the drill stem due to vessel roll and/or pitch. Such shear stresses can contribute greatly to catastrophic failure of the toe section(s) at the drill stem hook loads that are less than those revealed by the above formula that would otherwise seem to be acceptable with regard to pipe wall crushing.

The deceleration occurring at the moment when the slips become engaged and the drill stem comes to rest can have a large effect on the downward force exerted on the toe section of the slip segments, as well as the lateral crushing force on the pipe wall. Assuming “g” factors can be applied to the above formula to yield an operating range that carries a safety factor to account for this until such time as the hook loads increase to a point when this safety margin is absorbed. Rig drill crews should then take special additional precautions to ensure that the drill stem come to rest before setting the slips, and that the weight of the drill sting is transferred carefully and slowly to the slips.

In addition, this formula is only applicable to hand slips and some power slips that utilize hand slip segment used with drilling bowls. It is not applicable to certain types of power slips that do not incorporate drilling bowls. As such, the potential for pipe crushing or slip segment failure should be quantified through consultation with the slip manufacture for specific situations.

7.1.1.4 Stuck pipe and friction drag

Sufficient strength in excess of the buoyed weight of the drill stem should be provided to accommodate tension loads due to friction drag or due to pulling on stuck pipe. This should be accomplished by providing a minimum difference between the maximum allowable tension load of the drill pipe and the buoyed weight of the drill stem. This difference is called the Margin of Overpull. The magnitude of the margin of overpull should be based on consideration of both the history of friction drag in typical wells and the risk and consequence of stuck pipe.

7.1.2 Maximum allowable tension load

Tension stresses in steel drill pipe that exceeds the elastic limit will cause the drill pipe to suffer the Bauschinger effect. The Bauschinger Effect due to tension loads in excess of the elastic limit may reduce the compressive yield strength of the drill pipe by 70 Mpa to 100 MPa (10 000 psi to 15 000 psi). More significant, this Bauschinger Effect may reduce the compressive elastic limit to zero. This will make the drill pipe very prone to becoming crooked.

To avoid this potential problem, the maximum allowable tension load in the drill pipe should not exceed the elastic limit of the drill pipe. In the absence of actual stress-strain data on the drill pipe, it is reasonable to assume that the elastic limit is 90 % of the specified minimum yield strength of the drill pipe.

If the maximum allowable tension load on the drill pipe is to be equal to 90 % of the pipe body yield strength, the maximum allowable tension load in the drill pipe may be determined with the following formula:

$$F_{AT} = 0{,}9 \times F_Y \tag{47}$$

7.1.3 Buoyancy

7.1.3.1 Buoyancy is the apparent reduction in the weight of the drill stem submerged in a drilling fluid. The buoyed weight of the drill stem is determined by calculating K_{BF}, the buoyancy factor, and then multiplying the weight of the drill stem in air by the buoyancy factor. The buoyancy factor may be determined with any of the following formula.

7.1.3.2 For designs in SI units:

$$K_{BF} = \frac{7{,}842 - SG}{7{,}842} \tag{48}$$

7.1.3.3 For designs in USC units:

$$K_{BF} = \frac{490.1 - W_{\#F}}{490.1} \tag{49}$$

or

$$K_{BF} = \frac{65.44 - W_{\#G}}{65.44} \tag{50}$$

7.1.3.4 Buoyancy factors have been calculated and are tabulated in Table A.11 (Table B.11).

7.1.4 Maximum allowable buoyed tension load

7.1.4.1 The maximum allowable buoyed tension load on the drill pipe based on a the design factor or the slip crushing constant, whichever is applicable, is equal to the maximum allowable tension load determined with equation (47) divided by the design factor or the slip crushing constant, whichever is higher, and is determined with the following equation:

$$F_{DR} = \frac{0.9 \times F_Y}{DR} \tag{51}$$

If the design factor is greater than the slip-crushing constant, DR, the design ratio is equal to the design factor.

If the slip-crushing constant is greater than the design factor, DR, the design ratio is equal to the slip-crushing constant.

7.1.4.2 The maximum allowable buoyed tension load on the drill pipe based on a Margin of Overpull is equal to the maximum allowable tension load determined with equation (47) minus the Margin of Overpull and is determined with the following formula:

$$F_{MOP} = 0,9 \times F_Y - MOP \tag{52}$$

For calculations in SI units, the units of the Margin of Overpull are Newtons.

7.1.4.3 If F_{DR} determined with equation (51) is less than F_{MOP} determined with equation (52), F_{ABT}, the maximum allowable buoyed tension load is equal to F_{DR} determined with equation (51).

If F_{MOP}, the maximum allowable buoyed tension load based on the margin of overpull, determined with equation (52) is less than F_{DR}, the maximum allowable buoyed tension load based on the determined with equation (51), the maximum allowable buoyed tension load is equal to F_{MOP} determined with equation (52).

7.1.5 The maximum length of each section of drill pipe

7.1.5.1 Calculations in SI units should distinguish between Mass and Force. The force, expressed in Newtons, to hoist 1 m of drill pipe is determined with the following formula:

$$F_{DP-M} = 9,806\,650 \times W_{DP-M} \tag{53}$$

The force, expressed in Newtons, to hoist the drill collars and bottom hole assembly is determined with the following formula:

$$F_{C\&B-M} = 9,806\,650 \times W_{C\&B-M} \tag{54}$$

7.1.5.2 The maximum length of the first section of drill pipe just above the drill collars that can be used without exceeding the maximum allowable buoyed tension load can be determined using the following formula:

$$L_{DP1} = \frac{\left(\dfrac{F_{ABT1}}{K_{BF}} - F_{C\&B} \right)}{F_{DP1}} \tag{55}$$

7.1.5.3 The maximum length of the second section of drill pipe above the drill collars can be determined with one of the following formula, as applicable:

7.1.5.3.1 If L_{DP1} is based on the calculations in subclause 6.1.5.2:

$$L_{DP2} = \frac{F_{ABT2} - F_{ABT1}}{F_{DP2} \times K_{BF}} \tag{56}$$

7.1.5.3.2 If L_{DP1} is other than calculated in subclause 6.1.5.2:

$$L_{DP2} = \frac{F_{ABT2}}{F_{DP2} \times K_{BF}} - \frac{F_{C\&B} + F_{DP1} \times L_{DP1}}{F_{DP2}} \tag{57}$$

7.1.5.4 The maximum length of the third section of drill pipe above the drill collars can be determined with one of the following formula, as applicable:

7.1.5.4.1 If L_{DP2} is based on the calculations in subclause 6.1.5.3:

$$L_{DP3} = \frac{F_{ABT3} - F_{ABT2}}{F_{DP3} \times K_{BF}} \tag{58}$$

7.1.5.4.2 If L_{DP2} is other than calculated in subclause 6.1.5.3:

$$L_{DP3} = \frac{F_{ABT3}}{F_{DP3} \times K_{BF}} - \frac{F_{C\&B} + F_{DP1} \times L_{DP1} + F_{DP2} \times L_{DP2}}{F_{DP3}} \tag{59}$$

7.1.5.5 The maximum length of the fourth section of drill pipe above the drill collars can be determined with one of the following formula, as applicable:

7.1.5.5.1 If L_{DP3} is based on the calculations in subclause 6.1.5.4:

$$L_{DP4} = \frac{F_{ABT4} - F_{ABT3}}{F_{DP4} \times K_{BF}} \tag{60}$$

7.1.5.5.2 If L_{DP3} is other than calculated in subclause 6.1.5.4:

$$L_{DP4} = \frac{F_{ABT4}}{F_{DP4} \times K_{BF}} - \frac{F_{C\&B} + F_{DP1} \times L_{DP1} + F_{DP2} \times L_{DP2} + F_{DP3} \times L_{DP3}}{F_{DP4}} \tag{61}$$

7.1.6 Example calculation for a typical drill string for a vertical well

7.1.6.1 General

Two example calculations are shown, one in SI units and one in USC units. The SI and USC designs are similar except that the total depth, the margin of overpull, the length of the drill collars and the weight of the drill collars and bottom hole assembly have been rounded to nominal numbers in each system of units.

7.1.6.2 Design parameters

	For example calculations in SI units	For example calculations in USC units
Total depth	4 250 m	14 000 ft
Mud weight	1,32 SG	11 lb/gal
Tension design factor	1,30	1.30
Margin of overpull	450 000 N	100 000 lb
Length of drill collars	150 m	500 ft
Weight of drill collars and bottom hole assembly	30 000 kg	65 000 lb
Pipe inventory	New 4 1/2 in OD 16,60 lb/ft E-75 with new NC46 tool joints	
	New 5 in OD 19,50 lb/ft E-75 with new NC50 tool joints	
	New 5 1/2 in OD 24,70 lb/ft E-75 with new 5 1/2 FH tool joints	

7.1.6.3 Buoyancy

The buoyancy factor is determined using equation in subclauses 6.1.3.2 or 6.1.3.3, as follows.

7.1.6.3.1 For calculations in SI units:

$$K_{BF} = \frac{7,842 - W_{SG}}{7,842}$$

$$= \frac{7,842 - 1,32}{7,842} \tag{62}$$

$$= 0,832$$

7.1.6.3.2 For calculations in USC units:

$$K_{BF} = \frac{65.44 - W_{SG}}{65.44}$$

$$= \frac{65.44 - 11}{65.44} \tag{63}$$

$$= 0.832$$

7.1.6.3.3 The buoyancy factor may be determined by reference to Table A.11 (Table B.11).

7.1.6.4 Design of the first section of drill pipe above the drill collars

7.1.6.4.1 General

7.1.6.4.1.1 The maximum allowable buoyed tension load on the first section of drill pipe above the drill collars, based on either the design factor or the slip crushing factor, whichever is highest, is determined using equation (51), as follows:

$$F_{DR} = \frac{0{,}9 \times F_Y}{DR} \tag{64}$$

F_Y, the specified minimum yield strength of the drill pipe body for new 4 1/2 in OD 16,60 lb/ft E-75 drill pipe with new NC46 tool joints from column (11) of Table A.10 (Table B.10) is 1 470 395 N (330 558) lbs.

Reference to Table A.32 (Table B.32) shows that the slip crushing constant for 4 1/2 in OD drill pipe with 16 in slips and a coefficient of friction of 0,08 (0.08) is 1,37 (1.37). Since the slip crushing constant of 1,37 (1.37) is greater than the design factor of 1,30 (1.30), the slip crushing constant of 1,37 (1.37) is used for DR in formula (64) to determine the maximum allowable buoyed tension load based on the design factor or the slip crushing factor, whichever is greater, as follows:

7.1.6.4.1.2 For calculations in SI units:

$$F_{DR} = \frac{0{,}9 \times 1\ 470\ 395}{1{,}37}$$

$$= 965\ 953\,N \tag{65}$$

7.1.6.4.1.3 For calculations in USC units:

$$F_{DR} = \frac{0{,}9 \times 330\ 558}{1.37}$$

$$= 217\ 155\ lb \tag{66}$$

7.1.6.4.2 Maximum allowable buoyed tension load of the first section of drill pipe above the drill collars based on the margin of overpull

7.1.6.4.2.1 The maximum allowable buoyed tension load of the first section of drill pipe above the drill collars based on the margin of overpull is determined with equation (52), as follows:

$$F_{MOP} = 0{,}9 \times F_Y - MOP \tag{67}$$

7.1.6.4.2.2 For calculations in SI units:

$$F_{MOP} = 0{,}9 \times F_Y - MOP$$

$$= 873\ 356\ N \tag{68}$$

7.1.6.4.2.3 For calculations in USC units:

$$F_{MOP} = 0.9 \times F_Y - MOP$$

$$= 197\ 502\ lb \qquad (69)$$

7.1.6.4.2.4 Since F_{MOP} based on equation (67) is less than F_{DR} based on equation (64), F_{ABT1}, the maximum allowable buoyed tension load on the first section of drill pipe above the drill collars is 873 356 N (197 502 lbs).

7.1.6.4.3 The maximum length of the first section of drill pipe above the drill collars without exceeding the maximum allowable buoyed tension load

7.1.6.4.3.1 The maximum length of the first section of drill pipe above the drill collars that can be used without exceeding the maximum allowable buoyed tension load is determined using equation (55), as follows:

$$L_{DP1} = \frac{\left(\dfrac{F_{ABT1}}{K_{BF}} - F_{C\&B}\right)}{F_{DP1}} \qquad (70)$$

7.1.6.4.3.2 For calculations in SI units: The force, expressed in Newtons, to hoist the 300 000 kg mass of the drill collars and bottom hole assembly, expressed in kg, is determined with the following formula:

$$\begin{aligned} F_{D\&B} &= 9{,}806\ 650 \times W_{D\&B} \\ &= 9{,}806\ 650 \times 300\ 000 \\ &= 294\ 200\ N \end{aligned} \qquad (71)$$

W_{DP1} for new 4 1/2 in OD 16,60 lb/ft E-75 drill pipe with new NC46 tool joints from column (5) of Table A.8 is 27,34 kg/m. The force, expressed in Newtons, to hoist the mass of 1 m of drill pipe, expressed in kg/m, is determined with the following formula:

$$\begin{aligned} F_{DP-M} &= 9{,}806\ 650 \times W_{DP-M} \\ &= 9{,}806\ 650 \times 27{,}34 \\ &= 268{,}11\ N \end{aligned} \qquad (72)$$

Then

$$L_{DP1} = \frac{\left(\dfrac{873\ 356}{0{,}832} - 294\ 200\right)}{268{,}11}$$

$$L_{DP1} = 2\ 837\ m \qquad (73)$$

7.1.6.4.3.3 For calculations in USC units:

W_{DP1} for new $4^1/_2$ in OD 16,60 lb/ft E-75 drill pipe with new NC46 tool joints from column (5) of Table B.8 is 18,37 lbs

$$L_{DP1} = \frac{\left(\frac{197,502}{0.832} - 65\,000\right)}{18.37}$$

$$= 9,384 \ ft \tag{74}$$

7.1.6.4.3.4 The cumulative maximum length of the drill collars and bottom hole assemble, and the first section of drill pipe is:

Drill string elements	Lengths	
	For example calculations in SI units, m	For example calculations in USC units, ft
New 4 1/2 in OD 16,60 lb/ft E-75 with new NC46 tool joints	2 837	9 384
Drill collars and bottom hole assembly	150	500
Total length	2 987	9 884

7.1.6.4.3.5 Since the cumulative maximum length of the drill string is less than the 4 250 m (14 000 ft) total depth of the well, the length of the second section of drill pipe should be determined.

7.1.6.5 Design of the second section of drill pipe above the drill collars

7.1.6.5.1 The maximum allowable buoyed tension load on the second section of drill pipe above the drill collars

7.1.6.5.1.1 The maximum allowable buoyed tension load on the second section of drill pipe above the drill collars, based on either the design factor or the slip crushing factor, whichever is highest, is determined using equation (51), as follows:

$$F_{DR} = \frac{0,9 \times F_Y}{DR} \tag{75}$$

F_Y, the specified minimum yield strength of the drill pipe body for new 5 in OD 19,50 lb/ft E-75 with new NC50 tool joints from column 11 of Table A.8 (Table B.8) is 1 759 694 N (395 595 lbs).

Reference to Table A.32 (Table B.32) shows that the slip crushing constant for 5 in OD drill pipe with 16 in slips and a coefficient of friction of 0,08 (0.08) is 1,42 (1.42). Since the slip crushing constant of 1,42 (1.42) is greater than the design factor of 1,30 (1.30), the slip crushing constant of 1,42 (1.42) is used for DR in formula (52) to determine the maximum allowable buoyed tension load based on the design factor or the slip crushing factor, whichever is greater, as follows.

7.1.6.5.1.2 For calculations in SI units:

$$F_{DR} = \frac{0,9 \times 1\,759\,694}{1,42}$$

$$= 1\,115\,299 \ N \tag{76}$$

7.1.6.5.1.3 For calculations in USC units:

$$F_{DR} = \frac{0.9 \times 395\ 595}{1.42}$$

$$= 250\ 729\ lb \tag{77}$$

7.1.6.5.2 The maximum allowable buoyed tension load on the second section of drill pipe above the drill collars based on the margin of overpull

7.1.6.5.2.1 The maximum allowable buoyed tension load on the second section of drill pipe above the drill collars based on the margin of overpull is determined with equation (52), as follows:

$$F_{MOP} = 0{,}9 \times F_Y - MOP \tag{78}$$

7.1.6.5.2.2 For calculations in SI units:

$$F_{MOP} = 0{,}9 \times 1\ 759\ 694 - 450\ 000$$

$$= 1\ 133\ 724\ N \tag{79}$$

7.1.6.5.2.3 For calculations in USC units:

$$F_{MOP} = 0.9 \times 395\ 595 - 100\ 000$$

$$= 256\ 036\ lb \tag{80}$$

7.1.6.5.2.4 Since F_{DR} based on equation (75) is less than F_{MOP} based on equation (78), F_{ABT}, the maximum allowable buoyed tension load on the second section of drill pipe above the drill collars is 1 115 299 kg (250 729 lbs).

7.1.6.5.3 The maximum length on the second section of drill pipe above the drill collars without exceeding the maximum allowable buoyed tension load

7.1.6.5.3.1 Since the maximum length of the first section of drill pipe above the drill collars was determined in accordance with subclause 6.1.6.4.3.1, the maximum length of the second section of drill pipe just above the drill collars that can be used without exceeding the maximum allowable buoyed tension load is determined using equation (56), as follows:

$$L_{DP2} = \frac{F_{ABT2} - F_{ABT1}}{W_{DP2} \times K_{BF}} \tag{81}$$

7.1.6.5.3.2 For calculations in SI units:

W_{DP2} for new 5 in OD 19,50 lb/ft E-75 with new NC50 tool joints from column (5) of Table A.8 is 31,03 kg/m. The force, expressed in Newtons, to hoist the mass of 1 m of drill pipe, expressed in kg/m, is determined with the following formula:

$$F_{DP-M} = 9{,}806\ 650 \times W_{DP-M}$$

$$= 9{,}806\ 650 \times 31{,}03 \qquad (82)$$

$$= 304{,}30\ N$$

Then:

$$L_{DP2} = \frac{1\,115\ 299 - 873\ 356}{304{,}30 \times 0{,}832}$$

$$= 956\ m \qquad (83)$$

7.1.6.5.3.3 For calculations in USC units:

W_{DP1} for new 5 in OD 19,50 lb/ft E-75 with new NC50 tool joints from column 5 of Table B.8 is 20.85 lb.

$$L_{DP2} = \frac{250\ 729 - 197\ 502}{20.85 \times 0.832}$$

$$= 3\ 068\ ft \qquad (84)$$

7.1.6.5.3.4 The cumulative maximum length of the drill collars and bottom hole assemble, and the first, and second section of drill pipe is:

Drill string elements	Lengths	
	For example calculations in SI units, m	For example calculations in USC units, ft
New 5 in OD 19,50 lb/ft E-75 with new NC50 tool joints	956	3 068
New 4 1/2 in OD 16,60 lb/ft E-75 with new NC46 tool joints	2 837	9 384
Drill collars and bottom hole assembly	150	500
Total length	3 943	12 952

7.1.6.5.3.5 Since the cumulative maximum length of the drill string is less than the 4 250 m (14 000 ft) total depth of the well, the length of the third section of drill pipe should be determined.

7.1.6.6 Design of the third section of drill pipe above the drill collars:

7.1.6.6.1 The maximum allowable buoyed tension load on the third section of drill pipe above the drill collars

7.1.6.6.1.1 The maximum allowable buoyed tension load on the third section of drill pipe above the drill collars, based on either the design factor or the slip crushing factor, whichever is highest, is determined using equation (51), as follows:

$$F_{DR} = \frac{0,9 \times F_Y}{DR} \quad (85)$$

F_Y, the specified minimum yield strength of the drill pipe body for new 5 1/2 in OD 24,70 lbs/ft E-75 from column 11 of Table A.8 (Table B.8) is 2 211 754 N (497 222 lbf). Reference to Table A.32 (Table B.32) shows that the slip crushing constant for 5 1/2 in OD drill pipe with 16 in slips and a coefficient of friction of 0,08 (0.08) is 1,47 (1.47). Since the slip crushing constant of 1,47 (1.47) is greater than the design factor of 1,30, the slip crushing constant of 1,47 (1.47)is used for DR in formula (51) to determine the maximum allowable buoyed tension load based on the design factor or the slip crushing factor, whichever is greater, as follows:

7.1.6.6.1.2 For calculations in SI units:

$$F_{DR} = \frac{0,9 \times 2\ 211\ 754}{1,47}$$
$$= 1\ 354\ 135\ N \quad (86)$$

7.1.6.6.1.3 For calculations in USC units:

$$F_{DR} = \frac{0.9 \times 497\ 222}{1.47}$$
$$= 304\ 422\ lb \quad (87)$$

7.1.6.6.2 The maximum allowable buoyed tension load on the third section of drill pipe above the drill collars based on the margin of overpull

7.1.6.6.2.1 The maximum allowable buoyed tension load on the third section of drill pipe above the drill collars based on the margin of overpull is determined with equation (52), as follows:

$$F_{MOP} = 0,9 \times F_Y - MOP \quad (88)$$

7.1.6.6.2.2 For calculations in SI units:

$$F_{MOP} = 0,9 \times 2\ 211\ 754 - 450\ 000$$
$$= 1\ 540\ 579 \quad (89)$$

7.1.6.6.2.3 For calculations in USC units:

$$F_{MOP} = 0.9 \times 497\ 222 - 100\ 000$$
$$= 347\ 500\ lb \quad (90)$$

7.1.6.6.2.4 Since F_{DR} based on equation (85) is less than F_{MOP} based on equation (88), F_{ABT}, the maximum allowable buoyed tension load on the third section of drill pipe above the drill collars is 1 354 135 N (304 422 lbs).

7.1.6.6.3 The maximum length of the third section of drill pipe above the drill collars without exceeding maximum allowable buoyed tension load

7.1.6.6.3.1 Since the maximum length of the second section of drill pipe above the drill collars was determined in accordance with subclause 6.1.6.5.3.1, the maximum length of the third section of drill pipe just above the drill collars that can be used without exceeding the maximum allowable buoyed tension load is determined using equation (58), as follows:

$$L_{DP3} = \frac{F_{ABT3} - F_{ABT2}}{F_{DP3} \times K_{BF}} \tag{91}$$

7.1.6.6.3.2 For calculations in SI units:

W_{DP3} for new 5 1/2 in OD 24,70 lbs/ft E-75 with new 5 1/2 FH tool joints from column (5) of Table A.8 is 39,14 kg/m. The force, expressed in Newtons, to hoist the mass of 1 m of drill pipe, expressed in kg/m, is determined with the following formula:

$$\begin{aligned} F_{DP-M} &= 9{,}806\ 650 \times W_{DP-M} \\ &= 9{,}806\ 650 \times 39{,}14 \\ &= 383{,}83\ N \end{aligned} \tag{92}$$

Then,

$$\begin{aligned} L_{DP3} &= \frac{1\ 354\ 135 - 1\ 115\ 299}{383{,}83 \times 0{,}832} \\ &= 748\ m \end{aligned} \tag{93}$$

7.1.6.6.3.3 For calculations in USC units:

W_{DP3} for new 5 1/2 in OD 24,70 lbs/ft E-75 with new 5 1/2 FH tool joints from column (3) of Table B.8 is 26,30 lbs

$$\begin{aligned} L_{DP3} &= \frac{304\ 422 - 250\ 729}{26.30 \times 0.832} \\ &= 2\ 454\ ft \end{aligned} \tag{94}$$

7.1.6.6.3.4 The cumulative maximum length of the drill collars and bottom hole assemble, and the first, second and third section of drill pipe is:

Drill string elements	For example calculations in SI units	For example calculations in USC units
New 5 1/2 in OD 24,70 lbs/ft E-75 with new 5 1/2,5 FH tool joints	748 m	2 454 ft
New 5 in OD 19,50 lb/ft E-75 with new NC50 tool joints	956 m	3 068 ft
New 4 1/2 in OD 16,60 lb/ft E-75 with new NC46 tool joints	2 987 m	9 384 ft
Drill collars and bottom hole assemble	150 m	500 ft
	4 691 m	15 406 ft

Since the cumulative maximum length of the drill string exceeds the 4 250 m (14 000 ft) total depth of the well, the tension design of the drill sting can be considered complete.

7.2 Tension design of drill strings in directional wellbores

The tension design of directional wellbores should be handled differently from that of vertical wellbores because the limiting load factors have changed and the calculations are more complex.

The geometry of directional wellbores normally causes sideloads which increase torsional loads and decrease tensile loads. Higher dog legs and inclinations cause higher drill string side loads which causes higher torque and drag.

The calculation process for directional wellbores is cumbersome. It includes breaking the wellbore and drill stem into segments, performing load calculations for each segment and then summing each segment and comparing it to the capacity of the component as you travel up the drill stem. Because the likelihood for error is great and the calculations time consuming, hand calculation is not recommended. There are many commercial programs available that can improve calculation reliability and increase time for optimization.

The calculation process for directional wellbores is iterative. First, one should assume a drill string then perform the analysis. Based on the results, the drill string may need to be changed several times before finding one that will operate in a safe load range. Optimization for pump pressure, ECD, margin of overpull, or torsional strength may take many more iterations. In general, more iterations performed by the engineer will cause a more optimal design.

7.2.1 Design assumptions

7.2.1.1 Soft-string model

In 1983, Johancsik et al.1 (1984) presented a soft string model to predict the axial force profile of a bore hole. This approach assumes that the normal force between the drill pipe and the well bore wall is developed by tension and gravity. The soft-string model ignores the effects of drillstring stiffness, stabilizer placement, and borehole clearance. Consequently it generally shows reduced sensitivity to local borehole crookedness and underestimates the torque and drag. However, it has been shown that the assumption ignoring tubular stiffness results in significant error only for large dogleg severity (DLS) beyond 30° per 100 ft and for tubulars such as drill collar and casing with a significant moment of inertia.

7.2.2 Friction

Friction is defined as a force that resists the relative motion of two bodies in contact. The relative motion of a drill stem while tripping or sliding is 100 % axial. Friction resisting tripping or sliding is recorded as drag.

The relative motion of a drill stem while rotation off bottom is 100 % radial. Friction resisting rotation is recorded as torque.

The relative motion of the drill stem while rotary drilling ahead is approximately 95 % radial and only 5 % axial. Friction is recorded as torque since axial friction (drag) is negligible.

The theory behind the "soft-string" model for basic torque/drag prediction is well known in the industry. Proper application of the model requires a full understanding of the factors influencing torque and drag in the field. Total surface torque is comprised of frictional string torque, bit torque, mechanical torques, and dynamic torques. Separating these components allows more accurate definition of friction for torque projections and allows proper prioritization for torque reduction measures. Frictional torque is generated by contact loads between the drillstem and casing or open-hole. The magnitude of contact loads is determined by drillstem tension/compression, dogleg severities, drill pipe and hole size, drillstem weight, and inclination. Profile optimization and tortuosity control are therefore important measures to minimize contact loads.

Lubricity is a major factor controlling friction, and is itself largely controlled by mud and formation types. With means of predicting bit torque, the implications of using different bit types can be assessed. Dynamic torques can also significantly impact operations and should be minimized. Mechanical torque sources, such as cutting beds, borehole ledges, and stabilizer effects can be very significant and should also be minimized. The measurement of lubricity for torque and drag calculations is known as “drag coefficient”. Common drag coefficients for each mud type are as follows:

Drag Coefficient for water based mud: 0,20 to 0,30

Drag Coefficient for oil based mud: 0,18 to 0,25

7.2.2.1 Tortuosity

Directional wellbores are not drilled exactly as planned. The amount of wellbore directional control is accounted for in the application of tortuosity. It accounts for the crookedness of the wellbore not accounted for in the directional profile.

7.2.2.2 Stuck pipe and friction drag

Stuck pipe is accounted for in the margin of overpull as discussed in subclause 6.1.1.3. Because drag has a profound effect on directional wellbores, overpull should be applied to the worst case scenario which is generally “pick-up” weight or “tripping-out” weight.

7.2.3 Slip crushing and tensile capacity

7.2.3.1 Slip crushing

The issues with slip crushing are very similar to the ones stated in the subclause above. (see subclause 6.1.1.2) The difference is amount of drag in directional holes may provide some load relief while setting the slips while tripping in the hole. During standard drilling operations slips are set by slacking off on the drill stem and allowing some of the load to be absorbed by the drag encountered in the hole. The exception is when the pipe is brought in an upward motion and then slacked-off a very small distance, just enough to set the slips. The downward motion is so short that it does not allow a significant change in hook load from “pick-up weight”. This is why it is important to understanding what drilling processes (i.e. tripping out of hole) will incur heaviest slip crushing loads.

7.2.3.2 Tensile strength

Tension load is a function of weight, vertical depth, and drag. Selection of drill stem members should depend on worst case scenario. In most situations tripping out of the hole or pick-up weight will be the worst case scenario.

7.2.4 Compression

Whenever drilling high angle, extended reach, or horizontal well bores it is desirable to use compressively loaded portions of the drill stem. Drilling with drill pipe in compression causes no more damage to the drill pipe than conventional drilling operations as long as the operating conditions do not exceed the compressive service limits for the pipe (see buckling subclause 7.8.1).

7.2.5 Critical hole angle

In some wellbores such as horizontal wellbores the hole's angle is so extreme that the drill stem will not overcome axial friction and result in a downward movement. The available bit weight may come from drill stem section with an inclination below the critical hole angle. The critical hole angle equation is below:

$$\theta_{CRIT} = \arctan(1/f) \tag{95}$$

where

θ_{CRIT} = Critical Angle.

6.2.5.1 The critical sliding hole angle is the angle above which drill stem components may be pushed into the hole. Although many factors affect the coefficient of friction between the drill stem components and the wall of the hole, the type of drilling fluid used has the greatest impact (see Table A.19 (Table B.19)). Water-based drilling fluids generate the highest coefficient to friction and produce critical hole angles of about 73°. Synthetic-based drilling fluids provide the lowest coefficients of friction and produce critical hole angles of about 80°.

For most oil based muds: $75^\circ < \theta_{CRIT} < 80^\circ$

For most water based muds: $73^\circ < \theta_{CRIT} < 78^\circ$

7.2.6 Horizontal wells

Horizontal wells operate at angles above the critical hole angle. Heavyweight drill pipe in the vertical section of the hole is generally recommended to supply weight for weight on bit (see buckle subclause 7.8.1).

Operating significant lengths of the drill string in compression can cause the pipe to helically buckle and induce pipe curvatures larger than the curvature of the hole and may cause unacceptable bending stresses. Rotating drill pipe in curved portions of the hole generates cyclic bending stresses that can also cause fatigue failures. The most effective and efficient drill string design for extended reach and horizontal holes is the lightest weight drill stem that can withstand the operating environment. Using heavier components or thicker wall tubulars often increases the operating loads without reducing the bending stresses.

7.2.7 Sliding-mode drilling considerations.

The main considerations while slide mode drilling is cuttings removal, drag, differential sticking, high doglegs, and helical buckling.

In moderate to high angle wells cuttings removal while sliding is difficult while sliding because the drill stem on the low side of the hole will allow for cuttings build-up. Rotating the drill stem stirs the cuttings and allows for better cuttings removal in moderate to high angle wells. If the cuttings are allowed to build-up over an extended period of time the drill stem may become stuck.

Allowing the drill stem to remain still while circulating increases the chance of differential sticking. Precautions should be taken to prevent differential sticking.

Because all movement in drill stem is done in the axial direction all of the friction will be recorded as drag. The drill stem is said to be "stacking weight". When this happens the bit does not advance until a weight threshold is met to exceed the static coefficient of friction and weight is applied to the bit. The higher the hole angle the higher the drag will be. Drag will increase until the compression on the drill stem causes it to helically buckle. At this point no amount of additional weight will cause the drill stem to advance in the wellbore.

7.3 Torque

7.3.1 General

Do not allow your operating torque (rotating torque) to exceed your make-up torque.

7.3.2 Make-up torque and torsional strength

The make-up torque of drill pipe becomes critical when drilling deviated holes, deep holes, reaming, or when the pipe is stuck. As stated above, the well should be designed to where the operating torque does not exceed your make-up torque. Once the make-up torque limit is exceeded, the dynamic stored energy of the drill string will cause uncontrolled make-up of the connection, which may exceed the torsional strength and cause serious damage or failure.

7.3.3 Make-up torque

Most make-up torques, including API recommended make-up torques, are recommended values. With the use of a combined load curve, it is possible to increase or decrease the values of make-up torque, if needed, without adversely effecting the connection or your drill string design. See subclause 13.14 for more information.

Make-up torque and torsional strength is discussed under subclauses 13.13 and 13.14. Calculated values of torsional strength and make-up torque for various sizes, grades, and inspection classes of drill pipe are provided in Tables A.8, A.9 and A.10 (Tables B.8, B.9 and B.10). The basis for these calculations is shown in Annex E.

7.4 Pressure: internal and external

7.4.1 Burst of the pipe due to net internal pressure

Occasionally the drill pipe may be subjected to a net internal pressure. Tables A.3, A.5 and A.7 (Tables B.3, B.5 and B.7) contain calculated values of the differential internal pressure required to yield the drill pipe. Division by an appropriate safety factor will result in an allowable net internal pressure.

7.4.2 Collapse of the pipe due to net external pressure

The drill pipe may at certain times be subjected to an external pressure which is higher than the internal pressure. This condition usually occurs during the drill stem testing and may result in collapse of the drill pipe. The differential pressure required to produce collapse has been calculated for various sizes, grades, and inspection classes of drill pipe and appears in Tables A.3, A.5 and A.7 (Tables B.3, B.5 and B.7). The tabulated values should be divided by a suitable factor of safety to establish the allowable collapse pressure.

$$\frac{P_P}{SF} = P_{AC} \tag{96}$$

where

P_P is the theoretical collapse pressure from tables, expressed in (psi);

SF is the safety factor;

P_{AC} is the allowable collapse pressure, expressed in psi.

When the fluid levels inside and outside the drill pipe are equal and provided the density of the drilling fluid is constant, the collapse pressure is zero at any depth, i.e., there is no differential pressure. If, however, there should be no fluid inside the pipe, the actual collapse pressure may be calculated by the equation:

$$P_C = \frac{L_C W_G}{19.251} \tag{97}$$

or

$$P_C = \frac{L_C W_F}{144} \tag{98}$$

where

P_C is the net collapse pressure, expressed in (psi);

L_C is the depth at which P_C acts, expressed in (ft);

W_G is the weight of drilling fluid, expressed in (lb/gal);

W_F is the weight of drilling fluid, expressed in (lb/cu ft).

If there is fluid inside the drill pipe but the fluid level is not as high inside as outside or if the fluid inside is not the same weight as the fluid outside, the following equation may be used:

$$P_C = \frac{LW_G - (L_C - Y)W_G'}{19.251} \tag{99}$$

or

$$P_C = \frac{LW_F - (L_C - Y)W_F'}{144} \tag{100}$$

where

Y is the depth to fluid inside drill pipe, expressed in (ft);

W_G' is the weight of drilling fluid inside pipe, expressed in (lb/gal);

W_F' is the weight of drilling fluid inside pipe, expressed in (lb/cu ft).

7.5 Combined loading (Combined Torsion and Tension to Yield Rotary Shouldered Connection and Drill Pipe Body)

7.5.1 Introduction

Field-operating practice should always maintain operating torque below the make-up torque. Since there is no margin of safety applied in these curves, the actual dimensions of the tool joint and tube (inspection classification) should be used in the construction of the curve. Always be aware of the operating limits of the pipe in combination with the tool joint.

7.5.2 Drill Pipe Selection and Normal Operations

At torque levels up to the make-up torque of the tool joint, the Make-Up Torque THEN Tension curves are used to determine the capacity. These curves are used when the maximum torque a connection will experience is the torque applied before tension is applied. The torque value used in this set of curves is the make-up torque or the applied torque whichever is greater.

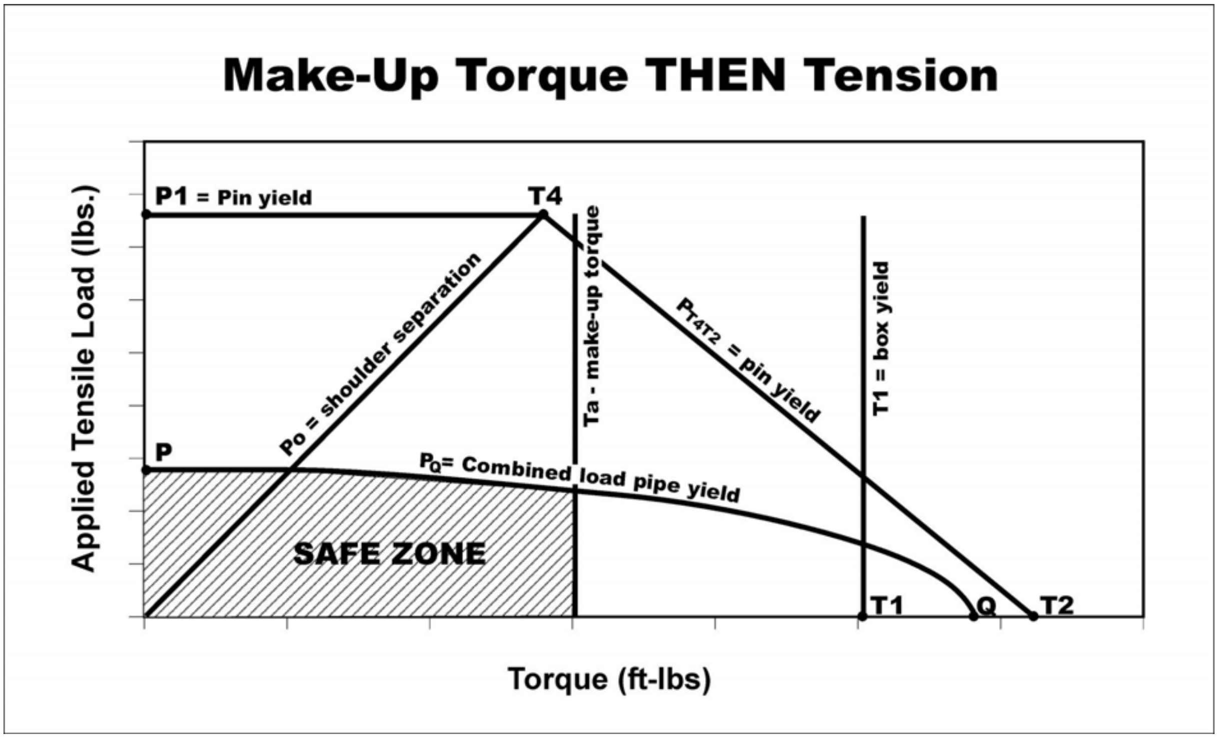


Figure 8 — Make-Up torque then tension

7.5.3 Special Operations

For operations where the applied downhole torque to the drill string might exceed the tool joint make-up torque (fishing, back-reaming, etc.), use the Tension THEN Torque curves. In these cases, the horizontal axis will represent the torque applied after tension is applied.

Caution: The dynamic loads considered in this simplified approach are different than static loads. Recommend always staying in the safe zone of operation. Going outside this zone or getting close to the boundaries can result in a catastrophic failure.

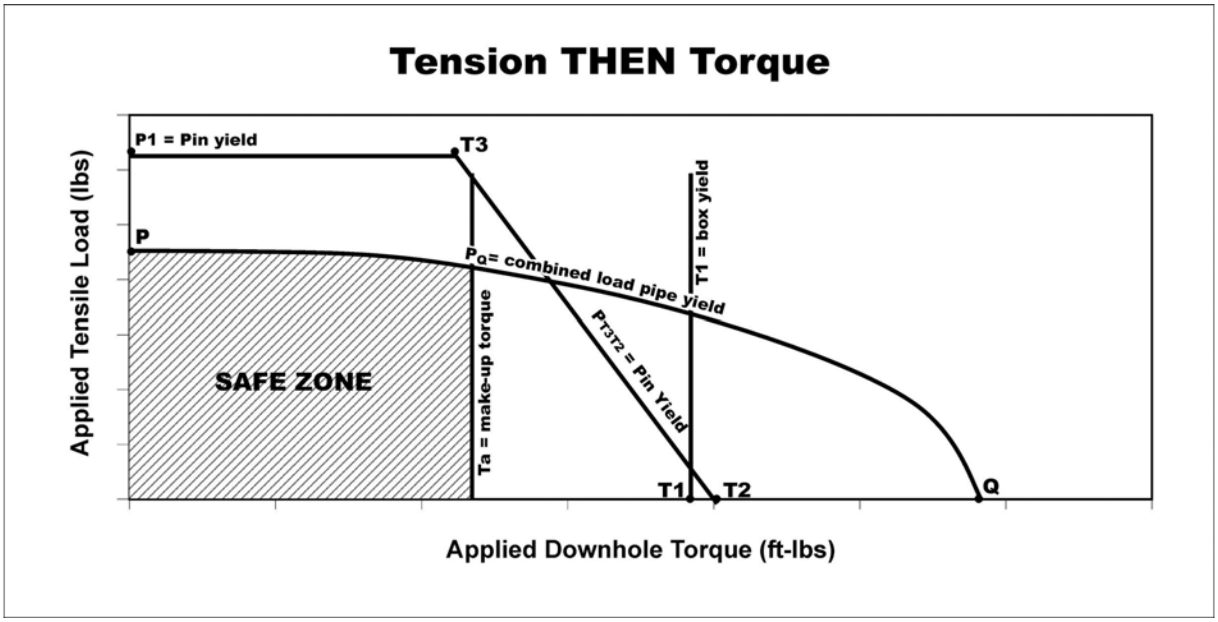


Figure 9 — Tension then torque

7.5.4 Equation Definitions:

The variables used in the equations below are defined in A.9.1.

$$P1 = Y_M A_P \tag{101}$$

$$P_O = \frac{12(A_B + A_P)T_A}{A_B\left(\frac{p}{2\pi} + \frac{R_T f}{\cos\theta} + R_S f\right)} \tag{102}$$

$$P_{T4T2} = (A_B + A_P)\left(Y_M - \frac{T_A 12}{A_P\left(\frac{p}{2\pi} + \frac{R_T f}{\cos\theta} + R_S f\right)}\right) \tag{103}$$

$$P_{T3T2} = \frac{Y_M A_P\left(\frac{p}{2\pi} + \frac{R_T f}{\cos\theta} + R_S f\right) - 12T_{DH}}{R_S f} \tag{104}$$

$$T1 = \left(\frac{Y_M}{12}\right)\left[A_B\left(\frac{p}{2\pi} + \frac{R_T f}{\cos\theta} + R_S f\right)\right] \tag{105}$$

$$T2 = \left(\frac{Y_M}{12}\right)\left[A_P\left(\frac{p}{2\pi} + \frac{R_T f}{\cos\theta} + R_S f\right)\right] \tag{106}$$

$$T3 = \left(\frac{Y_M}{12}\right)\left[A_P\left(\frac{p}{2\pi} + \frac{R_T f}{\cos\theta}\right)\right] \tag{107}$$

$$T4 = \left(\frac{Y_M}{12}\right)\left[\left(\frac{A_P A_B}{A_P + A_B}\right)\left(\frac{p}{2\pi} + \frac{R_T f}{\cos\theta} + R_S f\right)\right] \tag{108}$$

$$P_Q = A\sqrt{Y^2{}_M - \left(\frac{Q_T D}{0.096167 \cdot J}\right)^2} \tag{109}$$

A description of the values calculated above and those used to plot the curves are:

T_A = Torque that is applied to the tool joint before tension is applied, make-up torque, ft-lbs.

T_{DH} = Applied downhole torque, ft-lbs.

P1 = Yield strength of the tool joint pin at 5/8” from the make-up shoulder, lbs.

P_O = Tension required to separate the tool joint shoulders after T_A is applied, lbs. P_O is represented by the line from the origin to the point T4. Do not use this formula if T_A is greater than T_4 since P_O will be greater than P1.

P_{T4T2} = Tension required to yield pin after T_A is applied, lbs.

P_{T4T2} is represented by the line from T4 to T2.

P_{T3T2} = Tension required to yield pin after T_{DH} is applied, lbs.

P_{T3T2} is represented by the line from T3 to T2.

T1 = Torsional strength of the box of the tool joint and is represented by a vertical line at that value on the x-axis, ft-lbs.

T2 = Torsional strength of the pin of the tool joint, ft-lbs.

T3 = Torsional load required to produce additional make-up of the connection when the shoulders are separated by an external tensile load on the pipe that produces yield stress in the tool joint pin, ft-lbs.

T4 = Make-up torque at which pin yield and shoulder separation occur simultaneously with an externally applied tensile load, ft-lbs.

P_Q = Yield of the drill pipe tube in the presence of torsion represented by the elliptical curve, lbs.

7.5.5 Failure Modes:

The failure modes under combined torsion/tension loads are:

1) pin yield

2) box yield

3) shoulder separation (seal failure)

4) tube yield

7.5.6 Using the Curves:

Calculate the above values using the actual dimensions of the tool joints and tube. Use these values to plot a working curve.

EXAMPLE Drilling an Extended Reach Well

Assume:

Anticipated Maximum Drilling Torque = 25,000 ft-lbs

Drill Pipe string is 5”, 19.50 ppf, Grade S, Premium Class with NC50 tool joints (6-5/16” OD x 2-3/4” ID)

Calculated values:

P = 560,764 lbs

Q = 58,113 ft-lbs

T_A (make-up torque) = 28,381ft-lbs

P1 = 1,532,498 lbs

T1 = 47,302 ft-lbs

T2 = 62,608 ft-lbs

T4 = 26,945 ft-lbs

Questions

1 Is the drill pipe string adequate for the anticipated torque? Yes.

2 What is the allowable hook load at anticipated maximum drilling torque? P = 504,772 lbs at 25,000 ft-lbs.

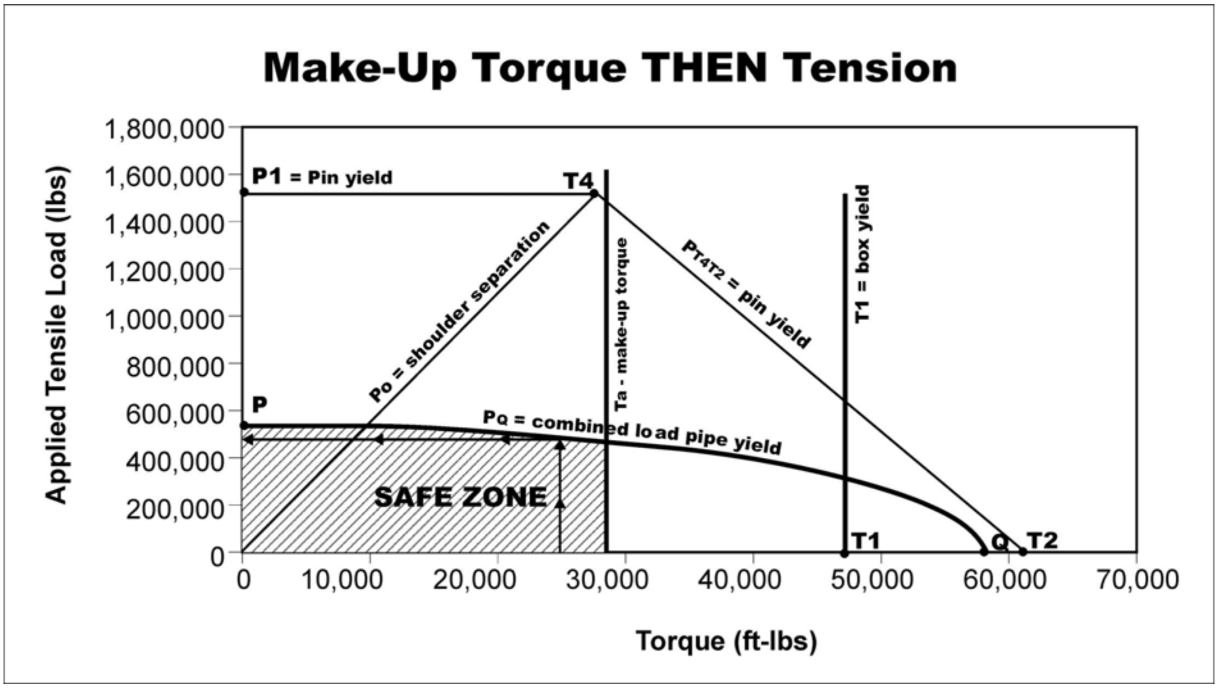


Figure 10 — Make-up torque then tension

7.5.7 Example 2 Fishing Drill Pipe String in Problem 1

Question:

What is the maximum pull without exceeding yield strength in the absence of torque?

Answer:

With a straight pull and no torque, the maximum pull is the tensile capacity of the tube which is P=560,764 lbs.

7.5.8 Example 3 Back-Reaming Hole with Drill Pipe Stinger

Assume:

Back-Reaming Pull = 300,000 lbs

Drill pipe stinger is 3-1/2”, 15.50 ppf, Grade S, Premium Class with NC38 tool joints (4-3/4” OD x 2-9/16” ID)

Calculated values:

P = 451,115 lbs

Q = 29,063 ft-lbs

Make-up Torque = 11,504 ft-lbs

P1 = 634,795 lbs

T1 = 19,174 ft-lbs

T2 = 20,062 ft-lbs

T3 = 10,722 ft-lbs

Question:

What is yield torque if back-reaming pull is 300,000 lbs?

Answer:

T_{DH} = 15,648 ft-lbs}

CAUTION — If a back-reaming pull of 300,000 lbs has a torque exceeding make-up torque (11,054 ft-lbs), the chance of pin failure increases as you approach P_{T3T2}. Always use a safety factor.

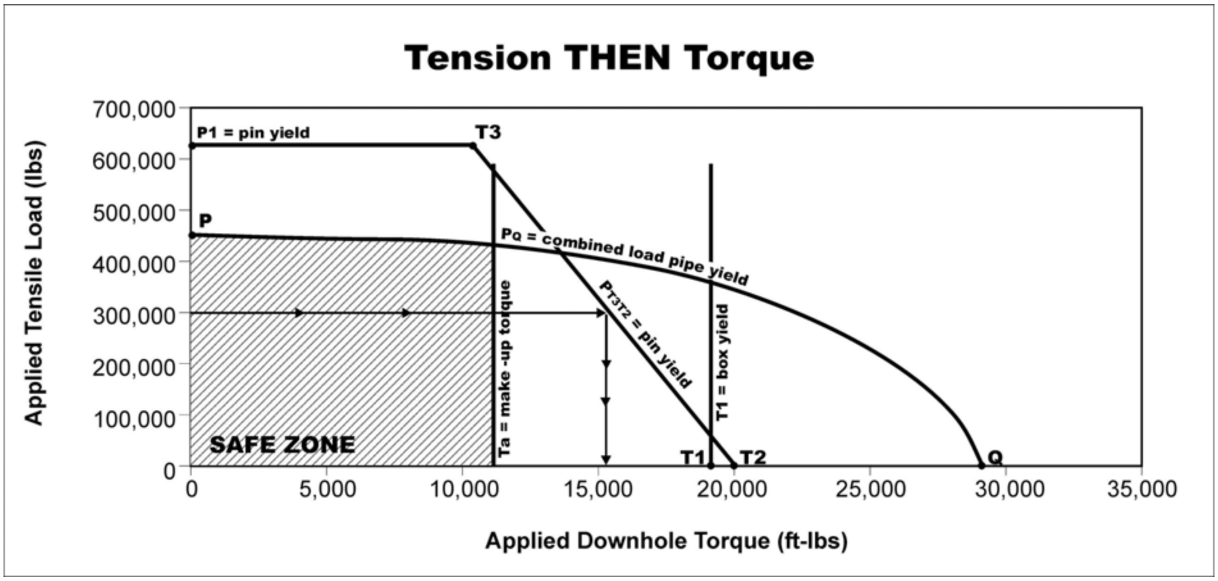


Figure 11 — Tension then torque

7.5.9 Caution

The loads considered in this simplified approach are torsion and tension. These curves are approximations that do not consider the effects of internal pressure or bending. For this reason, the answers from these curves should be derated. A safety factor of one was used.

7.6 Supplemental drill stem members

Machining of the connections to API specifications and the proper heat treatment of the material should be done on all supplemental drill stem members, such as cross-over subs, specialty tools, etc. Detailed material evaluation should be performed to assure compatibility with adjoining drill stem members.

Subs used to connect different drill pipe sections should meet or exceed the strength (tensile and torsional) of either adjacent section. Hydraulic pressure losses and pass through tolerances have the possibility of needing to be considered when selecting supplemental drill stem members.

8 BHA design

8.1 Mechanical and physical properties

8.1.1 Mechanical Properties

It is recommended that most BHA components follow the mechanical properties as specified in ISO 10424-1 for drill collars.

Drill Collar	**Minimum Yield Strength**		**Minimum Tensile Strength**		**Elongation Length 4 x Diameter**	**Minimum Brinell**
OD Range (inches)	N/mm^2	psi	N/mm^2	psi	(per cent)	Hardness
3-1/8 through 6-7/8	758	110,000	965	140,000	13	285
7 through 11	689	100,000	931	135,000	13	285

It is necessary that BHA components maintain toughness to reduce the rate of fatigue. Therefore it is recommended that all BHA components exhibit charpy impact values as specified in ISO 10424-1 for standard steel drill collars.

Specimen Size	**Minimum Average Charpy V-notch Impact Energy of Each Set of 3 Specimens**		**Minimum Charpy V-notch Impact Energy of Any Specimen of a Set**	
mm x mm	(J)	(ft/lb)	(J)	(ft/lb)
10.0 x 10.0	54	40	47	35
10.0 x 7.5	43	32	38	28
10.0 x 5.0	30	22	26	19

8.1.2 Physical Properties

8.1.2.1 Weight

One of the primary purposes of a BHA is to supply weight to the bit. The physical properties governing weight is length of BHA, OD, ID, and density. Since the density of steel is fixed, the only way for an engineer to design a BHA for available weight on bit is to adjust the length, OD, and ID of the components. Table A.13 (Table B.13) contains steel drill collar weights for a wide range of OD and ID combinations, in both API and non-API sizes. Values in the table may be used to provide the basic information required to calculate the weights of drill collar strings that are not made up of collars having uniform and standard weights.

8.1.2.2 Stiffness

In order to supply weight to the bit some of the BHA may be put into compression. Based on the stiffness of the component and the angle of the hole, a section of the BHA will be buckled. The stiffness of a component plays a large role in directional control (see section 7.6) and fatigue resistance (see section 7.8). A flexible BHA may be implemented to allow the assemble to bend or buckle more readily and thus give it the ability to change angle. Inversely, if an assembly needs to follow a straight path a stiff assembly with a higher “stiffness coefficient” may be required.

$$Stiffness\ Coefficient = E \times I$$

Where

E is Young's Modulus, expressed in pounds per square inch (psi)

I is the Moment of Inertia, expressed in inches to the fourth power (in^4)

$$I = \frac{\Pi \times (\mathrm{OD}^4 - ID^4)}{64} \tag{110}$$

8.2 Making up rotary shouldered connections

8.2.1 Application of thread compounds

See subclause 14.13.

8.2.2 Selecting the correct make-up torque

8.2.2.1 It has been observed that many of the problems associated with connections in the BHA are the result of incorrect application of make-up torque. Further, a majority of these problems are the result of insufficient make-up torque. To minimize problems in the BHA these connections should be made-up to the correct amount of torque or galling of the threads and shoulders will occur. Loose connections also accelerate the fatigue rate in the pin connection.

8.2.2.2 This section provides basic guidelines for the make-up torque requirements for equipment in the BHA. This includes all the equipment below the drill pipe.

8.2.2.3 The connections on rotary subs, reamers, stabilizers, jars, keyseat wipers and Heavy Weight drill pipe are manufactured from materials that have the same mechanical properties as drill collar material. Therefore, they should be made-up to the same make-up torque as drill collars. The recommended minimum make-up torque for drill collars is found in Table A.14 (Table B.14). The recommended range for make-up torque is the tabulated value to plus 10 %.

The one exception to this rule is the bit in that bits are not made up to the same torque as drill collars. Make-up torque for bits is found in Tables A.27 and A.28 (Tables B.27 and B.28).

8.2.2.4 If a special purpose tool is received at the rig and the crew has not had experience with that tool, contact the supplier for the recommended make-up torque.

8.2.3 Application of make-up torque

8.2.3.1 Before attempting to make-up the connections in the BHA, there are several steps that need to be taken.

a) Make sure a calibrated and working torque measuring device is used.

b) Make sure the correct make-up torque is known and used.

c) Make sure the tongs are in good working condition.

d) Make sure the length of the tongs is known and has been used in figuring the required line pull.

8.2.3.2 Care should be taken when stabbing the pin into the box connection to ensure the end of the pin is not set down on the sealing face of the box. The sealing face can easily be damaged by the impact. On connections that already have thread compound on the face, the damage can often be hidden from view by the lubricant. An unseen cut across can result in a washout downhole.

8.2.3.3 Once the small end of the pin is below the face of the box, care should be taken not to set the pin threads down on the box threads with such force so as to cause the threads to jam and cross-thread.

8.2.3.4 After the threads of the pin have engaged the threads of the box, rotate the connection slowly until the sealing faces contact.

8.2.3.5 Use both tongs to apply the full make-up torque.

8.2.3.6 Problems can occur in the initial stages of picking up the BHA. The tools immediately above the bit are the most difficult to ensure that sufficient torque has been applied because there is not enough weight below the tools to keep them straight in the rotary table. This is aggravated since there are no tools above to help stabilize the tools as they are suspended in the derrick. The result is that the tools will try to “lay over” when torque is applied, making it difficult to achieve full make-up torque.

8.2.3.7 For this reason, it is recommended that once the first 18 m (60 ft) of the BHA has been picked up, pull back out of the hole and again apply torque to all of the connections down to and including the bit. It is not necessary to break out these connections since they have not been rotated downhole. This should not be time consuming since only 2 to 5 connections are usually found in the first 18 m or so of the BHA.

8.3 [BC3]Available bit weight

8.3.1 General

8.3.1.1 Purpose

The purpose of this subclause is to design a drill stem with the correct amount of available bit weight to achieve the goals of the operator. One should consider the following:

— Weight on bit vs. optimum rate of penetration (ROP)

— Bit size and capacity

— BHA accessory capacity and requirements

— Fatigue accelerated by excessive loads

— Dynamic loading

Determine your maximum weight on bit (WOB). One should optimize rate of penetration without significantly increasing the risks of failure due to excessive loads. ROP should not be too excessive as to “overload” the hole with cuttings and cause well stability issues. Bit design and formation type are major factors in determining maximum WOB.

8.3.1.2 Limits

Manufacturers of bits and BHA components should specify the maximum load their component can withstand. Some manufacturers should also specify a minimum compressive load that their tools (i.e. certain jars and reamers) require to operate.

Beware; buckling is not permitted on some BHA equipment (i.e. jars). Buckling is a function of weight and hole angle (see Clause 8, Buckling).

8.3.1.3 Fatigue

Excessive loads due to an abundance of unnecessary available weight will cause the mean stress in the drill string to increase and will shorten the fatigue life of a drill string.

8.3.1.4 Design Factor

Due to dynamic loading of the drill string a design factor is recommended. The magnitude of the design should be adjusted for high ROPs, inconsistent formation strengths, hard formations, abrasive formations, and adverse hole conditions. Design factors between 1,2 and 1,4 should be effective for most applications.

8.3.2 Straight hole application

The maximum designed weight on weight (WOB_D) should meet the following conditions:

— Achieve the operators minimum ROP requirements.

— Be less than the maximum WOB specified by the bit manufacturer.

— Supply enough weight to allow function of BHA accessories.

Available weight on bit should be WOB_D times the design factor. This will allow the neutral point to be located safely within the drill stem members that can handle a buckled load. Accounting for buoyancy (B) and hole angle (θ), equation 105 will determine the length of BHA necessary for an available weight on bit (WOB_A).

$$WOB_A = WOB_D \times DF \tag{111}$$

$$Length_{BHA} = \frac{(WOB_A)}{[Wt \times B \times Cos\ \theta]} \tag{112}$$

For BHAs that have more that consist of one drill collar section and a heavy weight section use:

$$Length_{HWT} = \frac{[WOB_A - (Wt_{DC} \times B \times Cos\ \theta_{DC} \times L_{DC})]}{[Wt_{HWT} \times B \times Cos\ \theta_{HWT}]} \tag{113}$$

For vertical wells, Cos θ = 1.

8.3.3 Deviated hole application

The main difference between straight hole available bit weight and deviated hole available bit weight is that in some deviated applications the drill pipe can be utilized for weight as long as one does not buckle the drill pipe (see Clause 8, Buckling). Please see critical load factor for drill pipe in Clause 8, Buckling.

For horizontal well applications available bit weight may not come from directly above the bit. In holes where the drill string does not result in a downward force, the available bit weight should come from above the hole section with the inclination above the critical hole angle. The critical hole angle equation is below:

$$Critical\ Angle = \theta_{CRIT} = \arctan\left(\frac{1}{f}\right) \tag{114}$$

For most oil based muds: 75º < θ_{CRIT} < 80º

For most water based muds: 73º < θ_{CRIT} < 78º

Heavyweight drill pipe is generally recommended for hole sections on extended reach wells where the vertical drill stem member may buckle.

Utilize equations 104-106 to determine BHA lengths necessary for available bit weight in a directional application.

8.4 Fishability

When selecting drill collar sizes, there is always a possibility of having to fish them out of the hole. Fishing can mean running an overshot, or it can mean washing over. The length of time spent fishing will be determined by the cost of the material being fished for and the daily operating cost of the rig, which often makes fishing prohibitive. Only one run will be made with an overshot and fishing jars in an attempt to recover the fish. If the fish is not recovered, the hole will be plugged and sidetracked.

An important part of a drilling program should be to determine the availability of washpipe and overshots in the area where the well will be drilled. This can be a significant factor in selecting drill collar sizes.

Table A.36 (Table B.36) illustrates the maximum drill collar size that can be caught with an overshot and/or washed over. Often, some sizes of washpipe will not be available, so drill collar diameters should be reduced if washing over is a possibility.

8.5 Directional control

8.5.1 Packed hole assembly

8.5.1.1 Bottom hole assembly

The bottom hole assembly at its most basic consists of the drill bit, drill-collars and stabilizers as required. Its function is to apply weight to the bit while controlling the side force.

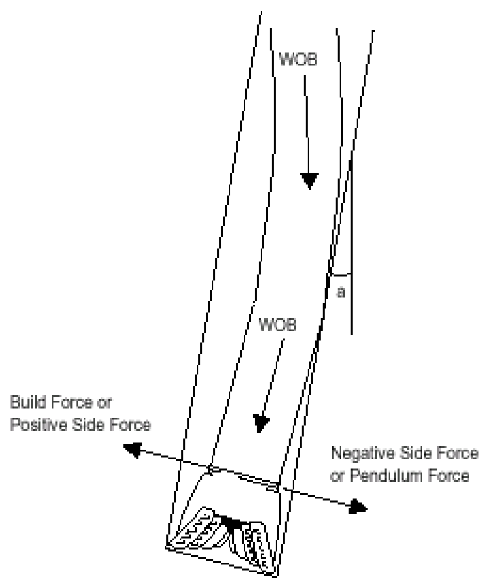


Figure 12 — BHA with Positive and Negative Side Forces

A positive side force leads to increasing inclination, while a negative side force will tend to return the hole to vertical (See Figure 12).

The simplest BHA is slick, with no stabilizers (See Figure 13).

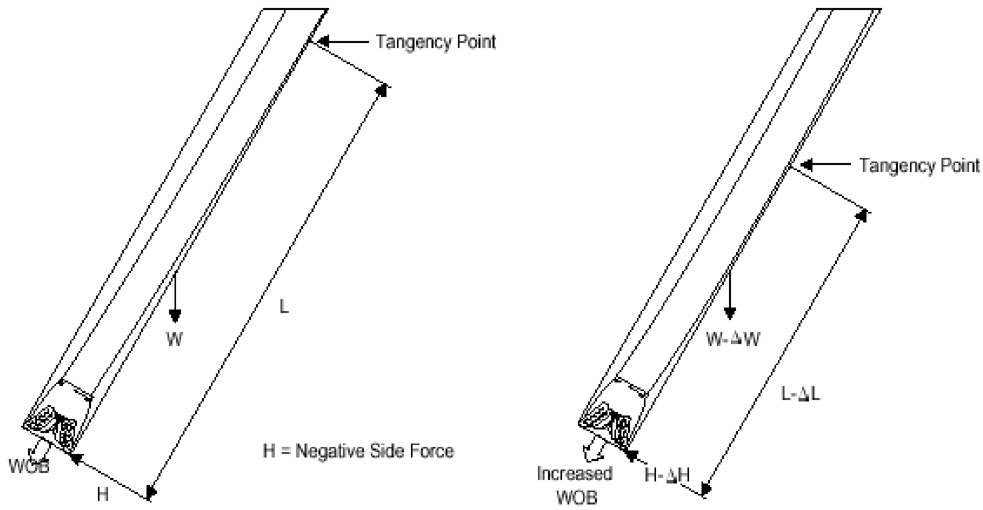


Figure 13 — Slick BHA with Pendulum Force

With zero weight on bit, a negative side force (pendulum force) only applies.

The maximum pendulum force at the bit is given by:

$$H = \frac{\left(W_C \times L_{TAN} \times BC \times \sin\theta\right)}{2} \tag{115}$$

where:

L_{TAN} is the tangency length;

BC is the Buoyancy Factor;

Wc is the weight of collars in air, expressed in pounds per foot (lbs/ft);

θ is the inclination.

The greater the hole inclination, the higher the pendulum force.

If we apply an axial load (weight on bit), a positive (bending) force is introduced. The tangency point moves closer to the bit. The pendulum force is thus reduced. A condition of zero net side force is achieved at some point. If we use stiffer drill collars, a larger pendulum force results. A higher weight on bit may be used to achieve a balanced condition. It may not even be possible. It is obvious that the uncertainty (lack of control) when using a slick assembly leads to unpredictable results. Thus, this type of BHA is not used in deviated wells.

8.5.1.1.1 Single stabilizer BHAs

An easy way to control the tangency point is to insert a stabilizer in the BHA (Figure 14). If the stabilizer is far enough back from the bit, it has no effect on BHA behavior. However, if the stabilizer is moved closer to the bit, the tangency point changes. The collar(s) between the bit and stabilizer bend when a certain weight on bit is applied. A point is reached where maximum negative (pendulum) side force occurs. Moving the stabilizer closer to the bit reduces the pendulum force. Eventually, a point is reached where zero side force occurs. Moving the stabilizer further down gives a positive side force. The collar directly above the stabilizer bends when weight is applied. The stabilizer forces the bit towards the high side of the hole. This is called the fulcrum effect. Increases in weight on bit (up to a certain point) lead to increased buildup rate. The more limber the collar directly above the near-bit stabilizer, the greater the buildup rate. The smaller the O.D. of the collar directly above the near-bit, the closer to the bit the contact point becomes. Thus, a higher positive side force is achieved. Single-stabilizer buildup BHAs are not normally used. Under no circumstances should a single stabilizer be run if, later in the hole, multi-stabilizer BHAs are to be run. More predictable BHA behavior and better hole condition results from using two or more stabilizers in every BHA.

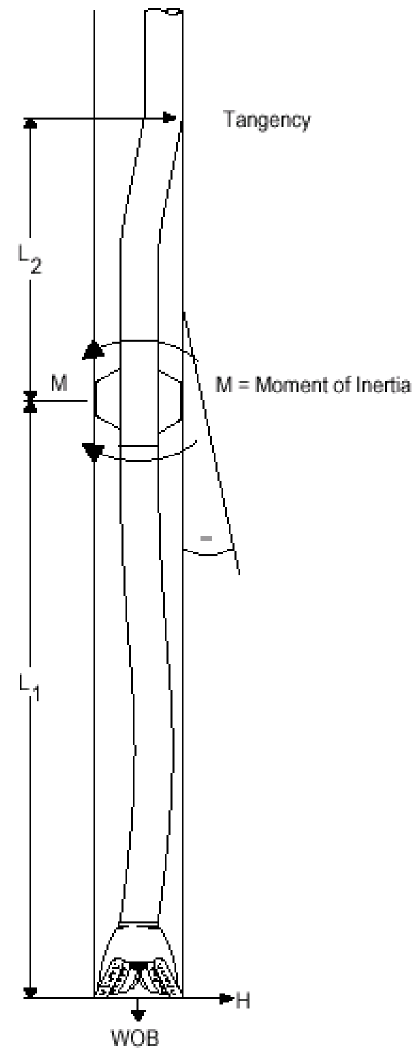


Figure 10-5 Single stabilizer BHA

Figure 14 — Single stabilizer BHA

8.5.1.1.2 Two stabilizer BHAs

The simplest multi-stabilizer BHA has a near-bit stabilizer (3'-6' from the bit to the leading edge of the stabilizer blade) and a second stabilizer at some distance above this (Figure 15). For a given weight on bit, the distance from bit to first stabilizer (L1) and between the stabilizers (L2) determines the tangency point. If tangency occurs between the bit and the bottom stabilizer, negative side force results (Figure 16). Figure 17 shows a two-stabilizer 27,4 m (90 ft) buildup BHA in which tangency occurs between the two stabilizers. Various bit and collar sizes are shown, together with the bit side forces achieved for WOB = 30 000 lb. in each case. In practice, weight on bit is one of the most important ways the DD has of controlling buildup rate.

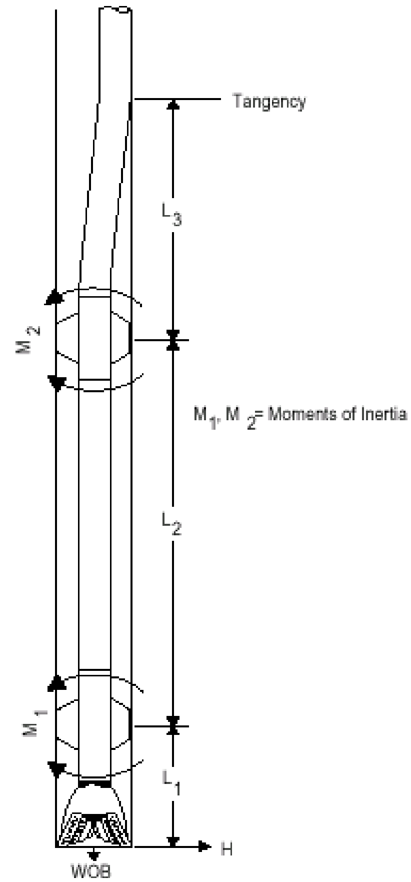


Figure 10-6 Two stabilizer BHA

Figure 15 — Two stabilizer BHA

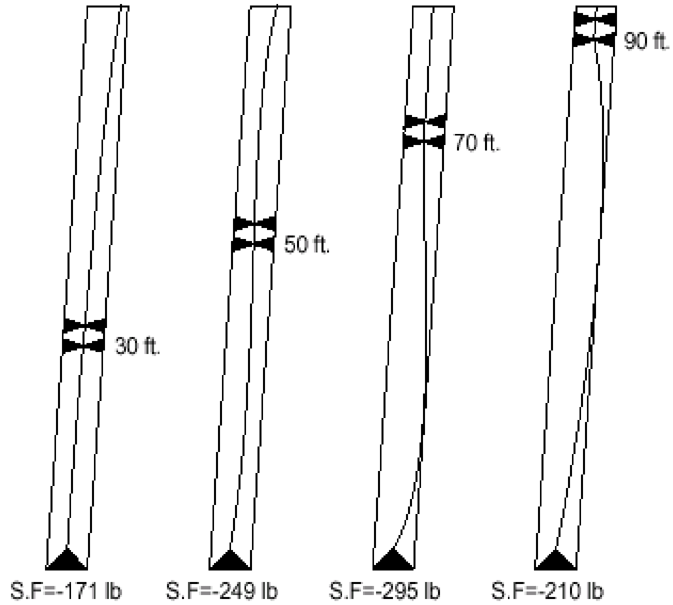


Figure 10-7 Negative side force

Figure 16 — Negative side force

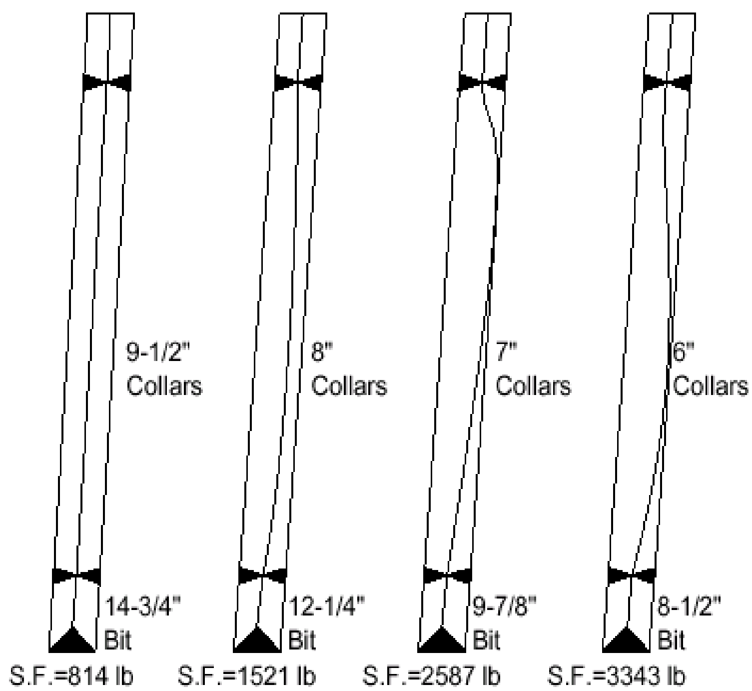


Figure 10-9 Buildup BHA using two stabilizers

Figure 17 — Buildup BHA using two stabilizers

8.5.1.1.3 Multi-stabilizer BHAs

Addition of a third stabilizer at 30' above the original top stabilizer has a significant effect on the response of a building BHA. Figure 18 is a plot of inclination versus side force at the bit for three 2-stabilizer BHAs. Figure 18 shows how the use of a third stabilizer increases the side force. In lock-up BHAs, use of the third stabilizer is essential. Otherwise, BHA behavior is erratic and unpredictable. However, in drop-off (pendulum) BHAs, two-stabilizer BHAs are normally sufficient. A third stabilizer would have negligible effect in most cases. Unless absolutely necessary (e.g. differential sticking problems), it is advisable to limit the number of stabilizers in any BHA to three. It helps keep rotary torque within acceptable limits and reduces mechanical wear on the hole. This is the approach in most locations worldwide.

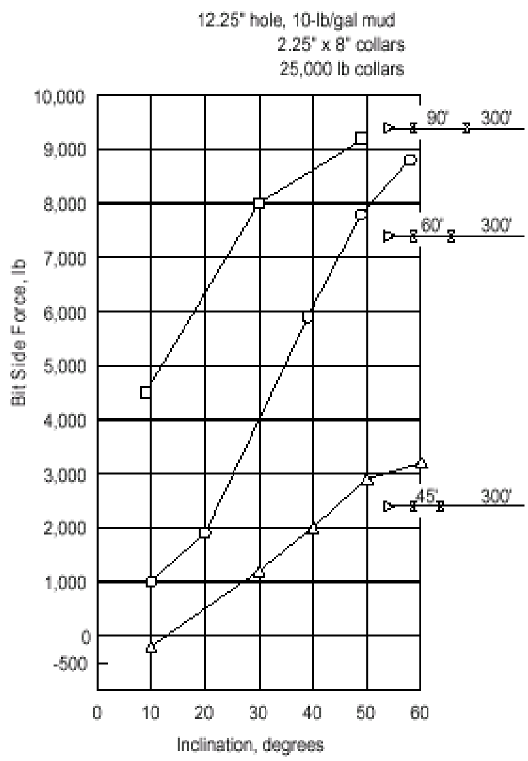


Figure 10-11 Inclination v sideforce for 3 BHAs

Figure 18 — Inclination v sideforce for 3 BHAs

8.5.1.1.4 Undergauge Second Stabilizer

If the second stabilizer is undergauge (Figure 20), it becomes easier to get a tangency point below it. It becomes easier to build angle. The more undergauge, the greater the effect. In holding (locked) BHAs, an undergauge second stabilizer is usually deliberately included in the BHA. The objective is to reach a condition of zero net side force at the bit.

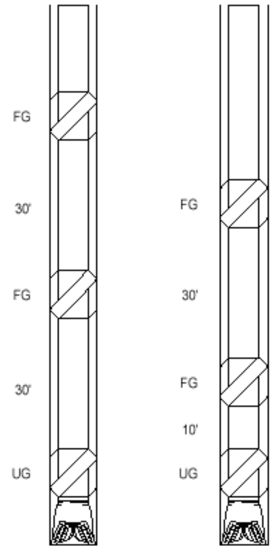


Figure 10-13 Undergauge near bit stabilizer

Figure 19 — Undergauge near bit stabilizer

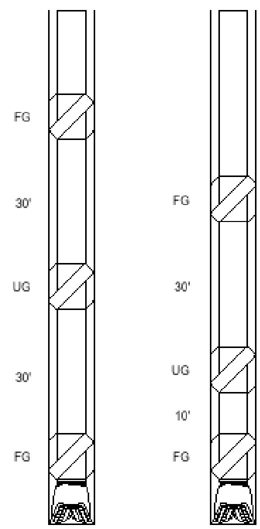


Figure 10-14 Undergauge second stabilizer

Figure 20 — Undergauge second stabilizer

8.5.1.2 Lockup BHA

A typical lockup BHA for 12-1/4" hole at 30° inclination is shown in Figure 21. If a slight build is called for (semi-build BHA), the second stabilizer should be reduced in gauge - typically down to 30,5 cm (12 in).

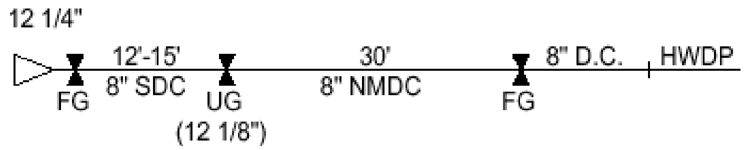


Figure 21 — Typical locked up BHA for 12-1/4 in hole

8.6 [BC4]Differential pressure sticking

8.6.1 General

Differential wall sticking is cause by the drill stem (normally drill collars) blocking the flow of fluid from the borehole into the formation. In a permeable formation, where the mud column hydrostatic head is higher than the pressure in the formation, the fluid loss can cause a vacuum type effect, locking the drill stem to the side of the wellbore.

The smooth surface of most drill stem members maximizes the cross section contact area between the wellbore and the drill stem member. Drill collars normally have the largest cross section exposed to the wellbore and are located low in the wellbore, where fresh cut formation has not had enough time, develop effective wall cake. Application of different BHA components can reduce the risks of becoming differentially stuck.

8.6.2 Stabilizers

Use of stabilizers will eliminate many of the conditions which result in sticking of the drill stem by holding the drill stem off the wall of the hole. Correct placement of the stabilizers depends mainly on drill collar stiffness and wellbore straightness. Additional stabilizers may be needed for limber drill collars in a deviated hole.

A potential adverse effect of too many stabilizers in the BHA is the abrasive effect the stabilizer may have rubbing the mud cake off the wall of the wellbore. This reduces the sealing effectiveness of the mud, increases the possibility of flow from the wellbore into the formation, and may increase the potential of becoming differentially stuck.

8.6.3 Heavy weight drill pipe

Replacing drill collars with heavy weight drill pipe (HWDP) reduces the cross section of drill stem member in contact with the well bore and reduces the vacuum effect of the under-pressured permeable wellbore. Additionally, the tool joints and center upset act as stabilizers helping to keep the tube of the HWDP off the wall of the wellbore.

8.6.4 Spiral or grooved equipment

Spiral or grooved equipment is another effective way of reducing the cross section of a drill stem member in contact with the wellbore. The depth and size of the groove and the condition of the filter cake will determine the effectiveness of this type of equipment.

8.7 Fatigue resistance

8.7.1 Stress points and fatigue

8.7.1.1 Bending Strength Ratio

A connection that has a bending strength ratio of 2,50:1 is generally accepted as an average balanced connection. However, the acceptable range may vary from 3,20:1 to 1,80:1 depending upon the drilling conditions, mud corrosivity, type of connection, and connection strengthening methods.

8.7.1.2 Drilling conditions

As the outside diameter of the box will wear more rapidly than the pin inside diameter, the resulting bending strength ratio will be reduced accordingly. When the bending strength ratio falls below 2,00:1, connection troubles may begin. These troubles may consist of swollen boxes, split boxes, or fatigue cracks in the boxes at the last engaged thread.

8.7.1.3 Mud corrosivity

The corrosive environment affects the box thread fatigue strength and, as long as sealing is assured, does not affect the pin fatigue strength. Therefore, just considering a more corrosive mud, “balanced” connections will start having box failures before pin failures.

8.7.1.4 Type of connection

In general, larger (smoother, less sharp) thread root radii will fatigue slower than thread roots with small radii. Sharp transitions cause stress concentration and decreases fatigue strength.

8.7.1.5 Connection strengthening methods

Cutting stress relief features and performing thread cold working methods to BHA connections will improve the fatigue strength of the component.

Finite element analysis and full scale testing agree that a one inch pin stress relief groove is optimal for the fatigue strength of most connections. The more tolerance that is allowed, the less fatigue strength the connection will have. However, a larger tolerance is desired for increasing the amount of re-cuts that is available for a particular connection. A recommended tolerance of 2,5 cm (1 in, +1/32 in, -5/32 in.

Cold working hardens threads by placing residual compressive stresses on the surface of the thread. As a result, connections generally gall less and do not, as easily, develop fatigue cracks.

The minimum bending strength ratio acceptable in one operating area may not be acceptable in another. Local operating practices experience based on recent predominance of failures and other conditions should be considered when determining the minimum acceptable bending strength ratio for a particular area and type of operation.

Certain other precautions should be observed in using these charts. It is imperative that adequate shoulder width and area at the end of the pin be maintained. The calculations involving bending strength ratios are based on standard dimensions for all connections.

8.7.1.6 Stiffness Ratio

Fatigue failures regularly occur at BHA transition points. A change in drill collar OD changes the stiffness coefficient of the two adjacent components. The difference in stiffness coefficient causes a stress concentration point. In order to regulate stress concentration due to drill collar transition it is recommended that a stiffness ratio of less than 3.5 be maintained throughout the BHA drill collar section.

Stiffness Ratio = Stiffness Coefficient$_{BIG}$ / Stiffness Coefficient$_{SMALL}$ < 3.5

If the components are made of the same material,

Stiffness Ratio = I_{BIG} / I_{SMALL} < 3.5

8.8 [BC5][BC6]Special problems

8.8.1 BHA buckling

BHA Buckling occurs when more weight on bit is applied than the drill collars can support. The column of drill collars “collapses” under the compressive load, and contacts the sides of the borehole to obtain side-support. As determined by Lubinski in the 1950’s, the “buckling load” can be relatively low in near vertical holes, and increases as the hole angle increases. A thorough discussion of the subject is presented in Clause 8.

As the severity of buckling increases, the side loading and curvature in the drill collars will lead to high stresses and fatigue failures in connections, and excessive localized wear on the collars. The contact forces against the side of the hole will also reduce the weight that gets to the bit, and result in a loss of rate of penetration.

Methods to mitigate buckling problems include:

— Reduce the weight on bit. This will also have the undesireable result in a loss of ROP.

— Increase the drill collar size. This will stiffen the assembly and increase the load carrying capacity of the BHA.

— Add stabilizers in the BHA. This will centralize the drill collar column in the borehole. This will reduce curvature in the collars, reducing bending stresses and fatigue problems. It will also lower the side forces required to support the load, and reduce wear damage.

8.8.2 Bit bounce

Bit bounce is a situation where large weight-on-bit fluctuations cause the bit to repeatedly lift off bottom and impact the formation. This condition often occurs when drilling with roller cone bits in hard formations.

Typical symptoms are large axial 1 to 10 Hz vibrations (shaking of hoisting equipment), large WOB fluctuations, cutter and/or bearing impact damage, fatigue cracks, and reduced ROP.

Remedial actions are:

— Run a shock sub or hydraulic thrusters

— Adjust the WOB and RPM Rev/Min

— Changing bit style

— Changing the length of the BHA

8.8.3 Bit and BHA whirl

BHA whirl is a condition where BHA whirls or orbits around the borehole. The violent whirling motion slams the drillstring against the borehole, causing torsional and lateral vibrations.

Typical symptoms are drillstring washouts/twist-offs, localized tool joint and/or stabilizer wear, increased average torque, and 5 Hz to 20 Hz lateral vibrations, even if the bit is off-bottom.

Remedial actions include:

— Lift the bit off bottom and stop rotation, then reduce the RPM Rev/Min

— Avoid drill collar weight in excess of 1,15 to 1,25 times WOB

— Use a packed hole assembly

— Reduce stabilizer rotational drag

— Adjust stabilizer placement

— Modify mud properties

— Consider drilling with a downhole motor

Bit whirl is a condition where the bit rotates in an eccentric pattern about a point other than its geometric center. This is caused by bit/wellbore gearing (analogous to a planetary gear). This mechanism induces high frequency lateral and torsional vibration of the bit and drillstring.

Symptoms include cutter impact damage, uneven bit gauge wear, over-gauge hole, reduced ROP, and 10 to 50 Hz lateral/torsional vibrations.

Remedial actions include:

— Lift the bit off bottom and stop rotation, then reduce RPM Rev/Min and increase WOB

— Consider changing bit (flatter profile, anti-whirl)

— Use slow RPM when tagging bottom and when reaming

— Pick up off bottom before stopping rotation

— Use stabilized BHA with full gauge near-bit stabilizer or reamer

8.8.4 Stick-slip

Stick-slip is a condition of non-uniform bit rotation in which the bit slows or even stops rotating momentarily, causing the drillstring to periodically torque up and then spin free. This mechanism sets up large torsional vibrations in the string.

Symptoms include surface torque fluctuations >15 % of average (below 1 Hz or stalling), increased MWD shock counts, cutter impact damage, drillstring washouts/twist-offs, connection over-torque or back-off.

Remedial actions include:

— Reduce WOB and increase RPM

— Consider a less aggressive bit

— Modify mud lubricity

— Reduce stabilizer rotational drag (change blade design or number of blades; use non-rotating stabilizer or roller reamer)

— Adjust stabilizer placement

— Smooth well profile

— Add rotary feedback system

9 [BC7]Buckling

9.1 Introduction

Buckling is the instance of lateral bending or bowing of a column due to an axial compressive load on the column. Leonhard Euler showed that there was a critical axial compressive load for buckling of a slender column. With any small compressive load, the column would remain straight and support it, not buckled or stable. With larger compressive loads, the column would bend sideways with an indefinitely large displacement; that is, it would buckle or be unstable. The greater the compressive load, the larger the lateral displacement. Euler further showed that the more slender the column (L/D ratio), the less compressive load it would take to reach the critical buckling point of the column.

Relative to drill string design, buckling analysis plays an important role. Considering the overall length and diameter of a typical drill string approximates a very slender column, drill strings are prone to buckle under compressive weights that are commonly used for weight on bit.

For rotary mode drilling, it is important to define what portions of the drill string are buckled and position buckling performance components in those areas to better withstand the cyclic loading induced by simultaneous rotation and bending due to buckling. For sliding mode drilling of little to no rotation of the drill string, it is important to define at what point the buckled shape of the drill string can prevent the application of additional weight to the bit, “lock-up.”

9.2 Rotary mode considerations

The most conservative method of designing for rotary mode drill string buckling is to assume standard drill pipe is not tolerant to simultaneous rotation and buckling. That is, the mean stress of the compressive load applied and the peak stress induced by the shape of the buckle and other stress concentrators found on drill pipe (slip cuts, internal upset geometry) will lead to rapid fatigue damage and potential washout or twist off. This does not mean that drill pipe cannot safely operate in simultaneous rotation and compression. It simply means that it is undesirable to operate drill pipe in rotation and compressive loads that exceed the drill pipe’s critical buckling load.

A second assumption is that when buckling the drill string is unavoidable, as often the case in drill string design, the use of heavy weight drill pipe and then drill collars is preferred. Heavy weight drill pipe, unlike standard drill pipe, does not have internal upsets and its wall thickness is generally thicker than standard drill pipe. The product’s stiffness ratio from the tube to the connection is more balanced than that of standard drill pipe. Drill collars with thicker wall sections also support more stress than that of drill pipe.

Employing this design methodology simply requires the definition of buckled areas within the string for the given wellbore geometry and operational parameters (such as WOB) and positioning of HWDP and drill collars in those areas of the string which are buckled. Alternatively, the designer may define the WOB value, that which begins to buckle drill pipe and safely maintain bit weight, beneath that value.

For rotary mode drilling, prediction of the onset and location of the first order of buckling, sinusoidal buckling, is required.

9.2.1 Drill pipe buckling in straight, vertical boreholes

The most common and most conservative equation used for the prediction of the onset of sinusoidal drill string buckling is the Dawson-Paslay equation[2].

$$F_{CRIT} = 2\sqrt{\frac{EIW_{DP}K_B \sin\vartheta}{h_C}} \tag{116}$$

Where

F_{CRIT} is the axial compressive force to induce buckling, expressed in pounds (lbs);

E is Young's modulus and is 30 X 10^6 psi for steel and 10.5 X 10^6 psi for aluminium, expressed in pounds per square inch (psi);

I is the Drill pipe moment of inertia with respect to its diameter, calculated by Equation 17, expressed in inches to the fourth power (in^4);

W_{DP} is the Air weight of the pipe, expressed in pounds per foot (lb/ft);

K_B is the Buoyancy factor;

ϑ is the wellbore inclination, expressed in radians;

h_C is the radial clearance between tool joint and hole, expressed in inches (in);

$$h_C = \left[\frac{D_H - D_{TJ}}{2}\right] \tag{117}$$

D_H is the hole diameter, expressed in inches (in);

D_{TJ} is the tool joint outside diameter, expressed in inches (in).

Note that a key variable in the Dawson-Paslay equation is ϑ, the wellbore inclination. If ϑ is zero, as that for a straight, vertical well, the critical buckling load is zero. This means that for any given straight, vertical well, all drill string components in mechanical compression (less than zero load) are assumed to be buckled. Though this may seem counter intuitive, that a large drill collar cannot withstand any compression before it buckles, the slenderness of the drill string as a whole supports the conservative approach of this methodology.

Designing for rotary mode drill pipe buckling in a straight, vertical well is simplified by defining only the location of the neutral point, the point at which components beneath are in compression and components above are in tension. For straight, vertical wells this neutral point may also be referred to as the zero buckling point.

Refer to Section 7.3 for BHA design methodologies that position the neutral point within the BHA and avoid placing drill pipe in compression.

9.2.2 Drill pipe buckling in straight, inclined boreholes

Another key component of the Dawson-Paslay equation is the product of the three variables $W_{DP}K_B\sin\vartheta$. This component of the equation defines the magnitude of load applied from the drill pipe directly to the wellbore or casing, commonly referred to as the side load. The equation predicts that as side load increases, the magnitude of compressive force required to buckle drill pipe increases with it. This emphasizes that in a

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2) Dawson, Rapier and Paslay, 1984.

straight, vertical wellbore where no side load is present; the magnitude of compressive force to buckle the drill string is less than zero.

Side loading improves the stability of the drill string and increases the drill string's resistance to buckling. The most buckling stable drill string would be a string at 90 degrees horizontal when $\sin\vartheta$ equals 1.0. This string could carry the most compressive load without buckling, other things equal.

Since drill pipe can support some level of compression and remain stable, or not buckled, in an inclined wellbore, it sustains the conclusion that drill pipe can be rotated under compression provided the compressive load does not exceed the critical buckling load. For straight, inclined wellbores, the “neutral point” and the zero buckling point no longer coexist in the same location. Some components may be in compression and buckled and some may be in compression and not buckled.

Similar to drill string design for straight, vertical wells, drill string design for straight, inclined wells should follow the same approach: determine which portions of the string are buckled under the operational parameters desired and position heavy weight drill pipe and drill collars in those locations rather than standard drill pipe. The only difference is not in the approach, but in the methodology used. All components in mechanical compression are buckled in a straight, vertical well and components in compression shall exceed their buckling critical load to buckle in straight, inclined wells.

The curves shown in Figures 46 through 66 predict the approximate axial compressive load at which sinusoidal buckling is expected to occur within drill pipe in straight, inclined wellbores. These curves are based on the Dawson-Paslay equation discussed elsewhere. The curves are reproduced here with permission from Standard DS-1, Drill Stem Design and Inspection[3]. The assumptions behind these curves include:

— Pipe weight is new nominal with X-grade tool joint dimensions (where applicable). If more than one tool joint is used on a particular pipe, the most common one was selected. Tool joint diameter is the minimum for Premium Class. Radial clearance is the distance between the tool joint OD and the hole ID.

— Pipe wall thickness is new nominal.

— The wellbore is straight.

— The effects of torque are neglected.

— Mud weight is 12.0 lb/gal.

9.2.2.1 Compensating for different mud weight

If the actual mud weight is not 12.0 lb/gal, critical buckling load may be adjusted by the following formula:

$$\left(F_{CRIT-ADJ}\right)=\left(F_{CRIT}\right)\left(f_{MW}\right) \tag{118}$$

Where

$F_{CRIT-ADJ}$ is the adjusted critical buckling load, expressed in pounds (lbs);

F_{CRIT} is the critical buckling load from curve, expressed in pounds (lbs);

f_{MW} is (Buoyancy factor/0,817) $^{0.5}$ (see below).

Mud Weight	Mud Weight

3) *Standard DS-1, Drill Stem Design and Inspection, Second Edition*, 1997.

(lb/gal)	f_{MW}	(lb/gal)	f_{MW}
8.0	1.04	14.0	0.98
9.0	1.03	15.0	0.97
10.0	1.02	16.0	0.96
11.0	1.01	17.0	0.95
12.0	1.00	18.0	0.94
13.0	0.99	19.0	0.93

9.2.2.2 Using the drill pipe buckling curves

Enter the curve for the correct pipe size and weight at the hole diameter. Read vertically to intersect the hole angle, then horizontally to read critical buckling load. Compensate the value obtained for mud weight being used.

Example

How much compressive load may be applied to 5-in, 19.50 lb/ft drill pipe in a 12 ¼-in horizontal hole before the drill pipe buckles? Mud weight is 9.0 lb/gal.

Solution

Reading from the figure for drill pipe in question (Figure 60), the critical buckling load is about 28 200 lbs. Adjusting to 9.0 lb/gal mud:

$$F_{CRIT-ADJ} = \text{28 200 lbs (1.03) = 29 000 lbs} \tag{119}$$

9.2.3 Drill pipe buckling in curved boreholes

The critical buckling load for compressively loaded drill pipe is also significantly influenced by the curvature of the borehole. In building and turning intervals, increasing curvature of the borehole increases the magnitude of compressive force required to buckle the pipe. In addition to the pipe's own weight acting on the wellbore or casing, a resultant side load is generated through compression of the pipe into the curvature of the wellbore. Increased side loading provides stability to the pipe and reduces its tendency to buckle. This enables drill string designers to disregard pipe buckling in building and walking boreholes since the tendency is to buckle in straight, vertical and straight, inclined boreholes adjacent to the building and/or walking borehole.

Drill pipe buckling in dropping boreholes may not be disregarded. When compressed, drill pipe resting on the low side of the hole in a dropping borehole has a tendency to lift off the low side. This lowers the resultant side load and increases the pipe's tendency to buckle. Calculation of the critical buckling force in dropping boreholes is complex and should be analyzed through computer buckling simulators. For rotary drilling, bit weights to prevent buckling of drill pipe in dropping boreholes should be utilized.

9.3 Sliding mode considerations

For rotary mode drilling, buckling analysis is performed to minimize fatigue damage to standard drill pipe. Since the drill string is confined to a wellbore or casing, the probability of compressing the string such that large lateral displacements yield the pipe plastically is low. Typical drill string and wellbore diameters do not permit large lateral deflections to generate stress levels greater than the pipe's yield strength. These stress levels combined with rotation, however, generate cyclic stress that rapidly fatigue drill pipe.

For sliding mode applications when rotation of the string is not present, buckling analysis is often directed at the prediction of the second order of buckling, helical buckling. Unlike a sinusoidal shape, helical buckling takes the shape of a helical spring. This form of buckling occurs during compressive loads well exceeding the sinusoidal critical buckling load predicted by the Dawson-Paslay equation. Helical buckling is most commonly found in long horizontal sections of wellbores.

Helical buckling is detrimental because the geometry of the helix redirects the axial compressive load in a radial direction toward the wellbore, increasing side load. As additional compressive weight is applied, the load is no longer transmitted axially to the bit. In extreme conditions, weight cannot be directed to the bit to continue drilling and “helical lock-up” of the string occurs.

Most drill string buckling simulators predict the onset of helical buckling. Due to the complexity of the analysis, readers are encouraged to use these simulators when predicting the onset of helical buckling during sliding mode drilling.

10 Cyclic loading of drill strings

10.1 Introduction

Most drill string failures are due to fatigue from fluctuating or cyclic downhole loading (see definitions 3.19, 3.20, and 3.21). Fatigue damage can occur at much lower average loads, or stresses, than the material yield strength used in designing the static limits of the drill string. Thus, cyclic loading should be considered, in addition to the properties of drill string elements (Clauses 4, 5, and 6) and the design calculations of Clause 7. Often cyclic, rather than static loading, is the constraint in drill string design and in setting operating limits in drilling and well design.

Cyclic loading arises from a variety of sources. When a drill string is rotated in a section of the hole in which there is a change of hole angle and/or direction, commonly called a dogleg, cyclic bending loads occur. Rotating a drill string, which has buckled under compressive loading, gives rise to cyclic bending loads. Drill string vibrations, which may be axial, lateral, or torsional, generate cyclic loading. Jarring, tripping, and drilling from a floating vessel also create cyclic loading. Even though it may not be possible to eliminate these conditions and operations, they can be controlled to reduce the severity of cyclic loading.

The magnitude of alternating stresses that result from cyclic loading are largest and most damaging at sudden changes in cross-section of either the drill string or within the design of drill string components. Alternating stresses are also large at localized stress risers such as threads, slip marks, formation cuts, and corrosion pits. Generally, the larger the alternating stresses, the shorter the fatigue life.

In non-corrosive conditions, steel drill string components can tolerate a reasonable level of alternating stresses without incurring fatigue damage. However, since most drilling environments are somewhat corrosive and since actual downhole conditions are rarely completely known, it is difficult to eliminate fatigue damage. Further, fatigue damage is cumulative and irreversible, ultimately leading to the initiation and propagation of a fatigue crack. Damage from prior wells can lead to unexpected failures in a well without large cyclic loading.

Despite these facts, the following Clauses and the references in the Annexes provide recommended practices to reduce drill string fatigue damage and to reduce the probability of downhole failure.

10.2 Fatigue limits

10.2.1 Introduction

Fatigue damage is irreversible and cumulative, and it occurs due to cyclic stresses on the drill pipe. Fatigue accumulation creates and propagates incipient microscopic cracks in the body of the drill stem component. Eventually, these microscopic cracks manifest as macroscopic cracks. If they are undetected by inspection techniques, these cracks can cause catastrophic failure. Assessment and prediction of fatigue damage in drill pipe has been a persistent problem since the nineteen forties[4)].

4) Grant, R. S. and Texter, H. G., 1941.

Fatigue damage can occur at much lower stresses than the material yield strength used in designing the static limits of the drill string. Thus, cyclic loading should be considered, in addition to the design calculations of Clause 7.

Cyclic loading arises from a variety of sources. It occurs when the drill string:

— is rotated in a section of the hole in which there is a change of hole angle and/or direction, commonly called a dogleg,

— is buckled and is rotated,

— experiences axial, torsional or flexural (bending) vibrations due to dynamic effects,

— experiences jarring, tripping, or other vibrational motion transmitted from a floating drilling vessel

Even though it is not possible to eliminate conditions that cause cyclic loading, they can be controlled to reduce the severity of cyclic stresses.

The alternating stresses that result from cyclic loading are largest and most damaging at sudden changes in cross-section of the drill string or sectional changes within a drill string component. Alternating stresses are also large at localized stress risers such as threads, slip marks, formation cuts, and corrosion pits. Generally, the larger the alternating stresses, the shorter the fatigue life.

In non-corrosive conditions, steel drill string components can tolerate a reasonable level of alternating stresses without incurring fatigue damage. However, since most drilling environments are somewhat corrosive and since actual downhole conditions are rarely completely known, it is difficult to assess fatigue accurately or eliminate fatigue damage entirely. Nevertheless, guidelines to minimize fatigue can be developed by rational assessment of the factors that cause fatigue.

Designing drill pipe to withstand fatigue loading safely is significantly different from other aspects of drill pipe structural design. Fatigue is an inherently scatter dependent phenomenon, and the designer ideally requires significant amounts of test data to assess fatigue damage or design a component to withstand the effects of fatigue. However, such data, especially for drill pipe and drill stem components is scant and far between. Due to the signifcant (and inevitable) scatter that characterizes high cycle fatigue data, the design cannot be based on strict deterministic principles. Due to the inherent randomness of fatigue data, the available fatigue models can, at best, be used to estimate conservative limits within which there is a high probability of avoiding fatigue damage. Therefore, this clause differs from the clauses in this document that address other aspects of drill pipe structural design. The following subclauses provide the rationale of fatigue assessment as opposed to merely listing the rules. The designer who uses these guidelines should note that the data, tables etc. provided in this clause are recommendations, and that they should be evaluated for the specific situation at hand. When faced with a situation that is not covered in these sections, the reader is urged to consult appropriate reference material cited throughout the text.

These facts notwithstanding, the following Clauses and references in the annexes provide recommended practices to assess probable fatigue damage to the drill string and to reduce the probability of downhole failure.

10.2.2 Fatigue Assessment- An Overview

10.2.2.1 General

Theoretical models to estimate fatigue of mechanical components subjected to cyclic loading are classified into models based on “cumulative damage” and models based on “linear elastic fracture mechanics (LEFM).” Historically, fatigue analyses of drillpipe have tended to use cumulative fatigue models [5), 6)] , though LEFM

5) Hansford, J. E. and Lubinski, A., 1964.

6) Hansford, J. E. and Lubinski, A., 1966.

based models[7),8),9)] have been used on occasion. The cumulative fatigue damage model (CFDM) relies on three basic notions, viz., the S-N curve, the failure criterion, and the damage accumulation rule. Given the S-N curve and the magnitude of the variable stresses, the calculation of fatigue damage requires careful tracking of the number and magnitude of stress cycles.

The S-N curve plots the peak value of cyclic stress (stress amplitude) as a function of the number of cycles required to fail the component. The stress amplitude required to cause failure at a given number of cycles is known as the fatigue strength. Figure 22 shows the S-N curve for a typical ferrous material operating in a non-corrosive environment. In this figure S_{UT} represents the ultimate strength of the material. When the number of cycles is greater than 1 000, the S-N curve is characterized by an inclined line and a horizontal line. The horizontal line represents the “endurance limit” and it is usually regarded as the fatigue strength that corresponds to a life of one million cycles (10^6). At stress amplitudes less than the endurance limit, the material can be cycled infinitely with a 50% probability of failure. The S-N curve is usually specified at a zero mean stress, though it is not necessary to do so. (The mean stress is the average of the maximum and minimum values of the cyclic stress.) Finally, it should be noted that the endurance limit need not always exist. For example, aluminum alloys and even ferrous materials operating in a corrosive environment do not exhibit an endurance limit.

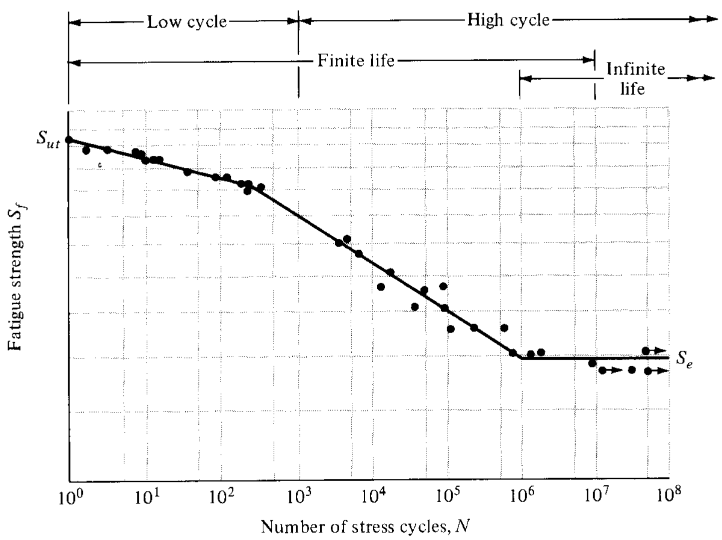

Figure 22 — A typical S-N curve for a material with an endurance limit

Since the S-N curve is specified at a specific mean stress, a rule to determine the S-N curve at an arbitrary mean stress is required. The “failure criterion” provides this rule. Typical examples of this rule are the Goodman Rule, the Gerber Rule and the Soderberg Rule. The Goodman rule [3)] or modifications thereof [4)] have been typically used in the analyzing drill pipe fatigue. The Goodman rule is discussed in further detail in subclause 9.2.2.2.3.

The damage accumulation or “life” rule is used to track fatigue damage caused by stress cycles that occur at different stress levels. The most commonly used rule for fatigue of ductile materials is Miner’s rule[10)].

7) Kral, E., Sengupta, P. K., Newlin, L., Quan, S. S., 1984a.

8) Kral, E., Sengupta, P. K., Newlin, L., Quan, S. S., 1984b.

9) Dale, B. A. 1988.

10) Miner, M. A., 1945.

If n_1, n_2, … represent the number of cycles at stress amplitudes S_{F1}, S_{F2},… respectively, then the total damage D imposed due to all the cycles $N = \sum_i n_i$ is given by

$$D = \sum_i \frac{n_i}{N_i} \tag{120}$$

where

N_1, N_2,…are the cycles to failure at S_{F1}, S_{F2},…respectively.

The component is said to have failed when $D = 1$. Miner's original article [8] indicates that failure can occur for values of D between 0.61 and 1.45, though for random loading $D = 1$ is known to give satisfactory results[11]. The oil industry has typically used a value of 1. It should be however remembered, that Miner's rule does not account for the effects of loading sequence. It assumes that n_1 cycles at a stress amplitude S_{F1} followed by n_2 cycles at S_{F2} is identical to n_2 cycles at S_{F2} followed by n_1 cycles at S_{F1}. The loading sequence is important if any of the stress levels are greater than the yield stress, i.e. cause plastic strains. In drilling situations, this is not usually a concern, and Miner's rule is generally applicable.

10.2.2.2 Fatigue limits for API drill pipe

10.2.2.2.1 Drill pipe fatigue data

A primary requirement for the calculation of drillpipe fatigue limits (or assessment of fatigue damage) is the S-N curve (rather than the endurance limit). There are three major sources of drillpipe S-N curve data in oilfield literature: Bachman's data set published in 1951[12], Morgan and Roblin's data set published in 1969[13] and Grondin and Kulak's data set published in 1991[14].

The first set of data was published in 1951 by Bachman of Hughes Tool Company. The study reported the results of a fifteen year investigation on fatigue of 4½ in. 16.6 ppf, Grade D[15] and E drillpipe. The 84 data points were obtained by subjecting full scale specimens of drill pipe to rotation as a cantilever. Since one of the goals of the study was to assess the fatigue resistance of the tool joints, the tool joint was located at the fixed end of the cantilever. Importantly, all the tests were conducted at zero mean stress in air.

The second major data set was published by Morgan and Roblin of the Youngstown Sheet and Tube Company in 1969[11]. This study published the results of fatigue testing on small scale coupons cut from "as produced" seamless drillpipe. Results were reported for grades E-75, X-95, G-105, S-135 and an experimental grade, then known as V-180. The Morgan-Roblin study was significantly different from the Bachman study in several aspects. Firstly, testing was done on small scale drillpipe (dog-bone) samples that were cut from the drillpipe body. Secondly, the testing was done by cycling the specimens in an axial load frame. Thirdly, the fatigue tests were complemented by data on tensile strength of the specimens tested. Finally, the results were analyzed to determine the effects of several test variables in some detail.

The third and most recent data set was generated during a study by Grondin and Kulak in 1991 at the University of Alberta[12]. This study was based on 29 tests in air and 27 tests in a 3.5 percent sodium chloride solution. All tests were conducted on 4.5 in., 16.6 ppf, E-75 drillpipe. The experimental set-up was devised to test drill pipe in rotating bending, with an axial preload so that the effect of mean stress on fatigue life could be

11) Juvinall, R. C., 1967.

12) Bachman, W. S., 1951.

13) Morgan, R. P., and Roblin, M.J., 1969.

14) Grondin, G. Y. and Kulak, G. L., 1991.

15) Grade D drill pipe is made with steel of minimum yield strength of 55 kpsi and minimum ultimate strength of 95 kpsi (Rollins, 1966). This grade is not listed in the 1995 edition of API RP 7G.

studied. A specially designed test machine was constructed for the study so that the test specimens were subjected to a constant bending moment due to four point bending.

Figure 23 shows the three test data sets. Only results of tests on Grade E drill pipe are shown from the Morgan-Roblin set, since the Bachman and the Grondin-Kulak data sets are based on tests on Grade E drill pipe. Though a total of 29 tests were performed in air by Grondin and Kulak only three of them were conducted at zero mean stress. Since the Bachman and Morgan -Roblin tests were conducted at zero mean stress, only the three relevant test points from the Grondin -Kulak set are shown. The figure also shows curve fits proposed to Bachman's data by Hansford and Lubinski [4)], and those proposed by Morgan and Roblin[11] and Grondin and Kulak[16)] to their respective data sets. An endurance limit of 20 kpsi to the Bachman data can be deduced from the Hansford-Lubinski curve fit. The endurance limit of 25 kpsi for the Grondin - Kulak data is based on a generalized curve fit to twenty nine tests as given in Grodin and Kulak (1994) of which twenty six tests were conducted at non-zero mean stress. However, examination of the Grondin-Kulak test data shows that the three tests at zero mean stress indicate an endurance limit between 20 kpsi and 21 kpsi. The Morgan-Roblin data and curve fit, indicate an endurance limit of 30 kpsi for Grade E drill pipe, and they consistently predict a larger number of cycles to failure for a given stress amplitude, i.e. the Morgan-Roblin data is less conservative in comparison with the Bachman data. The endurance limits for the higher grades (as obtained from the Morgan-Roblin[12)] tests) are shown in Table B.39 [12].

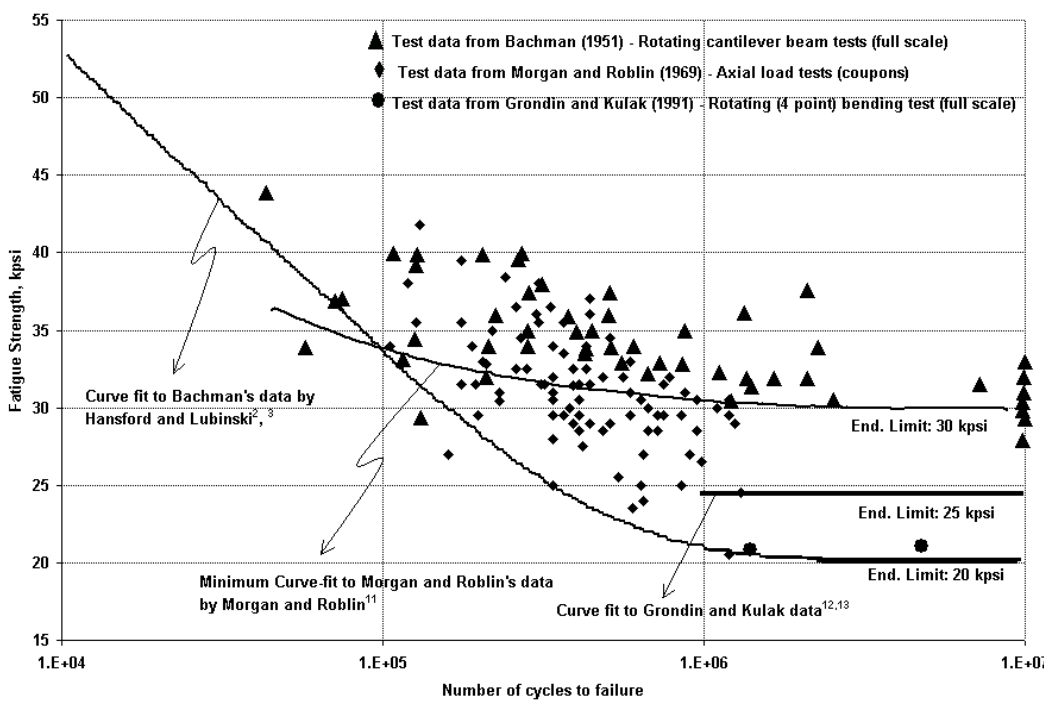

Figure 23 — Comparison of the fatigue S-N data sets

The discrepancy in the S-N data between the Bachman and the Morgan-Roblin data sets for drill pipe can be attributed to the effects of specimen size and the effect of residual stresses in the test specimens. Changing the size of a specimen changes the fatigue limit due to two effects – an increase in the volume/surface area of the specimen, and an increase in the stress gradient. In case of elastic bending, the stress gradient is proportional to the specimen diameter for a given curvature. This increases the volume of material that is highly stressed. Experimental data shows that the endurance limit reduces with increasing size. In general, a

16) Grondin, G. Y. and Kulak, G. L., 1994.

uniform microstructure cannot be obtained throughout very large sections of heat treated specimens. The higher probability of defects in the microstructure are potential sites of crack initiation, and they typically reduce fatigue strength. The Bachman data set which points to a lower endurance limit was generated on full scale specimens as opposed to dog-bone shaped samples cut from the drill pipe wall. Furthermore, the dog-bone shaped samples used in the Morgan-Roblin study were straightened before being axially cycled. The straightening is likely to have induced tensile and compressive residual stresses in the specimen. Compressive residual stresses can potentially improve the fatigue life. However, the effects of size, residual stresses, and other miscellaneous effects cannot be unambiguously isolated and quantified from the Morgan-Roblin data.

10.2.2.2.2 The S-N curve for drill pipe

Thus, the overwhelming problem in fatigue prediction of drill pipe is a lack of data. Despite the fact that the Morgan-Roblin study contains S-N data for drill pipe grades other than E, the data cannot be directly used until further test data or a more comprehensive analysis of their tests becomes available. The Bachman test data on the other hand is valid only for 4 ½ in Grade E drill pipe, and the Grondin-Kulak data is not extensive enough. While the Bachman and Grondin-Kulak data sets can be used for Grade E drill pipe, there remains the problem of obtaining data for other grades.

Therefore, the approach used to design against fatigue by machine designers who routinely consider its effects, is proposed. Since a majority of machine components are made of varying grades of structural steel (alloy or carbon steels), the approach used by the traditional machine designer, is in principle applicable to drill pipe. This has been confirmed partly by the work of Rollins in 1966 at the Battelle Memorial Institute[17)]. The details of this approach are discussed in Juvinall (1967) and Shigley and Mischke (1989). The comparison of this approach with existing data is discussed in Payne and Sathuvalli (2004).

The S-N curve of a material can be regarded as a material property, when it is obtained under specified and controlled conditions. These results are typically obtained in a machine known as the R. R. Moore rotating bending machine on small samples with tightly controlled geometry (see Figure 24). From a knowledge of the S-N curve of the material, the S-N curve of the component that may withstand specified cyclic loading is calculated by using specific rules. The rules involve choosing or calculating suitable factors that modify the endurance limit of the material, and subsequently determining the S-N curve of the component. The S-N curve of the component is a performance property that is specific to it. Having determined the S-N curve of the component, the designer applies well known theories to determine fatigue lives and damage accumulation.

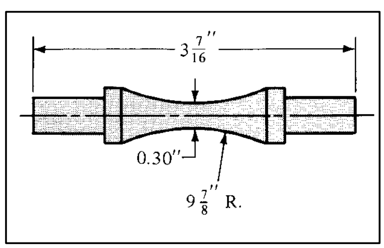

Figure 24 — Test Specimen Geometry for the R. R. Moore Machine[16]

The equation for the typical S-N curve (shown in Figure 22) is represented by the following equation :

$$S_F\left(S_M, N\right) = aN^B;\ 10^3 \le N \le 10^6 \qquad (121)$$

where

S_F is the Stress amplitude (also known as fatigue strength), expressed in psi;

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17) Rollins, H. M., 1966.

S_M is the Mean stress, expressed in psi;

N is the Number of cycles to failure at stress amplitude S_F and mean stress S_M;

a is the A constant that depends on the drill pipe size, grade and mean stress, expressed in psi;

b is the A constant that depends on the drill pipe size and grade.

The endurance limit S_E is the fatigue strength (or stress amplitude) for N = 10^6 cycles, so that

$$S_E(S_M) = S_F\left(S_M, 10^6\right) \tag{122}$$

The endurance limit is a function of the mean stress in the drill pipe. The endurance limit, is however, most often specified at zero mean stress, and it is denoted by the symbol S_{EO}, so that

$$S_{EO} = S_E(S_M = 0) \tag{123}$$

There is copious data to suggest that for ferrous materials, the fatigue strength at 1 000 cycles (N =10^3) is 90% of the tensile (ultimate) strength. Therefore,

$$S_F\left(S_M, 10^3\right) = 0.9 S_{ULT} \tag{124}$$

where

S_{ULT} is the Tensile (ultimate) strength of steel, expressed in pounds per square inch (psi).

If the endurance limit and the ultimate strength of steel are known, the constants *a* and *b* for the S-N curve described by Eq. (118) are determined as follows:

$$a = \frac{\left[0.9 S_{ULT}\right]^2}{S_E(S_M)} \tag{125}$$

$$b = \frac{1}{3}\log_{10}\left[\frac{S_E(S_M)}{0.9 S_{ULT}}\right] \tag{126}$$

For steels, the endurance limit at zero mean stress (S_M = 0 psi) is proportional to the ultimate strength. The constants of proportionality which are known as *k*-factors, are determined by following specific rules described in most textbooks on machine design [9), 18)]. The endurance limit S_{EO} at zero mean stress is given by

$$S_{EO} = k_A k_B k_C k_D k_E S_E'. \tag{127}$$

where

S_E' is the endurance limit at zero mean stress of a Moore machine sample;

k_A is the surface finish factor;

k_B is the size factor;

k_C is the load factor;

18) Shigley, J. E. and Mischke, C. R., 1989.

k_D is the temperature factor; and

k_E is the miscellaneous effects factor.

The Moore machine sample is shown in Figure 24. The specimen in this figure is carefully machined and polished, with a final polishing in the axial direction to avoid circumferential scratches. The specimen is subjected to high speed rotating bending in the R. R. Moore machine. This machine imposes pure bending on the specimen (no transverse shear) by means of weights. S_e' is the endurance limit obtained from an S-N curve generated with such samples, and it is usually regarded as a material property of the steel. The Moore machine endurance limit has been shown to be related to ultimate strength as follows[19]:

$$S_E' = \left| \begin{array}{l} 0.504 S_{ULT}, \; S_{ULT} \leq 200 \text{ kpsi} \\ 100 \text{ kpsi}, S_{ULT} > 200 \text{ kpsi} \end{array} \right. \tag{128}$$

The magnitude of the k-factors in Equation 127 is given in most textbooks on machine design [9), 16)]. The surface factor is a function of the surface finish and is given by

$$k_A = p S_{ULT}{}^{B} \tag{129}$$

where p and q are constants listed in Table B.40. Figure 25 shows the surface factors for different surface conditions. For drill pipe, k_A is usually calculated for the “hot rolled” condition.

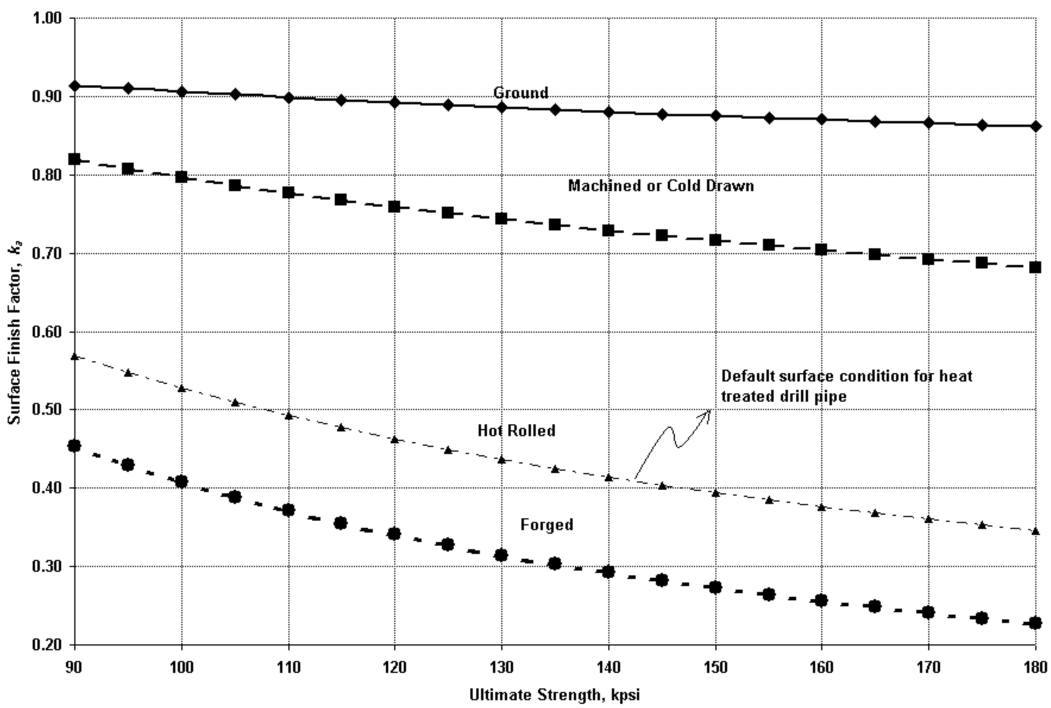

Figure 25 — Surface Finish Factors

19) Mischke, C. R., 1987.

The historically recommended size factor k_B varies between 0.6 and 0.75 for sizes larger than 2 in [9]. Here size refers to the drill pipe diameter. A more recent estimate of the size effect for sizes between 2 in and 9 in is given by the following equation[20],[21]

$$k_B = 1 - \frac{D - 0.03}{15}, \; 2 \le d \le 9 \text{ in.} \tag{130}$$

Where

D is the drill pipe OD, expressed in inches (in).

Table B.41 shows the values of k_B for API drill pipe.

The load factor k_C is usually set to 1 for rotating bending which corresponds to the case of drill pipe in a dogleg. For reversed axial loading this factor is usually set to 0.923, and to 0.577 for torsional and shear loading. (Axial and torsional load cycling may be important if fatigue due to downhole vibrations should be assessed.) The temperature factor k_D is equal to the ratio of the tensile strength at the operating temperature to the tensile strength at 70 °F. This ratio is very nearly equal to 1 for temperatures up to 400 °F. Unless temperatures higher than this are anticipated, k_D is set to unity for drill pipe.

The miscellaneous effects factor k_E is used to account for factors such as surface residual stresses, corrosion, electroplating, metal spray coatings, and stress concentrations. The effect of notches and other surface imperfections can be rigorously accounted for by using notch sensitivity factors [9]. The magnitude of each of these factors is found for specific situations by consulting standard machine design handbooks. The designer should account for these factors by consulting the references listed for the specific conditions that the drill pipe may encounter in a specific well.

Table B.42 lists the suggested endurance limits (S_{EO} of Equation 127) and S-N curve constants (*a* and *b* of Equation 121) for various grades and sizes., in a non-corrosive environment. The endurance limits and S-N curves are valid for rotational bending (caused in a dogleg) for hot rolled drill pipe that has no visible surface imperfections (such as gouges, notches, corrosion spots etc.). If such imperfections are present, these should be identified as “stress risers” with a potential to reduce the fatigue life. Suitable de-rating factors for k_e (which is set to unity in Table B.42) should be employed. Methods of assessing k_e in such situations are described in Bachman (1951).

The endurance limits and S-N curves described in Table B.42 are intended to provide a conservative guideline. It would be desirable to supplement this information in critical situations with actual tests on Moore machine and/or full scale samples. Due to vast improvements in the quality of steel and manufacturing processes, the data shown in Table B.42 may be over-conservative. However, such a determination cannot be made in the absence of data. In critical deepwater applications, where the drill string should be optimized (for structural and hydraulic perfromance), the designer would be well advised to make the final decision based on actual testing.

10.2.2.2.3 Effect of mean stress

Fatigue seldom occurs at non-zero mean stress. A non-zero tensile mean stress reduces the fatigue life of drill pipe. Since the S-N curve of the drill pipe is specified at a given mean stress (usually but not necessarily equal to zero), a method of determining the S-N curve at an arbitrary mean stress is necessary. Figure 26 is a geometrical representation of the problem. Assuming that the S-N curve $A_1B_1C_1$ at a given mean stress S_{M1} is known, it is required to determine the number of cycles to failure, N_X, at an arbitrary mean stress S_{MX} and stress amplitude S_{FX}. It is assumed that the line A_1B_1 is represented by

20) Castleberry, G., 1978.

21) Faupel, J. H. and Fisher, F. F., 1981.

$$S_F = a_1 N^B \tag{131}$$

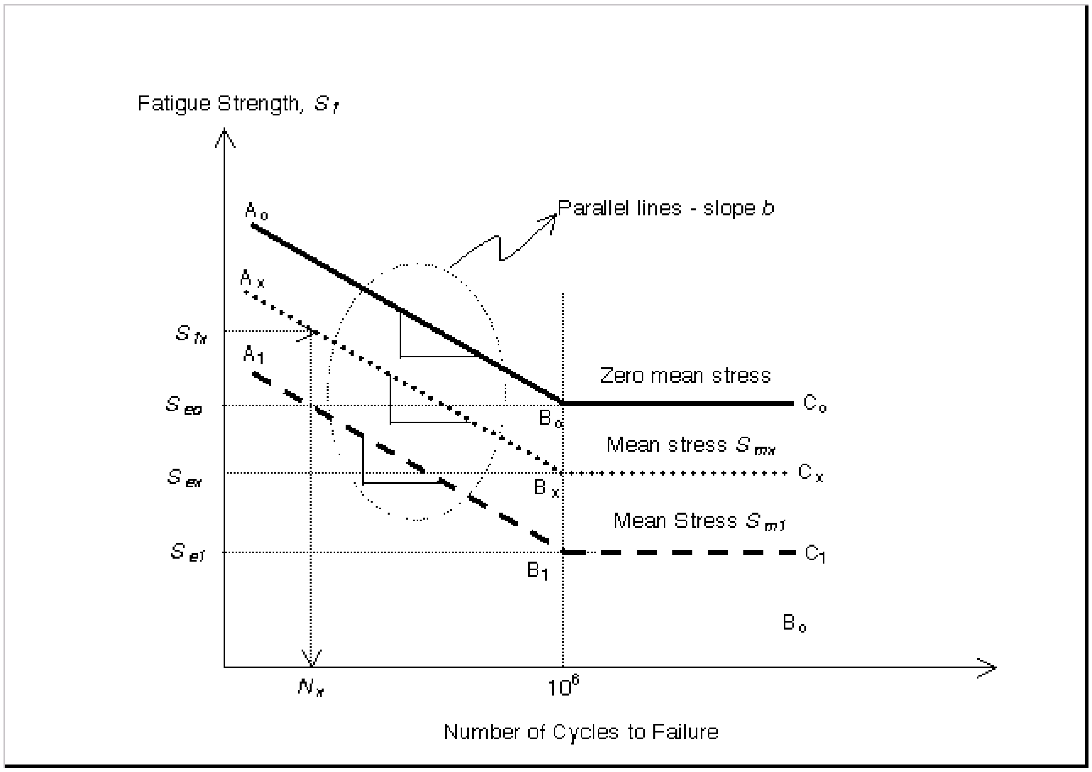

Figure 26 — Determination of Fatigue Life at an arbitrary mean stress

The number of cycles to failure N_X is determined as follows. The steps below are with reference to Figure 26 and Figure 27.

1) On an *x*-*y* plot where the mean stress is plotted on the *x*-axis and the fatigue stress is plotted on the *y*-axis, locate the point Q (S_{ULT}, 0) as shown in Figure 27.

2) On the same *x*-*y* plot, locate the point P (S_{M1}, S_{E1}). S_{E1} is the endurance limit at a mean stress S_{M1} and is known (see Figure 26).

3) Draw a straight line through points Q and P, and extend the line to intercept the *y*-axis at point S (S_{EO}, 0). The location of point S can also be determined algebraically by using the following equation:

$$S_{EO} = \frac{S_{E1}}{\left(1 - \frac{S_{M1}}{S_{ULT}}\right)} \tag{132}$$

4) Determine the endurance limit at the mean stress S_{MX}, by drawing a vertical line from the *x*-axis to the Goodman line and projecting it onto the y-axis, as shown in Figure 27. This is denoted by the point R (S_{MX}, S_{EX}) in Figure 27. The location of point R can be alternatively determined by the following equation:

$$S_{EX} = S_{E1} \frac{S_{ULT} - S_{MX}}{S_{ULT} - S_{M1}} \tag{133}$$

5) In Figure 26 draw the line B_XC_X parallel to the *x*-axis so that its distance from the *x*-axis represents the endurance limit S_{EX} at the mean stress S_{MX}.

6) Draw the line A_XB_X parallel to the line A_1B_1 as shown in Figure 26. $A_XB_XC_X$ is the S-N curve at the mean stress S_{MX}.

7) The number of cycles to failure N_X, at a stress amplitude S_{FX} and mean stress S_{MX} is obtained by drawing a horizontal line to the S-N curve $A_XB_XC_X$ and projecting it onto the *x*-axis. Alternatively, N_X is obtained algebraically as follows:

$$N_X = \left[\frac{S_{FX}}{S_{EX}}\frac{S_{E1}}{a_1}\right]^{\frac{1}{b}} \tag{134}$$

The same procedure is used if the S-N curve is specified at a zero mean stress, and it is required to determine the number of cycles to failure at an arbitrary mean stress. If the algebraic method (i.e. Equations 132 through 134) is preferred, it is merely necessary to set S_{M1} = 0 in these equations.

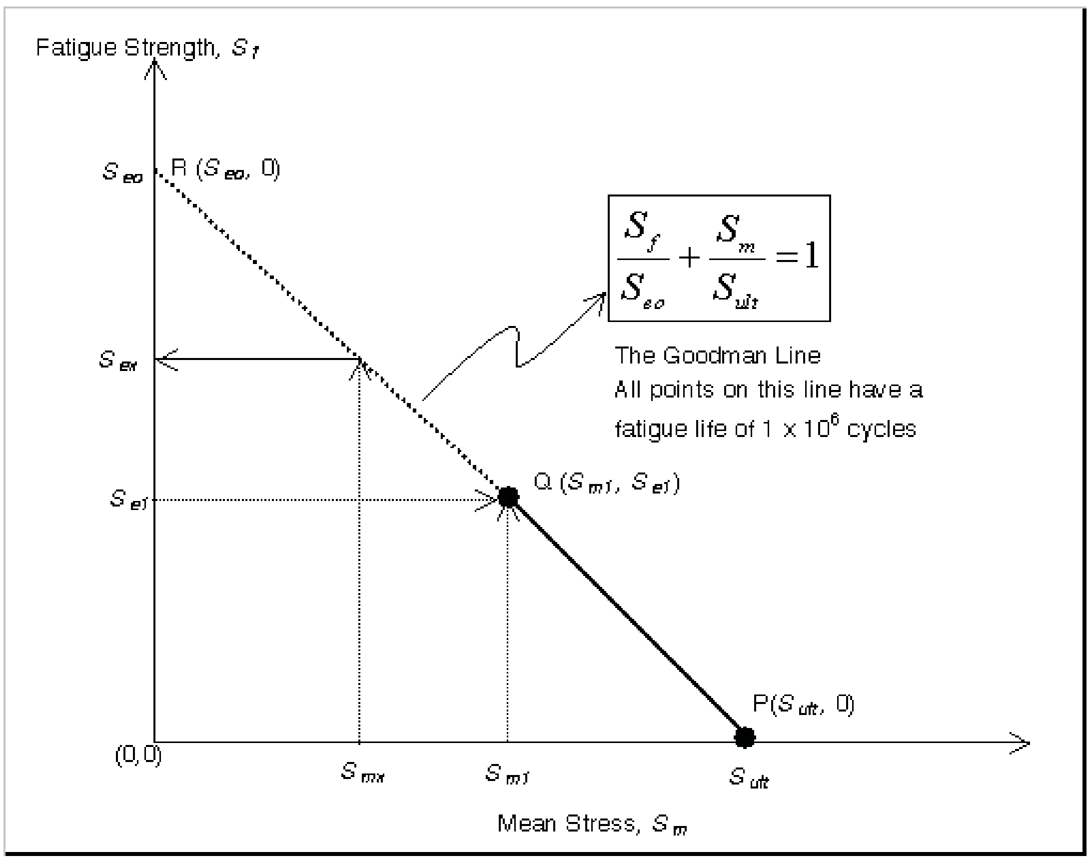

Figure 27 — Using the Goodman rule to determine fatigue life at an arbitrary mean stress

10.2.2.3 Fatigue in a corrosive environment

As noted in the previous subclauses, the endurance limit does not exist when the drill pipe operates in a corrosive environment. Though several works address the effects of corrosive environments on fatigue [12), 22)],

22) Azar, J. J. and Lummus, J. L., 1976.

[23), 24)], the data is insufficient generalize and quantify the effects of corrosive environment on the S-N curves of drill pipe across size and grade. Therefore, until further studies and test data become available, an approach similar to that used by Hansford-Lubinski [3),4)] is adopted. This approach has been demonstrated to consistently err on the conservative side[14)]. Figure 28 shows the S-N curves in corrosive and non-corrosive environments, as suggested by Hansford and Lubisnki [3)]. The S-N curve in a corrosive environment is de-rated by 40%. A similar approach of de-rating the ordinate of the S-N curve in a non-corrosive environment is adopted for other grades and sizes. Table B.42 shows the S-N curve fit constants for operation in a corrosive environment. The corrosive environment curves are obtained by de-rating the non-corrosive environment curves by 40%. The equations of the S-N curves are described as before by

$$S_F(S_M, N) = aN^B, \quad N \geq 10^3 \tag{135}$$

where the symbols are as defined in Equation 120 of section 9.2.2.2. However, Equation 135 is valid even when the number of cycles exceeds a million cycles. This implies that fatigue failure can occur at a finite (albeit, a large) number of cycles even at very low cyclic stress levels in the presence of a corrosive environment.

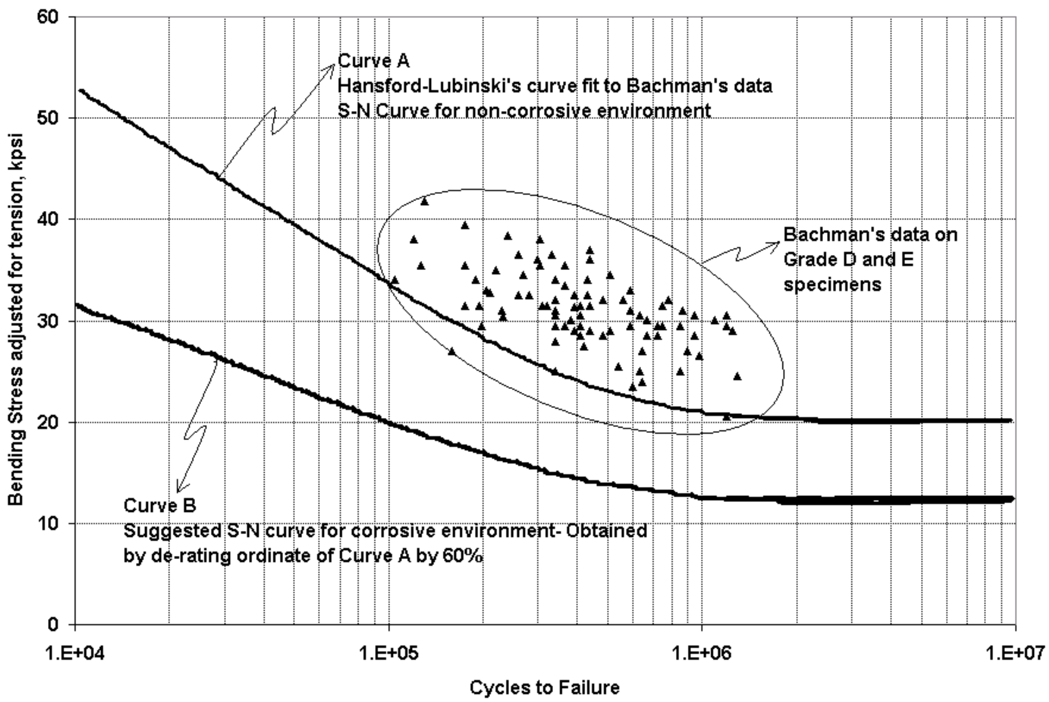

Figure 28 — S-N curves for Grade D and E drill pipe in a corrosive environment as suggested by Hansford and Lubinski[3]

The S-N curve fit constants shown in Table B.42 are applicable at zero mean stress. The effect of mean stress is determined by following the steps described in clause 9.2.2.3. However, it should be remembered that the S-N curve does not become parallel to the *x*-axis in a corrosive environment. Therefore, points B_o, B_1, and B_X in Figure 26 should not be interpreted as endurance limits. Rather they are regarded as points on an S-N curve that correspond to a fatigue life of a million cycles. Therefore, a finite value of fatigue life (number of cycles to failure) can be determined when the stress amplitude S_{FX} is less than the endurance limit, S_{EX}. This

23) Joosten, M. W., Shute, J., and Ferguson, R. A., 1985.

24) Helbig, R. and Vogt, G. H., 1987.

follows as geometrical consequence of extending the lines A_OB_O, A_XB_X and A_1B_1 towards the *x*-axis in Figure 26.

10.2.2.4 Maximum Permissible Dogleg Severity to Avoid Fatigue

Figure C.40 and Figure C.41 (Figure D.40 and D.41) show the maximum permissible dogleg through which a drill pipe can be rotated to avoid fatigue accumulation. The allowable dogleg severity depends on the drill pipe size and grade and the axial tension acting at the bottom of the dogleg. The axial tension creates a mean stress in addition to the cyclic bending stress caused by rotation in the dogleg. The curves in Figure C.40 and Figure C.41 (Figure D.40 and Figure D.41) are generated by determining the mean axial stress caused by the axial tension in the drill pipe at the bottom of the dogleg. The endurance limit at this given mean stress is determined as described in subclause 9.2.2.3. The dogleg severity that causes a peak bending stress equal to the endurance limit is then calculated. The last step accounts for the magnification of bending stress in the drillpipe due to the effect of tooljoints [4), 25)]. However, for the tension and dogleg severity combinations shown Figure C.40 and Figure C.41 (Figure D.40 and Figure D.41), the drill pipe is not in contact with the dogleg. Therefore, these curves are independent of the tool joint diameter.

These curves can be also derived by using the following equations:

$$c = \frac{432{,}000}{\pi}\frac{S_B}{ED}\frac{\tanh KL}{KL} \tag{136}$$

$$K = \sqrt{\frac{T_{BW}}{EI}} \tag{137}$$

where

c is the Maximum permissible dogleg severity (hole curvature), expressed in degree per 100 ft (°/100 ft);

E is the Young's modulus, expressed in pounds per square inch (psi);

= 30×10^6 psi for steel,

D is the drill pipe OD, expressed in inches (in);

S_B is the Maximum permissible bending stress, expressed in pounds per square inch (psi);

T_{BW} is the buoyant weight (including too joints) suspended below the dogleg, expressed in pounds force (lbf);

L_{TJ2} is the half the distance between tool joints, expressed in inches (in);

= 180 in. for range 2 drill pipe;

I is the drill pipe moment of inertia with respect to its diameter calculated by Equation 138, expressed in inches to the fourth power (in^4);

$$I = \frac{\pi}{64}\left(D^4 - d^4\right) \tag{138}$$

d is the drill pipe ID, expressed in inches (in).

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25) Paslay, P. R. and Cernocky, E. P., 1991.

The maximum allowable bending stress, S_B is calculated by Equation 139.

$$S_B = S_{EO}\left(1 - \frac{S_M}{S_{ULT}}\right) \tag{139}$$

where

S_{EO} is the drill pipe endurance limit (typical suggested values are listed in Table B.42), expressed in pounds per square inch (psi);

S_M is the Axial stress in drill pipe due to tension at the bottom of the dogleg, expressed in pounds per square inch (psi);

S_{ULT} is the Tensile strength of drill pipe, expressed in pounds per square inch (psi).

$$S_M = \frac{T}{A} \tag{140}$$

A_{DP} is the drill pipe cross sectional area, expressed in inches squared (in^2).

$$A_{DP} = \frac{\pi}{4}\left(D^2 - d^2\right) \tag{141}$$

10.3 Cumulative fatigue damage

10.3.1 Concept of cumulative fatigue damage (new)

The previous subclauses introduced the concept of an endurance limit and discussed its relation to fatigue failure. Whenever common oilfield materials are operated at stress levels that exceed their endurance limit, fatigue damage results. This damage is irreversible. It accumulates until either the operating loads are reduced to levels below the endurance limit, or a failure occurs. The number of cycles that occur between the time the endurance limit is exceeded and failure occurs is referred to as fatigue life. Another method for avoiding fatigue failures is to monitor the fatigue life that has been expended while operating above the endurance limit, and remove the tool from service before it's fatigue life has been expended.

10.3.2 Method for estimation of cumulative fatigue damage by monitoring buoyant weight below the dogleg

Hansford and Lubinski developed a method for estimating the cumulative fatigue damage to joints of pipe that have been rotated through severe doglegs (see Figures 31 and 32). These charts provide an estimation of the percentage of the fatigue life of a joint of pipe that is expended while rotating through a dogleg in a hole. A correction formula to use for other penetration rates and rotary speeds is as follows:

% Life Expended = % Life Expended from Figure 31 or 32 x

$$\frac{ActualRPM}{100RPM} \times \frac{10\ ft/hr}{Actual\ \ ft/hr} \tag{142}$$

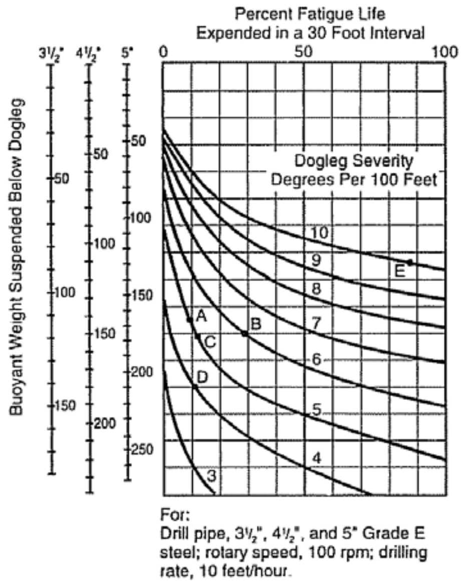


Figure 29 — Fatigue Damage in Gradual Doglegs (Non-corrosive Environment)

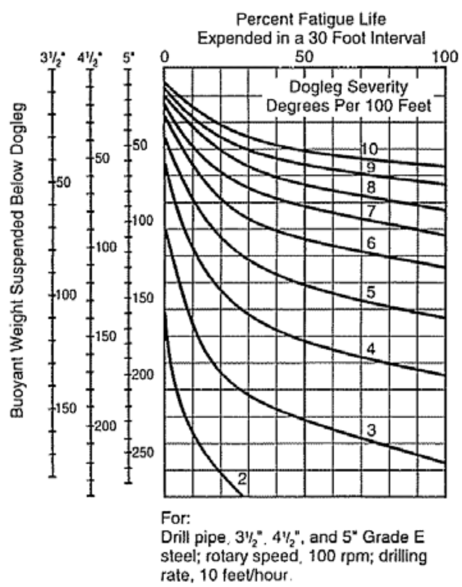


Figure 30 — Fatigue Damage in Gradual Doglegs (In Extremely Corrosive Environment)

10.3.3 Estimating Cumulative Fatigue Damage by Monitoring Stress vs. Revolutions

The key towards successful use of this technique is establishing the appropriate stress versus revolutions to failure curves for the various grades of API drill pipe.

Figure 33 provides an estimate of the median expected failure limits found by Morgan and Roblin for API drill pipe that has been manufactured to average API properties. One would expect that half of the drill pipe exposed to the limits shown in Figure 33 would fail.

Figure 33 is presented for guidance where experience and judgment might indicate less conservative values might be useful.

The median failure limits are based on an exponential relationship that connects the average tensile strength of the tested specimens representing one revolution to failure with the median fatigue endurance limits representing one million cycles to failure. The average tensile strength values and the median fatigue endurance limits from Table A.21 (B.21) were used to develop the stress versus revolutions to failure curves shown in Figure 33.

The equation is of the form:

$$S = \frac{TS}{N^X} \tag{143}$$

Where

- S is the bending stress limit, expressed in psi;
- TS is the tensile strength of pipe, expressed in psi;
- N is the revolutions to failure;
- X is the a fractional exponent of about 0,1.

Figure 34 is their estimate of the minimum failure limits of API drill pipe that has been manufactured to average API properties. The minimum failure limits in Figure 34 should avoid fatigue failures on typical API drill pipe. Thus, careful application of the limits from Figure 34 would be expected to avoid any failures in the field.

The curves of minimum failure limits are also based on the Morgan and Roblin data. Their report includes a table that defines the typical yield and ultimate tensile strengths for normalized Grade E75, normalized and tempered Grades X95 and G105, and quenched tempered Grade S135 API drill pipe. Using Casner's defined 0,22 ratio, the endurance limits for these tensile strengths were computed.

Table A.43 (Table B.43) summarizes the selected values for yield strength, ultimate strength, and fatigue endurance limit for the various grades of drill pipe. The SN curves were computed by using the exponential relationship described above with the tensile strength equal to the fatigue endurance limit for one revolution and the fatigue endurance limit values to represent the stress limit for one million revolutions. These values were used to prepare Figure 34.

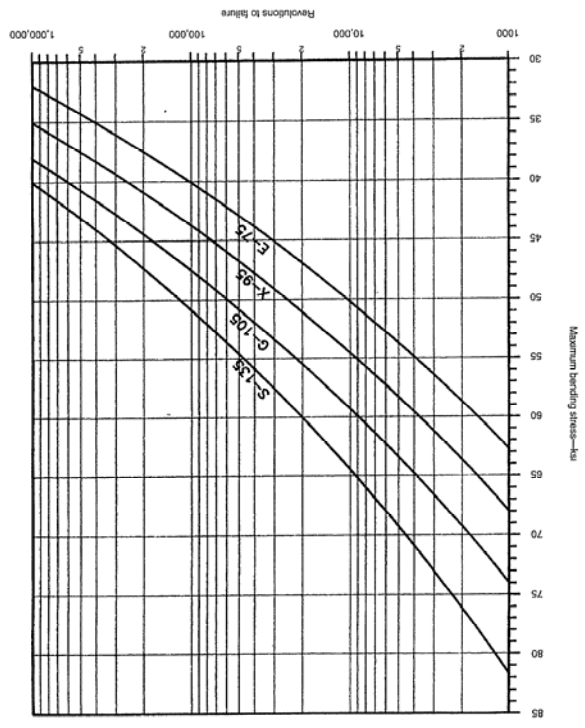


Figure 31 — Median Failure Limits for API Drill pipe in Non-corrosive Service

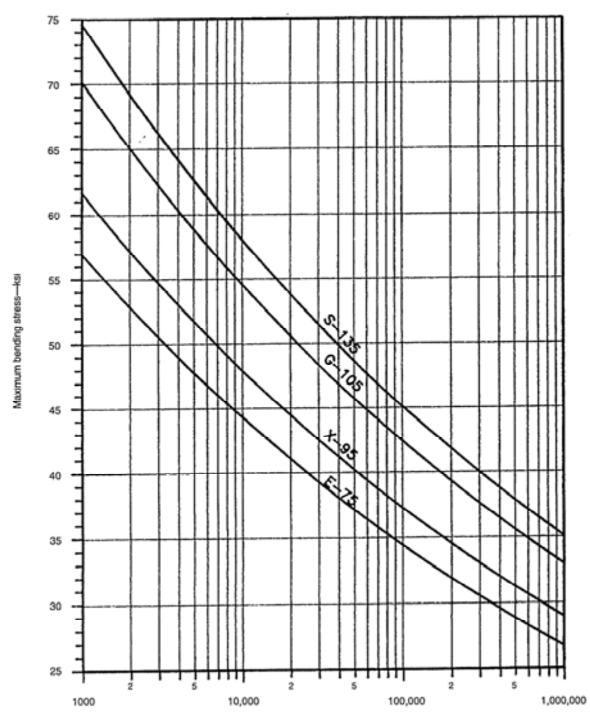


Figure 32 — Minimum Failure Limits for API Drill pipe in Non-corrosive Service

The cumulative fatigue damage is determined by counting the revolutions of pipe in highly curved portions of the borehole where the stresses exceed the fatigue endurance limit. If, for example, the revolutions in a particular section of hole represent 20 % of the predicted revolutions to failure for that dogleg severity it is judged that 20 % of the fatigue life has been consumed. Figures 33 and 34 can be used to judge inspection levels and ultimate retirement levels. The ultimate life can be judged from the plot of the median fatigue limits of Figure 33. The appropriate minimum inspection levels can be judged from the minimum fatigue limits of Figure 34. After exposure to this level of fatigue, inspection and removal of damaged joints can extend the remaining string life to or beyond the expected median life levels

An example of the cumulative fatigue damage calculations is given by the following. Consider drilling a 500 ft horizontal hole below a 100 ft radius build curve with 3,5 in S135 drill pipe. Drill pipe will be rotated at 30 RPM with 10 000 WOB, and is expected to drill at 15 ft per hour. The equivalent build rate is given by:

$$B = \frac{5730}{R} \tag{144}$$

Where

B is the build rate, expressed in degrees per hundred feet (°/100 ft);

R is the build radius, expressed in feet (ft).

For this case:

$$B = \frac{5730}{100} = 57.3 \deg/100\, ft \tag{145}$$

SOLUTION

Figure 78a shows that at 10 000 lb axial compressive load and a 57° per 100 ft build rate the maximum bending stress is 50 000 psi. Figure 33 predicts that half of the S135 pipe will fail under a 50 000 psi bending stress after 110 000 revolutions. The minimum failure limits of Figure 34 predict that S135 pipe can be rotated 39 000 revolutions without failure.

The number of revolutions of exposure is given by:

$$N = \frac{60 \times L_{HC} \times RPM}{ROP} \tag{146}$$

Where

N is the revolutions of exposure;

L_{HC} is the length of high curvature hole, expressed in ft;

RPM is the rotary speed, expressed in rev/min;

ROP is the penetration rate, expressed in ft/hr.

The length of L of our 90° build curve is equal to:

$$L = \frac{\pi}{2} \times R = \frac{\pi}{2} \times 100 = 157\, ft \tag{147}$$

Therefore, the number of revolutions of exposure for the pipe that is rotated through the build curve is equal to:

$$N = \frac{60 \times 157 \times 30}{15} = 18{,}850 \text{ revolutions} \qquad (148)$$

The cumulative damage can be computed by comparing the revolutions of exposure to the 110 000 revolutions required to cause half of the pipe to fail. This suggests that 17 % of the fatigue life of the affected pipe has been consumed in drilling one well. Comparing the revolutions of exposure to the minimum fatigue limit of 39 000 revolutions evaluates the risk of a failure. For this case, the 18 850 revolutions of exposure represent 48 % of the minimum fatigue life expected for the S135 pipe. This suggests that two wells could be drilled with this string before inspecting and downgrading or removing fatigue damaged joints from service. Continued use of the string will require removing significant portions of the affected pipe in order to prevent failures.

10.3.4 Identification of fatigued joints

The difficulty lies in identifying and recording each separate joint fatigue history. Joints which have been calculated to have more than 100 % of their fatigue life expended should be carefully examined and, if not downgraded or abandoned, watched as closely as possible. Such consideration should be finally governed by experience factors until such time as the analytical method for fatigue prediction gains more reliability.

10.4 Operating limits

10.4.1 General

As mentioned in subclause 9.1, fluctuating or cyclic loading causes drill string fatigue failures. Three key methods of generating cyclic loading to the drill stem include:

— Rotating pipe that is bent around a dog leg

— Rotating pipe that is buckled

— String and bit vibration

This subclause presents operating limits and mitigation methodologies for controlling the magnitude of cyclic loading to the drill string.

10.4.2 Tension state in drill pipe

Subclauses 6.1 and 6.2 present tension design of drill strings in vertical and directional wells. Analyzing cyclic loads often requires the calculation of the mean tension state in the drill pipe. For any point of interest, the mean tension state may be calculated to determine the mean stress level that which cyclic loading fluctuates about. For drill pipe, three suspect areas should be analyzed.

10.4.2.1 Immediately below the dogleg

Determining the buoyed weight of pipe beneath the dogleg permits review of the cyclic loads associated with rotating pipe that is bent around a dog leg.

Following is an example calculation:

Data:

1) 4 1/2-inch, 16.60 lb/ft, Grade E, Range 2 drill pipe (actual weight in air including tool joints, 17.8 lb/ft) 7 3/4-inch OD, 2 1/4-inch ID drill collars (actual weight in air 147 lb/ft).

2) 15 lb/gal (112.21 lb/cu. ft) mud.

3) buoyancy factor = 0.771

4) Dogleg depth: 3,000 ft.

5) Anticipated total depth: 11,600 ft.

6) Drill collar length: 600 ft.

7) Drill pipe length at total depth: 11,000 ft.

8) Length of drill collar string, whose buoyant weight is in excess of the weight on bit: 100 ft.

Solution:

Tensile load in the pipe at the dogleg:

[(11,000 – 3,000) 17.8 + 100 x 147] 0.771 = 121,124 lb

10.4.2.2 Top of bottom hole assembly (BHA)

Analyzing the tension state within the drill pipe immediately above the BHA is critical because of the compressive loading that occurs in the lower portion of the drill string. High bit weights may force the neutral point up into the drill pipe and buckle it, resulting in cyclic loading to the drill pipe. The following is an example calculation for determining the tension state of the drill pipe immediately above the BHA:

10.4.2.3 At the kick-off point

Pipe located just above a wellbore kick-off point is often in a straight vertical section. Discussed in subclause 8.2.1, straight, vertical wellbores provide no stability to the drill string and buckling of the string is common. For some wellbore types, such as short radius horizontal wells, buckling may occur just above the kick-off point before any buckling occurs throughout the rest of the string. Calculating the tension state above the kick-off point requires calculation of the weight of pipe in the build or drop section. Johancsik, Friesen and Dawson showed that the weight of the pipe in the build section may be calculated using the following formula[26] :

$$W_{Build} = K_B \times W_{DP}\left(\frac{5\,730(\sin\vartheta - \sin\sigma)}{DLS}\right) \tag{149}$$

The weight of the pipe in a dropping section may be calculated using the following formula:

$$W_{Drop} = K_B \times W_{DP}\left(\frac{5\,730(\sin\sigma - \sin\vartheta)}{DLS}\right) \tag{150}$$

Where:

W_{BUILD} is the Buoyed weight of pipe in the build section, expressed in pounds (lb);

W_{DROP} is the Buoyed weight of pipe in the dropping section, expressed in pounds (lb);

K_B is the Buoyancy factor;

W_{DP} is the Air weight of the pipe, expressed in pounds per foot (lb/ft);

ϑ is the Inclination at lower point of curved section, expressed in degrees (°);

σ is the Inclination at upper point of curved section, expressed in degrees (°).

26) Johancsik, Frieson and Dawson, 1984.

10.4.3 Cyclic loading from dog legs

Lubinski[27)] and Nicholson[28)] have published methods of calculating forces on tool joints and conditions necessary for fatigue damage to occur when rotating pipe is bent in a dogleg. Refer to Section 10.2.2.4 for determining the maximum permissible dogleg severity to avoid fatigue damage. Note within Figures C.40 and C.41 (Figures D.40 and D.41) that it is necessary to remain to the left of fatigue curves to reduce fatigue damage.

10.4.4 Cyclic loading from buckling

10.4.4.1 General

Subclause 8.2 presents the recommendation not to rotate buckled drill pipe. Sometimes bit weights that induce rotary mode drill pipe buckling may be used to accomplish project objectives. When this is required, drill pipe bending stresses should be analyzed.

10.4.4.2 Bending stresses on buckled drill pipe in straight, inclined boreholes

The bending stresses on buckled drill pipe may account for both the mechanics of buckling and the additional bending caused by the axial load and the tool joints. The curvature produced by buckling is given by:

$$B_{BUC} = \frac{F \times h_C \times 57.3 \times 12 \times 100}{2 \times E \times I} \tag{151}$$

$$B_{BUC} = \frac{17190 \times F(D_H - D_{TJ})}{E \times I} \tag{152}$$

Where

B_{BUC} is the curvature of buckled pipe, expressed in °/100 ft;

F is the axial load, expressed in pounds (lb);

D_H is the hole diameter, expressed in inches (in);

D_{TJ} is the tool joint OD, expressed in inches (in);

h_C is the radial clearance $= \dfrac{D_H - D_{TJ}}{2}$, expressed in inches (in);

I is the area moment of inertia of pipe, expressed in (in^4);

E is Young's modulus, expressed in psi = 30×10^6 psi for steel.

This curvature can be used in place of the hole curvature in the equations covered in subclause 9.2.2.4.

27) Lubinski, 1961.

28) Nicholson, 1974.

10.4.4.3 Bending stresses on compressively loaded drill pipe in curved boreholes

Rotating compressively loaded drill pipe in curved portions of the borehole generates cyclic bending stresses that, if large enough, may cause fatigue damage. Compressive loading may also cause a portion of the pipe body to contact the wall of the hole. In abrasive formations, pipe body contact can erode the body of the pipe and further magnify the bending stresses. The maximum bending stress caused by compressively loading drill pipe in curved boreholes is affected by the size of the tool joints, the size of the pipe body and the spacing of the tool joints, as well as the axial compressive load on the pipe and the curvature of the hole.

The application of compressively loads on tool jointed drill pipe in curved boreholes progresses through three stages. Under light loads, the maximum bending stress occurs in the center of the pipe span but only the tool joints are in contact with the wall of the borehole. As loading is increased, the center of the pipe comes into contact with the wall of the hole. Under this loading condition, the maximum bending stress occurs at two positions that are located on either side of the point of pipe body contact. As the load is further increased, the length of the pipe body contact increases from point contact to wrap contact along a length of pipe located in the center of the joint.

Figures 67 through 74 provide solutions to the bending stress, pipe body contact and lateral contact loads for the most common sizes of 6,63-in to 2,38-in drill pipe. There are four plots for each size of drill pipe. The plot in figures 67 – 74 (figures a) show the maximum bending stress as a function of axial compressive load for a range of hole curvatures. The type of loading is shown by the style of the plotted lines. For no pipe body contact, the bending stress curve is shown as a solid line. For point contact, the bending stress is shown as a dashed line and for wrap contact; the plotted stress is shown as a dotted line. The plots also include the fatigue endurance bending stress limits from subclause 11.5. The plots assume a 10-lb/gal mud, 90° hole angle and a defined hole size. The mud density, hole angle and hole size do not affect the bending stresses unless the pipe buckles. On most of the plots the critical buckling force is only exceeded for cases with zero or negative build rate. The bending stresses for buckled pipe are independent of the hole curvature and generally follow curves in which the bending stress increases more rapidly with increased axial load than for the constant hole curvature cases.

The plots in Figures 67 – 74 (figures b) show the pipe body contact length and the lateral contact forces between the pipe body and the tool joints with the wall of the hole.

Example

An 8 ½ in horizontal well will be drilled with 5-in 19,50 lb/ft drill pipe with 6,38 in tool joints in an 8 ½ in hole with 10 lb/gal mud. The maximum hole curvature will be 16°/ 100 ft. The horizontal interval will be drilled with surface rotation with loads up to 35 000 lb. What grade of drill pipe is required for this example?

Solution

Figure 69a shows the bending stresses and fatigue limits for 5 in, 19,50 lb/ft drill pipe with 6,38 in tool joints in a 90°, 8 ½ in hole with 10 lb/gal mud. For a hole curvature of 16 / 100 ft and an axial compressive load of 35 000 lbs, the maximum bending stress is 24 000 psi. A slightly higher bending stress of 25 000 psi is produced by a 24 000 lb axial load. The maximum bending stress exceeds the fatigue endurance limits for API Grade E75, X95 and D55 pipe, but is less than the fatigue endurance limits for grades G105 and S135 pipe. Figure 69b shows that at 35 000 lb axial load, the tool joints contact forces would be about 2 600 lb with a 16 / 100 ft curvature rate. The pipe body contact forces will be about 600 lbs under a 35 000 lb axial load. For this curvature rate the contact will be at the center of the span.

The maximum bending stresses for point and wrap contact are directly proportional to the hole curvatures and radial clearances between the pipe body and the tool joints. This allows us to use the existing bending stress plots to estimate the bending stresses for tool joint dimensions other than used in preparing the plots. No adjustment is necessary unless the loading conditions produces point or wrap contact. If this is the case, we can compute an adjustment factor from the actual size of the tool joints and use that to compute an equivalent hole curvature to determine the correct bending stress.

EXAMPLE Determine the maximum bending stresses for 4 in, 14 lb/ft drill pipe with 5,38 in tool joints in 20 ° per 100 ft curvature and a 20 000 lb axial compressive load.

Solution

Figure 71a shows the bending stresses for 4 in, 14 lb/ft drill pipe with 5 ¼ in tool joints. The actual tool joint outside diameter is 5,38 in or 0,125 in larger. Figure 75 shows the hole curvature adjustment factors for various sizes of drill pipe as a function of the difference in the tool joint outside diameters. For an actual tool joint, 0,125 in larger than nominal, Figure 75 shows that with a 4 in drill pipe the curvature adjustment factor is 1,1. To determine the maximum bending stress for the0,125 in larger tool joint in a 20 °/100 ft hole curvature at 20 000 lb load, multiply the actual hole curvature by the adjustment factor, 20 °/100 ft times 1,1 = 22 °/100 ft. Using 22 °/100 ft at 20 000 lbs on Figure 71a, the bending stress for 5,38 in tool joints is 24 500 psi.

10.4.5 Cyclic loading from vibration

Drill string vibration is one of the most contributing causes of cyclic loading. Refer to Section 11 for operating guidelines and mitigation methodologies.

10.5 Remedial action

10.5.1 Minimization of fatigue loading

There are three main contributors to fatigue damage: bending moment, cycles and tensile load.

Bending moment is controlled by hole curvature, and by deflection of the drill string within the collar. Every effort should be made to minimize drill string curvature, but this is required for directional wells and difficult to avoid completely in vertical wells. The design of bottom-hole assemblies for local conditions helps greatly to minimize accidental dog-legs, but is beyond the scope of this subclause.

When wells are being drilled using a mixture of rotary drilling and sliding, (as with mud motors or turbines), the local curvature may be much more severe than the average reported, especially when surveys are taken every 27,4 m (90 ft). This effect is reduced by the use of less aggressive steering tools, at the cost of reduced steering control.

Tensile load reduces the fatigue capacity of the drill string, so it is always advisable to avoid combining cyclic loading and high tensile loads. For this reason, it is advisable to kick-off a deviated section of the well as deep as possible, to avoid the later suspended load of the string adding to the fatigue loading. It is also advisable to minimize rotation with the bit off-bottom, since the weight of the bottom-hole assembly is added to the tensile load.

Finally, fatigue damage is proportional to the number of cycles. It can therefore be reduced by minimizing the rotational speed. During drilling this is not usually possible, but it is highly advisable to minimize rotary speed while off bottom. In directional work, it is important to minimize rotary speed while exiting a build section until the drill collars have exited the dog-leg. Conversely, very low rates of penetration may expose particular drill string components to much more than the usual number of cycles so that more attention is required to fatigue resistance.

10.5.2 Mechanical design

10.5.2.1 Transition from drill pipe to drill collars

Frequent failure in the joints of drill pipe just above the drill collars suggests abnormally high bending stresses in these joints. This condition is particularly evident when the hole angle is increasing with depth and the bit is rotated off bottom. Low rates of change of hole angle combined with deviated holes may result in sharp bending of the first joint of drill pipe above the collars. When joints are moved from this location and rotated to other subclauses, the effect is to lose identity of these damaged joints. When these joints later fail through accumulation of additional fatigue damage, every joint in the string becomes suspect. One practice to reduce failures at the transition zone and to improve control over the damaged joints is to use nine or ten joints of heavy wall pipe, or smaller drill collars, just above the collars. These joints are marked for identification, and used in the transition zone. They are inspected more frequently than regular drill pipe to reduce the likelihood of service failures. The use of heavy wall pipe reduces the stress level in the joints and ensures longer life in this severe service condition.

It has been found advisable to keep the ratio of section modulus between adjacent parts of the string to below 5 for moderate service, and below 3,5 in severe conditions. If several such transitions are required, they should be separated by at least three joints of pipe

10.5.2.2 Connection balance and torque

Fatigue failures of connections can be minimized by using those with a Bending Strength Ratio (BSR) in the recommended range of 2 to 3, as described in Clause 5. This range covers the strongest connections available for a particular box OD and pin ID. BSR is only an approximate guide to appropriate connections. In particular a poor choice of connection can not be made good by boring out the pin inside diameter to adjust its BSR!

An alternative approach providing similar results in most circumstances is to select the connection with the highest recommended make-up torque. Appropriate torque is very important. Insufficient torque may lead to connections loosening under torsional shock, giving washout and pin fatigue failure. Excessive torque increases the tensile stress in the pin, leading by the Goodman effect to reduced fatigue strength and pin failure.

10.5.2.3 Stress relief features

It has been clearly demonstrated by laboratory testing and field experience that stress relief features improve the fatigue life of connections, and they are therefore strongly recommended in the Bottom Hole Assembly whenever there is a risk of significant rotating bending. These features are described in API Specification 7: if they cannot be implemented as described, similar features are also useful.

Stress relief features are not recommended for tool joints, since the bending strength of pipe body is normally much lower than that of the connection,

10.5.3 Processing

Compressive treatment of thread roots by cold rolling or shot peening is advantageous, and is recommended for all connections in the bottom-hole assembly, and in other connections where fatigue may be a concern.

10.5.4 Materials issues

Steels vary considerably in their fatigue properties for the same strength levels. It has frequently been found that in alloy steels, quenched-and-tempered steels have better fatigue performance than normalized-and-tempered, and that steels with lower inclusion content perform better.

Fracture toughness is important to the ability of materials to survive fatigue. Higher toughness allows a component to survive with larger cracks, and therefore improves the chances of detecting cracks before they cause failures.

Harden ability is also an important issue especially for BHA components. Material properties are specified 2,54 cm (1 in) below the raw material surface, and will decrease towards the center of the bar. When components such as stabilizers and drilling tools are machined from bars much larger than the connections, there is a risk that the connection properties will be worse than expected. The selection of material with high harden ability, or heat treatment after rough machining will minimize this effect. Process review or full-section testing are the usual means of protecting against this possibility.

Non-magnetic drill collar materials often have laboratory fatigue test data available. This should not be used directly as a design criterion, but higher laboratory test values usually correlate with better field performance.

10.5.5 Maintenance and inspection

Stress concentrators such as slip marks, tong marks, gouges, notches, and scratches will significantly reduce fatigue resistance. Every effort should be made to avoid this kind of damage and to repair it if it occurs.

A regular inspection program can detect the majority of fatigue cracks before they cause service failures. Pre-existing cracks from causes such as heat-checking, and stress-corrosion or cracks in hard-banding may grow under fatigue loading. The inspection program should be designed to find such cracks, as well as fatigue cracks, before they can become dangerous.

10.5.6 Extending corrosion fatigue life

The fatigue life of drill pipe will be decreased considerably when it is used in a corrosive drilling fluid. This can be a significant effect even when there are no visible signs of corrosion, and can be equally dramatic on tool joints and drill collar connections.

For many water-base drilling fluids, the fatigue life of steel drill stems may be increased by maintaining a pH of 9.5 or higher, as well as taking other measures to minimize the corrosivity of the drilling fluid, notably the use of oxygen scavengers and H_2S inhibitors.

Drill pipe is significantly protected by plastic coating of the inside surface, where surface defects and damage are hard to control and inspect.

11 Drill string vibration

11.1.1 Cyclic loading from vibration

11.1.1.1 General

Downhole vibration is inevitable. In many cases, low levels of vibration go undetected and are harmless. However, severe downhole vibration can cause drill string fatigue failure (washout/twist-off), crooked drill strings, premature bit failure, and reduced penetration rates. The main sources of excitation are provided by the interaction of the bit with the formation and the drill string with the wellbore. The drill string response to these excitation sources is very complex.

Vibration can induce three components of motion in the drill string and the bit (Figure 35), namely: axial (motion along drill string axis), torsional (motion causing twist/torque) and lateral (side to side motion). All three dynamic motions may coexist and one motion may cause another.

While theories exist, there is no general agreement on how to predict (calculate) when damaging vibrations will occur. However, by observing the symptoms of severe downhole vibrations, probable mechanisms may be determined and appropriate corrective actions taken.

Severe downhole vibration is often accompanied by symptoms belonging to more than one mechanism. This fact makes the detection of the primary mechanism more difficult. For example, an increase in MWD shock counts, which is indicative of BHA lateral vibration, can be caused by BHA Whirl, Bit Bounce, or other mechanisms. Additional clues, such as bit tooth breakage, for this example, are required for the identification of the primary vibration mechanism.

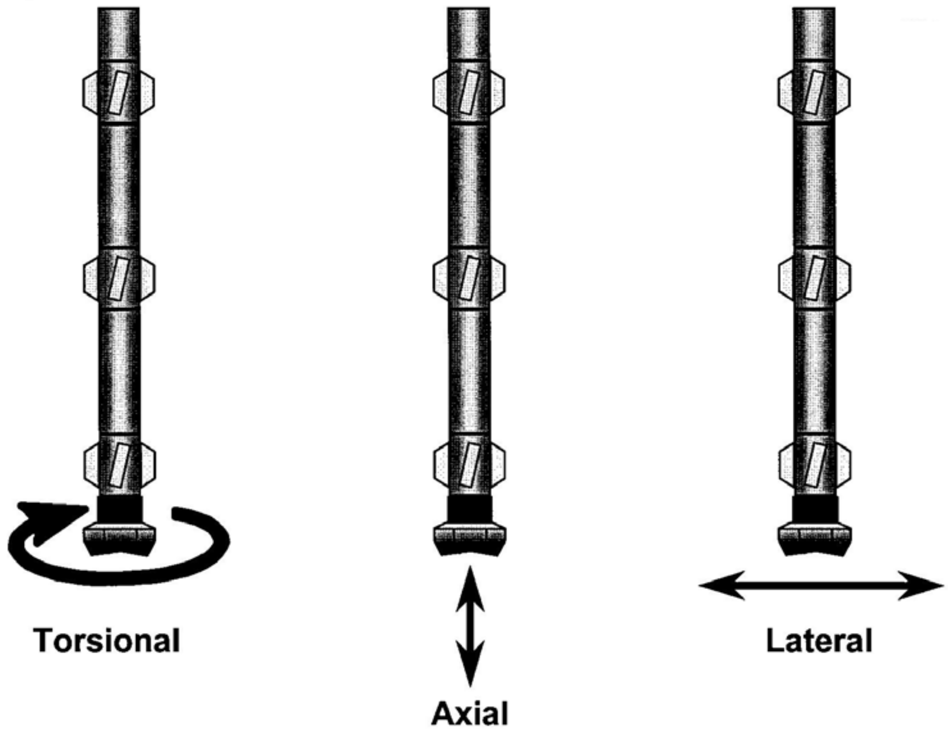


Figure 33 — Three Primary Motions Generated by Drilling Vibrations

The drill string may be simulated as a long series of interconnected masses and springs (see Figure 36). If the mass-spring system is excited on one end, the excitation will propagate through the drill string. The rate at which the excitation propagates through the system will depend on both the spring stiffness and the value of the masses. Stiffer springs will cause faster propagation of the excitation and larger masses will cause slower propagation.

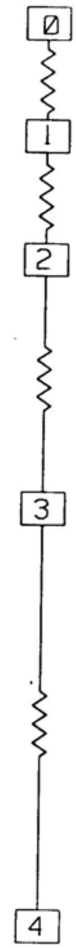


Figure 34 — Simulated drill string as a long series of masses and springs

For drill string tubulars, the stiffness of the spring is the axial stiffness of the tube, a product of the material's modulus of elasticity and its cross sectional area. The value for the drill string's mass relates to the mass per unit length of tube (material density) and it's cross sectional area. Since the material's cross sectional area is a characteristic of both, it is self-canceling. As a result, the velocity of axial excitation propagation is simply a function of the drill string's material density and its modulus of elasticity for each section.

For torsional excitation propagation, the property of stiffness is the drill string's torsional stiffness, or its shearing modulus of elasticity. The drill string's mass is the same as discussed above. However, since the torsional stiffness (shearing modulus of elasticity) differs from the axial stiffness (modulus of elasticity), the velocity of axial excitation propagation differs from that of torsional excitation propagation.

It can be expected that axial and torsional excitations will propagate through the drill string at a velocity predicted by the below equations.

$$\text{Torsional Velocity} = \left[\left(\frac{E}{2\rho}\right)(1+u)\right]^{0.5}$$
$$= 3{,}190 \text{ meters/second} \qquad (153)$$

$$\text{Axial Velocity} = \left(\frac{E}{\rho}\right)^{0.5}$$
$$= 5{,}140 \text{ meters/second} \qquad (154)$$

where

ρ is 7.8 x 10^3 kg/m^3;

E is 2.1 x 10^{11} N/m^2;

u is 0.3.

Propagation velocities are calculated for bars in air and room temperature. Field experience has shown that actual velocities of vibration propagation in drill strings are approximately 5-8% slower. This is probably due to the damping effect of the drilling mud environment.

Recognizing the differing excitation velocity values provides the ability to distinguish, or define "signatures" for different vibration types.

11.1.1.2 Axial Vibration

Axial vibration within drill strings is a common event. Axial vibration can be defined as axial force fluctuations in the drill string. Less severe axial vibration occurs when axial force fluctuations are observed while the bit remains in contact with the formation (Figure 37).

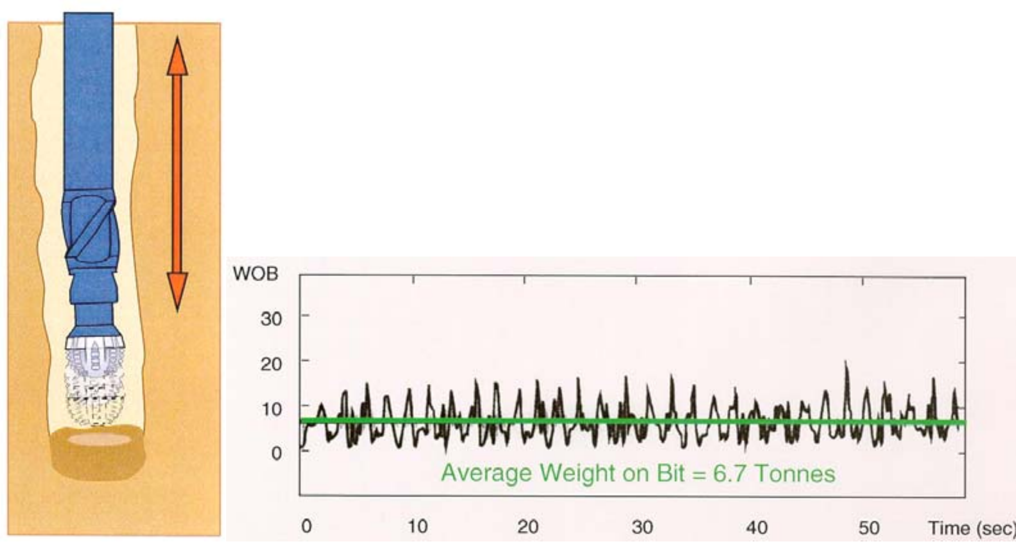


Figure 35 — Less Severe Axial Vibration

As seen in Figure 37, axial force fluctuations are noted in the drill string. An average of 6.7 tonne force is provided for WOB. However, the force fluctuates over time with similar amplitude and frequency. Characterized as a less severe form of axial vibration, rarely does the force at any point in time reach 0, indicating the bit remains in contact with the formation during the interval.

A more severe form of axial vibration is characterized as "bit bounce." Bit bounce can be defined as large axial force fluctuations causing the bit to repeatedly lift off bottom and impact the formation. This is damaging the bit and BHA. Figure 38 below illustrates a bit bounce event.

Unlike Figure 37 which provided [illegible] n, Figure 38 shows that although the same average WOB is pres[illegible] the bit to lift off bottom. This is indicated by the repeated 0 tonne [illegible]

11.1.1.2.1 Detrimental Impact

Axial vibration can be quite detrin[illegible] uses broken or prematurely worn bits, slow ROP, surface equipmer[illegible] ools like motors and MWD's), and the excitation of lateral vibration.

11.1.1.2.2 Typical Environmen[illegible]

Axial vibration occurs in typical e[illegible] les in hard rock formations. Axial vibration is much more common u[illegible] ri-lobe pattern developed by using tricone bits generates repeated up[illegible] ng, causing axial force fluctuations (see Figure 39). Because of the [illegible] ation with tricone bits will occur at 3 X RPM frequencies.

Figure 37 — Tri-lobe pattern from a roller cone bit

11.1.1.2.3 Factors Affecting

Factors, which affect axial vibration severity, include RPM, WOB, hole angle, BHA length, formation hardness, and bit type.

11.1.1.2.4 Identification Methods

There are several methods of detecting axial vibration. Some of the most common methods include:

- Extreme surface vibration (shaking of hoisting equipment, top drive shaking, kelly bouncing)
- Bit damage (broken teeth, damaged bearings, excessive wear of fixed cutters)
- Excessive downhole axial shocks (axial accelerometers)
- Excessive downhole WOB fluctuations
- Excessive surface WOH fluctuations
- Tri-lobing (3 X RPM frequency)
- Reduction in ROP
- BHA washout / twistoff

11.1.1.2.5 Mitigation Methods

Mitigation methods should be implemented to reduce the detrimental impact of axial vibration. Mitigation methods used to reduce drilling vibrations are often formation or area-specific and could be considered an art form of the drilling dynamics specialist. However, some general methods of mitigating axial vibration used in the industry include the following:

- Change bit type to PDC
- Use shock sub or hydraulic thruster (lowers resonant frequency)
- Change drilling parameters (reduce WOB, adjust RPM) and allow time for development of new tri-lobe pattern
- Design BHA so buckling neutral point is well within the BHA and does not fluctuate into the drill pipe
- STOP, lift off bottom, stop rotation, restart rotation to 1/2 targeted RPM, SLOWLY tag bottom, increase WOB to targeted value, and then increase RPM to target value
- Use automated feed-off control systems (automated drillers)

Several theories exist about the impact of using shock subs in BHA’s. Often, shock subs prove beneficial to reducing axial vibrations. As seen in Figure 40 below, shock subs help to dampen axial vibration by reducing the amplitude of 3 X RPM axial frequency.

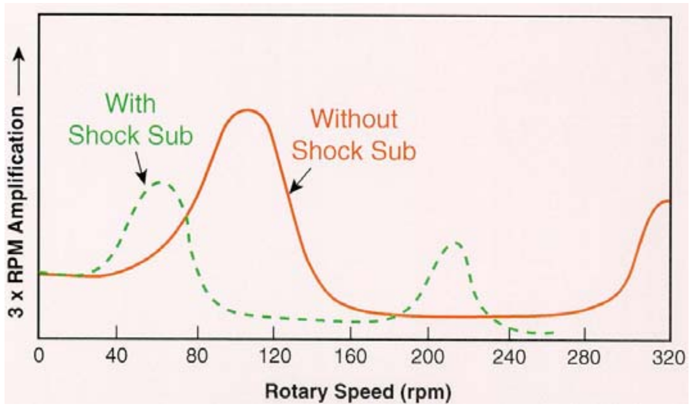


Figure 38 — Shock Sub Impact on Amplitude of Axial Force Fluctuations

However, some will argue the point that shock subs provide an excitation mechanism that maintains the axial vibration resonating in the string. In any case, when used, shock subs should only be run in their effective RPM/WOB windows.

When axial vibration is present, it is important to maintain the buckling neutral point within the BHA. This can be done in one of two ways, 1) have the proper amount of BHA in the hole for the given WOB, or 2) limit the amount of WOB application for the amount of BHA in the hole. Figure 41 illustrates this concept. As seen in the figure, axial vibration causes axial force fluctuations and therefore, causes constant oscillation of the neutral point. With the fluctuation of this point, the surrounding area cycles between compression and tension, generating cyclic stress in the string. More important, if this point is allowed to climb out of the BHA and into the drill pipe, significant damage may occur. The below equation provides a safe WOB limitation for vertical and near vertical (< 15°) wells.

$$WOB < \frac{(\mathrm{Total\ Air\ Weight\ of\ BHA})\mathrm{K_B}\cos\theta}{1.20}$$

Where:

K_B is the mud buoyancy factor

θ is the hole angle

Total Air Weight of BHA = drill collars and HWDP total air weight.

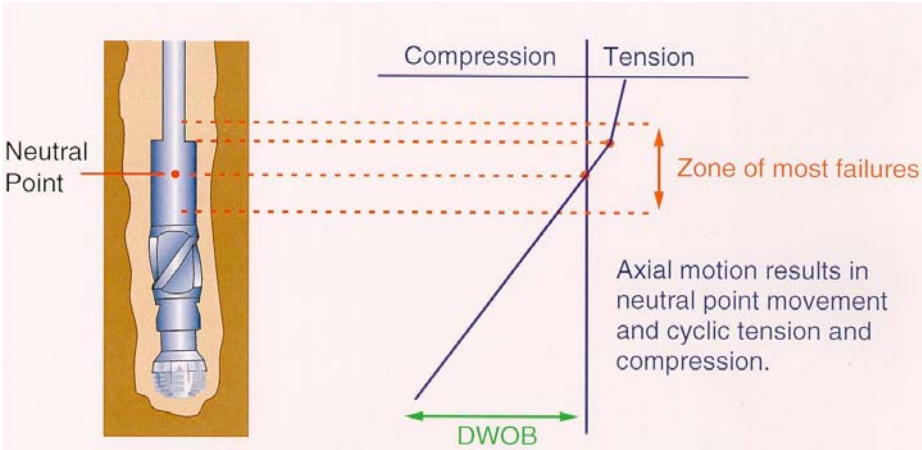


Figure 39 — Maintaining the Buckling Neutral Point in the BHA When Axial Vibration is Present

11.1.1.3 Lateral Vibration

Lateral vibration is an extremely damaging vibration type. The side to side impact of the drill string with the wellbore (see Figure 42) generates bending in the drill string and amplifies the number of bending cycles the drill string undergoes while rotating. Lateral vibration is often associated with very high acceleration levels, causing very high impact forces to the drill string. Lateral vibration is a much more complex mechanism than axial or torsional vibration, making detection and mitigation actions difficult. Lateral vibration often occurs in unstabilized BHA's.

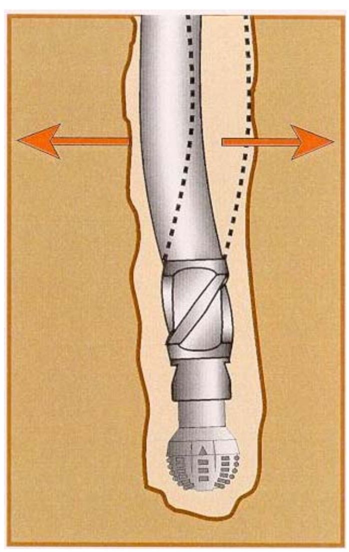


Figure 40 — Lateral vibration, the side to side impact of the drill string

11.1.1.3.1 Types of Lateral Vibration

11.1.1.3.1.1 General

Several types of lateral vibration occur during the drilling process. These types can be defined as bit whirl, BHA forward whirl, and BHA backward whirl. Noted within all these types is "whirl", which can be defined as the eccentric rotation of the bit and/or BHA about a point other than its center. The whirling motion is illustrated in Figure 43 below. As seen from the figure, the onset of whirl will generate lateral vibration.

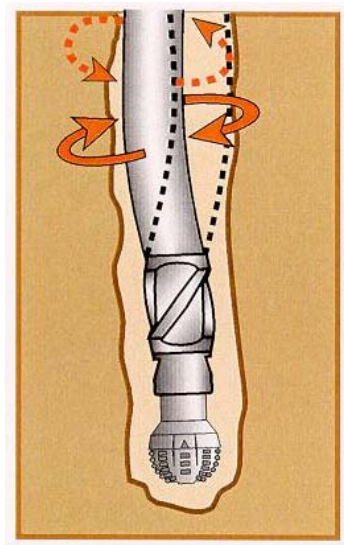


Figure 41 — Example of "Whirling"

11.1.1.3.1.2 Bit Whirl

Bit whirl is very common with PDC bits and not as prevalent, if at all, with tricone bits. Bit whirl is generated from excessively high side cutting forces. These forces are applied in an unbalanced nature, causing the bit to walk around the bore wall, often in a reverse direction to the string's clockwise rotation (see Figure 44). The motion is an eccentric rotation of the bit about a point other than its center. Detrimental effects of bit whirl are slow ROP, premature bit damage, and the initiation of other vibratory mechanisms like BHA whirl. Bit whirl often occurs with PDC bits in interbedded soft and hard formations in vertical and near vertical wells. Bit whirl also occurs during reaming when the bit is least constrained from gearing with the wellbore, making the practice of increasing to full speed after tagging bottom a good one. The factors that affect bit whirl include the presence of BHA whirl, RPM, WOB, changing of formation hardness, and the poor breaking-in of PDC bits.

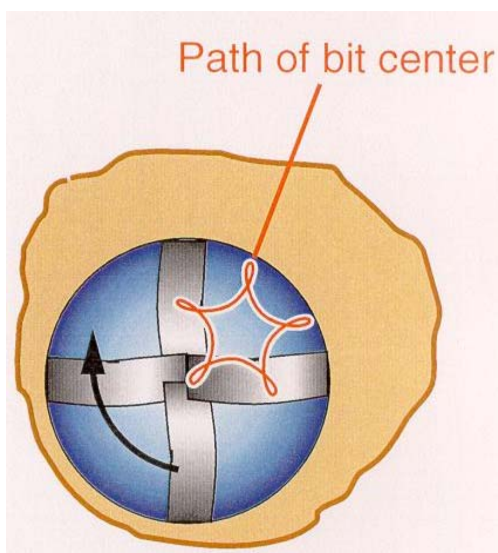


Figure 42 — Path of bit center

Methods of detecting bit whirl include:

— Premature cutter damage

— Over-gauge hole

— Reduction in ROP

— Increase in downhole on bottom torque

— High frequency downhole lateral / torsional vibration

— Excessive downhole shocks

— Increase in hole over-gauge

Most mitigation methods for whirl events are applicable to both BHA and bit whirl. One of the most common mitigation methods used specifically for bit whirl is the use of Anti-whirl bits. Anti-whirl bits are effective in minimizing bit whirl because they minimize side cutting, preventing the development of overgauge hole while drilling. The addition of bit wear plates on Anti-whirl bits enables the absorption of excessive side forces, providing a more stable cutting action. Figure 45 illustrates the design of Anti-whirl bits.

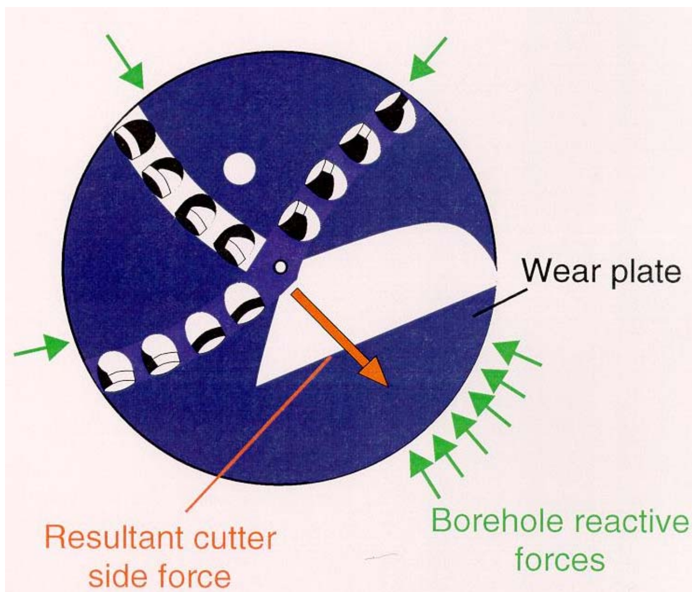


Figure 43 — Anti-whirl Bits Improve Bit Whirl Effects

11.1.1.3.1.3 Forward BHA Whirl

Another form of lateral vibration is BHA forward whirl. BHA whirl is the eccentric rotation of the BHA about a point other than its geometric center. Once the centerline of the BHA is offset from the hole centerline, the centrifugal motion induces bowing of the BHA. The resulting eccentricity causes dynamic imbalance and high lateral shocks. BHA whirl is often caused by friction driven gearing of stabilizers and tool joints with the wellbore. Two types of BHA whirl are often observed; forward and backward BHA whirl.

Forward BHA whirl can be defined as the event when the center line of the BHA is offset from the hole center line and the rotation of the BHA center line occurs at the same speed and in the same clockwise direction as the top drive (TD) / rotary table (RT). If the offset of the BHA centerline is sufficient, the same local circumferential point of the BHA will be in contact with the bore wall throughout the motion. Figure 46 illustrates BHA forward whirl. The friction force between the BHA and wellbore play a strong impact in whether or not forward BHA whirl will initiate and remain present. Mud properties like lubricity strongly impact forward BHA whirl.

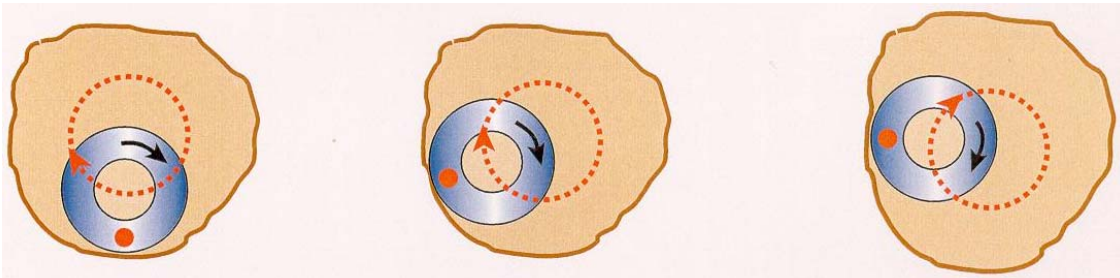


Figure 44 — Forward BHA Whirl

11.1.1.3.1.4 Backward BHA Whirl

Backward BHA whirl is also the eccentric rotation of the BHA about a point other than it centerline. Once the centerline of the BHA is offset from the hole centerline, if the offset is sufficient enough and the BHA is in contact with the wellbore, the friction between the BHA surface and the wellbore will oppose forward whirl movement. If the friction is high enough, the BHA centerline will rotate backwards (anti-clockwise) and generate backward BHA whirl. Figure 47 below illustrates the backward BHA whirl motion.

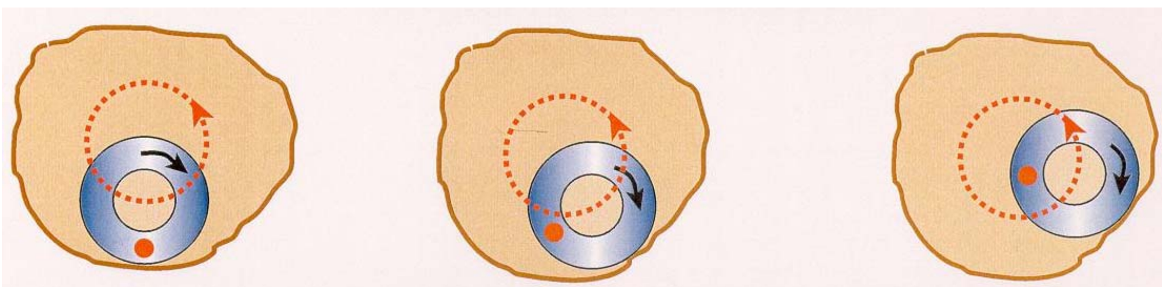


Figure 45 — Backward BHA Whirl

Often backward BHA whirl will result in an extremely erratic motion of planetary gearing. This motion yields high lateral impact shocks and is very damaging to the drill string. An illustration of the motion is provided below in Figure 48.

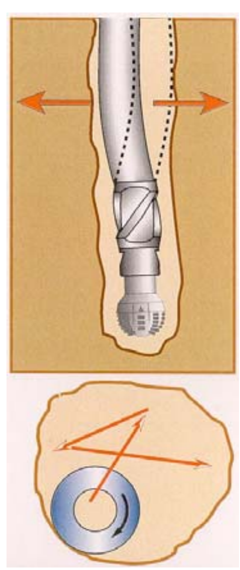


Figure 46 — Erratic BHA Whirl Generating Lateral Shocks

11.1.1.3.2 Detrimental Impact

Lateral vibration is quite detrimental to the drilling process. Some detrimental impacts associated with lateral vibration include external and internal impact damage to the drill string, especially specialty tools having internal circuitry like MWD/LWD tools; asymmetric wear to downhole tools like stabilizers; an increase in the number of bending cycles which increases the rate of fatigue damage.

11.1.1.3.3 Typical Environment

Lateral vibration will often occur in vertical to near vertical wells in washed out wellbores with unstabilized pendulum assemblies. Generally speaking, the BHA can be considered an imbalanced mass. Additionally, with non-linear friction interaction of the bit and BHA with the wellbore, lateral motion will result. Therefore, at any time the BHA cannot be stabilized, some magnitude of lateral vibration will almost assuredly be present. These environments would include bi-center bit runs in deepwater wells, over-gauge holes, washed out wellbores, etc.

11.1.1.3.4 Factors Affecting

Factors, which affect lateral vibration (bit and BHA whirl), include RPM, WOB, hole size, hole geometry, mud lubricity, and BHA stabilization.

11.1.1.3.5 Identification Methods

Lateral vibration is a complex mechanism, which can be difficult to identify. Some methods of detecting lateral vibration include:

— Localized wear on tool joints

— Asymmetric wear on stabilizer blades

— Surface torque and hookload vibrations

— Increase in mean surface and downhole torque

— High frequency downhole lateral vibrations

— Excessive downhole shocks (torque and lateral)

— BHA fatigue failures (washouts / twistoffs)

11.1.1.3.6 Mitigation Methods

It is important to keep lateral vibration to a minimum because of the significance of detrimental impacts associated with this vibration type. Some general methods of mitigating lateral vibration used in the industry include the following:

— Use anti-whirl bits (Bit whirl)

— Use bits with flatter profiles (Bit whirl)

— Reduce RPM / Change WOB according to forward or backward whirl mode

  — Forward Whirl – Reduce RPM / Increase WOB

  — Backward Whirl – Reduce RPM / Decrease WOB

— Use roller reamers instead of stabilizers or use non-rotating stabilizers

— Use non-rotating drill pipe protectors

— Change mud properties (mud weight, increase lubricity)

— Minimize BHA bending by using optimum WOB and RPM combinations

— Run packed assemblies to keep BHA centered in wellbore

— Replace straight stabilizers with spiraled stabilizers

— Run largest drill collar size for hole size and drill in-gauge hole

— Use downhole motors

— Increase flowrate.

— Reduce distance between bit and 1st stabilizer

— Ream at lower RPM

— STOP, lift off bottom, stop rotation, restart rotation to 1/2 targeted RPM, SLOWLY tag bottom, increase WOB to targeted value, and then increase RPM to target value

— Use automated feed-off control systems (automatic drillers)

One mitigation action of particular interest is the use of spiraled or non-rotating stabilizers instead of straight blade stabilizers. In inclined holes, straight blade stabilizers generate lateral shocks to the drill string as illustrated in Figure 49 below.

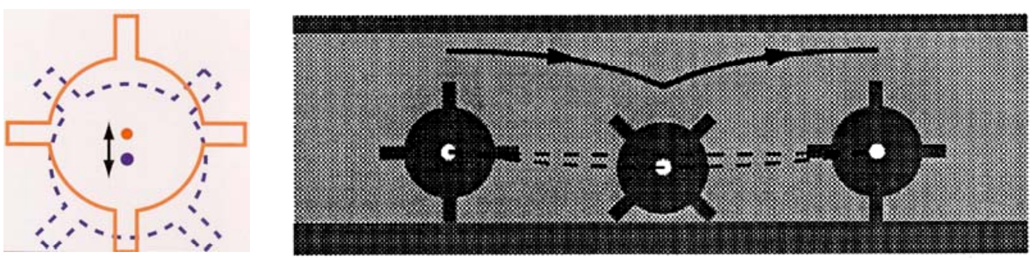


Figure 47 — Straight Blade Stabilizers Generate Lateral Shocks in Inclined Wellbores

Also specific to inclination wells is the expectation in the reduction of both axial and lateral shocks. Figure 50 below illustrates this observation. As hole angle increases, side loads increase and the amount of force available for WOB decreases. This increase in side load, reduces collar movement and whirling, and reduces axial movement due to the increase in friction forces (stabilization forces).

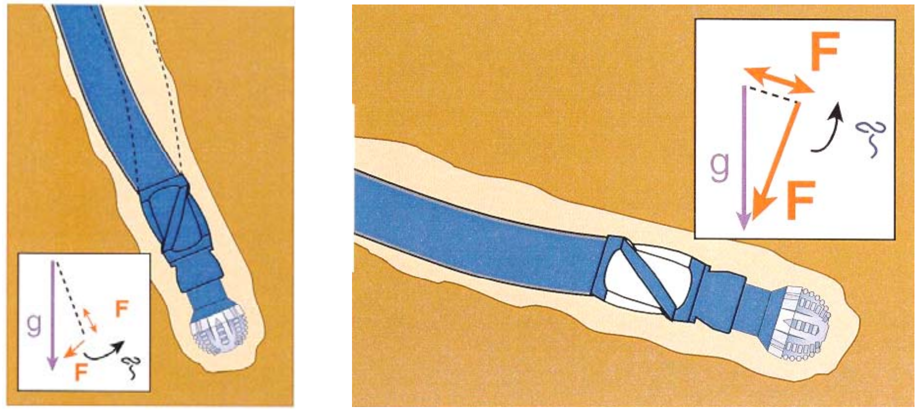


Figure 48 —Increased Hole Angle Reduces Axial and Lateral Shocks

Unfortunately, the increase in side load is not a cure all because along with the increase in axial friction, there is also an increase in rotational friction, leading to another detrimental form of vibration, torsional vibration.

11.1.1.4 Torsional Vibration

Because the drilling process requires some form of rotation, either the string and bit or simply bit rotation, torsional vibration is almost always present at some level. A less severe form of torsional vibration is the simple alternate slowing down and acceleration of the BHA.

A more severe form of torsional vibration is full stick-slip motion. This form of torsional vibration is quite damaging and should be avoided with all effort. Figure 51, below, illustrates the stick slip event.

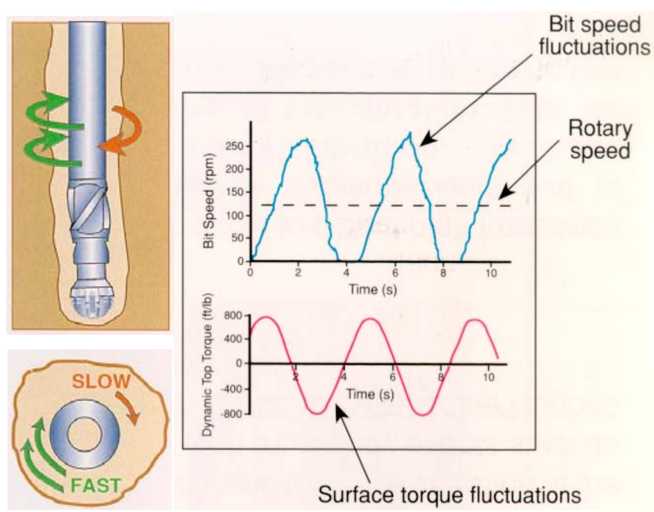


Figure 49 — Stick-Slip Torsional Vibration

During stick-slip, the bit stops rotating momentarily, causing the string to torque up and then spin free, accelerating the bit and BHA to high speeds. In extreme cases, the BHA RPM oscillation will be so extreme that the BHA will spin backwards and reach negative RPM values.

11.1.1.4.1 Detrimental Impact

Detrimental impact of torsional vibrations include damaged bit teeth and premature bit failure, reduction in ROP (as much as 50%), torsional damage to BHA connections and drill pipe tool joints, connection back-offs, and BHA and drill string fatigue failures (twistoffs and washouts).

11.1.1.4.2 Typical Environment

Although not limited to any one particular environment, typical environments for torsional vibration include high angle directional wells with sufficient drill string / wellbore interaction and aggressive PDC bits with high WOB.

11.1.1.4.3 Factors Affecting

Several factors affect torsional vibration. These factors include bit type, hole angle, WOB, RPM, lubricity of mud system and filter cake, and BHA stabilization.

11.1.1.4.4 Identification Methods

Methods for identifying the presence of torsional vibration include:

— Surface torque fluctuation > 15% of mean surface torque, or top drive stalling

— Excessive surface RPM fluctuation

— Excessive downhole torque fluctuation (bit torque)

— Increase in surface torque cyclicity (sigma torque)

— Increase in downhole shocks

— Increase in downhole torsional vibration

— Premature bit cutter damage

— Drill string connection torsional damage (stretched pins, belled boxes, seal and thread galling)

— Connection back-offs

— BHA fatigue failures (washouts / twistoffs)

11.1.1.4.5 Mitigation Methods

Mitigation methods for drilling vibration are often formation and / or area-specific. However, general mitigation methods for torsional vibration include:

— Reduce WOB / increase RPM

— Use roller reamers instead of stabilizers or use non-rotating stabilizers

— Use non-rotating drill pipe protectors

— Reduce or modify placement of stabilizers

— Smoother well trajectories

— Change top drive to higher gear

— Frequent borehole cleaning (wiper trips and reaming)

— Increase mud flowrate

— Change mud properties (mud weight; increase lubricity, especially in deviated wells)

— STOP, lift off bottom, stop rotation, restart rotation to 1/2 targeted RPM, SLOWLY tag bottom, increase WOB to targeted value, and then increase RPM to target value

— Use automated feed-off control systems

— Use automated rotary feedback systems

Of particular note, the mitigation method of reducing WOB and increasing RPM to reduce torsional vibration, especially stick-slip, is one of the most common methods to momentarily reducing the event. As seen in Figure 52 below, lowering WOB reduces bit torque and side forces while increasing RPM increases the rotational inertia and reduces the impact of rotational friction.

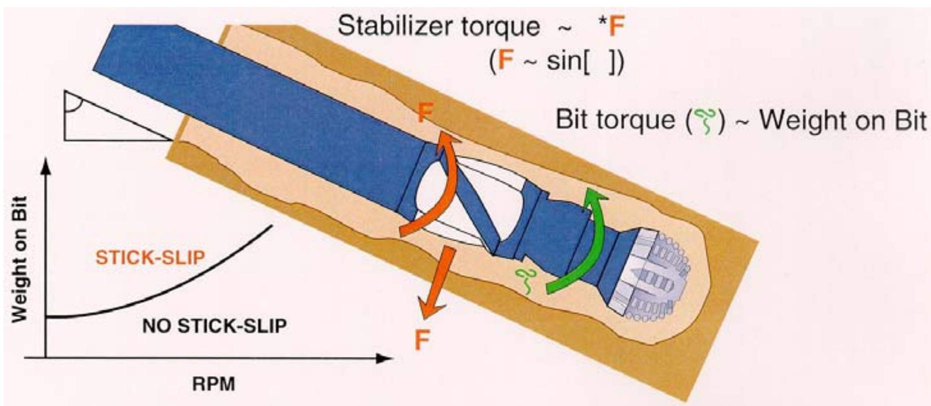
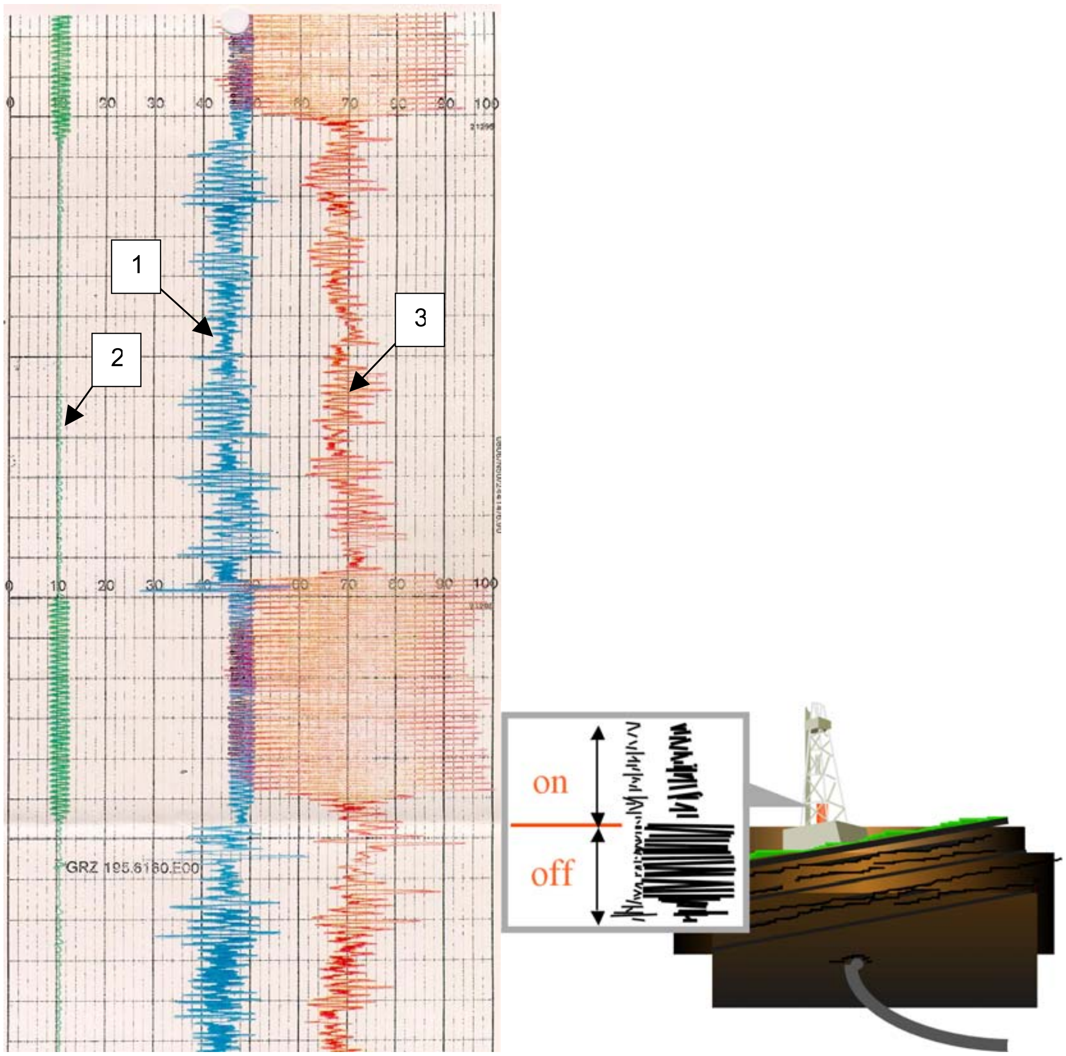


Figure 50 — Reducing WOB and Increasing RPM to Mitigate Stick-Slip

Often, the mitigation of torsional vibration requires more advanced methods than simply reducing WOB and increasing RPM. For example, one of the most impacting methods to reduce stick-slip is the use of soft-torque rotary systems (STRS). Soft-Torque Rotary Systems (STRS), also known as Rotary Feedback Systems, monitor torque data (via amperage) from the motor of the top drive or rotary table and force a variation in the top drive/rotary table motor RPM such that torsional oscillations within the drill string are dampened. Consider the drill pipe as a torsional spring and the BHA as a flywheel. Without STRS, the motor output of the top drive/rotary table is a constant RPM, providing a constant twist in the drill string. As the BHA interacts with the formation, the twist in the BHA will vary, sending torsional vibrations upward throughout the drill string. When reaching the top drive/rotary table, the top drive/rotary table has a constant RPM control and cannot alter the RPM like the torsional vibration did during its travel upward through the string. Therefore, the torsional vibration is reflected back downward through the string. Because the motor of the top drive/rotary table is attempting to maintain constant RPM, the torsional vibration is reflected down the string with increased amplitude. These torsional vibrations are resonant because the energy stored in the torsional vibrations cannot escape out of the drill string, leading to cyclic torsional loading causing stick-slip events, particularly with PDC bits.

STRS are wired into the rig's SCR unit and top drive/rotary table. Parameters of the drill string are entered into its software and the system calculates the drill string's predictable torsional vibration behavior. Most STRS center on current (torque) feedback to detect the amplitude and frequency of torsional oscillations at the surface. With this data, the STRS applies an electronic correction signal to the top drive throttle and adjusts the motor RPM to coincide with the cyclic torque (Σ torque). When large torque surges travel upward to the top drive, the system's correction signal is an increase in the top drive throttle to absorb the high torque surge. Likewise, when the torque is momentarily reduced from the mean at the top drive, the system is decreasing the top drive throttle to compensate for the drop in torque. The end result is a forced fluctuation of top drive RPM to absorb the torsional oscillations transmitted up through the string, providing a smooth application of constant surface torque. Figure 53 below provides an illustration of how STRS work.



Key

1 Top Drive Voltage Feedback (RPM)

2 STRS Output

3 Top Drive Current Feedback (Torque)

Figure 51 — Soft-Torque Rotary Systems (STRS)

STRS control the output of the top drive such that torsional vibrations are mitigated. SRTS dampen drill string vibration and prevent stick-slip oscillation of the bit. The parameters of the drill string are input into the STRS and the string's mass-inertia and elasticity values are calculated. From these parameters, the STRS calculates an electronic correction signal. The signal is looped into the rig's SCR-unit, which controls the speed and torque of the top drive motor. The drive system for the drill string rotation movement is now transformed from a stiff amplifier for torsional vibrations to a soft dampening system.

11.1.1.5 Mode Coupling of Vibration Mechanisms

In addition to simple axial, lateral, and torsional vibration events, often these events occur simultaneously, or coupled. One vibration event may trigger another event and vice-versa. For example, during the slip phase of stick-slip, the BHA spins with high rotational acceleration. During this phase, it is very common to observe high lateral shocks. Alternatively, when lateral shocks are occurring, they momentarily slow the rotational speed of the BHA, causing a reactive torque impulse during the release of the BHA from the wellbore. The vibration mode coupling of events is often characterized by more definitive names like parametric resonance or BHA forced vibration. However, specific naming of these events is not as important as understanding that vibration events occur simultaneously more often than singularly. Figure 54 below illustrates some common mode coupled vibration events.

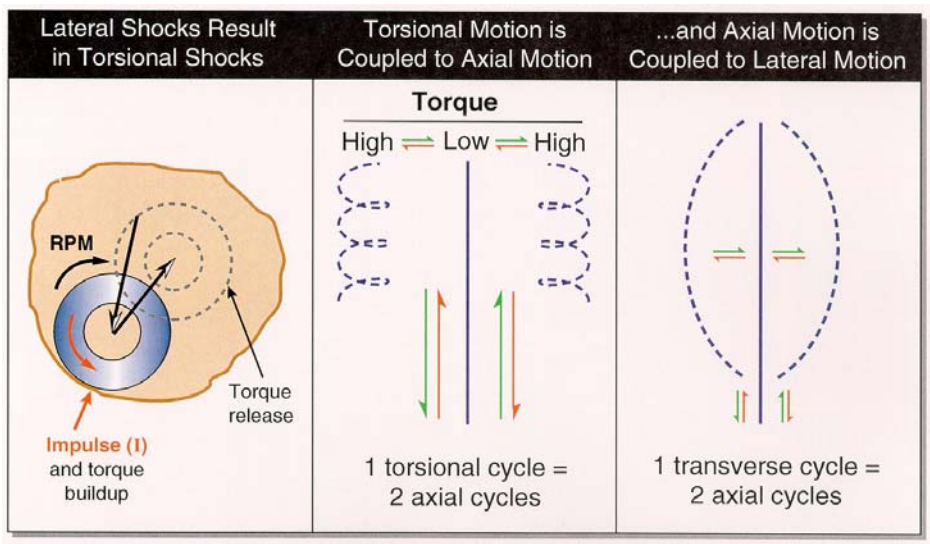


Figure 52 — Mode Coupling of Vibration Mechanisms

11.1.1.6 Detrimental Impact of Drilling Vibrations

Drilling vibrations carry significant detrimental impacts. Some of the most observed ones include:

— Low ROP – Rig energy is wasted wearing the drill string instead of breaking formation.

— Increased number of trips – More time is spent tripping for prematurely worn bits and / or damaged drill string components.

— Increased amount of rig down time – Extensive vibration increases the amount of drill string failures leading to fishing charges, trips for washouts, and sidetracking costs.

— Lost in hole charges – Tools left in the hole are quite costly.

Figure 55 provides illustrations of the detrimental impact of drilling vibrations.

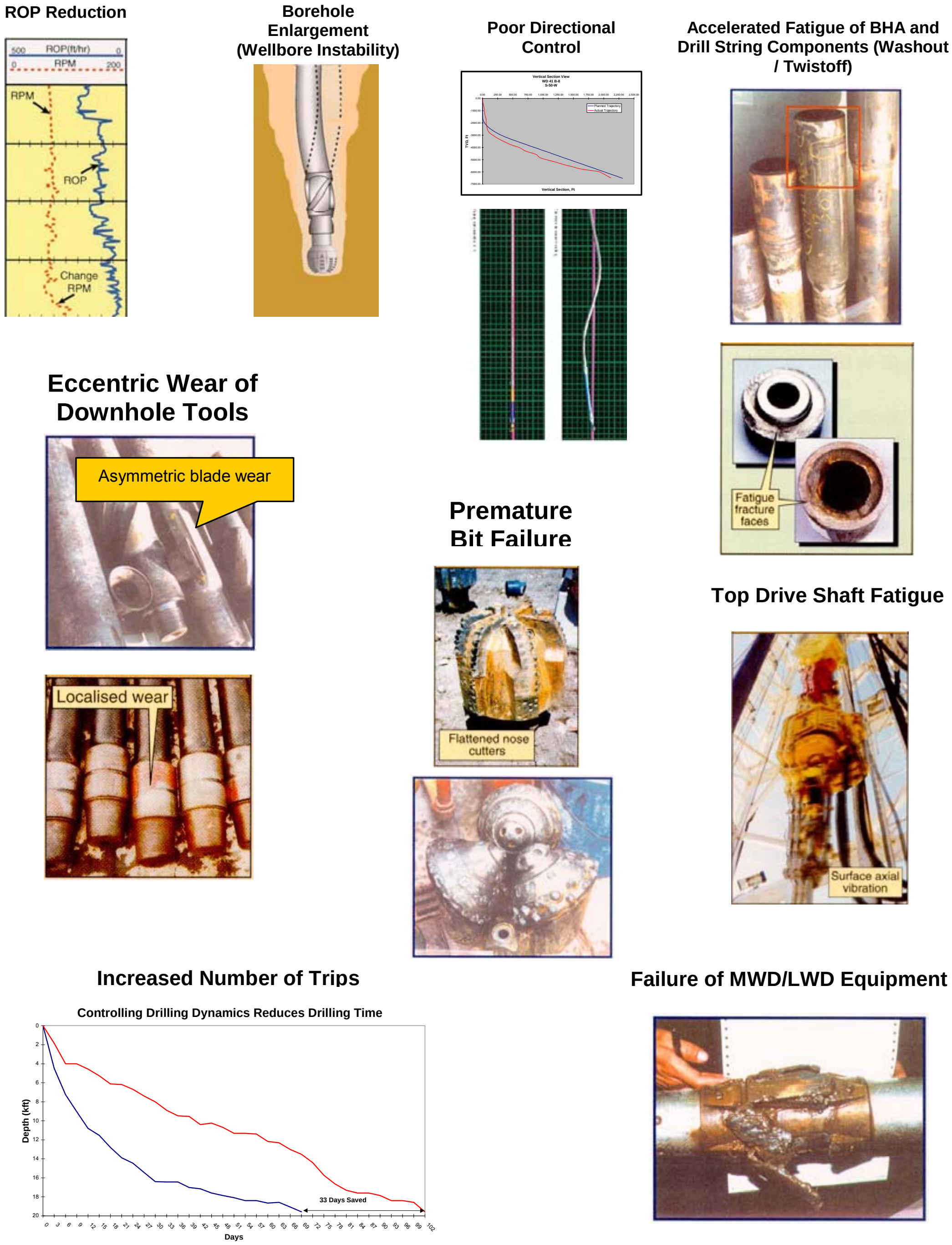


Figure 53 — Detrimental Impact of Drilling Vibrations

11.1.1.7 Measuring Drilling Vibrations

When combating drilling vibrations, it is essential to establish what vibration mechanism is occurring downhole so that the proper mitigation action can be used. As discussed, each vibration mechanism has its own symptom, which assists in their detection. However, several vibration mechanisms have similar symptoms, making correct identification difficult, especially during mode coupling events. To identify vibration mechanisms correctly requires the detection of more than one symptom and the use of more than one sensor source (surface and downhole). An effective drilling dynamics control program should use both surface and downhole detection sensors to collect the appropriate amount and type of data for the identification of specific vibration events.

The "closed-loop" drilling concept has been published and discussed by many authors (Figure 56) [29), 30)]. The closed-loop-drilling concept is a process by which drilling parameters are continuously monitored and dynamic dysfunctions are identified and eliminated in a manner such that drilling efficiency is increased. The closed-loop process is identified as [25)]:

— Motions are sensed by downhole and surface sensors

— Drilling phenomena are discriminated

— The severity of each phenomenon is quantified

— Drilling diagnostics are transmitted to the surface

— Severity levels are displayed on the rig floor

— The driller makes adjustments to the drilling controls

— Drill string and bit behavior changes

— The downhole tool senses the changed motions, etc.

— The driller sees the effect of control adjustments.

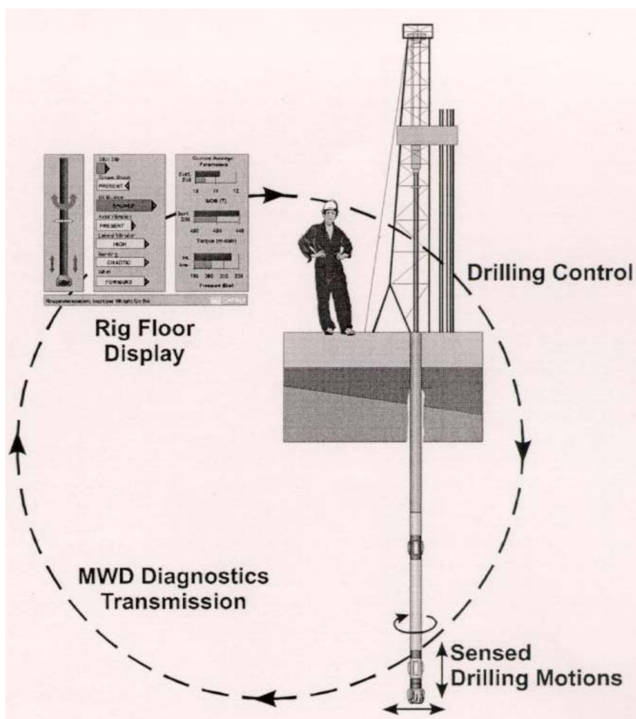


Figure 54 — Closed-loop Drilling Concept (SPE 30523) [25)] (SPE 49206) [24)]

29) Lubinski, 1961.

30) Hansford and Lubinski, 1966.

Implementation of the closed-loop-drilling concept is one of the most effective means of fine-tuning the drilling process to eliminate drilling vibration. For proper implementation of the process, both surface sensors and downhole sensors should be used to collect drilling parameters.

11.1.1.7.1 Surface Detection

11.1.1.7.1.1 General

A cost-effective means of gathering drilling vibration data is the collection of surface drilling mechanics data. Most mudlogging companies offer services of this type and the accuracy of detecting drilling vibration events using only surface data is improving as technology pushes forward in the industry. However, sole surface data collection to identify downhole drilling vibration events has its limitations. Some limitations of surface detection include:

— Downhole vibrations are damped at the surface by wellbore wall contact

— Hookload vibration sensors are difficult to install

— Topdrives are typically poorly instrumented (if at all)

— Surface sensors are insensitive to most damaging downhole lateral shocks / vibrations

— Surface vibrations are phase-shifted from downhole vibrations

— Interpretation of downhole conditions is difficult due to limitations of models.

In spite of the above limitations, surface measurements complement downhole data collection and assist in the proper identification of vibration events.

An array of sensors is used to collect surface drilling mechanics data. These sensors include proximity sensors, rotary torque sensors, and pressure sensors.

11.1.1.7.1.2 Proximity sensors

Proximity sensors (Figure 57) consist of an electronic detector which emits a pulse each time a metal activator passes in front of it. When used to measure surface RPM of the drill string, a metal activator is fitted to the top drive or rotary table and the proximity sensor detects the speed at which the metal activator passes the sensor. Proximity sensors are also used to measure pump strokes. For this application, the proximity sensor is placed near the end of the pump rod and the rod itself, serves as the metal activator.

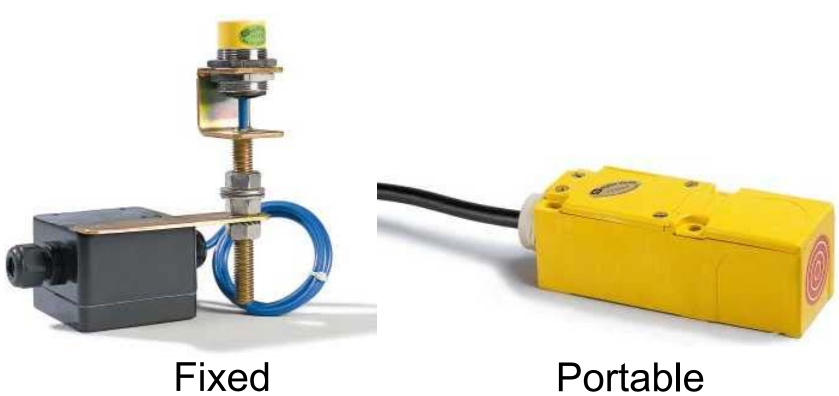


Figure 55 — Surface Proximity Sensors

11.1.1.7.1.3 Rotary torque sensors

Rotary torque sensors (Figure 58) are used to collect high frequency surface torque measurements. Some of the most modern rotary torque sensors are hall-effect current transducers used to measure the top-drive current (torque). The sensor is clamped around the cable feeding the electric motor driving the top drive or rotary table. The electric current drawn by the motor is proportional to the rotary torque applied to the drill string.

Figure 56 — Electrical Rotary Torque Surface Sensor

11.1.1.7.1.4 Pressure sensors

Pressure sensors (Figure 59) are used to measure surface weight on hook (WOH), stand pipe pressure, and even rotary torque. Pressure sensors are equipped with strain gauges. When fitted to the oil-filled pressure transducer on the dead line anchor, they are capable of measuring WOH. These same sensors are also fitted to the mud / oil diaphragm unit on the standpipe manifold to measure stand pipe pressure. Pressure sensors can also be fitted to the hydraulic circuit of the drive train tensioner for use in measuring rotary torque on chain-driven rotary tables.

Figure 57 — Surface Pressure Sensor

11.1.1.7.2 Downhole Detection

11.1.1.7.2.1 General

Downhole detection of vibration events is one of the most accurate forms of vibration detection. Sensors are positioned in downhole specialty tools like MWD/LWD and high-frequency downhole data collection tools. These sensors collect the downhole data, process and filter the data, categorize severity levels, and then transmit the appropriate severity levels to the surface through telemetry methods. Once at the surface, the tool's recorded raw data can then be download for post-vibration event analysis in higher data resolution. Unlike surface sensors however, downhole sensors used to collect downhole data are subject to the very damaging vibration they are collecting data on. Downhole sensors used to collect drilling vibration data generally include three types, accelerometers, magnetometers, and strain gauges.

11.1.1.7.2.2 Accelerometers

Most MWD/LWD packages (Figure 60) include an orthogonally mounted tri-axis accelerometer package. This accelerometer package detects dynamic accelerations of the string in the X, Y, and Z-axes. Measuring mean and RMS (root mean square) values of axial, lateral, and torsional accelerations provide a good indication of the relative magnitudes of dynamic motions. Often equipped with this sensor package is the standard gamma ray package. Gamma ray lithology measurements assist in identifying vibration dysfunctions associated with formation changes.

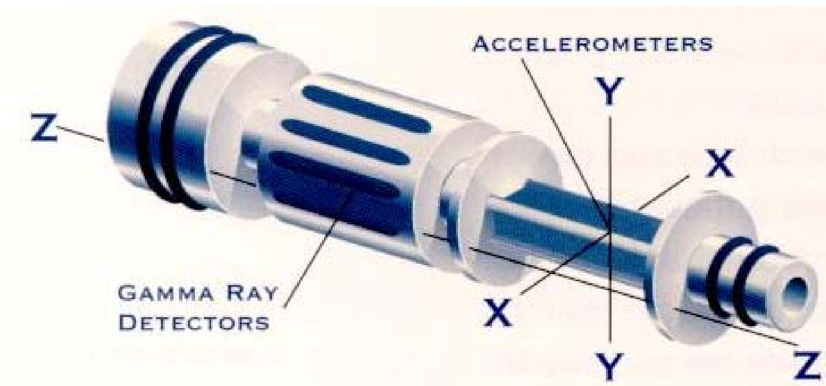


Figure 58 — Downhole Orthogonally Mounted Tri-Axis Accelerometer Sensor Package

Service companies offering MWD/LWD downhole data collection services use different methods for which to obtain and process the data obtained from accelerometers. For example, some companies commonly use mean, RMS, and actual g-force values for describing the magnitude of the vibration type. Other companies are known for monitoring "shock" counts. A g-force threshold is set for measuring g's in the respective X, Y, and Z axes, and the MWD/LWD tool counts the number of shocks above the preset threshold. Measuring shocks above a preset threshold level provides a valuable real time dynamic monitoring system. However, preset threshold values should be set low enough to provide sufficient data to characterize the vibration event. Figure 61 provides an illustration of this technique.

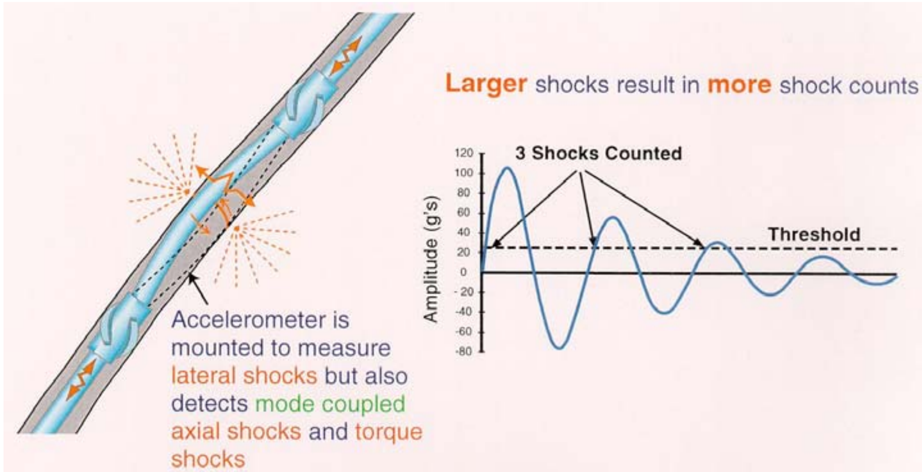


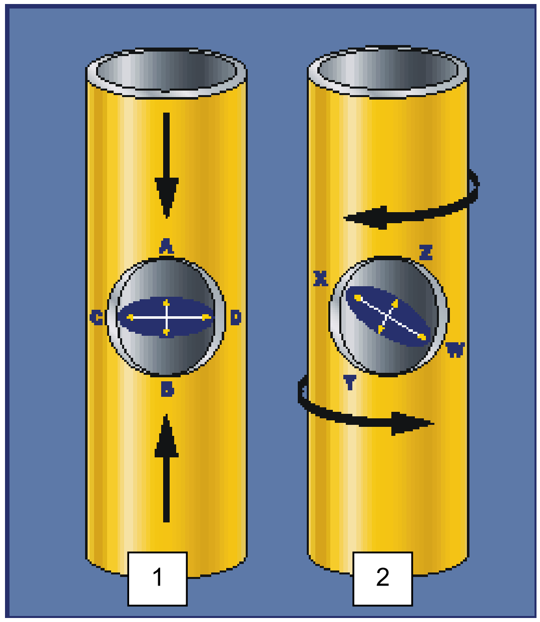
Figure 59 — Measuring Shock Counts with Downhole Accelerometers

11.1.1.7.2.3 Magnetometers

Magnetometers are mostly used for the detection of torsional vibration. Magnetometers are designed to measure RPM fluctuations in the X and Y-axes through the use of the earth's inherent magnetic field. Using specific stick-slip algorithms, the X and Y-axis RPM values are used to determine the bit and BHA rotational behavior and identify stick-slip events. Minimum, maximum, and average RPM are calculated and by observing separations in minimum downhole and maximum downhole RPM, torsional vibration events can be observed.

11.1.1.7.2.4 Strain Gauges

Strain gauge packages (Figure 62) are often included with standard MWD/LWD packages. Figure 62 below provides an illustration of how strain gauges work. When A-B is compressed and C-D is stretched, the strain gauge measures and quantifies the magnitude of axial force (WOB). In addition, when X-W is stretched and Z-Y is compressed, the strain gauge measures and quantifies the magnitude of torque applied.



Key

1. WOB results in compressing A-B and stretching C-D.

2. Torque results in stretching X-W and compressing Z-Y.

Figure 60 — Downhole Strain Gauges Detect Downhole WOB and Torque

12 Corrosion and sour service applications

12.1 Drill pipe corrosion

12.1.1 Corrosive agents

The principal corrosive agents affecting drill stem materials in water-base drilling fluids are dissolved gases, dissolved salts, bacterial degradation and acids. Dissolved gases include oxygen, carbon dioxide and hydrogen sulphide.

12.1.1.1 Oxygen

Oxygen is the most common corrosive agent. In the presence of moisture it causes rusting of steel, the most common form of corrosion. Oxygen causes uniform corrosion and pitting, leading to washouts, twist offs, and fatigue failures. Since oxygen is soluble in fresh water, and as most drilling fluid pits are open to the air, the drill stem is continually exposed to potentially severe corrosive conditions. Oxygen is compressed and dissolved as it passes from the pit to the high pressure pumps for the trip downhole. At the return flowline, the system is immediately re-saturated with oxygen.

A corrosion inhibitor and an oxygen scavenger along with a higher pH are usually needed to minimize the corrosive effect of oxygen.

12.1.1.2 Carbon dioxide

Carbon dioxide dissolves in water to form a weak acid (carbonic acid) that corrodes steel in the same manner as other acids (by hydrogen evolution), unless the pH is maintained above 6. At pH values higher than 6, carbon dioxide corrosion damage is similar to oxygen corrosion damage, but at a slower rate. When carbon dioxide and oxygen are both present, however, the corrosion rate is higher than the sum of the rates for each alone.

Carbon dioxide in drilling fluids may come from the makeup water, gas bearing formation fluid inflow, thermal decomposition of dissolved salts and some organic drilling fluid additives, or bacterial action on organic material in the makeup water or on other drilling fluid additives.

12.1.1.3 Hydrogen sulphide

Hydrogen sulphide dissolves in water to form hydrosulphuric acid, an acid somewhat weaker and less corrosive than carbonic acid. In solution it may cause pitting, particularly in the presence of oxygen and/or carbon dioxide. A more significant action of hydrogen sulphide is its effect on a form of hydrogen embrittlement known as sulphide stress cracking (see 11.2 for details).

Hydrogen sulphide in drilling fluids may come from the makeup water, gas-bearing formation fluid inflow, bacterial action (sulphate reducing bacteria) on dissolved sulphates, or thermal degradation of some sulphur-containing drilling fluid additives.

12.1.1.4 Dissolved salts

Dissolved salts (chlorides, carbonates, and sulphates) increase the electrical conductivity of drilling fluids. Since most corrosion processes involve electrochemical reactions, the increased conductivity may result in higher corrosion rates. However, concentrated or saturated salt solutions are usually less corrosive downhole than dilute solutions because of decreased oxygen solubility in the saturated salt water brines. Dissolved salts also may serve as a source of carbon dioxide or hydrogen sulphide in drilling fluids.

Dissolved salts in drilling fluids may come from the makeup water, formation fluid inflow, drilled formations, or drilling fluid additives.

The popular 3% KCl salt brine used as the external phase can be troublesome as it has high conductivity, but it is not salty enough to minimize the ppm of dissolved oxygen. It is desirable to have less than 1 ppm dissolved oxygen in the external phase and the use of both an oxygen scavenger and a corrosion inhibitor can be helpful.

10.1.1.5 Acids

Acids corrode metals by lowering the pH (causing hydrogen evolution) and by dissolving protective films. Dissolved oxygen in the presence of acids appreciably accelerates the corrosion rate and dissolved hydrogen sulphide (hydrosulphuric acid) greatly accelerates hydrogen embrittlement. The atomic hydrogen enters the metal and forms molecular hydrogen with increased stresses following**.**

Organic acids (formic, acetic, etc.) can be formed in drilling fluids by bacterial action or by thermal degradation of organic drilling fluid additives. Organic acids and mineral acids (hydrochloric, hydrofluoric, etc.) may be used during workover operations or stimulating treatments. Their presence requires the correct acid corrosion inhibitor be added at the start of the operation before damage is noted.

12.1.2 Important factors affecting corrosion rates of drill stem materials

12.1.2.1 pH

This is a scale for measuring hydrogen ion concentration. The pH scale is logarithmic; i.e., each pH increment of 1,0 represents a tenfold change in hydrogen ion concentration, The pH of pure water, free of dissolved gases, is 7,0. The pH values less than 7 are increasingly acidic, and pH values greater than 7 are increasingly alkaline. In the presence of dissolved oxygen, the corrosion rate of steel in water is relatively constant between pH 4,5 and 9,5. The corrosion rate and attack increases rapidly at pH values lower than 4,5 and decreases slowly at pH values greater than 9,5.

However, a high pH is not recommended with the use of aluminium drill pipe and can show increasing corrosion rates at pH values greater than 8,5 as sodium aluminate and hydrogen can be generated.

12.1.2.2 Temperature

In general, corrosion rates increase with increasing temperature, especially in the closed system from the pump suction to downhole and back up to the flowline.

12.1.2.3 Velocity

In general, corrosion rates increase with higher rates of flow because any protective film may be removed, leaving bare metal to react.

12.1.2.4 Heterogeneity

Localized variations in composition or microstructure may increase corrosion rates. Ringworm corrosion, sometimes found near the upset areas of drill pipe or tubing that has not been properly heat treated after upsetting, is an example of corrosion caused by non-uniform grain structure.

12.1.2.5 High stresses

Highly stressed areas can corrode faster than areas of lower stress. The drill stem just above drill collars often shows abnormal corrosion damage, partially because of higher stresses and high bending moments.

12.1.3 Corrosion damage

12.1.3.1 Forms of corrosion

Corrosion can take many forms and may combine with other types of damage (erosion, wear, fatigue, etc.) to cause extremely severe damage or failure. Several forms of corrosion can occur at the same time, but one type will usually predominate. Knowing and identifying the forms of corrosion can be helpful in planning corrective action. The forms of corrosion most often encountered with drill stem materials are listed in subclauses 11.1.3.2 to 11.1.3.5.

12.1.3.2 Uniform or general attack

During uniform attack, the material corrodes evenly, usually leaving a coating of corrosion products. The resulting loss in wall thickness can lead to failure from reduction of the material's load-carrying capability.

12.1.3.3 Localized attack (pitting)

Corrosion can be localized in small, well-defined areas, causing pits. Their number, depth, and size can vary considerably and they can be obscured by corrosion products. Pitting is difficult to detect and evaluate since it can occur under corrosion products, mill scale and other deposits, in crevices or other stagnant areas, in highly stressed areas, etc. For example, at the ends of drill pipe rubber protectors there is a shielded area where the elastomer contacts the pipe body that is hard to see and inspect where pitting and grooving can occur. Pits may serve as the origin of fatigue cracks that may cause washouts. Increased chlorides (up to saturation), oxygen, carbon dioxide, and hydrogen sulphide, and especially combinations of them, are major contributors to pitting corrosion.

12.1.3.4 Erosion-corrosion

Many metals resist corrosion by forming protective oxide films or tightly adherent deposits. If these films or deposits are removed or disturbed by high-velocity fluid flow, abrasive suspended solids, excessive turbulence, cavitation, etc., accelerated attack occurs at the fresh metal surface. This combination of erosive wear and corrosion can cause pitting, extensive damage and finally failure.

12.1.3.5 Fatigue in a corrosive environment (corrosion fatigue)

Metals subjected to cyclic stresses of sufficient magnitude will develop fatigue cracks that can grow until complete failure occurs. The limiting cyclic stress that a metal can sustain for an infinite number of cycles is known as the fatigue limit. Remedial action for reducing drill stem fatigue is discussed in Clause 9.

In a corrosive environment, no fatigue limit exists, since failure will ultimately occur from corrosion, even in the absence of cyclic stress. The cumulative effect of corrosion and cyclic stress (corrosion fatigue) is greater than the sum of the damage from each. Fatigue life will always be less in a corrosive environment, even under mildly corrosive conditions that show little or no visible evidence of corrosion.

12.1.4 Detecting and monitoring corrosion

The complex interactions between various corrosive agents and the many factors controlling corrosion rates make it difficult to accurately assess the potential corrosivity of a drilling fluid’s external phase. Various instruments and devices such as pH meters, oxygen meters, corrosion meters, hydrogen and galvanic probes, chemical test kits, drill pipe corrosion test coupons, etc. have been used to aid in field monitoring of corrosion agents and their effects.

The monitoring system described in Appendix A of API Recommended Practice 13B-1 may be used to evaluate corrosive conditions and to monitor the effect of remedial actions taken to correct undesirable conditions. Pre-weighed drill pipe corrosion ring coupons are placed in the recesses at the back of tool joint box threads at selected locations throughout the drill stem, exposed to the drilling operation for a period of time, then removed during a trip, cleaned, and reweighed in a testing laboratory.

An acceptable corrosion rate (weight loss) is generally considered to be less than 2 lb/ft^2/year. However, the degree and severity of pitting observed may be of greater significance than the weight loss measurement.

It is recommended that two locations be chosen for coupon placement; the first being at the kelly saver sub and the second being at the last joint of drill pipe before the drill collars. The coupon rings should be exposed for approximately 100 hours total time in the string.

The chemical testing of drilling fluids (see API Recommended Practices 13B-1 and 13B-2) should be performed in the field whenever possible, especially tests for pH, alkalinity and the dissolved gases (oxygen and carbon dioxide).

12.1.5 Procurement of samples for laboratory testing

When laboratory examination of drilling fluid is desired, representative samples (collect several samples over a short period of time to make sure they are representative) should be collected in a 2 1 to 4 l (0,5 gal to 1 gal) clean container, allowing an air space of approximately 1 % of the container volume and sealing tightly with a suitable stopper. Chemically resisting glass, polyethylene and hard rubber are suitable materials for most drilling fluid samples. Samples should be analyzed as soon as possible, and the elapsed time between collection and analysis reported. See ASTM D3370, *Standard Practices for Sampling Water*, for guidance on sampling and shipping procedures.

When laboratory examination of corroded or failed drill stem material is required, use care in securing the specimens. If torch cutting is needed, do it in a way that will avoid physical or metallurgical changes in the area to be examined. Specimens should not be cleaned, wire brushed, or shot blasted in any manner; and should be wrapped and shipped in a way that will avoid damage to the corrosion products or fracture surfaces. Whenever possible, both fracture surfaces should be supplied. The samples should be packed and shipped either perfectly insulated from one another, or in separate boxes. This is to insure that the fracture faces do not touch before metallurgical failure analysis.

12.1.6 Drill pipe coatings

Internally coating the drill pipe and attached tool joints can provide effective protection against corrosion in the pipe bore. In the presence of corrosive agents, however, the corrosion rate of the unprotected drill stem OD may be observed to increase. Drill pipe coating is a shop operation in which the pipe is cleaned of all grease and scale, sand or grit blasted to white metal, plastic coated and baked. After baking, the coating is examined for breaks or holidays. Some have reported the coated string lasting several times the life of a bare, uncoated string.

12.1.7 Corrective measures to minimize corrosion in water-base drilling fluids

The selection and control of appropriate corrective measures is usually performed by competent corrosion specialists. Generally, one or more of the following measures is used, but certain conditions may require more specialized treatments:

a) Control the drilling fluid pH. When practical to do so without upsetting other desired fluid properties, the maintenance of a pH of 9,5 or higher will minimize corrosion of steel in water-base systems containing dissolved oxygen.

   In some drilling fluids, however, corrosion of aluminium drill pipe increases at pH values higher than 8,5 and values as high as those run with steel pipe are not recommended.

b) Use appropriate inhibitors and/or oxygen scavengers to minimize weight loss corrosion. This is particularly helpful with low pH, low solids, drilling fluids. Inhibitors should be carefully selected and controlled because different corrosive agents and different drilling fluid systems (particularly those used for air or mist drilling) require different types of inhibitors.

c) Use plastic coated drill pipe. Care should be exercised to prevent damage to the coating with wireline tools.

d) Use degassers and desanders to remove harmful dissolved gases and abrasive material.

e) Limit oxygen intake by maintaining tight pump connections and by minimizing pit-jetting. Minimize the use of hoppers over extended periods of time.

f) Limit gas-cutting and formation fluid inflow by maintaining proper drilling fluid weight.

When the drill string is laid down, stored, or transported, wash out all drilling fluid residues with fresh water, clean out all corrosion products (by shot blasting or hydroblasting, if necessary), and coat all surfaces with a suitable corrosion preventive (see API 5C1).

Improper cleaning and application of storage compounds after lay down can result in a greater corrosion rate while in storage than when the pipe is downhole drilling.

12.1.8 Extending corrosion fatigue life

While generally not affecting corrosion rates, the following measures will extend corrosion fatigue life by lowering the cyclic stress intensity or by increasing the fatigue strength of the material:

a) Use thicker-walled components.

b) Reduce high stresses near connections by minimizing doglegs and by maintaining straight hole conditions, insofar as possible.

c) Minimize stress concentrators such as slip marks, tong marks, gouges, notches, scratches, etc. (Do not use hammers on the pipe to determine fluid level during trips.)

d) Use quenched and tempered components.

12.2 Sulphide stress cracking

12.2.1 Mechanism of sulphide stress cracking (SSC)

In the presence of hydrogen sulphide (H_2S), tensile-loaded drill stem components may suddenly fail in a brittle manner at a fraction of their nominal load-carrying capability after performing satisfactorily for extended periods of time. Failure may occur even in the apparent absence of corrosion, but is more likely if active corrosion exists. Embrittlement of the steel is caused by the absorption and diffusion of atomic hydrogen and is much more severe when H_2S is present. The brittle failure of tensile-loaded steel in the presence of H_2S is termed sulphide stress cracking (SSC).

12.2.2 Materials resistant to SSC

The latest revision of NACE Standard MR-01-75, *Sulphide Stress Cracking Resistant Metallic Material for Oil Field Equipment*, should be consulted for materials that have been found to be satisfactory for drilling and well servicing operations.

Other chemical compositions, hardnesses, and heat treatments should not be used in sour environments without fully evaluating their SSC susceptibility in the environment in which they will be used. Susceptibility to SSC depends on the following:

12.2.2.1 Strength of the steel

The higher the strength (hardness) of the steel, the greater is the susceptibility to SSC. In general, steels having strengths equivalent to hardnesses up to 22 HRC maximum are resistant to SSC. If the chemical composition is adjusted to permit the development of a well tempered, predominantly martensitic microstructure by proper quenching and tempering; steels having strengths equivalent to hardnesses up to 26 HRC maximum are resistant to SSC. When strengths higher than the equivalent of 26 HRC are required, corrective measures (as shown in a later section) should be used and, the higher the strength required, the greater the necessity for the corrective measures.

12.2.2.2 Total tensile load (stress) on the steel

The higher the tensile stress on the component, the greater is the possibility of failure by SSC. For each strength of steel used, there appears to be a critical or threshold stress below which SSC will not occur; however, the higher the strength, the lower the threshold stress.

12.2.2.3 Amount of atomic hydrogen and H_2S

The higher the amount of atomic hydrogen and H_2S present in the environment, the shorter the time before failure by SSC. The amounts of atomic hydrogen and H_2S required to cause SSC are quite small, but corrective measures to control their amounts will minimize the atomic hydrogen absorbed by the steel.

12.2.2.4 Time

Time is required for atomic hydrogen to be absorbed and diffused in steel to the critical concentration required for crack initiation and propagation to failure. By controlling the factors referred to above, time-to-failure may be sufficiently lengthened to permit the use of marginally susceptible steels for short duration drilling operations.

12.2.2.5 Temperature

The severity of SSC is greatest at normal atmospheric temperatures, and decreases as temperature increases. At operating temperatures in excess of approximately 135 °F (57 ºC), marginally susceptible materials (those having hardnesses higher than 22 to 26 HRC) have been used successfully in potentially embrittling environments. (The higher the hardness of the material, the higher the required safe operating temperature.) Caution should be exercised, however, as SSC failure may occur when the material returns to normal temperature after it is removed from the hole.

12.2.3 Corrective measures to minimize SSC in water-base drilling fluids

The selection and control of appropriate corrective measures is usually performed by competent corrosion technologists and specialists. Generally, one or more of the following measures is used, but certain conditions may require more specialized treatments:

a) Control the drilling fluid pH. When practical to do so without upsetting other desired fluid properties, maintain a pH of 10 or higher. Measure and maintain the correct filtrate and mud alkalinity.

NOTE In some drilling fluids, aluminium alloys show slowly increasing corrosion rates at pH values higher than 8,5 and the rate may become excessive at pH values higher than 10,5. Therefore, in drill strings containing aluminium drill pipe, the pH should not exceed 10,5.

b) Limit gas-cutting and formation fluid inflow by maintaining proper drilling fluid weight.

c) Minimize corrosion by the corrective measures shown in 11.1.7.

NOTE While use of plastic coated drill pipe can minimize corrosion, plastic coating does not protect susceptible drill pipe from SSC.

d) Chemically treat for hydrogen sulphide inflows with higher pH and hydrogen sulphide scavenger treatments, preferably prior to encountering the sulphide.

e) Use the lowest strength drill pipe capable of withstanding the required drilling conditions. At any strength level, properly quenched and tempered drill pipe will provide the best SSC resistance.

f) Reduce unit stresses by using thicker walled components.

g) Reduce high stresses at connections by maintaining straight hole conditions, insofar as possible.

h) Minimize stress concentrators such as slip marks, tong marks, gouges, notches, scratches, (do not use hammers to determine fluid levels during trips), etc.

i) After exposure to a sour environment, use care in tripping out of the hole, avoiding sudden shocks and high loads.

j) After exposure to a sour environment, remove absorbed hydrogen by aging in open air for several days to several weeks (depending upon conditions of exposure) or bake at 204 ºC to 316 ºC (400 ºF to 600 ºF) for several hours.

NOTE Plastic coated drill pipe should not be heated above 400 °F (204 °C) and should be checked subsequently for holidays and disbonding.

The removal of hydrogen is hindered by the presence of corrosion products, scale, grease, oil, etc. Cracks that have formed (internally or externally) prior to removing the hydrogen will not be repaired by the baking or stress relief operations.

k) Limit drill stem testing in sour environments to as brief a period as possible, using operating procedures that will minimize exposure to SSC conditions.

12.3 Drilling fluids containing oil or synthetics

12.3.1 Use of oil muds or synthetic-based fluids for drill stem protection

Corrosion and SSC can be minimized by the use of drilling fluids having oil or synthetics as the continuous phase. Corrosion does not occur if metal is completely enveloped and wetted by an oil or synthetic-based environment that is electrically nonconductive.

Oil systems used for drilling (oil, synthetic-based or invert emulsion muds) contain surfactants that stabilize water as emulsified droplets and cause preferential oil-wetting of the metal. Agents that cause corrosion in water (dissolved gases, dissolved salts, and acids) do not damage the oil-wet metal. Therefore, under drilling conditions that cause serious problems of corrosion damage, erosion-corrosion, or corrosion fatigue, drill stem life can be greatly extended by using an oil or synthetic-based mud.

12.3.2 Monitoring oil or synthetic-based muds for drill stem protection

An oil or synthetic-based mud should be properly prepared and maintained to protect drill stem from corrosion and SSC. Water will always be present in an oil mud, whether added intentionally, incorporated as a contaminant in the surface system, or from exposed drilled formations. Corrosion and SSC may occur if this water is allowed to become free and to wet the drill stem. Factors to be evaluated in monitoring an oil or synthetic-based mud include:

12.3.2.1 Electrical stability

This test measures the voltage required to cause current to flow between electrodes immersed in the oil mud (see API RP 13B-2 for details). The higher the voltage, the greater the stability of the emulsion, and the better the protection provided to the drill stem.

12.3.2.2 Alkalinity

The acidic dissolved gases (carbon dioxide and hydrogen sulphide) are harmful contaminants for most oil or synthetic-based muds. Monitoring the alkalinity of an oil or synthetic-based mud can indicate when acidic gases are being encountered so that corrective treatment can be instituted.

12.3.2.3 Drill pipe corrosion ring coupons (test rings)

Corrosion ring coupons placed in the drill pipe tool joint boxes are used to monitor the corrosion protection afforded by oil or synthetic-based muds. (see API RP 13B-2 for details). A properly functioning oil mud should show little or no visual evidence of corrosion on the corrosion coupon.

13 Drill stem failure mechanisms

13.1 Tension

13.1.1 Appearance

The fracture face of the ductile tension failure is normally angular (approximately 45º to the axis of the tool) and has not been affected by fatigue or corrosion. When the component is not of brittle material, a necked-down or “bottlenecked” section occurs adjacent to the fracture face.

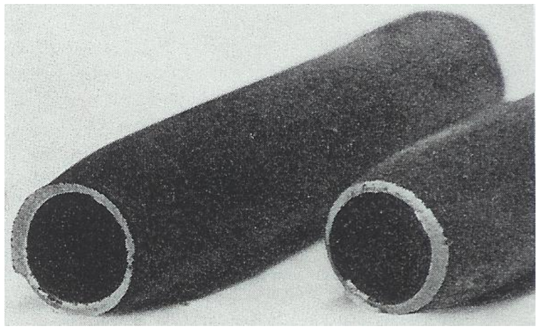


Figure 61 — Tension failure

13.1.2 Location

Occurs in thin cross sections of drill pipe tubes between the upsets. When drill pipe section has a uniform wall thickness the failure will occur high in the hole where the loads are the highest.

13.1.3 Cause

Tensile stresses exceed the tensile capacity of the component. You pulled too hard.

13.1.4 Prevention

Calibrate hook load devices. Inspect the drill string to ensure minimum wall thickness. Maintain and review grade markings and mill test certificates. Limit the maximum tensile load to less than the tensile capacity plus a safety factor.

13.2 Torque

13.2.1 Appearance

Pin fractures occur in a 45º angle initiating from a thread root and “jumping” the helix. Pin failures can range from stretched threads to an angular fracture of pin occurring within the first three threads from the pin shoulder. Box failures can appear as a belled or split box.

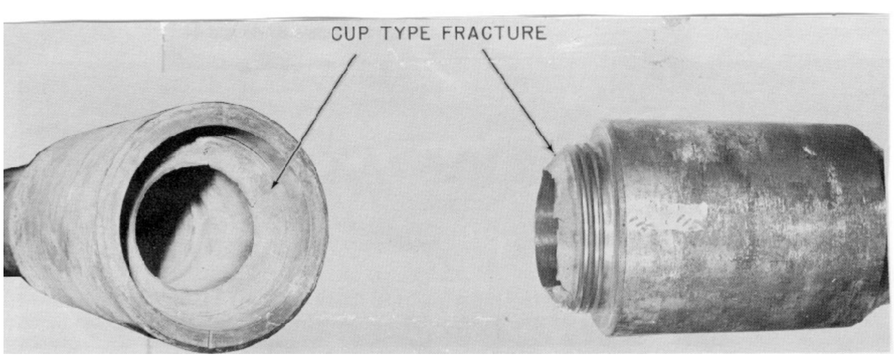


Figure 62 — Torsional failure

13.2.2 Location

By design, drill pipe tubes and drill collars bodies are significantly stronger in torque than the tool joints or connection to which they are attached. Drill pipe tubes and drill collar bodies should not failed solely due to torsional effects (see combine loading 6.5).

In single shouldered rotary pins, the failure generally occurs within the first three threads from the pin shoulder. In single shouldered rotary boxes, failure occurs in sealing shoulder of the box.

13.2.3 Cause

Excessive torque. Excessive make-up torque from improperly calibrated make-up devices can damage connections before they go downhole. Make-up torques that are too low may experience excessive operating torque, that may cause uncontrolled downhole make-up, that will exceed the torque capacity of the connection.

13.2.4 Prevention

Calibrate all torque devices (operating, make-up, break-out…) on a regular basis and set the torque limiting device on your top drive or kelly to 80% of the make-up torque on your drill pipe. Adjust surface parameters to prevent or reduce the severity of stick-slipping. Measure Pin ID and Box OD to determine correct make-up torque and torque capacity.

13.3 Combined Loading

13.3.1 Appearance

In drill pipe tube, the appearance of most combine loading failures is an angular fracture face following a helical path.

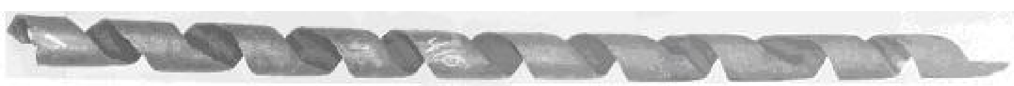


Figure 63 — Combined Tension/Torque Loading Failure

In a drill pipe pin connection, the appearance is similar to that of a pure torsional failure (see figure 64). Compressive forces are applied to box connection during make-up. Therefore boxes will very rarely fail in combine tension/torsion loading.

13.3.2 Location

Can occur in the tube of the drill pipe or in the pin of a connection. Boxes are held in compression and should not fail due to combined loading.

13.3.3 Cause

Most companies will apply torque while backreaming or jarring on stuck pipe. Excessive torsional load will significantly reduce the tensile capacity of the pipe and failure can occur. Over-torqued pin connections can reduce the pin-neck tensile capacities below operating hook loads.

13.3.4 Prevention

Refer to combine loading subclause 6.5 for combines load capacities. Make sure all torsional and tensile load devices are calibrated and functioning properly.

13.4 Burst

13.4.1 Appearance

Generally, a longitudinal split surrounded by bulged plastically deformed material.

13.4.2 Location

Generally occurs in the thin cross section of the component. In the drill pipe, it will occur in the thinnest weakest section of the tube.

13.4.3 Cause

Differential pressure. The internal pressure exceeds the external pressure enough to outwardly deform and rupture the body of the component.

13.4.4 Prevention

Do not exceed the burst pressure rating for the component. (See Tables A.3,A. 5 and A.7 (Tables B.3, B.5 and B.7))

13.5 Collapse

13.5.1 Appearance

“Pancaked” tube section in drill pipe or thin cross-sectioned tool. From a section view the component will look like a bow tie or a crescent moon.

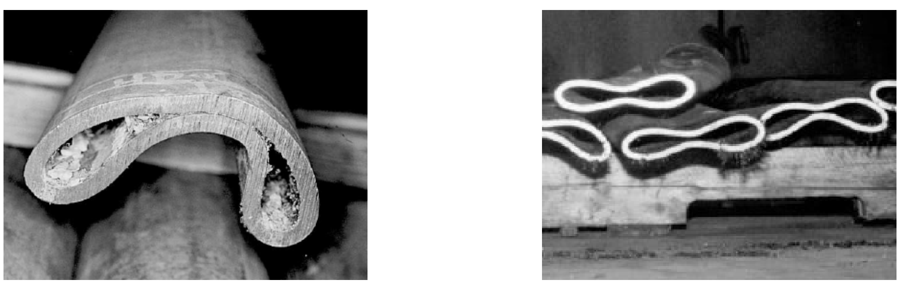


Figure 64 — Collapse failure

13.5.2 Location

Will generally be found low in the hole section or where differential pressures (high external and low internal) are extreme. Occurs in the thin cross section of the tube between the upsets.

13.5.3 Cause

The pressure differential across a thin cross section exceeds the collapse resistance of the component.

13.5.4 Prevention

Calibrate all pressure monitoring gauges. Monitor potential external pressures. Be aware of fluid level inside drill pipe. Do not allow the differential pressure to exceed the collapse rating plus a safety factor.

13.6 Fatigue

13.6.1 General

Fatigue is the #1 contributor to failures in the drilling industry. It is estimated that 75% to 85% of all failures are due to fatigue. Fatigue is cumulative, irreversible damage done by cyclic stress. Fatigue failures can occur at low stress levels. Generally fatigue will destroy enough of a cross section of a component to where the remaining cross section will fail due to mechanical overload.

13.6.2 Appearance

A single pure fatigue crack appears flat, planar, and transverse to the axis of the pipe. Generally at the origin of the fatigue crack, the texture of the fracture is smooth and may progressively get rougher further from the origin.

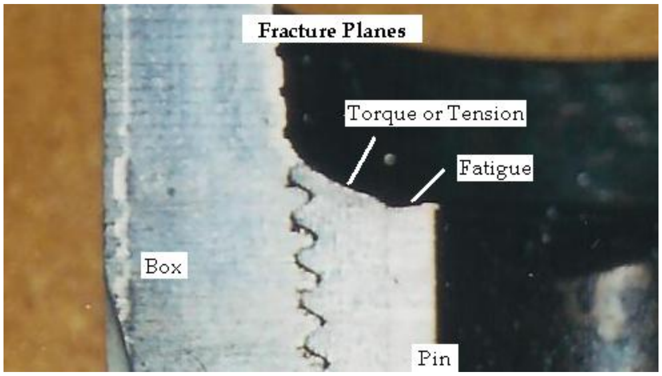


Figure 65 — Fatigue Failure cross section

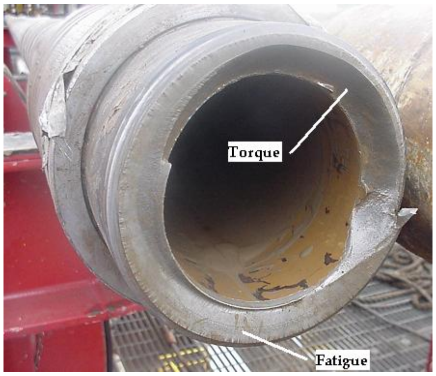


Figure 66 — Pin Fatigue Failure

High Load conditions and corrosive environments will accelerate fatigue crack propagation and cause a rougher (striated) fracture face. Striations indicate individual load applications and may not be visible to the necked eye.

If the propagation of the fracture is not applied in a continuous time frame “beach marks” can appear within the fracture face. A beach mark is a macroscopically visible mark or ridge caused by either a hardening of the crack tip or change in color due to a change in environments.

Both striations and beach marks expand from the origin of the fatigue crack often in a circular or semi-circular fashion. It resembles the ripple effect of a stone thrown into a pond.

Ratchet marks are created when different adjacent fatigue cracks propagate close enough for them to join under enough applied stress. The joining of fatigue cracks will cause a ridge where the cracks connect.

Fatigue fracture faces are generally accompanied by mechanical overload fracture faces. A fatigue fracture will propagate reducing the cross section of a component until the remaining cross section is unable to withstand the operating loads. The remaining cross section will fail due to mechanical overload.

13.6.3 Location

Fatigue failures occur in high stressed areas, generally near or in a stress concentrator. Fatigue attacks weaker sections of components such as thin cross sections adjacent to upsets, or in thread roots, slip cuts, pits, and gouges.

M_{IU} Transition (Drill Pipe Upset Area). In drill pipe, fatigue commonly occurs in the thin cross section of tube adjacent to upset area. Bending stresses are concentrated on the thin cross section next to the transition area. (See Figure 69)

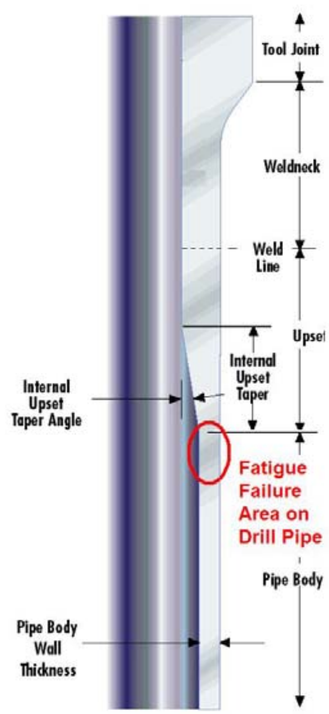


Figure 67 — Drill Pipe Upset Geometry Fatigue Location

M_{IU} refers to the length of the internal upset taper. API Specification 5 requires 7,6 cm (3 in) taper lengths for internal/external upset drill pipe to allow for smooth transition between tube and tool joint. Short or sharp transition areas magnify the stress concentration and accelerate fatigue.

Slip Cuts and Formation Gouges. Adjacent from the box connection M_{IU} transition area is approximately the same area where slip cuts are generated. They act as a stress concentrator also. Fatigue cracks may form in the root of the gouge. All slip cuts should be inspected for fatigue cracks and removed prior to utilization. If a fatigue crack exists, the component should be rejected. In the same fashion, some abrasive formations or debris in the hole can create transverse gouges in the drill string, which can act as stress concentrators. Sharper deeper gouges cause more stress concentration than rounder shallower gouges.

13.6.4 Thread Roots

In drill collars, fatigue generally occur in the first engaged thread root of the pin or box. (See Figure 70) Lack of proper stress relief features in BHA components accelerates the fatigue process. Unengaged threads in a BHA connection create a large stress riser in the weak part of the component. Properly cut stress relief features will eliminate these unnecessary threads.

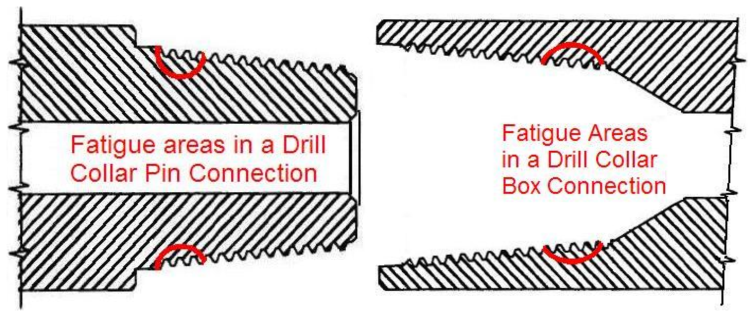


Figure 68 — BHA Connection Fatigue Locations

13.6.5 Corrosion Pit

Similar to gouges, corrosion pits can also act as a stress concentrator. The weaken surface at the root of the pit creates a good environment for fatigue to grow. In addition to creating a stress riser, corrosive environments have been known to accelerate fatigue crack propagation.

13.6.6 Cause

Cyclic stresses, such as bending, rotating, and vibration, cause fatigue. Corrosion accelerates fatigue. Some industries operate in benign environments where properly design equipment may not encounter fatigue. The oil industry does not operate in a benign environment.

13.6.7 Prevention

It is impossible to prevent fatigue. It is possible to reduce the risk and slow the accumulation of fatigue in a component. If you are able to slow the fatigue propagation, then it you may be able to detect fatigue cracks during inspection and before they fail downhole.

Monitor and reduce operating parameters such as RPM and tensile load. By reducing the tensile load you can reduce the magnitude of the fatigue stresses. You can reduce tensile load by reducing the number of excess drill collars or by using lighter drill pipe. By slowing the rotary speed you can reduce the fatigue cycle frequency. It may be necessary to utilize alternative means such as straight hole drilling motors to maintain an efficient bit rotary speed.

Do not buckle fragile components such as drill pipe and jars. Most BHA equipment is designed to be buckled and have thick cross sections to handle excessive bending loads. Components such as drill pipe and jars have thin cross sections and are accessible to rapid fatigue in buckled rotary applications.

Monitor the usage and inspection of different groups of drilling equipment. Increase inspection frequency when fatigue cracks are being detected during the inspection process. It may be necessary to down grade the application to reduce the liability of failures. (i.e. from a QR3-type well to a QR2-type well or QR2 type well to a QR1-type well). Utilize new strings on long deep-water wells and old strings on shallow quick land wells.

Reduce the effects of environment on fatigue life (see corrosion). Since corrosive fluids decrease fatigue life, monitor fluid coming out of the hole and treat with the appropriate additive or scavenger until the fluid coming out of the hole has acceptable properties. If possible, drill wells slightly over balanced to prevent native corrosive fluids from invading the well bore.

13.7 Washout

13.7.1 General

A wash out is not a hole washed out through the pipe, but is a crack (usually a fatgue crack) which has formed under mechanical stress and propagated through the wall, but which has been enlarged and rounded off at the edges by the eroding action of the mud fluid.

13.7.2 Appearance

Eroded pathway adjoining the ID and OD in the pipe.

13.7.3 Location

A wash out can be located anywhere a flaw is possible, but are most likely found where fatigue cracks are located. Slip/upset area on drill pipe and in the connection in BHA components are common areas for wash outs.

13.7.4 Cause

Generally cause by a flaw (most commonly a fatigue crack) that has propagated and opened a flow pathway from the ID to the OD of the drill string.

13.7.5 Prevention

A wash out is generally a product of another failure mechanism (most commonly fatigue). To prevent wash outs simply prevent the mechanism which creates them (see fatigue section 12.7).

13.8 Corrosion

13.8.1 General

13.8.1.1 Appearance

Corrosion as such is rarely the direct cause of drill string failures. It is however frequently a reason for equipment to be retired from service, since corrosion damage can accelerate several other kinds of failure.

Usually appears as a pit which is the origin of a fatigue failure.

13.8.1.2 Location

Can occur anywhere.

13.8.1.3 Cause

Improper control of mud – pH, oxygen, etc.

13.8.1.4 Prevention

To prevent or slow the effects of corrosion, coat the component to insulate it from the environment. Use corrosion inhibitors in your mud system.

13.8.2 Direct corrosion Failures

13.8.2.1 Seal Failures

The chief direct failure mode by corrosion is damage to sealing surfaces. The contact faces of rotary shouldered connections are well protected in service, but are vulnerable at surface if thread protectors are not properly used.

Float valve bores are subject to crevice corrosion, and drilling tools frequently contain seals that are subject to similar problems. For this reason, seals require regular disassembly for cleaning and inspection.

13.8.2.2 Penetration Failures

It is possible for a localised region of corrosion to penetrate completely through a drill string component, leading to a wash-out. This is however, quiterare

13.8.3 Failure Acceleration by Corrosion

The principal contribution of corrosion to drill string failure is by the acceleration of tensile and fatigue failure.

13.8.4 Accelerated Tensile Failures

Loss of cross section due to corrosion gives a direct reduction in tensile strength of drill string members, notably drill pipe. The corrosion is typically on the inside surface of the pipe, so it has much less effect on the torsional strength.

13.8.5 Accelerated Fatigue Failures

Corrosion in drilling fluids is rarely uniform. It commonly manifests as localised regions of pits, surrounded by unharmed surface. These regions of pitting have sharp roots, and therefore generate significant stress concentrations. These stress concentrations can initiate fatigue cracks, especially if they occur in areas where bending stresses are already high, such as thread roots, upsets, grooves and holes. This effect is purely geometric, and distinct from the effect of corrosive media in accelerating fatigue.

13.8.6 Accelerated Brittle Fracture

Pits can also serve as the initiation sites of stress corrosion cracks, although this is quite rare. Most materials used in the drill stem have sufficient toughness that brittle fracture will not be initiated by a corrosion pit before it penetrates the wall of the tube.

13.8.7 Accelerated Erosion

The mud velocity in Bottom Hole Assembly components can be high enough that erosion is a significant problem. The geometry changes created by corrosion pits can drastically increase the rate of localised erosion, leading to washout or occasionally tensile failure. If this happens, the initiation site will be removed by the erosion, so it is rare to positively identify the cause.

13.9 Sulphide stress cracking (SSC) and stress corrosion cracking (SCC)

13.9.1 General

SSC and SCC are the fracturing of the material through a combined action of corrosion and tensile forces. It is brittle fracture of a material that is normally ductile.

It is difficult to distinguish with certainty between SSC and SCC. It is necessary to consider the history of the component, crack origin, crack pattern, corrosion on the fracture surfaces, and microscopic features.

13.9.1.1 Appearance

The appearance of the fracture is rough throughout the fracture face. This is in contrast to a single fatigue failure where the origin appears smooth. It is orientated transverse to the axis of the stress. Usually appears as the origin of a fatigue failure. Sometimes there will be additional multiple cracks near the origin. Can also appear as a split tool joint boxes.

13.9.1.2 Location

Because it is a stress and environment related, fractures will originate in high stress concentration areas and generally from the OD of the component where the corrosion from the influx or formation fluids are higher.

13.9.1.3 Cause

Improper control of mud, material of too high a yield strength for conditions, excessive stresses.

13.9.1.4 Prevention

Limit the stress applied to a component (see Tension subclause 12.2) and/or limit the corrosive environment (see Corrosion subclause 12.8).

13.9.2 Sulphide Stress Cracking

Sulphide Stress Cracking (SSC) is possible with high strength steels in the presence of mud containing H_2S. Cracks will sometimes form due to residual stress in the steel. More often they will occur due to service stress and these may lead to complete failure. The fractures appear totally brittle, with little or no branching.and when failures occur it is quite common to find that an entire string of pipe is cracked.

H_2S exposure can lead to delayed failure, where the material becomes charged with hydrogen during one run, and then cracks on a subsequent run, where there is higher stress or lower temperature. It is therefore very important to track any exposure to H_2S.

The use of steels of the lowest possible strength will minimize the risk of SSC, but it is not practical to use steels of the very low hardness (22 HRC maximum) that are considered immune to SSC. The materials used to make non-magnetic drill collars are much less subject to SSC than the alloy steels used in the rest of the drill string.

13.9.3 Stress Corrosion Cracking

Stress Corrosion Cracking occurs in water-base mud, primarily on high strength alloy steels and stainless steels, given exposure to a tensile stress above threshold. It is not common in the medium-strength alloy steels used for most drill stem components.

Stress Corrosion Cracking can initiate without any stress concentration, but is aided by any stress concentration that may exist, such as corrosion pits. Conversely, SCC cracks may serve as the initiation sites for corrosionpits.

Stress corrosion cracks start at the exposed surface and are heavily branched. They are perpendicular to the stress that drives them. In stainless steels, it is quite easy for residual stresses from material fabrication to exceed the threshold, so that cracking can occur without any service stress. In this case the residual stresses are normally circumferential, and tensile on the inside surface, so the most common SCC cracks are longitudinal on the inside surface.

However, SCC can arise from service stresses, such as hoop stress in a box connection from excessive torque, or from service damage, such as localised heating.

High strength alloy steels are sometimes used for highly stressed components of drilling tools and mud-motors. These steels are sometimes subject to SCC.

SCC cracks are easily identified by their branched character, and frequently by their orientation, when it is not perpendicular to the axis of the string.

SCC can be mitigated by material selection, and by the use of compressive treatment, such as shot peening or cold rolling, on exposed surfaces. Stainless Steel non-magnetic drill collars are normally supplied with compressive treatment of the inside diameter.

13.10 Friction induced failures

13.10.1 General

Friction induced failures are failure mechanisms caused by overheating that is attributable to friction. In these cases, high load forces between the drill string and the wellbore while rotating the drill string can cause temperatures to meet or exceed normalizing temperatures resulting in drastically reducing mechanical properties such as yield strength, ultimate tensile strength, and surface hardness.

13.10.2 Heat checking

13.10.2.1 General

Tool joints rotated under high lateral force against the side of the well bore may be damaged as a result of heat checking. The heat generated at the surface of the tool joint by this friction can raise the temperature of the tool joint surface above critical levels (in excess of 1 050 °F) (approximately 550 °C). When the drilling fluid circulates past the tool joint, it quenches the heated surface, and quench cracks form parallel to the tool joint axis. Metallurgical examination of heat checked surfaces has revealed higher hardness levels as deep as 0,476 cm (0,188 in) below the tool joint surface.

13.10.2.2 Appearance

Longitudinal surface cracking on the OD surface of the component. This may not be visible to the naked eye but will normally appear during a magnetic particle inspection as shown in Figures 71 and 72.

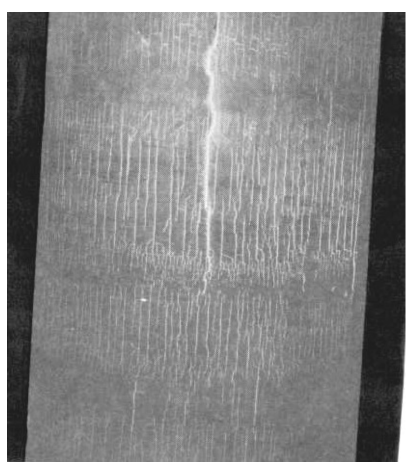


Figure 69 — Heat checking of box tool joint shown under black light

Figure 70 — Pin tool joint with heat checking revealed by dry magnetic particle inspection

13.10.2.3 Location

Usually occurs on the O.D. surface of drill pipe and HWDP tool joints, and rarely occurs to only one tool joint.

13.10.2.4 Cause

There can be many causes of heat checking but the most common cause is constant rotation of drill string in deviated well bores with no vertical movement of the drill string. The severity of the friction is directly dependent on the following factors:

— Well bore deviation or dogleg curvature severity

— Amount of tension in the drill string at these points

— Duration of time the drill string is rotated without any vertical movement

The worst case scenario in this regard is when doglegs exceeding 3° per 100 ft of curvature are located in the upper section of a deep well bore and there is a significant amount of time spent rotating off-bottom without vertically moving the drill string. The quench cracks shown in the figures above have been known to propagate further below the surface of box tool joints resulting in box washouts and drill string failures. This is caused primarily by hoop stresses in the box when the connection is made up. Since the quench cracks are perpendicular to the hoop stresses in the box, they can continue to grow and propagate deeper into the box until the cracks have grown completely through the box.

13.10.2.5 Prevention

As a general rule, minimizing doglegs in excess of 3° per 100 ft of well bore curvature in wells over 10 000 ft TVD will limit the lateral force between the tool joint and the well bore to acceptable levels. In addition, maintaining continuous vertical movement of the drill string while rotating off-bottom is highly recommended.

13.10.3 Eccentric Wear

13.10.3.1 General

Eccentric wear is usually incurred to drill pipe and HWDP tool joints. It is manifested by excessive wear to only one side of the tool joint as depicted in the figure below. Eccentric wear to the tool joints is the most common result as shown in the figure below.

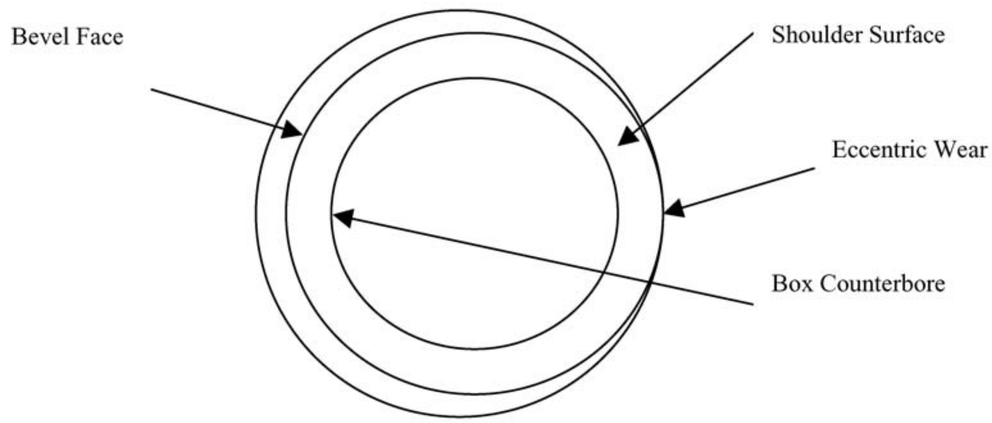


Figure 71 — Eccentric wear of box connection end view

13.10.3.2 Appearance

Confirmation of eccentric wear can easily be determined via visual inspection of the bevel on the tool joint shoulder. If the width of the bevel is thin on one side of the tool joint, it is a good indication of eccentric wear as shown in Figure 73.

13.10.3.3 Location

Eccentric wear normally occurs to the OD of the drill pipe and HWDP tool joints.

13.10.3.4 Cause

Eccentric wear is usually caused by "precession" in the well bore. Precession occurs when pipe is buckled and rotated into the shape of a helix that is contained within the confines of the well bore. When the helix revolves in the hole at the same RPM as the drill string itself, the part of the pipe that is in contact with the well bore remains in contact with it. The root cause is drilling with the drill pipe and/or HWDP in compression. This is most likely to occur in highly deviated well bores where the total "drag" (the additional hoisting force required to overcome friction between the drill string and well bore) is high. In some cases, the use of drill collars or additional HWDP in the upper section of the drill string is necessary to create the additional weight required to push the drill string down into the well bore. Eccentric wear can also be caused by sliding and not rotating the drill string in severely inclined or horizontal well bores.

13.10.3.5 Prevention

There are several potential remedies for eccentric wear caused by drilling with the drill pipe in compression as listed below:

— Revise the BHA design to increase or decrease BHA weight. The greater the inclination of the hole angle towards a horizontal attitude, the more impact a lighter BHA will have to reduce the friction between the BHA and the well bore

— Utilize a mud motor or turbine to reduce, but not eliminate rotation of the drill string

— Alter or replace the drilling fluid to increase lubricity

— Special tools can be used in the BHA that will induce pulsation forces on the drill string to reduce the friction between the well bore and the BHA

13.10.4 Overheat Pull Failures

13.10.4.1 General

Overheat pull failures occur when friction increases the heat of the drill string component to critical levels that in turn, cause the yield and ultimate tensile strength to be severely reduced to the extent that the drill string component is not capable of supporting the applied tensile load.

13.10.4.2 Appearance

In overheat pull failures, the heat is high enough to decrease the yield strength of the material, and if sufficient upward force is applied, simply fails in tension. The appearance resembles that of a simple tension failure (see subclause 11.2).

13.10.4.3 Location

This commonly occurs to the area just above the pin tong space on Heavy-Weight Drill Pipe, but it can also happen to drill pipe in the same manner. The photo below (Figure 74) shows a failure to a joint of 5" HWDP that failed just above the pin tool joint.

Figure 72 — Overheat pull failure of HWDP above the pin tool joint

The next photo below (Figure 75) shows the same type of failure that occurred just above the center wear pad on 5 in HWDP. The globs of rough-looking metal close to failure area are the remnants of tungsten-carbide hard-banding materials that were applied to the center wear pad.

Figure 73 — Overheat pull failure of HWDP above the center wear pad

The next example of an overheat pull failure (Figure 76, below) shows the failure of the pipe body of a joint of 5-1/2” drill pipe as a result of rotating the drill string with a closed annular BOP.

Figure 74 — Failure of Drill Pipe Body Rotated against Annular BOP Element

13.10.4.4 Cause

Overheat pull failures most commonly occur in sloughing formations when the hole caves-in on the drill string while it is being rotated. Attempts to prevent sticking the drill string by hoisting it upwards against the formation and rotating it causes a tremendous amount of heat to be generated which resulted in the loss of the HWDP to support the tensile load. In both examples above, the cross-section is tapered down to the failure point due to the material being stretched before it failed in tension. Metallographic examination of both failure areas revealed surface hardness of the material at the failure point is high (50 to 60 Rockwell C). This is clearly over 20 Rockwell C points in excess of the maximum hardness the HWDP pipe body is typically manufactured to. Hardness readings taken at certain intervals going away from the failure point were found to gradually reduce until normal readings are obtained several inches away from the failure point. The hardness increase at the failure point is caused by the rapid cooling of the overheated area by the drilling fluid as it flows past it when the HWDP parted. It should be noted that analysis of other failures of this type have not revealed the same pattern of hardness readings. It is assumed the reason for this was due to the fact that at the time of failure, no drilling fluids were being circulated to cause the quenching effect.

13.10.4.5 Prevention

These types of failures can be prevented by:

— Resisting the temptation of pulling up against a sloughing formation and rotating the drill string for more than a few seconds at a time and maintain circulation.

— Prevent sloughing of formations by optimizing well bore hydraulics and mud properties.

13.10.5 High rates of tool joint wear

As explained in subclause 6.1 drill strings in high tension inside curved well bores can cause high side-loading on drill pipe tool joints, and sometimes, even the drill pipe tube body. In some formations, this can cause rapid O.D. loss to the tool joint unless it is protected with raised hardbanding. One mode of failure is called “heat checking” which is covered in subclause 12.10.2. Another mode of failure is the loss of torsional strength of the tool joint due to excessive wear. The photograph below (Figure 77) shows unusually heavy wear to a box tool joint such that the O.D. has worn down into the thread roots of the connection. It was extremely fortunate in this case that the rig was able to pull the drill string out of the hole without a failure.

Figure 75 — Excessive tool joint OD wear

14 Special service problems

14.1 Wear of tool joints and drill pipe

14.1.1 When drill pipe in a dogleg is in tension it is pulled to the inside of the bend with substantial force. The lateral force will increase the wear of the pipe and tool joints. When abrasion is a problem it is desirable to limit the amount of lateral force to less than about 907,2 kg (2 000 lb) on the tool joints by controlling the rate of change of hole angle. Values either smaller or greater than 907,2 kg (2 000 lb) might be in order, depending on formation at the dogleg. Figure 38 shows curves for 453,6 kg (1 000 lb), 907,2 kg (2 000 lb), or 1360,8 kg (3 000 lb) lateral force on the tool joints; points to the left of these curves will have less lateral force, and points to the right more lateral force on the tool joints. Figures 41, 42, 43, and 44, developed by Lubinski, show lateral force curves for both tool joints and drill pipe for three popular pipe sizes. The first three figures are for three pipe sizes of Range 2 drill pipe. Figure 44 is for 5-in, 19,5 lb per ft, Range 3 drill pipe.

14.1.2 For conditions represented by points located to the left of Curve No. 1, such as Point A in Figure 41, only tool joints and not drill pipe between tool joints contact the wall of the hole. This should not be construed to mean the drill pipe body does not wear at all, as Figure 41 is for a gradual and not for an abrupt dogleg. In an abrupt dogleg, drill pipe does contact the wall of the hole half way between tool joints, and the pipe body is subjected to wear. This lasts until the dogleg is rounded off and becomes gradual.

14.1.3 For conditions represented by points located on Curve No. 1, theoretically the drill pipe contacts the wall of the hole with zero force at the midpoint between tool joints.

14.1.4 For conditions represented by points located between Curve No. 1 and Curve No. 2, theoretically the drill pipe still contacts the wall of the hole at midpoint only, but with a force which is not equal to zero. This force increases from Curve No. 1 toward Curve No. 2. Practically, of course, the contact between the drill pipe and the wall of the hole will be along a short length located near the midpoint of the joint.

14.1.5 For conditions represented by points located to the right of Curve No. 2, theoretically the drill pipe contacts the wall of the hole not at one point, but along an arc with increasing length to the right of Curve No. 2.

14.1.6 On each of the Figures 41, 42, 43, and 44, there are in addition to curves No. 1 and No. 2, two families of curves: one for the force on tool joint, and the other for the force on drill pipe body. As an example, consider Figure 41; Point B indicates that if the buoyant weight suspended below the dogleg is 77110,7 kg (170 000 lb), and if dogleg severity (hole curvature) is 10,1 °/100 ft, then the force on tool joint is 2721,6 kg (6 000 lb), and the force on drill pipe body is 1360,8 kg (3 000 lb).

14.1.7 As explained in subclause 6.1 drill strings in high tension inside curved well bores can cause high side-loading on drill pipe tool joints, and sometimes, even the drill pipe tube body. In some formations, this can cause rapid O.D. loss to the tool joint unless it is protected with raised hardbanding. One mode of failure is called “heat checking” which is covered in subclause 12.10.2. Another mode of failure is the loss of torsional strength of the tool joint due to excessive wear. The photograph below shows unusually heavy wear to a box tool joint such that the O.D. has worn down into the thread roots of the connection. It was extremely fortunate in this case that the rig was able to pull the drill string out of the hole without a failure.

Figure 76 — Excessive wear to a box tool joint

14.2 Pulling on stuck pipe

14.2.1 General

It is normally not considered good practice to pull on stuck drill pipe beyond 90 ° of the minimum yield strength for the size, grade, weight, and classification of the pipe in use (see Tables A.2, A.4, and A.6 (Tables B.2, B.4 and B.6). For example, assuming a string of 5-in., 19,5 lb/ft Grade E drill pipe is stuck, the following approximate values for maximum hook load would apply:

Premium Class: 311 535 lbs

Class 2: 270 432 lbs

The stretch in the drill pipe due to its own weight suspended in a fluid should be considered when working with drill pipe and the proper formulas to use for stretch when free or stuck should be used.

14.2.2 Example I

See E.6 for derivation.

Determine the stretch in a 10 000 ft string of drill pipe freely suspended in 10 lb/gal drilling fluid.

$$Formula? \tag{155}$$

= 53,03 in,

where

L1 is the length of free drill pipe, expressed in feet;

Wg is the weight of drilling fluid, expressed in lb/gal;

e is the stretch, expressed in inches.

14.2.3 Example II

See E.4 for derivation.

Determine the free length in a 10 000 ft string of 4,5 in OD 16,60 lb/ft drill pipe which is stuck, and which stretches 49 in due to a differential pull of 80 000 lb.

$$Formula? \tag{156}$$

= 7476 ft,

where

L1 is the length of free drill pipe, expressed in feet;

e is the differential stretch, expressed in inches;

Wdp is the weight of drill pipe, expressed in pounds per foot;

P_{DIFF} is the differential pull, expressed in pounds.

14.3 Jarring

It is common practice during fishing, testing, coring and other operations to run rotary jars to aid in freeing stuck assemblies. Normally, the jars are run below several drill collars which act to concentrate the blow at the fish. It is necessary to take the proper stretch to produce the required blow. The momentum of the moving mass of drill collars and stretched drill pipe returning to normal causes the blow after the jar hammer is tripped. A hammer force of three to four times the excess of pull over pipe weight is possible depending on type and size of pipe, number (weight) of drill collars, drag, jar travel, etc. This force may be large enough to damage the stuck drill pipe and should be considered when jarring operations are planned.

14.4 Torque in washover operations

Although little data are available on torque loads during washover operations, they are significant. Friction and drag on the wash pipe cause considerable increases in torque on the tool joints and drill pipe, and should be considered when pipe is to be used in this type service. This is particularly true in directionally drilled wells and deep straight holes with small tolerances. (See 12.6.)

14.5 Allowable hookload and torque combinations

Allowable hook loads and torque combinations for fishing on stuck drill strings should be determined by the equations and curves provided in subclause 6.5.

14.6 Welding on down hole drilling tools

14.6.1 Usually the materials used in the manufacture of down hole drilling equipment (tool joints, drill collars, stabilizers and subs) are AISI-4135, 4137, 4140, or 4145 steels.

14.6.2 These are alloy steels and are normally in the heat treated state, these materials are not wieldable unless proper procedures are used to prevent cracking and to recondition the sections where welding has been performed.

14.6.3 It should be emphasized that areas welded can only be reconditioned and cannot be restored to their original state free of metallurgical change unless a complete heat treatment is performed after welding, which cannot be done in the field.

14.7 Landing string considerations

The purpose of this subclause is to discuss landing string tubulars which are defined as tubulars with rotary shoulder connections with wall thickness greater than those offered in this part of ISO 10407. These tubulars are used to lower casing strings in offshore applications to the subsea wellhead. They may be produced as either welded assemblies or manufactured integrally from steel bars. These tubulars may also be used for normal drilling or completion operations. However, these are beyond the scope of this part of ISO 10407. This subclause will not discuss other parts of the landing string such as casing hanger, subs, top drive, and valves. All components of the landing string both pipe body and connection should be analyzed for tensile capacity and its respective safety factor.

It is recommended that landing string tubulars be specified by nominal wall thickness. This practice will help eliminate confusion between all parties that use and manufacture this equipment. For example, a common landing string tubular currently in use is 5,5” S-135 0,750” wall.

The first step in determining the correct landing string tubular is to determine the largest casing load required, including the landing string tubular with an appropriate safety factor. A minimum Safety Factor of 1,1 (pin tensile yield capacity/pipe yield capacity) should be applied when selecting the tool joint connection to be used with the selected pipe. A weld area S.F. of 1,1 should also be used (versus 1,0 requirement in API Specification 7) The cross-sectional area of the pipe determines the lifting capacity of a tubular along with the handling tools. Using the minimum wall standard that is utilized on drill pipe to determine the useful life of landing string tubulars will always underestimate cross-sectional area. Consequently, landing string tubulars should be selected and inspected for the estimated load as calculated from the cross-sectional area.

The use of conventional rig handling equipment is a great benefit for landing string tubulars. However, as the strings become heavier, slip capacity (slip crushing) may become the limiting factor. As the limits for landing string tubulars increase, the handling system (elevators and slips) capacity should be verified. The issue of slip crushing is one that is currently being investigated.

Inspection equipment used to determine cross-sectional area for landing string tubular classification should be a direct indicating instrument that the operator can demonstrate to be within 2% accuracy along the length of the tube by use of a tube section approximately the same as the tube being inspected. The tube section should have not less than a 7% variation in cross-sectional area. Cross-sectional area reduction should be representative of the wall loss substantially from one side. The minimum cross-sectional area throughout the length of the tube should be used to calculate lifting capacity of the landing string tubular.

14.8 Dynamic loading of drill pipe at the slips

NOTE For quantitative results, see Reference 15, Appendix C.

14.8.1 When running a stand of drill pipe into or out of the hole, the pipe is subjected not to its static weight, but to a dynamic load.

14.8.2 The dynamic load oscillates between values which are greater and smaller than the static load (the greater values may exceed the yield), which results in fatigue, i.e., shortening of pipe life.

14.8.3 Dynamic loading can exceed yield in long strings, such as 3 048 m (10 000 ft).

14.8.4 Dynamic loading increases with the length of drill collar string.

14.8.5 In the event the smallest value of the dynamic load tries to become negative, the pipe is kicked off the slips, and the string can be dropped into the hole.

14.8.6 The likelihood of dynamic loading resulting in a jumpoff (kicking of the slips) increases as the drill pipe string becomes shorter and the collar string becomes longer.

14.8.7 For a long drill pipe string, such as 3 048 m (10 000 ft), a jump-off is possible only if drill pipe, after having been pulled from the slips, is dropped at a very high velocity, such as 16 ft/sec.

14.8.8 Dynamic phenomena are severe only when damping is small, which can be the case in exceptional holes, in which there are no doglegs, the deviation is small, the cross-sectional area of the annulus is large, and the mud viscosity and weight are low.

14.8.9 In case of small damping, the running time of a stand of drill pipe should not be less than 15 s.

14.9 Factors considering longevity of drill stem members

14.9.1 General factors affecting the life extension of the drill string

Factors that affect the useful life of drill string components are driven by the forces exerted on the drill string while drilling, as well as corrosion. Forces induced in the drill string include, but are not limited to tensile, torsion, internal pressure, collapse pressure, bending, and multi-directional drill string vibrations. For the User/Owner to extend the useful life of this asset actions should be taken to reduce stress levels as much as possible. Unless wells are drilled on a turnkey or footage basis, drilling contractors and rental companies (which make up most of the drill pipe Owners/Users) do not typically have control over the drilling conditions to which their drill string components are exposed. As such, remedial actions that are available to the Owner/User to extend the life-cycle of drill string components are limited to:

— Enhance drill string component purchase specifications to:

— Add material where the stresses, erosion, and corrosion are anticipated to be the highest

— Ensure stress risers are not created

— Recognize when drill string vibrations or other adverse drilling conditions are encountered while drilling, and take corrective actions within acceptable and/or established operating parameters

— Take preventive actions to reduce or eliminate corrosion to the drill string throughout its useful life

14.9.2 Factors affecting the life-cycle of drill collars

14.9.2.1 The effect of BSR ratios on the life-cycle of drill collar connections

In almost every instance, drill collars are in compression while drilling as a part of the BHA (Bottom-hole Assembly). By design, drill collars are stiff members when compared to drill pipe. When drill collars are placed in compression to the point where they buckle within the well bore, most of the bending takes place in the weakest part of the drill collar, which is the connection. Due to the fact that the box connection cross-sectional area is nearer to the O.D. of the drill collar than the pin cross-sectional area, the box is more affected by bending as compared to the pin. Hence the need to run a “balanced” connection that has a Bending Strength Ratio (BSR) within the range specified in subclause 5.5.3, 5.8.4, of this standard. Therefore, with all other contributing factors notwithstanding, life extension of drill collars can be achieved by maximizing the O.D. of a drill collar within the optimum BSR range specified. In addition, greater O.D. dimensions provide a greater wear allowance before the BSR is reduced below the recommended BSR range. In some instances where the formation is extremely abrasive, it may be necessary to apply hard-banding to drill collars to reduce the rate of wear to the O.D. in order to extend its useful life. It is important to note that the recommended BSR ranges specified in this standard are provided as broad guidelines. Smaller BSR values can be successfully applied to smaller diameter drill collars because they are not as stiff as larger diameter drill collars, and less bending takes place in the connection. By the same token, large diameter drill collars may require a BSR value that is higher within the ranges recommended.

14.9.2.2 The effect of stress-relief features and cold-rolling thread root on the life cycle of drill collar connections

The useful life of drill collar connections can be further enhanced by employing the use of stress-relief features specified in subclause 5.8.5 of this standard (Note: BSR guidelines do not account for any benefits that can be realized by cold-rolling the threads or employing the use of the stress-relief features provided in this standard). These features reduce the stress levels in the thread root of the last engaged thread of the pin and box when bending takes place in the connection. In addition, by cold-rolling the surface of the thread roots of the connections, fatigue life can be further extended.

14.9.2.3 The effect of poor handling practices on the life-cycle of drill collars

The useful life of drill collar connections can also be shortened by excessive handling damage, corrosion, and other types of deterioration that would cause them to be re-cut too frequently. Each time a connection is re-cut, the amount of available box tong space and fishneck length is reduced, along with the overall length of drill collar. Therefore, taking care to ensure the minimum amount of material is removed during a re-cut is important to maximize drill collar life-cycles. In cases where handling damage or minor pitting or galling exists, only a small amount of connection length should be sacrificed during a re-cut. This is sometimes referred to as a “chase-and-face”, or “minor” re-cut. When stress-relief features are employed on a drill collar connection that is to be re-cut, the length of the pin stress-relief feature may be extended to no more than 1,25 in (31,75 mm). If the resultant length of the pin stress relief groove is greater than this dimension, then a “full” re-cut should be performed to ensure stress-relief feature dimensions for the pin are maintained. The same care should be taken when re-cutting a box connection with stress-relief features. However, the cylinder bore-back length cannot be compromised. In cases where fatigue cracks are discovered in the thread root of the last-engaged thread of a pin or box, it is important to cut the connection off at the location where the crack is located. Depending on the type of connection employed, the location of the last-engaged thread is approximately 0,625 to 0,75 in (approximately16 to 22 mm) from the shoulder surface. Cutting the connection off at this location removes the fatigued area away from the new location of the last-engaged thread. This is commonly referred to as a “full” re-cut. In summary, for severe drilling conditions where the maximum life-cycle should be achieved for the connections, it is very important to ensure that the dimensional tolerances of the stress-relief features of both the pin and box are maintained, that the thread roots are cold-rolled after any machining, and anti-galling treatment is applied to the threads and shoulders of drill collar connections after machining. Handling damage can be prevented by the following:

— Control the descent of the blocks to allow the floor hands to swing the elevators out of the way before they contact the box shoulder in preparation for latching the elevators

— Ensure the shoulders are visible by the Driller during make-up and break-out to prevent the tong dies from overlapping the shoulders

— Ensure thread protectors are in good condition and make them up firmly against the connection shoulder to prevent moisture from entering in between the protector and the connection while racked in storage

— Ensure lift caps and subs are in good condition so that they do not damage the threads and shoulders when they are made-up to the connection

— Use a good quality thread compound that has a friction factor of 1,0 so there is no confusion as to what the make-up torque should be

— If a connection is made-up properly, no connection damage exists, and the connections are within API tolerances, then the break-out torque should not be greater than 90% of the make-up torque. If the break-out torque is higher than this, stop and look for galling damage to the connections, and cull them out of the string if repairs are required.

— Implement proper cleaning and doping procedures while tripping that are consistent between all crews

— Rotate connection breaks on every trip

— Use stabbing guides while running in the hole (RIH) whenever possible to protect damage to threads and shoulders while stabbing

— Look for signs of impact against the shoulder bevel that can deform the shoulder surface. File the deformed area down flush with the shoulder surface before attempting to make-up the connection

— Ensure that the instrumentation and gauges on the rig floor that are used to indicate make-up and break-out torque are calibrated at regular intervals to establish high reliability

— Do not attempt to reface the entire shoulder surface of a connection with manual abrasive tools on the rig. If a shoulder needs to be refaced, always use a qualified third party

— Don’t hook the ends of a sling into the connections. Either hitch the slings around the body of the drill collar near each end or shackle the slings to the bails on the cast steel lift caps that are made-up firmly to each connection to handle drill collars in and out of the V-door

— Ensure that the bore of any zip-lift elevator is suited for the O.D. of the elevator recess of the collar to which the elevators are to be engaged. Elevator bores for new elevators are specified in API Specification 8C, and drill collar elevator recess O.D. dimensions are specified in API Specification 7. As a rule-of-thumb, the difference between the bore of the elevator and the O.D. of the elevator recess is approximately 6,3 mm to 7,9 mm (approximately 0,25 in to 0,31 in)

14.9.2.4 The effect of corrosion on the life-cycle of drill collar connections

Most of the corrosion to drill collar connections occurs while drill collars are removed from the well bore. This is when the connection has the greatest exposure to oxygen. Therefore, it is important to clean and re-dope the connections before drill collars are racked. This applies to when drill collars are racked back in the derrick during round trips, as well as after inspections and repairs are performed. Because thread compounds in general are not designed for long-term corrosion protection, it is important to use preservative compounds on the connections when drill collars are stored for periods exceeding 6 months. In addition, it is extremely important to use undamaged thread protectors, and to make them up firmly against the shoulder to prevent any moisture from entering in-between the connection and the thread protector while the drill collars are racked.

14.9.3 Factors affecting the life-cycle of drill pipe

14.9.3.1 The effect of upset geometry and pipe body wall thickness on the life-cycle of drill pipe

Because the drill pipe body is more flexible than the tool joint, most of the bending force concentrates in the pipe body section adjacent to the tool joint. This also happens to be where the internal and/or external upsets are located. Therefore, stress concentrations will occur in this area if the configuration of the upset has an abrupt change in cross-sectional area. The life cycle of drill pipe can therefore be enhanced if purchase specifications for new drill pipe require a minimum length of approx. 100 mm (4 in), and a minimum transition radius of approx. 200 mm (8 in) as shown in Figure 79.

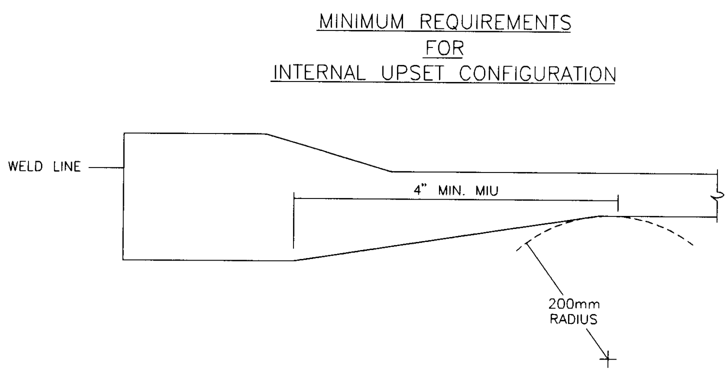


Figure 77 — Minimum requirements for internal upset configuration

In addition, API Spec 5D provides a tolerance of 12,5% of the nominal wall thickness of the pipe body when it is rolled at the mill. This means that 5”, 19,50 lbs./ft. drill pipe with a nominal wall thickness of 9,195 mm (0,362 in) may be rolled to a wall thickness of 8,05 mm (0,317 in), and still meet API specifications. This leaves a margin of only 0,686 mm (0,027 in) of wall thickness remaining before the tube is re-classified as API Class 2 condition 7,366 mm (0,290 in). However, it is possible to order drill pipe that is manufactured to tighter tolerances and thus increase this margin. As such, all Owners/Users can extend the life cycle of drill pipe by requiring drill pipe body wall thickness to meet or exceed 95% of the specified nominal wall thickness when new pipe is purchased.

14.9.3.2 The effect of internal plastic coating on the life-cycle of the drill pipe body

The useful life of the pipe body can be extended further by applying and maintaining an internal coating onto the I.D. surface of the drill pipe after it is manufactured, and then again at some point in time when the coating deteriorates. The average lifetime of these coatings is approximately 4 years. These coatings protect the I.D. surface of the pipe body from corrosion that would otherwise reduce its wall thickness over time. Furthermore, internal coatings will prevent the formation of localized corrosion pits where high stress concentrations can lead to premature fatigue failures. Several types of coatings are available to choose from.

14.9.3.3 The effect of longer tool joints and hard-banding in extending the life-cycle of the drill pipe

14.9.3.3.1 General

Extending the useful life of the drill pipe body through the actions specified above in subclause 13.9.3.2 drives the necessity for extending the life of the tool joints also. Tool joint life is mainly a function of two variables:

— The number and frequency of re-cuts that may be performed to the connection

— The rate of wear to the O.D. of the tool joint

14.9.3.3.2 The effect of longer tool joints on the life-cycle of the drill pipe

The number and frequency of re-cuts that may be performed to a drill pipe connection wil greatly determine the life-cycle of the drill pipe. Generally speaking, this is determined by the minimum tong space required to efficiently make-up the connection with standard tongs. Since the dies in standard tongs are approximately 139,7 mm (5,5 in) in length, tong space lengths that are approaching this length are usually deemed to be too short for further use. All other factors notwithstanding, the life-cycle of tool joints can be extended by increasing the length of the tool joint when drill pipe is purchased. As a general rule, ordering tool joints that are approx. 100 mm (4 in) longer than standard is sufficient for extending tool joint life to equal the life-cycle of the pipe body manufactured and maintained in accordance with the recommendations in subclause 13.9.3.2 above.

14.9.3.3.3 The effect of tool joint hard-banding on the life-cycle of the drill pipe

Drill pipe life-cycles can also be shortened by O.D. wear to the box. This can be prevented by applying and maintaining “casing-friendly” hard-banding that is raised approximately 2,38 mm (0,09 in) above the surface of the tool joint. This can be applied to the O.D. of the box and pin. A total length of approximately 114 to 127 mm (4,5 to 5 in) of the raised, hard-banded surface is recommended for each pin/box assembly. “Casing-friendly” hard-banding materials are much less abrasive than conventional spherical tungsten carbide hard-banding materials, and have been field-proven to minimize wear to casing and marine riser. There are a number of these products to choose from. These types of hard-banding products are discussed in more detail in subclause 13.15 of this standard. While the application of raised hard-banding does encroach on usable tong space, tool joints that are made approx. 200 mm (4 in) longer than standard will provide sufficient tong space until such time as the tool joint is shortened due to re-cuts. At that time, the Owner/User may elect to cease applying raised hard-banding, and allow the tool joint O.D. to wear. By the time the tool joint does wear down to the minimum dimension, it is most likely that the pipe body will be close to rejection as well, thus providing the maximum life-cycle of the pipe.

14.9.3.4 Other factors affecting the life-cycle of drill pipe

14.9.3.4.1 General

After drill pipe is manufactured and placed into service, its useful life can be extended by the Owner/User through proper handling, corrosion prevention, and minimizing the forces exerted onto the drill pipe while drilling. The same precautions that are recommended for drill collars for preventing handling damage to and corrosion of the connections on drill collars applies to drill pipe. However, additional handling precautions for drill pipe are required for the tube section. These precautions are focused on tong operations and setting the slips. The”dos and don’ts” of tong operations are provided as follows.

14.9.3.4.2 Bending pipe in the slips

This can occur when only one tong is used. Without a back-up tong engaged, there is no force to counter the bending moment on the pipe in the slip area, especially when the connection is unusually high above the rotary bushing, or when connections are difficult to break-out while pulling out of the hole (POOH). Bending the pipe against the slips also forces the slip dies to penetrate deeper into the O.D. of the tube. In some cases, depending on the nominal wall thickness of the tube section, heavy slip die marks can exceed 10% of the tube wall thickness, causing the pipe to be downgraded during the next inspection. Continued operations with this pipe can also cause a washout to occur due to fatigue cracks that can be formed in the sharp bottom of a slip die mark. In addition, the bend in the pipe that is formed when this occurs is very abrupt and is therefore very difficult to straighten. Even if the pipe is straightened successfully, the force required results in yielding the material yet a second time; thus contributing further to the cause of premature fatigue failures. Preventive actions include, but are not limited to using two tongs at all times, ensuring the connection is no more than approx. 913 mm (3 ft) above the rotary bushing, or employing the use of power tongs.

14.9.3.4.3 Slip Crushing

Setting the slips before the downward travel of the pipe is stopped can result in crushing and “necking” the tube section, severely reducing its tensile capacity. Even if the pipe is not crushed, heavy slip die marks can lead to fatigue failures as discussed above. In addition, when hook loads approach the tensile capacity of the pipe in use, all drill pipe made to API specifications will begin to crush in the slips at hook loads that are less than the pipe is rated to handle when it is engaged in the elevators. The formula used to determine the hook load that the pipe in use will begin to crush is provided in subclause 6.1.1.2 of this standard.

Preventive actions include precautions to ensure the pipe is at rest before setting the slips, using power slips to minimize crew fatigue that may lead to dropping the slips inadvertently, and ensuring that maximum hook loads do not exceed those determined by the formula provided above. In addition, it is important to determine the extent of wear to the slip segments and drilling bowls that may result in local crushing. The procedure used for several years with success is called the “slip paper test” as described below:

14.9.3.5 Slip paper test

1) Hook load should be 100 000 lb to insure correct slip die insert marking on pipe.

2) Ensure the section of pipe to be tested is free of previous insert marks and is clean.

3) Wrap waterproof paper around cleaned section of pipe and tape seam.

4) Place rotary slips around wrapped section of pipe and lower pipe and slips with normal setting speed.

5) After the slips have been set and full hook load has been transferred to the pipe, hoist the drill string up, and hold slip together and raise pipe. Remove slips carefully from the pie when they clear the bushings.

6) Remove paper carefully from the pipe.

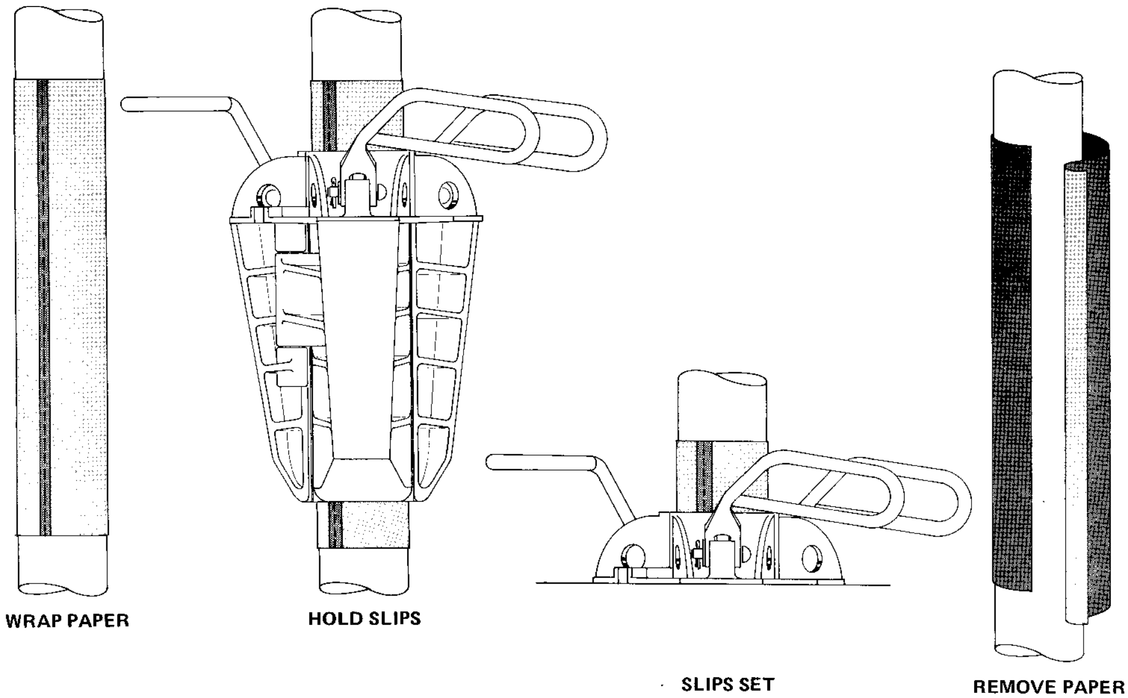


Figure 78 — Slip paper test

7) If full contact is evident between the slip dies and the pipe body O.D., it indicates that little wear has taken place and no components are in need of replacement

8) If slip die insert contact is showing only on the on upper section of the impression made on the paper, it indicates that either one or more of the following components are worn:

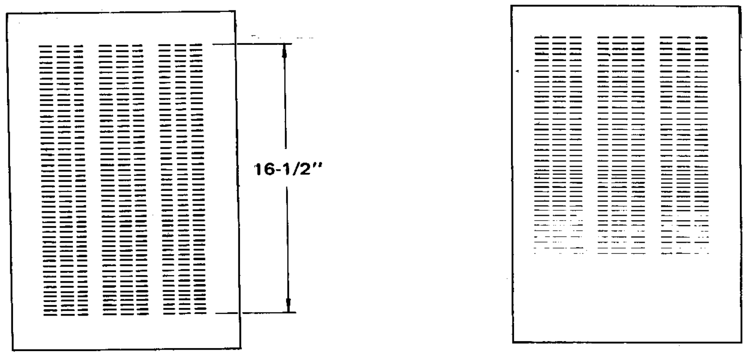
    i) Slip Segments

    ii) Drilling Bowls

    iii) Master Bushing

9) Repeat test with new slips. If re-testing shows insert contact on upper section of the impression, replace the drilling bowls with a new set and re-test.

10) If there is still uneven contact between the pipe and the slips after replacing the slips and the drilling bowls, check the inside diameters of the master bushing to verify they are within tolerance. If not, replace the master bushing



FULL CONTACT

TOP CONTACT

Evaluation should be done using the second layer of the paper because the outside layer will have misleading slip impressions. If full insert contact is indicated, the master bushing and slips are in good condition and no further analysis is necessary.

If there is not full contact, the test should be run again with new slips. If the second test results in full contact, discard the slips because they are worn, crushed or otherwise distorted. Cut off the toes of discarded slips so they cannot be refurbished and used again. If the results of the second test indicate top contact only, the master bushings and/or bowls are worn and should be inspected for replacement.

Figure 79 — Slip paper test results

14.9.3.6 Other types of slip damage to the pipe body

14.9.3.6.1 Pipe body damage caused by rotating pipe in the slips

This can occur when there is very little string weight below the rotary, and only one tong is used to make-up or break-out the connections, or when the top drive motor is used to make-up connections under these same conditions. It can also be caused by turning the rotary table with the pipe engaged in the slips to orient tools or for other reasons. The use of one tong should be strongly discouraged, and a back-up tong should always be used when making-up or breaking out connections. When making-up connections with the Top Drive motor, ensure that drill string weight below the rotary is in excess of 100 000 lbs. If it becomes necessary to rotate the drill string with the pipe in the slips for any reason or if rotation of the string is detected with the pipe engaged in the slips, inspect the slip area that was engaged for damage before continuing operations.

14.9.3.6.2 Pipe body damage caused by improper slip segment contact

This type of damage can occur with certain types of power slips where the slip segments are positioned below the rotary bushing and remain hidden from view during operation. In some cases when the slip segments become disengaged from their actuating linkage, they can become cocked or misaligned. If this occurs, then it is possible to damage the tube body when the sips are set as shown in the photo below:

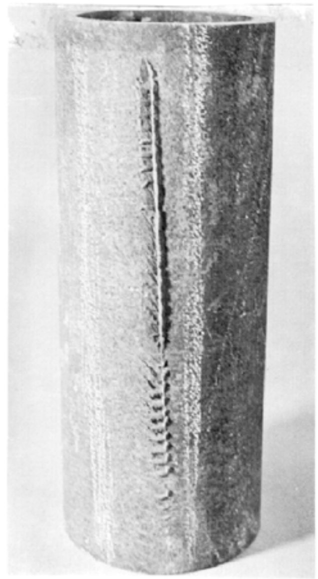


Figure 80 — Slip segment damage to pipe

13.9.3.1 The effect of internal coating on the life-cycle of the drill pipe body

Most coating damage is caused by running tools on a wire-line through the I.D. of the drill pipe. Damage can also be caused by acidizing or gravel-packing through drill pipe. Whenever available, work strings should be used instead of drill pipe for most all types of well stimulation operations creating corrosive or abrasive conditions. Drill pipe with damaged coating should be removed from the rig and re-coated whenever feasible. A grading system that has been widely used n the past is provided below. When the coating reaches a condition of “C” or worse, the pipe should be re-coated.

Figure 81 — Plastic coating condition Grade A – all coating intact

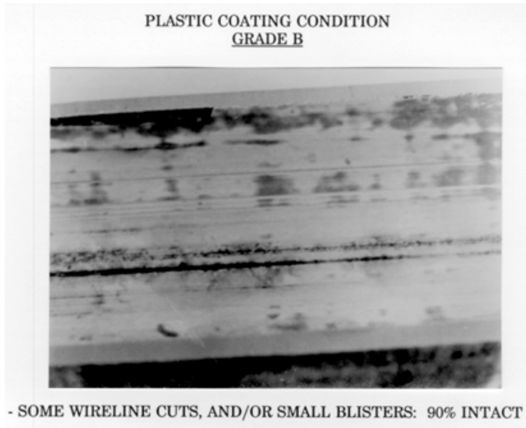


Figure 82 — Plastic coating condition Grade B – some wireline cuts and/or small blisters: 90% intact

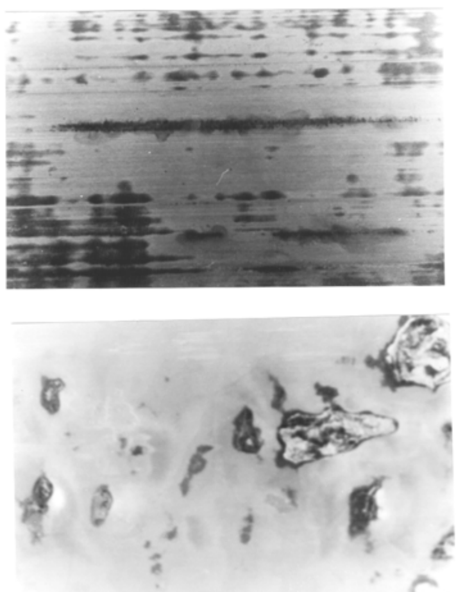


Figure 83 — Plastic coating condition Grade C – heavy wireline damage in tool joint I.D. and/or large blisters: 60-80% intact

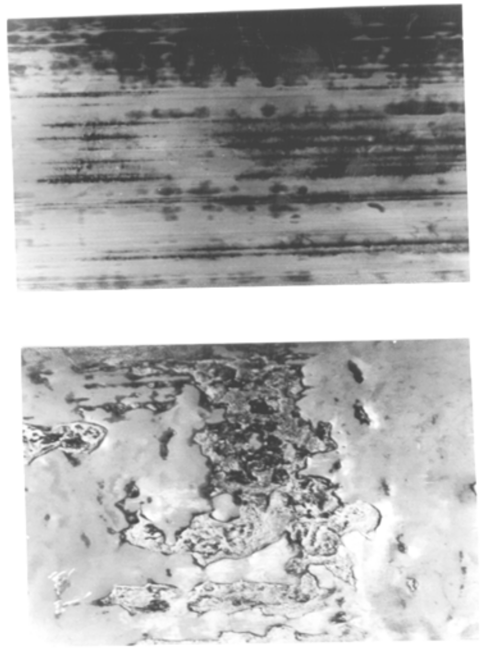


Figure 84 — Plastic coating condition Grade D – less than 50% of coating remaining - corrosion taking place in internal upset area

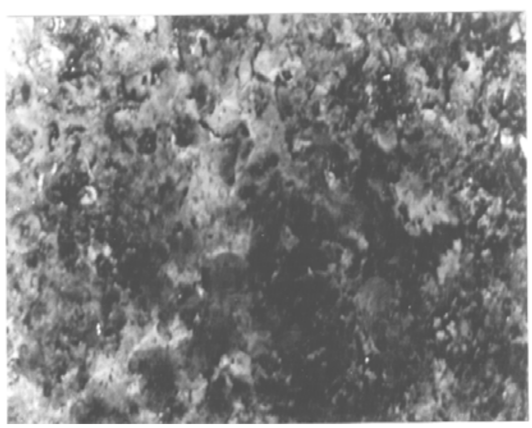


Figure 85 — Plastic coating condition Grade F – virtually no coat remaining - heavy corrosion in upset areas and throughout rest of tube I.D.

13.9.4 Factors affecting the life-cycle of heavy-weight drill pipe

Most of the drilling hours that heavy-weight drill pipe (HWDP) is exposed to is in open hole. In addition, since it functions as a transition between the drill collars and the drill pipe, it is exposed to varying degrees of compression and tension, especially in deviated well bores. Since the pipe body is still more flexible than the tool joints, most of the bending stress takes place in the tube body next to the tool joint. Since HWDP is machined from one piece of bar stock, has no internal upset, and usually has a wall thickness of 1 in (25.4 mm) it is not prone to the same concerns as drill pipe with regard to corrosion or fatigue failures in the pipe body. The same precautions and guidelines provided in the drill collar section for preventing excessive connection damage applies to HWDP. The tool joints on HWDP are usually substantially longer than drill pipe, and therefore can absorb several re-cuts before the tool joint becomes too short to use. Therefore, the factor that most greatly affects HWDP life is wear to the tool joint O.D. As such, the application and maintenance of hard-banding on HWDP tool joints is the most effective preventive action that can be taken to extend the useful life of HWDP. Insofar as drilling conditions are concerned, the greatest risk to HWDP are the friction-induced failures described in subclause 12.90.

14.10 Top drive considerations

14.10.1 General

There are four areas of concern in a drill string when drilling with top drives or power swivels. First, the top drive main shaft assembly is always at the top of the drill string, and is therefore subject to the maximum tensile and torsional loads. Secondly, by the nature of how top drives and power swivels can be used as opposed to kelly drives, unique forms of wear and tear to the drill string components below the main shaft assembly can be incurred. Thirdly, the devices used to make-up connections to the top drive / power swivel main shaft assembly are critical to the integrity of the drill string. Lastly, the duration of the life-cycle of the main shaft can be affected by internal erosion and corrosion.

14.10.2 Top drive main shaft assembly

The top drive / power swivel main shaft assembly includes, but is not limited to the following components:

— Swivel Stem

— Top drive / power swivel main shaft

— One or two IBOP valves

— Saver Sub

Some top drives / power swivels incorporate an integral swivel. Therefore, the swivel stem and the top drive main shaft are the same shaft. Most top drives / power swivels have a main shaft that has a landing collar that absorbs all of the tensile loads when the drill string or casing string is suspended on the elevators. In this operating mode, all of the main shaft assembly components below the main shaft are not under load. When the drill string is made up to the top drive / power swivel for drilling purposes, the entire main shaft assembly absorbs all tensile and torsional loads. The primary influence that affects the fatigue life of main shaft assembly components is high total mean stress level. To a large extent, tensile and torsional loads incurred by drilling conditions cannot be controlled. However, cyclic bending stress resulting from misalignment of the main shaft assembly and the drill string can be prevented. Misalignment can be caused by any one or more of the following conditions:

— The vertical axis of the derrick misaligned with the natural gravitational axis of the drill string

— The plane of the traveling equipment guide rails (if fitted) is misaligned with the derrick and/or the drill string

— The plane of the guide dolly structure is misaligned with the plane of the guide rails, and/or vertical axis of the derrick

— The axis of the top drive/power swivel main shaft is misaligned with the plane of the guide dolly structure

Identification and alleviation of one or more of the above conditions should be carried out by specialists who have the equipment and expertise to identify the root cause of the misalignment, and know what actions need to be performed to correct it. The initial signs of misalignment are fatigue cracks in the last-engaged threads of any box connection that exists in the main shaft assembly. These are generally more susceptible to cyclic bending fatigue failure than pin connections; much the same as fatigue failures that are incurred to drill collar box connections when subjected to extended cyclic bending stress caused by drilling with a buckled bottom hole assembly (BHA). However, some connections of some components of the main shaft assembly cannot accommodate stress-relief features, such as the IBOP valves and the Swivel Stem box connection. Hence the need to perform appropriate non-destructive examination (NDE) to the connections of the main shaft assembly on a regular basis, even though the size of the connections used in the main shaft assembly are much larger than those typically used for the drill string that is connected to it.

14.10.3 Unique wear and tear to the drill string

Drilling with treble stands of drill pipe at higher drilling torque as opposed to drilling with a kelly allows rotating and circulating while pulling out of the hole, backreaming through tight spots, and drilling extended-reach horizontal well bores. While this capability increases drilling efficiency, these drilling practices can shorten the fatigue life of the drill string by increasing the total mean stress in the following ways:

— Rotating and Circulating While Pulling Out of the Hole and Backreaming / Rotating Off-Bottom:

  These drilling practices increase both the total mean stress and the number of bending cycles incurred throughout the drill string in deviated well bores due to the fact the bit is off bottom and the entire weight of the BHA is supported by the drill pipe above it. Drill pipe that is rotated through high doglegs in deep well bores is subject to the greatest loss of fatigue life because that is where the total mean stress levels are the highest. When total mean stress levels amount to a high percentage of the yield strength of the material, only a small amount of bending stress and/or few bending cycles (drill string rotations) can cause rapid expenditure of fatigue life. The specific sections of drill pipe that are most susceptible to the loss of fatigue life are:

— Sharp transitions in cross-sectional area, such as the internal upsets of the drill pipe tube body

  — Heavy slip die marks and transverse formation cuts on the tube body OD surface

  — Localized areas of heavy pitting on the ID of the tube body at or near the internal upset (M_{IU}) area where it transitions to the nominal tube body wall thickness.

  — Over-torqued connections causing high tensile stress in the pin at the last engaged thread, which is increased further due to high tensile loads below the connection in question.

— Drilling Extended-Reach Horizontal Well Bores:

  Extended reach drilling increases drilling torque to much higher levels as opposed to drilling a straight well bore with the same drill string. Higher drilling torque requires higher make-up torque to prevent down-hole make-up which, if incurred, can result in over-torquing the connections. This can cause galled threads and shoulders, stretched pins, and belled boxes in API standard rotary-shouldered connections. Because of this, drill pipe with special proprietary double-shouldered connections possessing higher torsional capacities may have to be utilized to drill these types of wells. Alternatively, the use of thread compounds having a higher friction factor than 2,54 cm (1,00 in) will allow higher make-up torque to be applied to the connection than what is specified in this standard (see subclause 13.13). In addition, “stress balancing” techniques covered in subclause 13.14 can be utilized to increase make-up torque to meet high drilling torque.

  In addition, the probability of buckling a portion of the drill string is much higher when drilling extended-reach horizontal well bores due to a greater friction created between the drill string and the well bore. Drilling with buckled drill pipe will cause much higher bending stress in the same areas listed above and thus reduce the fatigue life of the drill string. In addition, greater contact force is created between the drill pipe and the formation, which causes accelerated wear. Further, a buckled drill string that assumes the shape of a helix within the confines of the wellbore can result in eccentric wear to the tool joints. Accelerated wear can also be incurred to the portion of the drill string that occupies the horizontal well bore section. This is caused by the weight of the drill string lying on the low side of the well bore. In horizontal, abrasive formations, tool joints can become completely worn-out during the drilling of a single well. In addition, when a significant portion of the drill string occupies a horizontal well bore section, much greater drag is incurred when pulling the drill string out of the hole. Consequently, greater contact force is created between the drill string and the high side of a deviated well bore section located above the horizontal section. Depending on the tensile load on the drill string and the well bore curvature in the deviated hole section, wear to the tube section of the drill pipe can also occur (ref. subclause 9.2.3).

14.10.4 Make-up/break-out operations with top drives

Most top drive / power swivels have a self-contained device to make-up and break-out some of the main shaft assembly components as well as the drill string connection to it. Most of these devices depend on the proper operation of sequencing valves and hydraulic pressure to grip the tool joints and apply the torque required for proper make-up and break-out of the connections. Since these devices are subject to wear and tear and can go out of calibration, it is important to verify that proper make-up torque is being applied by comparing the make-up results with a calibrated tong gauge and/or E-Z torque device. In addition, the following operating precautions should be taken:

— For most proprietary double-shouldered connections, the gripping dies should not be closer than 5 cm (2 in) from the box shoulder. When the gripping dies are located any closer to the box shoulder, the box can be egg-shaped from the gripping force, and false make-up torque can result. In some cases, this may require the saver sub to be of a specific shoulder-to-shoulder length to ensure that the gripping dies are in the correct position every time a connection is made to the saver sub. Further, inspections should be carried out at frequent intervals to ensure that the gripping pressure is at the correct setting.

— For drill pipe that has raised hard-banding, it is critical that the remaining tong space be sufficient to engage the gripping dies without encroaching on the raised hard-banded areas. Failure to do so will result in only partial contact between the gripping dies and the tool joint, causing undue wear and tear to the gripping dies, and gouging and tearing of the portion of the hard-banded area gripped by the dies. As a result, sharp burrs protruding out from the hard-banded area can cut BOP (Blowout Preventer) ram packers and annular elements, as well as cause damage to casing and marine riser, and injury to personnel handling the pipe.

Drilling torque should never exceed make-up torque. These readings are measured differently. Make-up torque is usually indicated by hydraulic pressure in the cylinders that actuate the make-up dies, while drilling torque is usually indicated by amperage of the top drive / power swivel motor which is converted to joules or foot-pounds. Therefore, proper maintenance and calibration procedures should be followed to ensure accurate torque readings in both cases, and that they indicate the same reading.

NOTE Some top drive motors are hydraulic motors, and the torque output of these may be related to hydraulic pressure and not amperage.

14.10.5 Top drive main shaft erosion and corrosion

Due to high flow velocities and the corrosive nature of drilling fluids being pumped through the main shaft assembly, erosion and corrosion can combine to:

— Reduce the tensile load capacity of pin connections by increasing the inner diameter (ID)

— Induce corrosion fatigue cracking on the exposed surfaces of the bottom of box connections that initiate from corrosion pits

Inspection of the pin and box connections of the main shaft assembly should include measuring the pin ID and a visual assessment of the exposed surfaces of the bottom of box connections. The acceptance and rejection criteria for pin ID erosion should be obtained from the manufacturer of the top drive / power swivel. Other than taking steps to reduce the corrosivity of the drilling fluid, steps that can be taken to reduce these effects are:

— Maintain an internal coating on the internal surfaces of the main shaft (as well as the swivel stem if the swivel is separate from the top drive) similar to that which is used for drill pipe internal coating. Please note that it is usually not necessary to internally coat the other main shaft assembly components, as these are replaced on a far more frequent basis with minimal impact to the drilling operation.

— Swivel stems in some swivels that are separate from the top drive / power swivel and which employ the use of a box connection at the bottom utilize an elastomer donut-type seal in the bottom of the box that is energized when the pin is made-up to it. The seal insulates the surfaces of the box stress-relief feature from erosion and corrosion. Drilling crews should ensure these seals are replaced after this box connection is inspected and before it is placed back into service.

14.11 External pressure design for well control situations

Drilling personnel should be familiar with all potential loading situations that may be imposed on a drillstring. This section discusses a loading scenario that can occur during a well control event or during displacement operations to control a kick. After a kick is detected, standard procedures are usually to shut-in the well and take pressure measurements to estimate the intensity and size of the kick. If detection of the kick was efficient, the shut-in pressure in the annulus may be only a few hundred psi. Such a pressure will not endanger the drillstring or other rig equipment. However, if the well is left shut-in and the kick is allowed to migrate up to the BOPs with the well shut-in, the pressure in the annulus (i.e. outside the DP) will increase substantially. Since floats are used in most drillstrings, an accompanying pressure increase inside the drillstring may not occur and thus a significant pressure imbalance between the drillpipe external pressure and internal pressure may develop. Various pressure scenarios are possible depending on the nature of the kick (gas vs. oil vs. saltwater), size of the kick, intensity of the kick, well geometry, fluid weights, etc. This section will not attempt to cover the details of these calculations so interested parties are referred to other appropriate literature. Well control pressures in the annulus outside the drillstring can form an effective collapse load which may cause a collapse failure of the drillpipe and hence also threaten the pressure integrity of the seal between the BOPs and the drillstring.

In such cases, users should be aware that the effective pressure limit of their well may be limited by the collapse rating of the drillstring. In other words, despite the fact that a 10 ksi or 15 ksi BOP system is in use, if the drillstring is limited to a differential collapse pressure of 8,000 psi, then the well really cannot sustain a differential collapse pressure across the drillpipe above 8,000 psi and thus may effectively be the upper limit for well control loads. Again, users will need to consider the details of their drilling configuration. The use of non-ported vs. ported drillpipe floats, for example, may be a significant issue in determining potential pressure differentials across the drillpipe.

In addition to considering the potential collapse differentials across the drillstring, users should be aware that this design scenario also involves the drillstring being in tension. Tension tends to decrease collapse resistance and specific collapse ratings should be used which account for the tension in the drillstring. Depending on well depth, wellbore trajectory and drillstring design, the tension load will vary but should be estimated as accurately as possible when reviewing the design case being discussed here. Users are referred to ISO 10400 (API 5C3) for specific rating equations for drillpipe collapse and collapse under tension loads. It is important that operations personnel consider this potential loading scenario and account for it during their planning for well control events.

14.12 Shear ram capacity for well control situations

Due to the need for higher strength drill pipe required for deep drilling, advances in steel chemistry along with thicker-walled drill pipe manufactured with high minimum yield strength materials have in some cases exceeded the capacity of some ram BOP (Blowout Preventer) shear rams to successfully and/or reliably shear drill pipe. Typically (but not always), the shear rams that are most affected by this are utilized in smaller bore surface ram BOP stack applications as opposed to sub-sea BOP stacks that typically have larger bores and larger ram operating cylinders. Regardless of the type of BOP, all Owners/Users should contact the ram BOP OEMs (Original Equipment Manufacturers) and investigate into the shear ram capacities of their equipment for the drill pipe that is intended for use. Most operating cylinders of shear rams can be modified to increase the shearing force of the rams up to ram shaft design load limits without an increase in BOP Control System working pressure. However, such modifications will increase the volume of fluid required to operate the shear rams, and as such, the Owner/User should also examine the volume capacity of the BOP Control System in use, and determine whether additional system volume is required to meet applicable standards governing minimum system volume/pressure requirements.

14.13 Thread compound friction performance considerations

14.13.1 Purpose

This section provides an overview of service issues regarding thread compounds. This includes a description of friction factors, compound types, and considerations that should be taken into account during drilling operations that may affect the performance of the thread compound and in turn impact the make-up and break-out of drill string connections.

14.13.2 Thread Compound Design Factors

14.13.2.1 Thread compound manufactured for use on drill string connections is formulated to prevent galling of connections and exhibit features to include but not necessarily limited to the following performance characteristics:

— The connections are preloaded to the prescribed stress after make-up, assuming the equipment used to make-up the connections is calibrated, maintained, and operated to deliver the specified make-up torque

— The connections will resist further make-up while drilling, assuming the average drilling torque does not exceed make-up torque, and peak drilling torque does not exceed the make-up torque by more than 5%

— The constituents in the thread compound will act to augment the ability of the mating sealing surfaces of the connection to withstand high differential pressures without leaking, and prevent galling

— The compound will exhibit consistant frictional properties and connections will generally break-out at a range of approximately 80 to 90 % of the make-up torque, assuming the threads and shoulders of the connections are in serviceable condition

— The mechanical performance characteristics of the compound will not degrade as long as ambient service conditions do not exceed the limits published by the thread compound manufacturer

— The compound will resist water contamination, and adequately adhere to the surfaces of the connections during and after application

— The compound will be “spreadable” at temperatures down to 0 °F so as not to cause unnecessary waste of productive rig time to apply it to the connections

— The shelf life of the thread compound is sufficient to minimize product wastage

14.13.2.2 Most thread compounds are formulated with an alkali metal soap-base grease such as lithium, calcium, or aluminum-complex base grease. Various ingredients are added to the grease base to provide the features described above. For many years, most thread compounds used for drill string connections were formulated primarily with powdered metallic lead or zinc. The small metal particles suspended in the grease base would deform under contact pressure, and “plate-out” on the loaded surfaces of threads and shoulders to prevent metal-to-metal contact of the connection surfaces and subsequent wear and galling. Thread compounds with either of these metals performed very consistently and very well. Subsequently, concerns for protecting the environment and work place from potentially hazardous substances resulted in a decrease in use of zinc and lead in the formulation of thread compounds. In addition, deeper drilling translates to higher down-hole temperatures that can adversely affect the physical properties of lead and greatly increase the chemical reactivity of zinc. This resulted in a need for constituents that could withstand these higher temperatures. As a result, several types of thread compounds were formulated with varying degrees of success using constituents such as Copper, graphite, synthetic fibers, and other materials as well as combinations of two or more of these in an attempt to duplicate the mechanical performance qualities of lead and zinc based compounds.

14.13.3 Preparation for application of thread compound

Thread protectors will prevent most of the mechanical damage to threads and shoulders when drill string components are handled. However, experience has shown that it is impossible to prevent all damage. Mechanical damage to the threads and sealing shoulders of one connection will cause damages to the mating connection. A simple pre-make-up inspection as provided below will minimize the affect of mechanical damage.

1) Remove the thread protector and thoroughly clean the threads and sealing shoulder.

2) Visually inspect the threads and shoulders for mashes, dings, galls, cuts and any metal protruding above the normal surface.

3) Field dress all discrepancies. Damaged connections should never be run into the hole as this will only cause damage to mating connections. It is best if this preparation is done on racks before the tools are lifted to the rig floor, but it can be done on the rig floor if time is not a constraint.

14.13.4 Application of thread compound

14.13.4.1 After the visual inspection and necessary repairs are finished, apply thread compound to the connection. If the drill string component (s) are still on the cat-walk, re-apply the thread protectors before handling them to the rig floor.

14.13.4.2 Rotary shouldered connections are subjected to high stresses both during make-up and in the drilling operation down-hole. These high stresses require the use of a thread compound formulated to satisfy most, if not all of the design factors listed in 13.13.2.1. Unless unusual drilling conditions dictate otherwise, use a thread compound that has a Friction Factor of 1.0 (see section 13.13.5 for details) that will allow make-up torque to be applied in accordance with the values specified in this standard.

14.13.4.3 For best results, thread compound should be applied to threads and shoulders that are clean and dry. Thread compounds should not be thinned to aid in application. Thinning will reduce the concentration of anti-galling constituents.

14.13.4.4 Thread compound should be applied thoroughly to the threads and shoulders so that all of the surfaces are completely coated. Apply compound to both the pin and the box connection. Do not rely on the rotation during make-up to spread the compound.

14.13.4.5 Doping procedures for proprietary double-shouldered connections are different than for conventional rotary-shouldered connections where there is no secondary shoulder. For double-shouldered connections, it is important that the secondary shoulder at the bottom of the box as well as the nose of the pin is thoroughly cleaned and doped prior to make-up of the connection. This can be particularly difficult when pulling out of the hole when there is a plugged bit, or when slugging the string with fresh water is not allowed. When these difficulties are encountered, the best results are achieved when cleaning and doping activities are performed while running in the hole. Also for double-shouldered connections, it is important that the correct amount of thread compound is applied. Insufficient amounts will cause steel-to-steel contact, and excessive amounts can cause interference as the connections is made up that can provide false torque readings. A paint brush is often recommended for use in applying thread compound to double-shouldered connections as opposed to the conventional bristle brush.

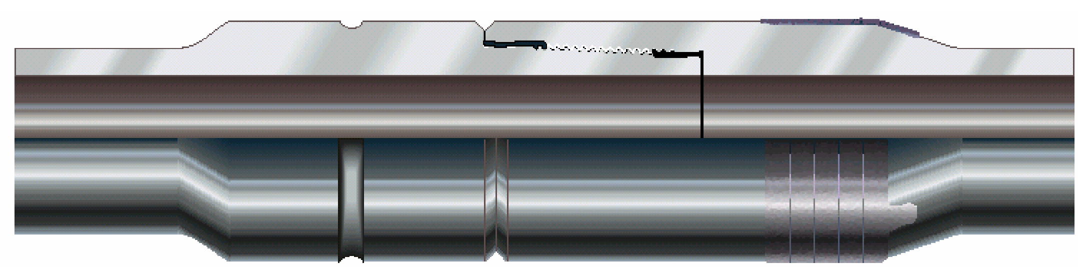


Figure 86 — Cross-section of a Double-Shouldered Connection

14.13.5 Thread Compound Friction Factor

14.13.5.1 Thread Compound Friction Factor Definition

Thread compound friction factor is a term used to quantify the frictional characteristics of a thread compound. The make-up torque values specified in this standard for all drill string connections are based on thread compounds containing either 50% metallic zinc or 60% metallic lead. Compounds containing 60% metallic lead is specified by API RP 7A1 as the “reference compound” because of its demonstrated consistent frictional properties. The friction factor of a thread compound that duplicates the frictional characteristics of the “reference compound” is considered to possess a friction factor of 1.0. Use of this compound allows make-up of the connection to the same torque value specified by this standard and/or the connection manufacturer’s recommendation. If a compound possesses a friction factor under 1.0, less applied make-up torque is required to achieve the proper preload stress in the connection. A compound possessing a friction factor exceeding 1.0 requires more applied torque. Thus, for the convenience of the user, the actual torque to be applied to a connection when a thread compound is used that does not have a friction factor of 1.0 is the product of the specified torque times the friction factor of the thread compound.

14.13.5.2 The Effect of Contamination on the Friction Factor of Thread Compounds

Since thread compounds are formulated to provide specific performance characteristics, contamination with foreign materials can affect its friction factor as well as its galling and wear resistance. Dirt, sand, and solids suspended in drilling fluids are not only abrasive causing unnecessary wear and tear, but if solid particles are large enough, they can cause deformation to the surfaces of the threads and shoulders of the connection that can lead to connection washouts. Solids contamination can also alter the friction characteristics of the thread compound by interfering with the proper mating of the surfaces of the connection. In addition to solids contamination, some drilling fluids (invert, high pH, polymer-based) may react chemically with the grease base of the thread compound or affect the adherence of the compound to the connection surfaces. These reactions can greatly affect the lubricity and the film strength of the compound, and thus alter the friction factor. The compatibility of a given type of drilling fluid and a given thread compound, and the effect that contamination of the thread compound has on its friction factor can only be determined by conducting certain tests such as that which is specified API RP 7A1, or other testing which should be performed in consultation with the thread compound manufacturer. In the absence of any test results, the user should establish connection cleaning and doping practices that will ensure that the thread compound is not contaminated when the connections are made-up. If thread compound contamination does occur, it may be manifested by one or more of the following symptoms:

— Galling of threads and shoulders

— High breakout torque (break-out torque that exceeds make-up torque values)

— Connection washouts caused by the loss of sealing capability of the connection

— Fatigue cracking of pin connections at the last-engaged thread caused by under-torque or over-torque

— Belled or otherwise deformed box counterbore caused by over-torque

— Flared pin nose on double-shouldered connections caused by over-torque

— Unusual wear or abrasion to connection threads and shoulders

These symptoms are not exclusive to thread compound contamination issues. Therefore care should be taken to ensure that other causes of these symptoms are ruled out. Conversely, connections that are over-torqued which can be caused by lowering the friction factor of the thread compound for any reason can be tested for on the rig. This test requires that certain connections be selected that are not damaged. After they are made up to the specified torque, the box and pin tool are punch-marked on the tool joint surface next to the shoulder and in-line vertically with each other. During the next trip, these connections are visually examined. If the punch-mark applied to the pin tool joint has moved to the left of the punch-mark on the box, the connection has definitely made up further during drilling. This test should be done on connections without any changes to the established doping procedures. If further make-up whioe drilling is discovered, the test should be repeated on undamaged connections where all surfaces of the pin and box are completely cleaned and re-doped prior to make-up. Comparison of any changes to the relative positions of these punch-marks may reveal whether established doping practices need to be changed, or whether there are other reasons causing the connections to make up further while drilling.

14.13.5.3 The Effect of Ambient Temperatures on the Friction Factor of Thread Compounds

Ambient temperatures can affect both the application and the frictional properties of thread compound. Environmental concerns notwithstanding, zinc and lead melting points generally exceed down-hole temperatures. However, as technology allows deeper drilling higher down-hole temperatures have been encountered that exceed the melting point or increase the reactivity of these metals. By substituting copper or other suitable materials in place of these metals to address environmental concerns, it will also improve its resistance to higher ambient down-hole temperatures. In addition, a "thermal grade" greases that have improved resistance to higher temperatures are also available. In any case, high down-hole temperatures will lower the viscosity of the grease base. In dynamic conditions while drilling, lowering the viscosity of the grease base on the compound may also result in lowering its film strength which can cause metal-to-metal contact of the mating surfaces of the connection. This may increase the possibility of galling the connection during break-out. In contrast, cold ambient temperatures on the rig floor cause an increase in compound viscosity, and can adversely affect the ability to apply it to the connections properly and in an efficient manner.

14.13.5.4 Special Factors Affecting the Friction Factor of Thread Compounds

14.13.5.4.1 Friction Factor Variance Between Thread Compounds

Aside from factors caused by field conditions, various tests conducted for the API in connection with work being done to revise API RP - 7A1 have revealed that the friction factors of various compounds vary greatly, and in some cases, the friction factor actually changes as the connection is made up to the maximum recommended torque. The graph below shows how the Friction Factor of the lead-based reference compound behaves as a function of Applied Torque, and Contact Pressure between the mating surfaces of the connection. For this test, the apparatus used for determining the friction factor of a compound as specified in API RP 7A1 was modified to incorporate a load cell to measure the axial load. Make-up of the connection was stopped after 600 ft.-lbs of torque was reached.

NOTE The turn-down at the upper end of the friction curve is an artifact of the trend line polynomial, not a reflection of actual data points.

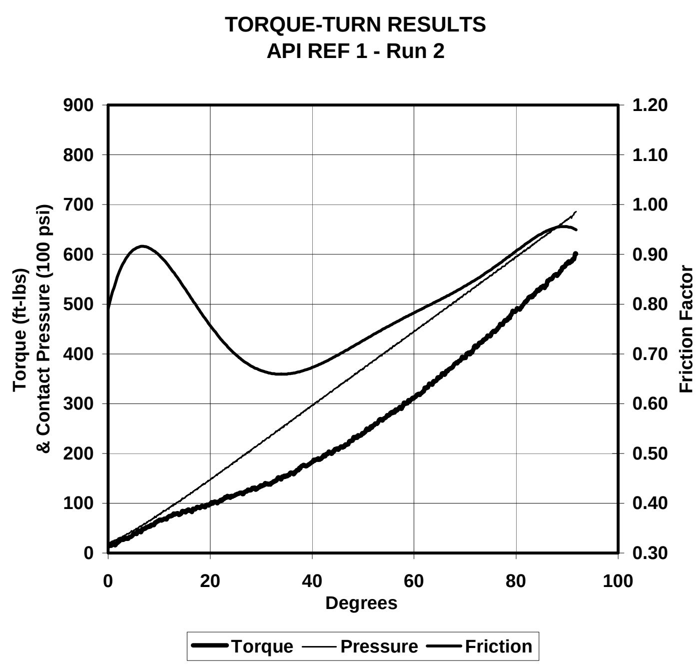


Figure 87 — Friction Factor Behavior as a Function of Applied Torque vs Contact Pressure for Lead-Based API Reference Thread Compound

The next graph shows how a popular copper-based compound compares to the graph above:

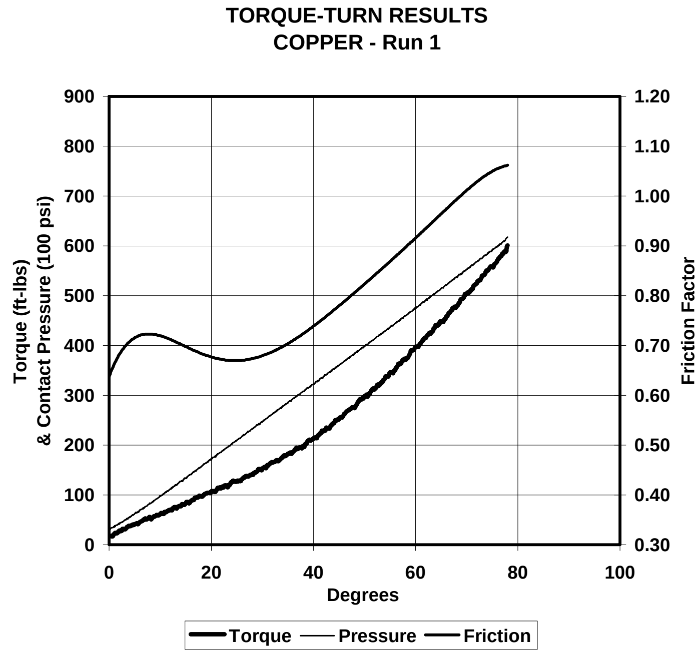


Figure 88 — Friction Factor Behavior as a Function of Applied Torque vs Contact Pressure for Copper-Based Thread Compound

The steeper slope of the Friction Factor curve for the copper-based compound shows that greater friction is created as contact pressure increases for each degree of make-up as compared to the lead-based API reference compound. In comparing these two graphs, connections with the copper-based thread compound require 22% more make-up torque to achieve the same amount of preload on the connection. This illustrates why the user should be aware of the friction factor of the compound in use. In some cases, the user can utilize these differences to the advantage of the drilling program if drilling torque approaches the maximum recommended make-up torque of the connection in use. By substituting a compound that has a higher friction factor, the user may be able to avoid down-hole make-up of the connections when drilling torque is high without damaging the connections and threatening the integrity of the drill string.

14.13.5.4.2 Variance in Friction Factor of Thread Compounds Among Connection Sizes and Types

Another important consideration is that the same thread compound may exhibit a different friction factor between two different sizes or types of connections. The reason for this is the difference in contact pressure between two different connections. For example, an NC46 connection (6-1/4” O.D. X 3-1/4” I.D.) that is made up to the maximum recommended make-up torque will result in a contact pressure of approximately 78,200 psi on the mating shoulders. By comparison, an NC50 connection (6-5/8” O.D. X 3-1/4” I.D.) made up to the maximum recommended make-up torque will create a much higher contact pressure between the mating shoulders of approximately 111,895 psi. Since the Friction Factor curves above show that the Friction Factor of a given thread compound changes as the contact pressure increases, the Friction Factor of a given compound may have to be revised, depending on what connection it is to be used with. In other words, the Friction Factor of a given compound is only consistant under the same contact pressure. Further testing is required utilizing full-size tool joint specimens to quantify these differences. Until this information is available to the industry, the User should rely on the Friction Factor assigned to the thread compound in use to determine make-up torque.

14.14 Connection stress balancing considerations

When drilling torques dictate higher make-up torques, and tool joint dimensional changes can result in undesirably large ODs or small IDs, one measure is available through engineering a higher make-up torque. Make-up torques for tool joints are based on stressing the tool joint to 60% of yield, with the remaining 40% of pin yield capacity available for the external tension rating. Thus, margins for higher make-up torques and hence drilling torques exist if these specific stress levels can be optimized for a given application.

As make-up torque is increased, the pin is exposed to higher tension at make-up and is thus able to support less subsequent applied tension. As drilling torque is limited by the make-up torque, the optimization task is to maximize the make-up torque while balancing the design margin on tool joint tensile capacity. When operating tension can be confidently forecast below the maximum tension available at nominal make-up torque, the opportunity exists to increase torque capacity by an engineered reduction in tension capacity.

This optimization has been termed “stress balancing” and can substantially increase tool joint torsional capacity. As shown in Figure 91, make-up torque on a 6-3/8” OD by 2-3/4” ID NC50 connection can be increased from 30 ft-kips to 40 ft-kips by reducing allowable tension from 1 440 kips to 1 000 kips. Since such a make-up torque increase would be made knowing the maximum subsequent field loads, the structural integrity of the tool joint is not compromised. It is simply stressed to a higher degree at make-up and accompanied by a lower subsequent field tension capacity. Moreover, since new 5” 19.50 lb/ft S-135 drill pipe only has a pipe body tension rating of 712 kips, the higher tool joint tension rating is actually unnecessary.

Torque increases through stress balancing can, however, carry some risks. One of them is galling of the connection. For this reason, engineers are encouraged to ensure that proper tool joint plating and drill stem thread compounds are used and to test elevated torque levels so that any problems can be observed early and different thread compounds tried, as required. Also, increased make-up torque, resulting in higher mean stresses in the pin critical section, can affect fatigue characteristics of the tool joint. Stress balancing, therefore, should be applied with caution for drilling wells with severe doglegs.

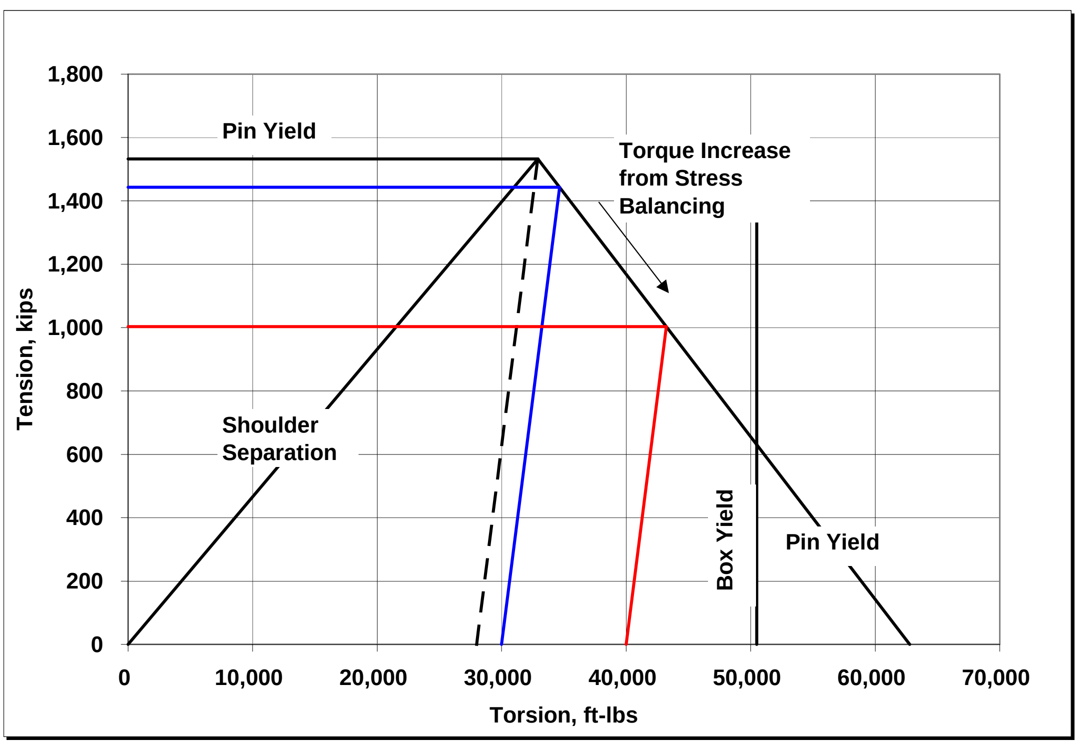


Figure 89 — Stress balancing technique to increase make-up torque with an engineered reduction in 6-3/8” OD x 2-3/4” ID NC50 tool joint tensile capacity.

14.15 Hardbanding

14.15.1 General

Hard-banding is applied to the OD of drill pipe and heavy-weight tool joints, as well as drill collars to reduce the rate of OD wear, and thus extend the useful life of the drill stem elements it is applied to. It is most commonly applied to the OD of drill pipe box tool joints, but it can also be applied to the pin. “Casing-friendly” hard-banding is applied to tool joints reduce the amount of casing wear caused by casing/tool joint interaction. It is applied by a variety of welding processes, and is available in a variety of metal mixtures and alloys.

14.15.2 Tungsten-Carbide Hard-banding

This type of hard-banding is applied via the submerged-arc welding process using mild steel welding wire with a “hopper” that feeds tungsten carbide particles into the weld puddle during the welding process. As the particles are fed into the molten metal, the tungsten carbide particles disperse fairly evenly and are trapped inside the weld deposit as it cools. The matrix of tungsten carbide particles in the weld deposit provides a very abrasion-resistant surface. This type of hard-banding is usually applied flush to the tool joint surface. This requires the area that is to be hard-banded to be “undercut” so that the hard-band weld deposit is flush with the original tool joint surface after it is applied. For many years, this was the standard type of hard-banding for the industry, and in many areas of the world it still is. In extremely abrasive drilling conditions, it is almost exclusively used to minimize OD wear to the connections in the drill string. However, with the advent of top drives, directional drilling, and drilling in deeper water offshore, increased side-loads were created between the drill string and casing as well as marine riser. An increase in incidents involving excessive casing and marine riser wear then ensued, which prompted the search for hard-banding products that would be less abrasive.

14.15.3 “Casing Friendly” Hard-banding

Joint industry projects were formed to determine solutions to excessive casing and marine riser wear. The result of these projects promoted the development of what is known as “casing-friendly” hard-banding that is less abrasive than tungsten-carbide hard-banding. Some of the more important conclusions that have been reached by these joint industry projects are discussed below.

14.15.4 Raised Hard-banding

Wear comparison tests between casing and “casing-friendly” hard-banding, versus casing and tool joint steel have conclusively shown that in spite of greater hardness, the “casing friendly” hard-banding is less abrasive to casing than plain steel tool joints without any hard-banding. This conclusion is important because it promotes the application of these hard-banding materials so that they are raised from the surface of the tool joint to ensure the tool joint does not come into contact with the casing or marine riser. Furthermore, by maintaining the hard-banding proud to the tool joint surface by re-applying it when it becomes worn flush, the original OD of the tool joints can be maintained indefinitely. This ensures that the original tensile and torsional capacity of the connection is maintained throughout its life. This is especially important for proprietary, double-shouldered connections, because less wear to the box OD can be tolerated before the torsional strength of the connection is reduced. Since raised hard-banding will interfere with tonging operations as well as some types of fishing tools, it is important to specify longer tool joints when the drill pipe is manufactured. This will provide sufficient tong space and fish neck length to allow re-cutting the connection at least twice and still leave sufficient remaining tong space for make-up and break-out and engaging fishing tools. Optimum protection of both the tool joint and the casing or marine riser requires a total of 10,2 cm to 12,7 cm (4 in to 5 in) of linear contact between the hard-banding and the casing for each pin and box assembly in the hole. Since more tong space is available on the box, it is common practice to apply a 76 mm (3 in) wide band of hard-banding near the 18° elevator taper. Applying a 38 mm (1,5 in) wide band on the uppermost part of the tong space on the pin will provide the added surface contact area to obtain optimum protection.

14.15.5 Hard-band Cracking and Spalling of Chromium-Carbide Hard-banding

Most (not all) of the casing friendly hard-banding materials in use are made with chromium carbides. The high chrome content of the weld deposit is what provides the hardness required to resist abrasion and wear.

However, after the weld is applied, and as the tool joint cools down, the weld shrinks around the tool joint OD, causing extremely high tensile loading on the weld deposit until it eventually cracks. These cracks are usually superficial and oriented longitudinally in parallel with the axis of the tool joint, and are therefore of no concern as regards to detracting from the physical capacity of the connection or the rest of the joint of pipe.

Figure 90 — Photo of new pin tool joint with longitudinal crack in hard-banding

Figure 91 — Photo of worn out box tool joint with longitudinal crack in hard-banding

Transverse cracking is a concern, mainly due to the fact that these are perpendicular to the tensile loads exerted on the pipe, and it is possible for a transverse crack in the hard-banding material to grow into the base metal of the tool joint. Failure to pre-heat the tool joint to the prescribed temperature, maintain prescribed inter-pass welding temperatures, and to ensure that the tool joint cools slowly after welding is completed will not only cause more cracking to occur, but can also cause problems with adhesion between the weld deposit and the parent tool joint material. Without proper adhesion, parts of the hard-band weld deposit will gradually disbond and spall-off the surface of the tool joint (see photo below).

Figure 92 — Photo of new box tool joint with newly-applied hard-banding spalling-off

Figure 93 — Photo of spalling problems on worn hard-banding on box tool joint

A spalled section creates a cutting edge that gouges casing inner diameter and defeats the purpose of “casing friendly” hard-banding. Spalled hard-banding should not be allowed in cased holes where casing wear may be an issue. See part 2 of ISO 10407 and manufacturer’s specifications for hard-banding inspection criteria.

Lastly, it is extremely important to ensure that the proper current and voltage settings on the welding machine are calibrated and maintained to suit the requirements of the welding procedures specified by the hard-banding material manufacturer to ensure proper adhesion between the weld deposit and the tool joint parent material. Of particular importance during the period after all hard-banding is completed, and the tool joint is cooling slowly is to ensure that the insulation used to wrap the tool joint covers the drill pipe tube body for a distance of 15,2 cm to 22,9 cm (6 in to 9 in) past the tool joint tong space. This will ensure that the hard-banded area closest to the pipe body will cool as slowly as the rest of the hard-banded area. Failure to do so may result in accelerated cooling in this area, and result in poor adhesion of the hard-banding.

14.15.6 Hard-banding Removal and Re-Application

After raised hard-banding has worn flush with the tool joint, it should be re-applied as long as sufficient tong space remains to perform make-up and break-out of the connection. If insufficient tong space remains (less than 15,2 cm (6 in) for standard 14,0 cm (5,5 in) long tong dies), then raised hard-banding should not be re-applied. In some cases such as (but not limited to) spalled hard-banding, it may be necessary to remove any remaining hard-banding by the plasma-arc process or by machining or grinding before new hard-banding may be applied. In these cases, the amount of metal removed from the surface of the tool joint may be as deep as 6,35 mm (0,25 in) below the surface of the tool joint to ensure all old hard-banding is removed.

Figure 94 — Photo of box tool joints after removal of old hard-banding by machining

14.15.7 Removal

After removal, the surface is examined via wet MPI (see magnetic particle inspection ISO 10407-2), and then filled-in with mild steel weld deposit that is proud of the tool joint surface. The photo below shows a crack that was found in the tool joint material after removal of the old hard-banding. Additional material was removed in this instance, but the crack was still present. This is why it is important to conduct MPI inspections at this stage of the re-hard-banding process.

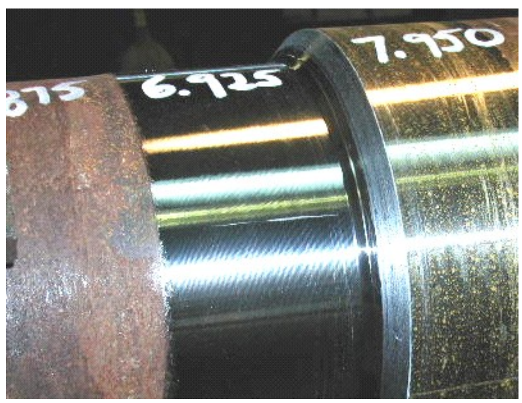


Figure 95 — Photo of machined area in way of old hard-banding removal showing longitudinal crack

14.15.8 Internal Plastic Coating

If the drill pipe has internal plastic coating, procedures are available to include running water through the ID of the tool joint while the weld build-up is applied. As long as the water exiting the tool joint that is being welded does not exceed approx. 71 °C (160 °F), the probability is high that no damage will be done to the internal coating. After the weld build-up process is completed, the tool joint is allowed to stand for 24 hours, after which the built-up area is then machined flush to the tool joint OD, and inspected again via wet MPI before the re-application of new hard-banding can commence. Any cracks revealed on or next to the built-up surface by the MPI inspection at this stage isare grounds for rejection and scrapping of the joint of pipe. Further use of the pipe should not be continued, because any cracks forming at this point in the process is a strong indication that an error of some sort was made during the build-up process.

14.15.9 Re-Application

Alternative, some types of hard-banding can be applied over the top of existing worn (not spalled) hard-banding, depending on the product. These types of hard-banding products are therefore generally cheaper to maintain than those that require complete removal before re-application is possible. Therefore, the drill pipe Owner/User is encouraged to research all of the available types of hard-banding products available in order to select the one that best meets their needs.

14.16 Drill pipe bending resulting from tonging operations

14.16.1 General

It is generally known that the tool joint on a length of drill pipe should be kept as close to the rotary slips as possible during make-up and break-out operations to prevent bending of the pipe. This section does not apply to tool joint make-ups and break-outs with iron rough neck systems.

There is a maximum height that the tool joint may be positioned above the rotary slips and the pipe to resist bending, while the maximum recommended make-up or break-out torque is applied to the tool joint.

14.16.2 Many factors govern this height limitation

Several of these which should be taken into most serious considerations are:

— The angle of separation between the make-up and break-out tongs, illustrated by Case I and Case II, Figure 98. Case I indicates tongs at 90º and Case II indicates tongs at 180º.

— The minimum yield strength of the pipe.

— The length of the tong handle.

— The maximum make-up or break-out torque.

Case I

$$H_{MAX} = \frac{\left(0.053 \times Y_M \times L_T \times \left[\frac{I}{C}\right]\right)}{T} \tag{157}$$

Case II

$$H_{MAX} = \frac{\left(0.038 \times Y_M \times L_T \times \left[\frac{I}{C}\right]\right)}{T} \tag{158}$$

Where:

H_{MAX} is the height of the tool joint above the shoulder of the slips (ft);

Y_M is the minimum yield stress of the pipe (psi);

L_T is the tong arm length (ft);

P is the line pull (lbs);

T is the maximum make-up or break-out torque applied to the tool joint (ft-lbs); and

I/C is the section modulus of pipe (in^3) (see Table B.18).

Constants 0,053 and 0,038 include a factor of 0,9 to reduce Y_M to proportional limit

Common tong arm lengths:

14.16.3 Sample Calculation

5” 19.50# S135 dp with NC50 (6-5/8” box OD, 2-3/4” pin ID)

5 foot tong arms (SDD)

Tongs at 180º

Using equation #:

$$H_{MAX} = \frac{\left(0.038 \times Y_M \times L_T \times \left[\frac{I}{C}\right]\right)}{T} \tag{159}$$

where

Y_M = 135000 psi for grade S

L_T = 5 ft

(I/C)= 5.71 cu.in. (from Table B.18)

T = 38044 ft-lbs. (from Table B.10)

$$H_{MAX} = \frac{(0.038 \times 135000 \times 5 \times [5.71])}{38044} \tag{160}$$

= 3.85 ft

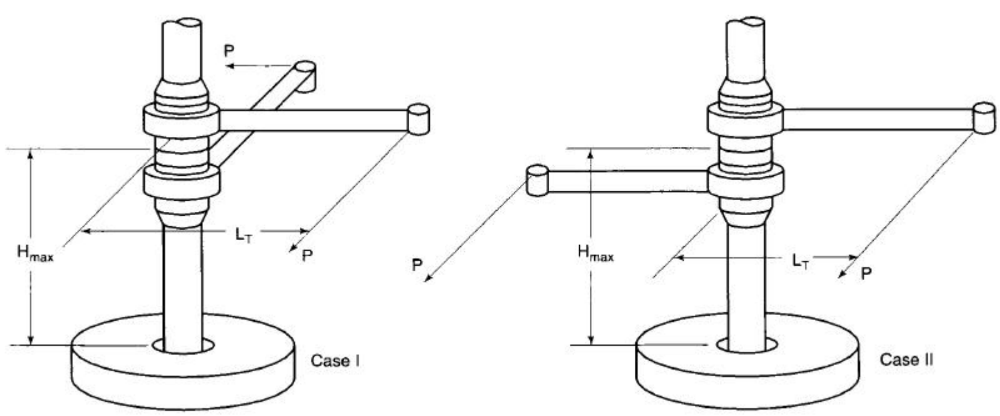


Figure 96 — Maximum height of tool joint above slips to prevent bending during tonging[BC8]

Annex A
(normative)

Tables in SI units

Table A.1 —New drill pipe dimensional data

Labels[a]		Outside diameter D	Nominal linear mass	Plain end linear mass[b]	Wall thickness t	Inside diameter d	Section area body of pipe A_{DP}[c]	Polar sectional modulus Z[d]
1	2	mm	kg/m	kg/m	mm	mm	mm^2	Cm^3
1	2	3	4	5	6	7	8	9
2 3/8	4,85	60,32	7,22	6,61	4,83	50,67	841,4	21,65
	6,65	60,32	9,90	9,33	7,11	46,10	1 189,0	28,40
2 7/8	6,85	73,02	10,19	9,18	5,51	62,00	1 169,0	36,72
	10,40	73,02	15,48	14,47	9,19	54,64	1 843,8	52,50
3 1/2	9,50	88,90	14,14	13,12	6,45	76,00	1 671,1	64,29
	13,30	88,90	19,79	18,34	9,35	70,21	2 336,1	84,30
	15,50	88,90	23,07	21,80	11,40	66,09	2 776,6	95,82
4	11,85	101,60	17,63	15,58	6,65	88,29	1 985,0	88,49
	14,00	101,60	20,83	19,27	8,38	84,84	2 454,7	105,83
	15,70	101,60	23,36	21,89	9,65	82,30	2 788,1	117,28
4 1/2	13,75	114,30	20,46	18,23	6,88	100,53	2 322,8	117,72
	16,60	114,30	24,70	22,32	8,56	97,18	2 843,5	139,99
	20,00	114,30	29,76	27,85	10,92	92,46	3 547,2	167,67
	22,82	114,30	33,96	31,82	12,70	88,90	4 053,7	185,91
5	16,25	127,00	24,18	22,15	7,52	111,96	2 822,1	159,25
	19,50	127,00	29,02	26,71	9,19	108,61	3 403,0	187,06
	25,60	127,00	38,10	35,80	12,70	101,60	4 560,4	237,46
5 1/2	19,20	139,70	28,57	25,13	7,72	124,26	3 201,5	200,27
	21,90	139,70	32,59	29,52	9,17	121,36	3 760,1	230,43
	24,70	139,70	36,76	33,58	10,54	118,62	4 277,2	257,08
6 5/8	25,20	168,28	37,50	33,05	8,38	151,51	4 210,4	320,73
	27,70	168,28	41,22	36,07	9,19	149,89	4 595,3	346,68

[a] Labels are for information and assistance in ordering.

[b] kg/m = 0,0246616(D – t)t

[c] $A_{DP} = 0,7854\ (D^2 - d^2)$

[d] $Z = 0,19635\ (D^4 - d^4)/1000D$

Table A.2 — New drill pipe torsional and tensile data

Labels [a]		Outside diameter D mm	Nominal linear mass T&C kg/m	Torsional data [b] torsional yield strength N·m				Tensile data based on minimum values load at the minimum yield strength [c] kN			
1	2			E75	X95	G105	S135	E75	X95	G105	S135
1	2	3	4	5	6	7	8	9	10	11	12
2 3/8	4,85	60,33	7,22	6 458	8 180	9 041	11 638	435,112	551,144	609,157	783,203
	6,65	60,33	9,90	8 474	10 734	11 865	15 254	614,807	778,759	860,731	1 106,655
2 7/8	6,85	73,03	10,19	10 959	13 881	15 342	19 726	604,522	765,730	846,332	1 088,142
	10,40	73,03	15,48	15 665	19 842	21 932	28 198	953,450	1 207,706	1 334,831	1 716,213
3 1/2	9,50	88,9	14,14	19 179	24 294	26 852	34 523	864,129	1 094,565	1 209,783	1 555,436
	13,30	88,9	19,79	25 152	31 859	35 213	45 273	1 207,999	1 530,135	1 691,201	2 174,402
	15,50	88,9	23,07	28 589	36 211	40 024	51459	1 435,775	1 818,647	2 010,085	2 584,395
4	11,85	101,6	17,63	26 403	33 445	36 965	47 527	1 026,449	1 300,171	1 437,029	1 847,613
	14,00	101,6	20,83	31 574	39 994	44 204	56 833	1 269,340	1 607,828	1 777,074	2 284,811
	15,70	101,6	23,36	34 994	44 324	48 991	62 989	1 441,749	1 826,218	2 018,447	2 595,151
4 1/2	13,75	114,3	20,46	35 125	44 493	49 176	63 226	1 201,171	1 521,483	1 681,637	2 162,107
	16,60	114,3	24,70	41 769	52 907	58 476	75 184	1 470,395	1 862,502	2 058,553	2 646,710
	20,00	114,3	29,76	50 031	63 372	70 043	90 055	1 834,260	2 323,395	2 567,963	3 301,666
	22,82	114,3	33,96	55 469	70 260	77 656	99 844	2 096,176	2 655,157	2 934,643	3 773,115
5	16,25	127,0	24,18	47 513	60 183	66 519	85 524	1 459,342	1 848,499	2 043,077	2 626,813
	19,50	127,0	29,02	55 815	70 698	78 140	100 466	1 759,694	2 228,946	2 463,572	3 167,445
	25,60	127,0	38,10	70 851	89 744	99 190	127 531	2 358,198	2 987,048	3 301,475	4 244,756
5 1/2	19,20	139,7	28,57	59 756	75 690	83 658	107 560	1 655,544	2 097,021	2 317,759	2 979,975
	21,90	139,7	32,59	68 754	87 088	96 255	123 756	1 944,389	2 462,896	2 722,147	3 499,901
	24,70	139,7	36,76	76 704	97 158	107 386	138 067	2 211,754	2 801,552	3 096,456	3 981,154
6 5/8	25,20	168,28	37,50	95 694	121 213	133 971	172 249	2 177,245	2 757,844	3 048,144	3 919,039
	27,70	168,28	41,22	103 442	131 026	144 819	186 194	2 376,236	3 009,894	3 326,723	4 277,215

[a] Labels are for information and assistance in ordering.

[b] Based on the shear strength equal to 57,7 % of minimum yield strength and nominal wall thickness. Minimum torsional yield strength calculated from equation E.15

[c] Minimum tensile strength calculated from equation E.13

Table A.3 — New drill pipe collapse and internal pressure data

Labels [a]		Outside diameter D mm	Nominal linear mass T&C kg/m	Collapse pressure based on minimum values[b] MPa				Internal pressure at minimum yield strength[b] MPa			
1	2			E75	X95	G105	S135	E75	X95	G105	S135
1	2	3	4	5	6	7	8	9	10	11	12
2 3/8	4,85	60,33	7,22	76,118	96,416	106,565	131,242	72,395	91,700	101,353	130,311
	6,65	60,33	9,90	107,551	136,234	150,575	193,598	106,689	135,137	149,361	192,040
2 7/8	6,85	73,03	10,19	72,167	89,218	96,664	117,445	68,306	86,515	95,623	122,947
	10,40	73,03	15,48	113,826	144,176	159,352	204,885	113,943	144,328	159,524	205,098
3 1/2	9,50	88,9	14,14	68,954	83,268	90,011	108,579	65,673	83,185	91,942	118,211
	13,30	88,9	19,79	97,306	123,258	136,227	175,154	95,148	120,520	133,207	171,266
	15,50	88,9	23,07	115,653	146,493	161,916	208,180	116,094	147,051	162,530	208,966
4	11,85	101,6	17,63	57,785	68,796	73,829	86,998	59,274	75,077	82,985	106,689
	14,00	101,6	20,83	78,283	99,160	109,599	138,867	74,656	94,568	104,518	134,386
	15,70	101,6	23,36	88,915	112,626	124,485	160,048	85,971	108,896	120,355	154,746
4 1/2	13,75	114,3	20,46	49,456	57,999	61,749	70,899	54,496	69,030	76,297	98,099
	16,60	114,3	24,70	71,650	88,012	95,320	115,646	67,769	85,840	94,879	121,989
	20,00	114,3	29,76	89,384	113,219	125,133	160,889	86,474	109,530	121,058	155,649
	22,82	114,3	33,96	102,146	129,380	143,004	183,862	100,546	127,360	140,770	180,987
5	16,25	127,0	24,18	47,836	55,903	59,405	67,782	53,572	67,858	75,001	96,430
	19,50	127,0	29,02	68,686	82,916	89,625	108,055	65,521	82,992	91,728	117,935
	25,60	127,0	38,10	93,079	117,900	130,311	167,543	90,494	114,625	126,691	162,889
5 1/2	19,20	139,7	28,57	41,637	47,863	50,421	55,799	50,021	63,356	70,023	90,032
	21,90	139,7	32,59	58,006	69,079	74,139	87,419	59,398	75,236	83,158	106,917
	24,70	139,7	36,76	72,147	89,170	96,616	117,369	68,279	86,488	95,596	122,906
6 5/8	25,20	168,28	37,50	33,012	36,687	37,921	41,617	45,078	57,095	63,108	81,138
	27,70	168,28	41,22	40,638	46,574	48,973	53,869	49,449	62,632	69,223	89,004

[a] Labels are for information and assistance in ordering.

[b] Calculations are based on formulas in API Bulletin 5C3.

Table A.4 — Used drill pipe torsional and tensile data API premium class

Labels [a]		Outside diameter D mm	Nominal linear mass T&C kg/m	Torsional data [b] torsional yield strength N·m				Tensile data based on minimum values load at the minimum yield strength [c] kN			
1	2			E75	X95	G105	S135	E75	X95	G105	S135
1	2	3	4	5	6	7	8	9	10	11	12
2 3/8	4,85	60,33	7,22	5 050	6 398	7 071	9 091	342,037	433,248	478,851	615,665
	6,65	60,33	9,90	6 523	8 261	9 131	11 740	478,700	606,350	670,178	861,661
2 7/8	6,85	73,03	10,19	8 585	10 874	12 019	15 452	475,720	602,578	666,010	856,296
	10,40	73,03	15,48	12 010	15 212	16 813	21 619	740,785	938,330	1 037,099	1 333,417
3 1/2	9,50	88,9	14,14	15 041	19 052	21 057	27 073	680,485	861,950	952,680	1 224,876
	13,30	88,9	19,79	19 471	24 664	27 260	35 048	943,690	1 195,340	1 321,166	1 698,643
	15,50	88,9	23,07	21 891	27 729	30 648	39 404	1 114,813	1 412,097	1 560,739	2 006,660
4	11,85	101,6	17,63	20 758	26 292	29 059	37 362	809,648	1025,555	1133,509	1457,371
	14,00	101,6	20,83	24 670	31 249	34 538	44 406	997,211	1263,130	1396,092	1794,978
	15,70	101,6	23,36	27 207	34 462	38 090	48 972	1 129,186	1430,299	1580,858	2032,531
4 1/2	13,75	114,3	20,46	27 663	35 040	38 728	49 792	948,619	1 201,585	1 328,066	1 707,512
	16,60	114,3	24,70	32 728	41 455	45 820	58 910	1 157,272	1 465,876	1 620,180	2 083,089
	20,00	114,3	29,76	38 889	49 260	54 446	70 001	1 436,402	1 819,438	2 010,961	2 585,520
	22,82	114,3	33,96	42 826	54 246	59 957	77 086	1 635,015	2 071,021	2 289,024	2 943,033
5	16,25	127,0	24,18	37 430	47 412	52 402	67 375	1 152,779	1 460,187	1 613,891	2 075,002
	19,50	127,0	29,02	43 773	55 446	61 282	78 791	1 385,777	1 755,322	1 940,092	2 494,403
	25,60	127,0	38,10	54 970	69 629	76 959	98 946	1 844,633	2 336,535	2 582,486	3 320,344
5 1/2	19,20	139,7	28,57	47 134	59 703	65 988	84 840	1 308,934	1 657,986	1 832,512	2 356,085
	21,90	139,7	32,59	54 047	68 461	75 667	97 285	1 533,658	1 942,632	2 147,121	2 760,584
	24,70	139,7	36,76	60 090	76 114	84 126	108 162	1 740,523	2 204,659	2 436,732	3 132,941
6 5/8	25,20	168,28	37,50	75 609	96 971	107 177	137 799	1 723,535	2 183,143	2 412,947	3 102,359
	27,70	168,28	41,22	81 609	104 821	115 855	148 956	1 879,013	2 380,083	2 630,621	3 382,223

[a] Labels are for information and assistance in ordering.

[b] Based on the shear strength equal to 57,7 % of minimum yield strength and nominal wall thickness. Minimum torsional yield strength calculated from equation E.15

[c] Minimum tensile strength calculated from equation E.13

Table A.5 — Used drill pipe collapse and internal pressure data API premium class

Labels [a]		Outside diameter D mm	Nominal linear mass T&C kg/m	Collapse pressure based on minimum values[b] MPa				Internal pressure at minimum yield strength[b] MPa			
1	2			E75	X95	G105	S135	E75	X95	G105	S135
1	2	3	4	5	6	7	8	9	10	11	12
2 3/8	4,85	60,33	7,22	58,757	70,058	75,236	88,880	66,190	83,840	92,666	119,141
	6,65	60,33	9,90	92,238	116,832	129,132	166,026	97,540	123,554	136,558	175,575
2 7/8	6,85	73,03	10,19	52,676	62,170	66,417	77,125	62,446	79,104	87,426	112,405
	10,40	73,03	15,48	98,064	124,216	137,288	176,520	104,180	131,959	145,845	187,517
3 1/2	9,50	88,9	14,14	48,774	57,116	60,763	69,589	60,046	76,056	84,061	108,075
	13,30	88,9	19,79	82,841	104,924	115,970	149,106	86,991	110,192	121,789	156,587
	15,50	88,9	23,07	99,781	126,388	139,688	179,602	106,138	134,441	148,596	191,054
4	11,85	101,6	17,63	39,328	44,871	47,071	51,331	54,193	68,644	75,870	97,547
	14,00	101,6	20,83	62,136	74,429	80,131	95,396	68,258	86,460	95,561	122,865
	15,70	101,6	23,36	75,249	95,320	104,731	128,194	78,600	99,560	110,040	141,480
4 1/2	13,75	114,3	20,46	32,309	35,784	36,901	40,734	49,828	63,115	69,754	89,687
	16,60	114,3	24,70	51,883	61,143	65,273	75,594	61,963	78,483	86,743	111,530
	20,00	114,3	29,76	75,670	95,844	105,835	129,663	79,062	100,139	110,682	142,308
	22,82	114,3	33,96	87,253	110,523	122,161	157,063	91,928	116,446	128,704	165,474
5	16,25	127,0	24,18	30,957	34,026	34,936	39,031	48,980	62,039	68,575	88,163
	19,50	127,0	29,02	48,546	56,820	60,433	69,148	59,902	75,877	83,861	107,820
	25,60	127,0	38,10	79,000	100,071	110,606	141,411	82,737	104,800	115,832	148,927
5 1/2	19,20	139,7	28,57	25,759	28,475	29,896	32,502	45,733	57,923	64,025	82,317
	21,90	139,7	32,59	39,507	45,106	47,333	51,683	54,303	68,789	76,028	97,747
	24,70	139,7	36,76	52,641	62,129	66,369	77,063	62,432	79,076	87,398	112,371
6 5/8	25,20	168,28	37,50	20,209	22,422	23,118	23,642	41,210	52,200	57,695	74,181
	27,70	168,28	41,22	24,925	27,779	29,110	31,454	45,209	57,268	63,294	81,379

[a] Labels are for information and assistance in ordering.

[b] Calculations are based on formulas in API Bulletin 5C3.

Table A.6 — Used drill pipe torsional and tensile data API Class 2

Labels [a]		Outside diameter D mm	Nominal linear mass T&C kg/m	Torsional data [b] torsional yield strength N·m				Tensile data based on minimum values load at the minimum yield strength [c] kN			
1	2			E75	X95	G105	S135	E75	X95	G105	S135
1	2	3	4	5	6	7	8	9	10	11	12
2 3/8	4,85	60,33	7,22	4 371	5 536	6 119	7 866	296,634	375,737	415,286	533,942
	6,65	60,33	9,90	5 600	7 094	7 839	10 079	413,111	523,271	578,353	743,596
2 7/8	6,85	73,03	10,19	7 435	9 418	10 409	13 383	412,799	522,884	577,922	743,044
	10,40	73,03	15,48	10 292	13 036	14 408	18 525	638,573	808,860	894,004	1 149,434
3 1/2	9,50	88,9	14,14	13 032	16 508	18 245	23 458	590,693	748,209	826,969	1 063,245
	13,30	88,9	19,79	16 765	21 236	23 472	30 178	815,795	1033,340	1142,112	1 468,429
	15,50	88,9	23,07	18 748	23 747	26 247	33 746	960,669	1216,847	1344,938	1 729,206
4	11,85	101,6	17,63	18 007	22 809	25 210	32 414	703,406	890,983	984,770	1 266,133
	14,00	101,6	20,83	21 338	27 028	29 874	38 409	864,570	1 095,121	1 210,397	1 556,219
	15,70	101,6	23,36	23 476	29 736	32 866	42 258	977,443	1 238,096	1 368,420	1 759,396
4 1/2	13,75	114,3	20,46	24 018	30 423	33 626	43 233	824,651	1 044,563	1 154,514	1 484,376
	16,60	114,3	24,70	28 347	35 906	39 686	51 025	1 004,280	1 272,089	1 405,994	1 807,704
	20,00	114,3	29,76	33 552	42 499	46 972	60 394	1 243,287	1 574,826	1 740,598	2 237,914
	22,82	114,3	33,96	36 825	46 646	51 556	66 286	1 412,297	1 788,910	1 977,217	2 542,137
5	16,25	127,0	24,18	32 504	41 173	45 507	58 509	1 002,256	1 269,523	1 403,156	1 804,057
	19,50	127,0	29,02	37 930	48 045	53 102	68 274	1 202,942	1 523,730	1 684,119	2 165,297
	25,60	127,0	38,10	47 382	60 018	66 335	85 288	1 595,715	2 021,236	2 233,999	2 872,284
5 1/2	19,20	139,7	28,57	40 957	51 878	57 339	73 721	1 138,540	1 442,149	1 593,954	2 049,371
	21,90	139,7	32,59	46 887	59 390	65 641	84 396	1 332,389	1 687,695	1 865,344	2 398,303
	24,70	139,7	36,76	52 040	65 919	72 858	93 673	1 510,318	1 913,074	2 114,449	2 718,575
6 5/8	25,20	168,28	37,50	65 753	83 288	92 055	118 356	1 500,101	1 900,129	2 100,144	2 700,186
	27,70	168,28	41,22	70 920	89 832	99 288	127 657	1 634,521	2 070,394	2 288,330	2 942,139

[a] Labels are for information and assistance in ordering.

[b] Based on the shear strength equal to 57,7 % of minimum yield strength and nominal wall thickness. Minimum torsional yield strength calculated from equation E.15

[c] Minimum tensile strength calculated from equation E.13

Table A.7 — Used drill pipe collapse and internal pressure data API Class 2

Labels [a]		Outside diameter D mm	Nominal linear mass T&C kg/m	Collapse pressure based on minimum values[b] MPa				Internal pressure at minimum yield strength[b] MPa			
1	2			E75	X95	G105	S135	E75	X95	G105	S135
1	2	3	4	5	6	7	8	9	10	11	12
2 3/8	4,85	60,33	7,22	47,243	55,130	58,543	66,631	57,916	73,360	81,082	104,249
	6,65	60,33	9,90	83,689	106,007	117,163	150,644	85,350	108,110	119,493	153,629
2 7/8	6,85	73,03	10,19	41,748	48,008	50,573	56,006	54,641	69,216	76,497	98,354
	10,40	73,03	15,48	89,204	112,991	124,885	160,565	91,156	115,460	127,615	164,081
3 1/2	9,50	88,9	14,14	38,225	43,444	45,478	49,208	52,538	66,548	73,553	94,568
	13,30	88,9	19,79	74,863	94,824	103,711	126,836	76,118	96,416	106,565	137,013
	15,50	88,9	23,07	90,832	115,046	127,160	163,488	92,872	117,638	130,021	167,170
4	11,85	101,6	17,63	29,723	32,419	33,619	37,480	47,422	60,067	66,390	85,357
	14,00	101,6	20,83	50,297	59,088	62,977	72,533	59,729	75,656	83,620	107,510
	15,70	101,6	23,36	65,714	79,069	85,316	102,318	68,775	87,115	96,285	123,795
4 1/2	13,75	114,3	20,46	23,421	26,510	27,689	29,558	43,596	55,227	61,039	78,476
	16,60	114,3	24,70	41,031	47,077	49,539	54,627	54,213	68,672	75,904	97,588
	20,00	114,3	29,76	66,403	79,965	86,322	103,649	69,175	87,625	96,851	124,519
	22,82	114,3	33,96	79,000	100,071	110,606	141,411	80,441	101,898	112,612	144,790
5	16,25	127,0	24,18	22,580	25,483	26,545	28,027	42,858	54,289	59,998	77,145
	19,50	127,0	29,02	38,018	43,175	45,174	48,808	52,414	66,390	73,381	94,348
	25,60	127,0	38,10	71,278	87,150	94,355	114,363	72,395	91,700	101,353	130,311
5 1/2	19,20	139,7	28,57	19,547	21,567	22,167	22,511	40,017	50,683	56,020	72,030
	21,90	139,7	32,59	29,882	32,633	33,777	37,680	47,519	60,191	66,528	85,529
	24,70	139,7	36,76	41,713	47,967	50,532	55,951	54,627	69,189	76,477	98,326
6 5/8	25,20	168,28	37,50	15,355	16,154	16,175	16,175	36,060	45,678	50,483	64,907
	27,70	168,28	41,22	19,064	20,939	21,463	21,705	39,555	50,104	55,379	71,202

[a] Labels are for information and assistance in ordering.

[b] Calculations are based on formulas in API Bulletin 5C3.

Table A.8 — Mechanical properties of new tool joints and New Grade E75 drill pipe

1	2	3	4	5	6	7	8	9	10	11	12	13	14
										Mechanical properties			
Drill pipe data						Tool joint data				Tensile yield		Torsional yield	
Labels		Nominal size in	Nominal weight	Approx. weight[a]	Type upset	Conn.	OD	ID	Drift dia.[b]	Pipe[c]	Tool Joint[d]	Pipe[e]	Tool Joint[f]
1	2	mm	Kg/m	Kg/m			mm	mm	mm	N	N	N·m	N·m
2 3/8	4,85	60,33	7,22	7,83	EU	NC 26	85,73	44,45	41,28	435 112	1 395 323	6 458	9 321 b
		60,33	7,22	7,37	EU	OH	79,38	50,80	45,90	435 112	918 184	6 458	6 136 p
		60,33	7,22	7,52	EU	SLH90	82,55	50,80	46,99	435 112	901 521	6 458	6 881 p
		60,33	7,22	7,66	EU	WO	85,73	50,80	45,90	435 112	870 415	6 458	5 742 p
	6,65	60,33	9,90	10,40	EU	NC 26	85,73	44,45	41,28	614 807	1 395 323	8 474	9 321 b
		60,33	9,90	10,25	EU	OH	82,55	44,45	41,28	614 807	1 310 535	8 474	8 548 b
		60,33	9,90	9,99	IU	PAC	73,03	34,93	31,75	614 807	1 060 919	8 474	6 334 p
		60,33	9,90	10,09	EU	SLH90	82,55	50,80	42,42	614 807	902 322	8 474	6 881 p
2 7/8	6,85	73,03	10,19	11,16	EU	NC31	104,78	53,98	50,80	604 522	1 988 934	10 959	15 985 p
		73,03	10,19	10,31	EU	OH	95,25	61,91	57,23	604 522	996 121	10 959	7 408 p
		73,03	10,19	10,49	EU	SLH90	98,43	61,91	58,32	604 522	1 160 021	10 959	10 186 p
		73,03	10,19	10,88	EU	WO	104,78	61,91	57,23	604 522	1 234 617	10 959	9 511 p
	10,40	73,03	15,48	16,18	EU	NC31	104,78	53,98	49,86	953 450	1 988 934	15 665	15 985 p
		73,03	15,48	15,76	EU	OH	98,43	54,77	49,86	953 450	1 537 154	15 665	11 740 p
		73,03	15,48	15,28	IU	PAC	79,38	38,10	34,93	953 450	1 214 089	15 665	7 736 p
		73,03	15,48	15,76	EU	SLH90	98,43	54,77	50,95	953 450	1 702 624	15 665	15 222 p
		73,03	15,48	16,65	IU	XH	107,95	47,63	44,45	953 450	2 246 592	15 665	17 745 p
		73,03	15,48	15,40	IU	NC26	85,73	44,45	41,28	953 450	1 395 323	15 665	9 321 B
3 1/2	9,50	88,90	14,14	15,74	EU	NC38	120,65	68,26	65,10	864 129	2 612 476	19 179	24 501 p
		88,90	14,14	14,64	EU	OH	114,30	76,20	71,22	864 129	1 744 019	19 179	16 003 p
		88,90	14,14	14,87	EU	SLH90	117,48	76,20	72,31	864 129	1 631 185	19 179	16 891 p
		88,90	14,14	15,09	EU	WO	120,65	76,20	71,22	864 129	1 867 350	19 179	17 250 p
	13,30	88,90	19,79	21,38	EU	H90	133,35	69,85	66,52	1 207 999	2 953 842	25 152	31 784 p
		88,90	19,79	20,73	EU	NC38	120,65	68,26	62,41	1 207 999	2 612 476	25 152	24 501 p
		88,90	19,79	20,46	EU	OH	120,65	68,26	61,32	1 207 999	2 489 145	25 152	23 275 p
		88,90	19,79	19,94	IU	NC31	104,78	53,98	50,80	1 207 999	1 988 934	25 152	15 985 P
		88,90	19,79	20,70	EU	XH	120,65	61,91	58,75	1 207 999	2 539 663	25 152	22 869 p
	15,50	88,90	23,07	24,61	EU	NC38	127,00	65,09	61,32	1 435 775	2 887 599	28 589	27 245 p
4	11,85	101,60	17,63	19,35	IU	H90	139,70	71,44	68,28	1 026 449	4 064 376	26 403	47 871 p
		101,60	17,63	20,12	EU	NC46	152,40	82,55	79,38	1 026 449	4 008 578	26 403	45 051 p
		101,60	17,63	18,01	EU	OH	133,35	88,11	83,49	1 026 449	2 763 934	26 403	29 696 p
		101,60	17,63	19,21	EU	WO	146,05	87,31	84,15	1 026 449	3 482 900	26 403	38 835 p
	14,00	101,60	20,83	22,38	IU	NC40	133,35	71,44	68,28	1 269 340	3 165 404	31 574	31 562 p
		101,60	20,83	22,96	IU	H90	139,70	71,44	68,28	1 269 340	4 064 376	31 574	47 871 p
		101,60	20,83	23,59	EU	NC46	152,40	82,55	79,38	1 269 340	4 008 578	31 574	45 051 p
		101,60	20,83	22,35	EU	OH	139,70	82,55	79,38	1 269 340	3 380 093	31 574	36 688 p
		101,60	20,83	21,36	IU	SH	117,48	65,09	61,93	1 269 340	2 277 645	31 574	20 373 p

1	2	3	4	5	6	7	8	9	10	11	12	13	14
										Mechanical properties			
Drill pipe data						**Tool joint data**				**Tensile yield**		**Torsional yield**	
Labels		Nominal size in	Nominal weight	Approx. weight[a]	Type upset	Conn.	OD	ID	Drift dia.[b]	Pipe[c]	Tool Joint[d]	Pipe[e]	Tool Joint[f]
1	2	mm	Kg/m	Kg/m			mm	mm	mm	N	N	N·m	N·m
	15,70	101,60	23,36	25,00	IU	NC40	133,35	68,26	65,10	1 441 749	3 453 626	34 994	34 615 p
		101,60	23,36	25,43	IU	H90	139,70	71,44	68,28	1 441 749	4 064 376	34 994	47 871 p
		101,60	23,36	26,10	EU	NC46	152,40	82,55	78,61	1 441 749	4 008 578	34 994	45 051 p
4 1/2	13,75	114,30	20,46	22,66	IU	H90	152,40	82,55	79,38	1 201 171	4 174 225	35 125	52 259 p
		114,30	20,46	22,86	EU	NC50	168,28	95,25	92,08	1 201 171	4 177 307	35 125	50 530 p
		114,30	20,46	20,89	EU	OH	146,05	100,81	95,76	1 201 171	2 468 069	35 125	28 036 p
		114,30	20,46	21,98	EU	WO	155,58	98,43	95,25	1 201 171	3 774 846	35 125	45 409 p
	16,60	114,30	24,70	27,00	IEU	FH	152,40	76,20	73,03	1 470 395	4 342 159	41 769	46 595 p
		114,30	24,70	26,67	IEU	H90	152,40	82,55	79,38	1 470 395	4 174 225	41 769	52 259 p
		114,30	24,70	26,71	EU	NC50	168,28	95,25	92,08	1 470 395	4 177 307	41 769	50 530 p
		114,30	24,70	25,40	EU	OH	149,23	95,25	92,08	1 470 395	3 175 937	41 769	36 520 p
		114,30	24,70	24,99	IEU	NC38	127,00	68,26	65,10	1 470 395	2 612 476	41 769	24 501 p
		114,30	24,70	27,34	IEU	NC46	158,75	82,55	79,38	1 470 395	4 008 578	41 769	45 051 p
	20,00	114,30	29,76	32,20	IEU	FH	152,40	76,20	73,03	1 834 260	4 342 159	50 031	46 595 p
		114,30	29,76	32,20	IEU	H90	152,40	76,20	73,03	1 834 260	4 829 279	50 031	60 941 p
		114,30	29,76	32,13	EU	NC50	168,28	92,08	87,68	1 834 260	4 563 787	50 031	55 473 p
		114,30	29,76	32,87	IEU	NC46	158,75	76,20	73,03	1 834 260	4 663 632	50 031	52 874 p
	22,82	114,30	33,96	35,88	EU	NC50	168,28	92,08	87,68	2 096 176	4 563 787	55 469	55 473 p
		114,30	33,96	36,55	IEU	NC46	158,75	76,20	73,03	2 096 176	4 663 632	55 469	52 874 p
5	19,50	127,00	29,02	33,16	IEU	5 1/2 FH	177,80	95,25	92,08	1 759 694	6 442 836	55 815	85 285 b
		127,00	29,02	31,03	IEU	NC50	168,28	95,25	92,08	1 759 694	4 177 303	55 815	50 530 p
	25,60	127,00	38,10	42,07	IEU	5 1/2 FH	177,80	88,90	85,73	2 358 198	7 202 699	70 851	85 285 p
		127,00	38,10	39,96	IEU	NC50	168,28	88,90	85,73	2 358 198	4 937 171	70 851	60 274 p
5 1/2	21,90	139,70	32,59	35,39	IEU	FH	177,80	101,60	98,43	1 944 389	5 630 568	68 754	75 501 p
	24,70	139,70	36,76	39,14	IEU	FH	177,80	101,60	98,43	2 211 754	5 630 568	76 704	75 501 p
6 5/8	25,20	168,28	37,50	40,60	IEU	FH	203,20	127,00	123,83	2 177 245	6 439 678	95 694	99 278 p
	27,70	168,28	41,22	43,25	IEU	FH	203,20	127,00	123,83	2 376 231	6 439 678	103 442	99 278 p

[a] Tool joint plus drill pipe, for range 2 steel pipe (See appendix A for method of calculation).

[b] See subclause 4.4.

[c] The tensile yield strength of Grade E drill pipe is based on 517,1 MPa minimum yield strength.

[d] The tensile strength of the tool joint pin is based on 827,4 MPa minimum yield and the cross sectional area at the root of the thread 15,875 mm from the shoulder.

[e] The torsional yield strength is based on a shear strength of 57,7 % of the minimum yield strength.

[f] p = pin limited yield; b = box limited yield; P or B indicates that tool joint could not meet 80 % of tube torsional yield.

Table A.9 — Mechanical properties of new tool joints and new high strength drill pipe

1	2	3	4	5	6	7	8	9	10	11	12	13	14
Drill pipe data						Tool joint data				Mechanical properties			
Labels [a]		Nominal size	Nominal linear mass	Approx. weight [b]	Type upset & pipe grade	Conn.	OD	ID	Drift diameter [c]	Tensile yield		Torsional yield	
										Pipe [d]	Tool joint [e]	Pipe [d]	Tool joint [f]
1	2	mm	kg/m	kg/m			mm	mm	mm	kN	kN	N·m	N·m
2 3/8	6,65	60,33	9,90	10,58	EU-X95	NC26	85,73	44,45	41,28	778,76	1 395,32	10 734	9 321 b
		60,33	9,90	10,40	EU-X95	SLH90	82,55	46,04	42,42	778,76	1 202,01	10 734	9 304 p
		60,33	9,90	10,58	EU-G105	NC26	85,73	44,45	41,28	860,73	1 395,32	11 865	9 321 b
		60,33	9,90	10,40	EU-G105	SLH90	82,55	46,04	42,42	860,73	1 202,01	11 865	9 304 p
2 7/8	10,40	73,03	15,48	16,50	EU-X95	NC31	104,78	50,80	47,63	1 207,71	2 205,10	19 842	17 840 p
		73,03	15,48	16,30	EU-X95	SLH90	101,60	50,80	47,63	1 207,71	1 974,88	19 842	17 787 p
		73,03	15,48	16,50	EU-G105	NC31	104,78	50,80	47,63	1 334,83	2 205,10	21 932	17 840 p
		73,03	15,48	16,30	EU-G105	SLH90	101,60	50,80	47,63	1 334,83	1 974,88	21 932	17 787 p
		73,03	15,48	17,19	EU-S135	NC31	111,13	41,28	38,10	1 716,21	2 774,10	28 198	22 790 p
		73,03	15,48	16,76	EU-S135	SLH90	104,78	41,28	38,10	1 716,21	2 544,78	28 198	23 225 p
3 1/2	13,30	88,90	19,79	21,73	EU-X95	H90	133,35	69,85	66,52	1 530,14	2 953,84	31 859	31 784 p
		88,90	19,79	21,76	EU-X95	NC38	127,00	65,09	61,93	1 530,14	2 887,60	31 859	27 245 p
		88,90	19,79	20,92	EU-X95	SLH90	120,65	65,09	61,93	1 530,14	2 651,43	31 859	28 078 p
		88,90	19,79	21,89	EU-G105	NC38	127,00	61,91	58,75	1 691,20	3 149,62	35 213	31 231 p
		88,90	19,79	20,92	EU-G105	SLH90	120,65	65,09	61,93	1 691,20	2 651,43	35 213	28 078 p
		88,90	19,79	22,20	EU-S135	NC38	127,00	53,98	50,80	2 174,40	3 747,36	45 273	35 933 p
		88,90	19,79	21,80	EU-S135	SLH90	127,00	53,98	50,80	2 174,40	3 511,20	45 273	37 704 p
		88,90	19,79	22,52	EU-S135	NC40	136,53	61,91	58,75	2 174,40	3 990,77	45273	40 355 p
	15,50	88,90	23,07	25,03	EU-X95	NC38	127,00	61,91	58,75	1 818,65	3 149,62	36 211	29 875 p
		88,90	23,07	25,34	EU-G105	NC38	127,00	53,98	50,80	2 010,09	3 747,36	40 024	35 933 p
		88,90	23,07	25,25	EU-G105	NC40	133,35	65,09	61,93	2 010,09	3 728,75	40 024	37 547 p
		88,90	23,07	26,15	EU-S135	NC40	139,70	57,15	53,98	2 584,40	4 359,24	51 459	44 326 p
4	14,00	101,60	20,83	22,83	IU-X95	NC40	133,35	68,26	65,10	1 607,83	3 453,63	39 994	34 615 p
		101,60	20,83	23,26	IU-X95	H90	139,70	71,44	68,28	1 607,83	4 064,38	39 994	47 871 p
		101,60	20,83	24,09	EU-X95	NC46	152,40	82,55	79,38	1 607,83	4 008,58	39 994	45 051 p
		101,60	20,83	23,68	IU-G105	NC40	139,70	61,91	58,75	1 777,07	3 990,77	44 204	40 355 p
		101,60	20,83	23,26	IU-G105	H90	139,70	71,44	68,28	1 777,07	4 064,38	44 204	47 871 p
		101,60	20,83	24,09	EU-G105	NC46	152,40	82,55	79,38	1 777,07	4 008,58	44 204	45 051 p

4	14,00	101,60	20,83	24,09	IU-S135	NC40	139,70	50,80	47,63	2 284,81	4 804,68	56 833	49 165 p
		101,60	20,83	23,26	IU-S135	H90	139,70	71,44	68,28	2 284,81	4 064,38	56 833	47 871 p
		101,60	20,83	24,44	EU-S135	NC46	152,40	76,20	73,03	2 284,81	4 663,63	56 833	52 874 p
	15,70	101,60	23,36	26,07	IU-X95	NC40	139,70	61,91	58,75	1 826,22	3 990,77	44 324	40 355 p
		101,60	23,36	25,64	IU-X95	H90	139,70	71,44	68,28	1 826,22	4 064,38	44 324	47 871 p
		101,60	23,36	26,49	EU-X95	NC46	152,40	82,55	79,38	1 826,22	4 008,58	44 324	45 051 p
		101,60	23,36	26,07	IU-G105	NC40	139,70	61,91	58,75	2 018,45	3 990,77	48 991	40 355 p
		101,60	23,36	25,64	IU-G105	H90	139,70	71,44	68,28	2 018,45	4 064,38	48 991	47 871 p
		101,60	23,36	26,49	EU-G105	NC46	152,40	82,55	79,38	2 018,45	4 008,58	48 991	45 051 p
		101,60	23,36	26,82	EU-S135	NC46	152,40	76,20	73,03	2 595,15	4 663,63	62 989	52 874 p
4 1/2	16,60	114,30	24,70	27,28	IEU-X95	FH	152,40	76,20	73,03	1 862,50	4 342,16	52 907	46 595 p
		114,30	24,70	26,95	IEU-X95	H90	152,40	82,55	79,38	1 862,50	4 174,23	52 907	52 259 p
		114,30	24,70	27,32	EU-X95	NC50	168,28	95,25	92,08	1 862,50	4 177,30	52 907	50 530 p
		114,30	24,70	27,96	IEU-X95	NC46	158,75	76,20	73,03	1 862,50	4 663,63	52 907	52 874 p
		114,30	24,70	27,28	IEU-G105	FH	152,40	76,20	66,68	2 058,55	4 342,16	58 476	46 595 p
		114,30	24,70	27,28	IEU-G105	H90	152,40	76,20	79,38	2 058,55	4 829,28	58 476	60 941 p
		114,30	24,70	27,32	EU-G105	NC50	168,28	95,25	92,08	2 058,55	4 177,30	58 476	50 530 p
		114,30	24,70	27,96	IEU-G105	NC46	158,75	76,20	73,03	2 058,55	4 663,63	58 476	52 874 p
		114,30	24,70	28,56	IEU-S135	FH	158,75	63,50	60,33	2 646,71	5 495,05	75 184	59 946 p
		114,30	24,70	27,28	IEU-S135	H90	152,40	76,20	73,03	2 646,71	4 829,28	75 184	60 941 p
		114,30	24,70	27,71	EU-S135	NC50	168,28	88,90	85,73	2 646,71	4 937,17	75 184	60 274 p
		114,30	24,70	28,28	IEU-S135	NC46	158,75	69,85	66,68	2 646,71	5 266,29	75 184	60 143 p
	20,00	114,30	29,76	33,32	IEU-X95	FH	152,40	63,50	60,33	2 323,40	5 495,05	63 372	59 946 p
		114,30	29,76	32,41	IEU-X95	H90	152,40	82,55	79,38	2 323,40	4 174,23	63 372	52 259 p
		114,30	29,76	32,86	EU-X95	NC50	168,28	88,90	85,73	2 323,40	4 937,17	63 372	60 274 p
		114,30	29,76	33,74	IEU-X95	NC46	158,75	69,85	66,68	2 323,40	5 266,29	63 372	60 143 p
		114,30	29,76	33,32	IEU-G105	FH	152,40	63,50	60,33	2 567,96	5 495,05	70 043	59 946 p
		114,30	29,76	32,74	IEU-G105	H90	152,40	76,20	73,03	2 567,96	4 829,28	70 043	60 941 p
		114,30	29,76	32,86	EU-G105	NC50	168,28	88,90	85,73	2 567,96	4 937,17	70 043	60 274 p
		114,30	29,76	34,02	IEU-G105	NC46	158,75	63,50	60,33	2 567,96	5 816,53	70 043	66 838 p
		114,30	29,76	34,27	EU-S135	NC50	168,28	76,20	73,03	3 301,67	6 299,68	90 055	78 006 p
		114,30	29,76	34,27	IEU-S135	NC46	158,75	57,15	53,98	3 301,67	6 314,37	90 055	72 943 p

1	2	3	4	5	6	7	8	9	10	11	12	13	14
Drill pipe data						**Tool joint data**				**Mechanical properties**			
Labels [a]		**Nominal size**	**Nominal linear mass**	**Approx. weight** [b]	**Type upset & pipe grade**	**Conn.**	**OD**	**ID**	**Drift Diameter** [c]	**Tensile yield**		**Torsional yield**	
										Pipe [d]	**Tool Joint** [e]	**Pipe** [d]	**Tool Joint** [f]
1	2	mm	kg/m	kg/m			mm	mm	mm	kN	kN	N·m	N·m
4 1/2	22,82	114,30	33,96	37,40	IEU-X95	FH	158,75	57,15	53,98	2 655,16	5 992,89	70 260	65 787 p
		114,30	33,96	36,07	EU-X95	NC50	168,28	88,90	85,73	2 655,16	4 937,17	70 260	60 274 p
		114,30	33,96	36,86	IEU-X95	NC46	158,75	69,85	66,68	2 655,16	5 266,29	70 260	60 143 p
		114,30	33,96	36,79	EU-G105	NC50	168,28	82,55	79,38	2 934,65	5 644,63	77 656	69 441 p
		114,30	33,96	37,14	IEU-G105	NC46	158,75	63,50	60,33	2 934,65	5 816,53	77 656	66 838 p
		114,30	33,96	37,81	EU-S135	NC50	168,28	69,85	66,68	3 773,12	6 902,33	99 844	85 949 p
5	19,50	127,00	29,02	33,66	IEU-X95	5 ½ FH	177,80	95,25	92,08	2 228,95	6 442,84	70 698	85 285 b
		127,00	29,02	32,64	IEU-X95	H90	165,10	82,55	79,38	2 228,95	5 232,29	70 698	69 445 p
		127,00	29,02	31,92	IEU-X95	NC50	168,28	88,90	85,73	2 228,95	4 937,17	70 698	60 274 p
		127,00	29,02	33,66	IEU-G105	5 ½ FH	177,80	95,25	92,08	2 463,57	6 442,84	78 140	85 285 b
		127,00	29,02	32,96	IEU-G105	H90	165,10	76,20	73,03	2 463,57	5 887,34	78 140	78 648 p
		127,00	29,02	32,64	IEU-G105	NC50	168,28	82,55	79,38	2 463,57	5 644,63	78 140	69 441 p
		127,00	29,02	34,94	IEU-S135	5 ½ FH	184,15	88,90	85,73	3 167,45	7 202,70	100 466	97 908 p
		127,00	29,02	33,65	IEU-S135	NC50	168,28	69,85	66,68	3 167,45	6 902,33	100 466	85 949 p
	25,60	127,00	38,10	42,55	IEU-X95	5 ½ FH	177,80	88,90	85,73	2 987,05	7 202,70	89 744	85 285 b
		127,00	38,10	41,48	IEU-X95	NC50	168,28	76,20	73,03	2 987,05	6 299,68	89 744	78 006 b
		127,00	38,10	43,39	IEU-G105	5 ½ FH	184,15	88,90	85,73	3 301,48	7 202,70	99 190	97 908 p
		127,00	38,10	42,14	IEU-G105	NC50	168,28	69,85	66,68	3 301,48	6 902,33	99 190	85 949 b
		127,00	38,10	43,80	IEU-S135	5 ½ FH	184,15	82,55	79,38	4 244,76	7 910,16	127 531	106 725 b
5 ½	21,90	139,70	32,59	36,50	IEU-X95	FH	177,80	95,25	92,08	2 462,90	6 442,84	87 088	85 285 b
		139,70	32,59	36,91	IEU-X95	H90	177,80	88,90	79,38	2 462,90	5 644,25	87 088	78 682 p
		139,70	32,59	37,77	IEU-G105	FH	184,15	88,90	85,73	2 722,15	7 202,70	96 255	97 908 p
		139,70	32,59	39,44	IEU-S135	FH	190,50	76,20	73,03	3 499,90	8 565,21	123 756	117 638 p
	24,70	139,70	36,76	41,45	IEU-X95	FH	184,15	88,90	85,73	2 801,55	7 202,70	97 158	97 908 p
		139,70	36,76	41,45	IEU-G105	FH	184,15	88,90	85,73	3 096,46	7 202,70	107 386	97 908 p
		139,70	36,76	41,33	IEU-S135	FH	190,50	76,20	73,03	3 981,15	8 565,21	138 067	117 638 p

See end of table for notes.

1	2	3	4	5	6	7	8	9	10	11	12	13	14
Drill pipe data						**Tool joint data**				**Mechanical properties**			
Labels [a]		**Nominal size**	**Nominal linear mass**	**Approx. weight** [b]	**Type upset & pipe grade**	**Conn.**	**OD**	**ID**	**Drift diameter** [c]	**Tensile yield**		**Torsional yield**	
										Pipe [d]	**Tool joint** [e]	**Pipe** [d]	**Tool joint** [f]
1	2	mm	kg/m	kg/m			mm	mm	mm	k N	k N	N·m	N·m
6 5/8	25,20	168,28	37,50	40,40	IEU-X95	FH	203,20	127,00	123,83	2 757,84	6 442,88	121 213	99 278 p
		168,28	37,50	41,97	IEU-G105	FH	209,55	120,65	117,48	3 048,14	7 464,76	133 971	115 878 p
		168,28	37,50	44,09	IEU-S135	FH	215,90	107,95	104,78	3 919,04	9 351,32	172 249	146 907 p
	27,70	168,28	41,22	44,81	IEU-X95	FH	209,55	120,65	117,48	3 009,90	7 464,76	131 026	115 878 p
		168,28	41,22	44,81	IEU-G105	FH	209,55	120,65	117,48	3 323,93	7 464,76	144 819	115 878 p
		168,28	41,22	46,94	IEU-S135	FH	215,90	107,95	104,78	4 277,22	9 351,32	186 194	146 907 p

[a] Labels are for information and assistance in ordering.

[b] Tool joint plus drill pipe, for Range 2 steel pipe (see appendix A for method of calculation).

[c] See subclause 4.4.

[d] The torsional yield strength is based on a shear strength of 57,7 % of the minimum yield strength.

[e] The tensile strength of the tool joint pin is based on 827,4 MPa minimum yield and the cross sectional area at the root of the thread 15,875 mm from the shoulder.

[f] p = pin limited yield; b = box limited yield; P or B indicates that tool joint could not meet 80 % of tube torsional yield.

Table A.10 — – Recommended minimum OD and make-up torque of weld-on type tool joints Based on torsional strength of box and drill pipe

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
		Drill pipe data			New tool joint data				Premium class			Class 2		
Labels[a]		Nom OD	Nom linear mass	Type upset and pipe grade	Connection	New OD	New ID	Make-up torque[d]	Min OD tool joint[c,f]	Min box shoulder with eccentric wear[e]	Make-up torque for min OD tool joint[d]	Min OD tool joint[c,f]	Min box shoulder with eccentric wear[e]	Make-up torque for min OD tool joint[d]
1	2	mm	Kg/m			mm	mm	N·m	mm	mm	N·m	mm	mm	N·m
2 3/8	4,85	60,3	7,22	EU-E75	NC26	85,7	44,5	5 593 B	79,4	1,2	2 637	78,6	0,8	2 290
		60,3	7,22	EU-E75	W.O.	85,7	50,8	3 445 P	77,8	1,6	2 704	77,0	1,2	2 367
		60,3	7,22	EU-E75	2 3/8 OHLW	79,4	50,8	3 682 P	76,2	1,6	2 336	75,4	1,2	2 008
		60,3	7,22	EU-E75	2 3/8 SL-H90	82,6	50,8	4 124 P	75,4	1,6	2 706	74,6	1,2	2 340
	6,65	60,3	9,90	IU-E75	2 3/8 PAC[b]	73,0	34,9	3 800 P	70,6	3,6	3 329	69,1	2,8	2 786
		60,3	9,90	EU-E75	NC26	85,7	44,5	5 593 B	81,0	2,0	3 345	80,2	1,6	2 988
		60,3	9,90	EU-E75	2 3/8 SL-H90	82,6	50,8	4 124 P	77,0	2,4	3 456	75,4	1,6	2 706
		60,3	9,90	EU-E75	2 3/8 OHSW	82,6	44,5	5 129 B	77,8	2,4	3 004	77,0	2,0	2 667
		60,3	9,90	EU-X95	NC26	85,7	44,5	5 593 B	82,6	2,8	4 074	81,8	2,4	3 707
		60,3	9,90	EU-G105	NC26[b]	85,7	44,5	5 593 B	83,3	3,2	4 446	82,6	2,8	4 074
2 7/8	6,85	73,0	10,19	EU-E75	NC31	104,8	54,0	9 591 P	93,7	2,0	4 276	92,9	1,6	3 802
		73,0	10,19	EU-E75	2 7/8 WO	104,8	61,9	5 707 P	92,1	2,0	4 360	91,3	1,6	3 899
		73,0	10,19	EU-E75	2 7/8 OHLW[b]	95,3	61,9	4 461 P	88,9	2,8	4 461	87,3	2,0	3 802
		73,0	10,19	EU-E75	2 7/8 SL-H90	98,4	61,9	6 107 P	88,9	2,4	4 606	87,3	1,6	3 615
	10,40	73,0	15,48	EU-E75	NC31	104,8	54,0	9 591 P	96,8	3,6	6 233	95,3	2,8	5 243
		73,0	15,48	IU-E75	2 7/8 XH	108,0	47,6	10 647 P	94,5	3,6	5 907	92,9	2,8	4 968
		73,0	15,48	IU-E75	NC26[b]	85,7	44,5	5 593 B	85,7	4,4	5 593	84,9	4,0	5 205
		73,0	15,48	EU-E75	2 7/8 OHSW[b]	98,4	54,8	7 042 P	91,3	4,0	5 981	90,5	2,8	5 530
		73,0	15,48	EU-E75	2 7/8 SL-H90	98,4	54,8	9 127 P	91,3	3,6	6 140	89,7	2,8	5 111
		73,0	15,48	IU-E75	2 7/8 PAC[b]	79,4	38,1	4 642 P	79,4	6,0	4 642	79,4	6,0	4 642
		73,0	15,48	EU-X95	NC31	104,8	50,8	10 704 P	99,2	4,8	7 763	97,6	4,0	6 737
		73,0	15,48	EU-X95	2 7/8 SL-H90[b]	98,4	54,8	9 127 P	93,7	4,8	7 731	92,1	4,0	6 664
		73,0	15,48	EU-G105	NC31	104,8	50,8	10 704 P	100,0	5,2	8 284	98,4	4,4	7 247
		73,0	15,48	EU-S135	NC31	111,1	41,3	13 675 P	103,2	6,7	10 432	101,6	6,0	9 346
3 ½	9,50	88,9	14,14	EU-E75	NC38	120,7	76,2	10 297 P	111,9	3,2	7 827	110,3	2,4	6 504
		88,9	14,14	EU-E75	NC38	120,7	68,3	14 701 P	111,9	3,2	7 827	110,3	2,4	6 504
		88,9	14,14	EU-E75	3 1/2 OHLW	120,7	76,2	9 602 P	108,7	3,2	7 240	108,0	2,8	6 600
		88,9	14,14	EU-E75	3 1/2 SL-H90	117,5	76,2	10 127 P	106,4	2,8	7 485	105,6	2,4	6 783
	13,30	88,9	19,79	EU-E75	NC38	120,7	68,3	14 701 P	114,3	4,4	9 862	112,7	3,6	8 498
		88,9	19,79	IU-E75	NC31[b]	104,8	54,0	9 591 P	101,6	6,0	9 346	100,0	5,2	8 284
		88,9	19,79	EU-E75	3 1/2 OHSW	120,7	68,3	13 965 P	111,9	4,8	9 868	110,3	4,0	8 540
		88,9	19,79	EU-E75	31/2 H90	133,4	69,9	19 040 P	115,1	3,2	9 577	114,3	2,8	8 795
		88,9	19,79	EU-X95	NC38	127,0	65,1	16 347 P	116,7	5,6	11 961	115,1	4,8	10 555
		88,9	19,79	EU-X95	3 ½ SL-H90[b]	117,5	68,3	15 013 P	111,1	5,2	11 853	109,5	4,4	10 368
		88,9	19,79	EU-X95	3 1/2 H90	133,4	69,9	19 040 P	117,5	4,4	11 966	115,9	3,6	10 367
See end of table for notes														

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
		Drill pipe data			**New tool joint data**				**Premium class**			**Class 2**		
Labels[a]		**Nom OD**	**Nom linear mass**	**Type upset and pipe grade**	**Connection**	**New OD**	**New ID**	**Make-up torque[d]**	**Min OD tool joint[c,f]**	**Min box shoulder with eccentric wear[e]**	**Make-up torque for min OD tool joint[d]**	**Min OD tool joint[c,f]**	**Min box shoulder with eccentric wear[e]**	**Make-up torque for min OD tool joint[d]**
1	2	mm	kg/m			mm	mm	N·m	mm	mm	N·m	Mm	mm	N·m
3 1/2	13,30	88,9	19,79	EU-G105	NC38	127,0	61,9	17 925 P	118,3	6,4	13 394	116,7	5,6	11 961
		88,9	19,79	EU-S135	NC40	136,5	61,9	24 212 P	127,0	7,1	17 041	124,6	6,0	14 599
		88,9	19,79	EU-S135	NC38	127,0	54,0	21 560 P	122,2	8,3	17 102	119,9	7,1	14 856
	15,50	88,9	23,07	EU-E75	NC38	127,0	65,1	16 347 P	115,1	4,8	10 555	113,5	4,0	9 178
		88,9	23,07	EU-X95	NC38	127,0	61,9	17 925 P	118,3	6,4	13 394	116,7	5,6	11 961
		88,9	23,07	EU-G105	NC38	127,0	54,0	21 560 P	119,9	7,1	14 856	117,5	6,0	12 674
		88,9	23,07	EU-G105	NC40	133,4	65,1	22 528 P	125,4	6,4	15 406	123,0	5,2	13 009
		88,9	23,07	EU-S135	NC40	139,7	57,2	26 596 P	129,4	8,3	19 550	126,2	6,7	16 220
4	11,85	101,6	17,63	EU-E75	NC46	152,4	82,6	27 031 P	132,6	2,8	10 634	131,0	2,0	8 780
		101,6	17,63	EU-E75	4 WO	146,1	87,3	23 301 P	132,6	2,8	10 634	131,0	2,0	8 780
		101,6	17,63	EU-E75	4 OHLW	133,4	88,1	17 878 P	127,0	3,6	10 665	125,4	2,8	8 939
		101,6	17,63	IU-E75	4 H90	139,7	71,4	28 723 P	123,8	2,8	10 345	123,0	2,4	9 439
	14,00	101,6	20,83	IU-E75	NC40	133,4	71,4	18 938 P	122,2	4,8	12 225	120,7	4,0	10 680
		101,6	20,83	EU-E75	NC46	152,4	82,6	27 031 P	134,1	3,6	12 518	132,6	2,8	10 634
		101,6	20,83	IU-E75	4 SH[b]	117,5	65,1	12 224 P	112,7	6,0	11 907	111,1	5,2	10 598
		101,6	20,83	EU-E75	4 OHSW	139,7	82,6	22 013 P	128,6	4,4	12 380	127,0	3,6	10 628
		101,6	20,83	IU-E75	4 H90	139,7	71,4	28 723 P	125,4	3,6	12 183	123,8	2,8	10 345
		101,6	20,83	IU-X95	NC40	133,4	68,3	20 770 P	125,4	6,4	15 406	123,0	5,2	13 009
		101,6	20,83	EU-X95	NC46	152,4	82,6	27 031 P	136,5	4,8	15 406	134,9	4,0	13 473
		101,6	20,83	IU-X95	4 H90	139,7	71,4	28 723 P	127,8	4,8	15 002	126,2	4,0	13 115
		101,6	20,83	IU-G105	NC40	139,7	61,9	24 212 P	127,0	7,1	17 041	124,6	6,0	14 599
		101,6	20,83	EU-G105	NC46	152,4	82,6	27 031 P	138,1	5,6	17 372	135,7	4,4	14 435
		101,6	20,83	IU-G105	4 H90	139,7	71,4	28 723 P	129,4	5,6	16 922	127,8	4,8	15 002
		101,6	20,83	EU-S135	NC46	152,4	76,2	31 725 P	141,3	7,1	21 404	139,7	6,4	19 372
	15,70	101,6	23,36	IU-E75	NC40	133,4	68,3	20 770 P	123,8	5,6	13 801	121,4	4,4	11 449
		101,6	23,36	EU-E75	NC46	152,4	82,6	27 031 P	134,9	4,0	13 473	133,4	3,2	11 572
		101,6	23,36	IU-E75	4 H90	139,7	71,4	28 723 P	126,2	4,0	13 115	124,6	3,2	11 260
		101,6	23,36	IU-X95	NC40	139,7	61,9	24 212 P	127,0	7,1	17 041	124,6	6,0	14 599
		101,6	23,36	EU-X95	NC46	152,4	76,2	31 725 P	138,1	5,6	17 372	135,7	4,4	14 435
		101,6	23,36	IU-X95	4 H90	139,7	71,4	28 723 P	129,4	5,6	16 922	127,8	4,8	15 002
		101,6	23,36	EU-G105	NC46	152,4	76,2	31 725 P	138,9	6,0	18 367	137,3	5,2	16 385
		101,6	23,36	IU-G105	4 H90	139,7	71,4	28 723 P	131,0	6,4	18 876	128,6	5,2	15 958
		101,6	23,36	IU-S135	NC46	152,4	66,7	36 583 B	143,7	8,3	24 517	140,5	6,7	20 385
		101,6	23,36	EU-S135	NC46	152,4	73,0	33 947 P	143,7	8,3	24 517	140,5	6,7	20 385
See end of table for notes														

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
		Drill pipe data			**New tool joint data**				**Premium class**			**Class 2**		
Labels[a]		**Nom OD**	**Nom linear mass**	**Type upset and pipe grade**	**Connection**	**New OD**	**New ID**	**Make-up torque[d]**	**Min OD tool joint[c,f]**	**Min box shoulder with eccentric wear[e]**	**Make-up torque for min OD tool joint[d]**	**Min OD tool joint[c,f]**	**Min box shoulder with eccentric wear[e]**	**Make-up torque for min OD tool joint[d]**
1	2	mm	kg/m			mm	mm	N·m	mm	mm	N·m	mm	mm	N·m
4 ½	16,60	114,3	24,70	IEU-E75	4 1/2 FH	152,4	76,2	27 957 P	136,5	5,2	16 439	134,1	4,0	13 656
		114,3	24,70	IEU-E75	NC46	158,8	82,6	27 031 P	137,3	5,2	16 385	135,7	4,4	14 435
		114,3	24,70	IEU-E75	4 1/2 OHSW	149,2	95,3	21 913 P	138,1	5,2	16 083	136,5	4,4	14 067
		114,3	24,70	EU-E75	NC50	168,3	95,3	30 317 P	145,3	4,0	15 714	144,5	3,6	14 606
		114,3	24,70	IEU-E75	4 1/2 H-90	152,4	82,6	31 355 P	135,7	4,8	16 561	134,1	4,0	14 429
		114,3	24,70	IEU-X95	41/2 FH	152,4	69,9	32 126 P	139,7	6,7	20 263	137,3	5,6	17 383
		114,3	24,70	IEU-X95	NC46	158,8	82,6	27 031 P	140,5	6,7	20 385	138,1	5,6	17 372
		114,3	24,70	EU-X95	NC50	168,3	95,3	30 317 P	148,4	5,6	20 237	146,8	4,8	17 958
		114,3	24,70	IEU-X95	4 1/2 H-90	152,4	76,2	36 565 P	138,9	6,4	20 935	136,5	5,2	17 643
		114,3	24,70	IEU-G105	4 1/2 FH	152,4	69,9	32 126 P	141,3	7,5	22 223	138,9	6,4	19 295
		114,3	24,70	IEU-G105	NC46	158,8	76,2	31 725 P	142,1	7,5	22 433	139,7	6,4	19 372
		114,3	24,70	EU-G105	NC50	168,3	95,3	30 317 P	150,0	6,4	22 551	147,6	5,2	19 093
		114 3	24,70	IEU-G105	4 1/2 H-90	152,4	76,2	36 565 P	139,7	6,7	22 051	138,1	1,5	19 829
		114,3	24,70	IEU-S135	NC46	158,8	69,9	36 085 P	146,8	9,9	28 784	143,7	8,3	24 517
		114,3	24,70	EU-S135	NC50	168,3	88,9	36 165 P	154,0	8,3	28 495	151,6	7,1	24 902
	20,00	114,3	29,76	IEU-E75	4 1/2 FH	152,4	76,2	27 957 P	138,9	6,4	19 295	136,5	5,2	16 439
		114,3	29,76	IEU-E75	NC46	158,8	76,2	31 725 P	139,7	6,4	19 372	137,3	5,2	16 385
		114,3	29,76	EU-E75	NC50	168,3	92,1	33 284 P	147,6	5,2	19 093	146,1	4,8	16 832
		114,3	29,76	IEU-E75	4 1/2 H-90	152,4	76,2	36 565 P	137,3	5,6	18 731	135,7	4,8	16 561
		114,3	29,76	IEU-X95	4 1/2 FH	152,4	63,5	35 967 P	142,9	8,3	24 216	140,5	7,1	21 239
		114,3	29,76	IEU-X95	NC46	158,8	69,9	36 085 P	143,7	8,3	24 517	141,3	7,1	21 404
		114,3	29,76	EU-X95	NC50	168,3	88,9	36 165 P	150,8	6,7	23 723	149,2	6,0	21 389
		114,3	29,76	IEU-X95	4 1/2 H-90	152,4	76,2	36 565 P	141,3	7,5	24 308	138,9	6,4	20 935
		114,3	29,76	IEU-G105	NC46	158,8	63,5	40 102 P	145,3	9,1	26 634	142,9	7,9	23 471
		114,3	29,76	EU-G105	NC50	168,3	88,9	36 165 P	153,2	7,9	27 289	150,0	6,4	22 551
		114,3	29,76	EU-S135	NC50	168,3	76,2	46 803 P	158,0	10,3	34 667	154,8	8,7	29 711
5	19,50	127,0	29,02	IEU-E75	NC50	168,3	95,3	30 317 P	149,2	6,0	21 389	147,6	5,2	19 093
		127,0	29,02	IEU-X95	NC50	168,3	88,9	36 165 P	153,2	7,9	27 289	150,8	6,7	23 723
		127,0	29,02	IEU-X95	5 H-90	165,1	82,6	41 667 P	148,4	7,5	26 929	146,1	6,4	23 206
		127,0	29,02	IEU-G105	NC50	168,3	82,6	41 664 P	154,8	7,9	29 711	152,4	7,5	26 091
		127,0	29,02	IEU-G105	5 H-90	165,1	76,2	47 189 P	150,0	8,3	29 458	147,6	7,1	25 679
		127,0	29,02	IEU-S135	NC50	168,3	69,9	51 570 P	160,3	11,5	38 479	157,2	9,9	33 414
		127,0	29,02	IEU-S135	5 1/2 FH	184,2	88,9	58 745 P	171,5	9,5	38 962	168,3	7,9	33 098

See end of table for notes

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
		Drill pipe data			New tool joint data				Premium class			Class 2		
Labels[a]		Nom OD	Nom linear mass	Type upset and pipe grade	Connection	New OD	New ID	Make-up torque[d]	Min OD Tool Joint[c,f]	Min box shoulder with eccentric wear[e]	Make-up torque for min OD tool joint[d]	Min OD tool joint[c,f]	Min box shoulder with eccentric wear[e]	Make-up torque For min OD tool joint[d]
1	2	mm	kg/m			mm	mm	N·m	mm	mm	N·m	mm	mm	N·m
5	25,60	127,0	38,10	IEU-E75	NC50	168,3	88,9	36 165 P	153,2	7,9	27 289	150,8	6,7	23 723
		127,0	38,10	IEU-E75	5 1/2 FH	177,8	88,9	51 171 B	165,1	6,4	27 394	162,7	5,2	23 221
		127,0	38,10	IEU-X95	NC50	168,3	76,2	46 803 P	158,0	10,3	34 667	154,8	8,7	29 711
		127,0	38,10	IEU-X95	5 1/2 FH	177,8	88,9	51 171 B	169,1	8,3	34 550	166,7	7,1	30 227
		127,0	38,10	IEU-G105	NC50	168,3	69,9	51 570 P	159,5	11,1	37 200	156,4	9,5	32 171
		127,0	38,10	IEU-G105	5 1/2 FH	184,2	88,9	58 745 P	170,7	9,1	37 482	168,3	7,9	33 098
		127,0	38,10	IEU-S135	5 1/2 FH	184,2	82,6	64 035 B	176,2	11,9	48 058	173,0	10,3	41 953
5 ½	21,90	139,7	32,59	IEU-E75	5 1/2 FH	177,8	101,6	45 301 P	164,3	6,0	25 994	162,7	5,2	23 221
		139,7	32,59	IEU-X95	5 1/2 FH	177,8	95,3	51 171 B	168,3	7,9	33 098	165,9	6,7	28 806
		139,7	32,59	IEU-X95	5 1/2 H-90	177,8	88,9	47 210 P	157,2	8,3	33 101	154,8	7,1	28 945
		139,7	32,59	IEU-G105	5 1/2 FH	184,2	88,9	58 745 P	170,7	9,1	37 482	167,5	7,5	31 658
		139,7	32,59	IEU-S135	5 1/2 FH	190,5	76,2	70 583 P	176,2	11,9	48 058	173,0	10,3	41 953
	24,70	139,7	36,76	IEU-E75	5 1/2 FH	177,8	101,6	45 301 P	166,7	7,1	30 227	164,3	6,0	25 994
		139,7	36,76	IEU-X95	5 1/2 FH	184,2	88,9	58 745 P	170,7	9,1	37 482	167,5	7,5	31 658
		139,7	36,76	IEU-G105	5 1/2 FH	184,2	88,9	58 745 P	172,2	9,9	40 452	169,9	8,7	36 011
		139,7	36,76	IEU-S135	5 1/2 FH	190,5	76,2	70 583 P	178,6	13,1	52 743	174,6	11,1	44 986
6 5/8	25,20	168,3	37,50	IEU-E75	6 5/8 FH	203,2	127,0	59 567 P	188,9	6,4	36 349	187,3	5,6	32 675
		168,3	37,50	IEU-X95	6 5/8 FH	203,2	127,0	59 567 P	193,7	8,7	47 642	190,5	7,1	40 067
		168,3	37,50	IEU-G105	6 5/8 FH	209,6	120,7	69 526 P	195,3	15,9	51 498	192,9	8,3	45 732
		168,3	37,50	IEU-S135	6 5/8 FH	215,9	108,0	88 144 P	200,8	12,3	65 356	197,6	10,7	57 367
	27,70	168,3	41,22	IEU-E75	6 5/8 FH	203,2	127,0	59 567 P	190,5	7,1	40 067	188,1	6,0	34 507
		168,3	41,22	IEU-X95	6 5/8 FH	209,6	120,7	69 526 P	195,3	9,5	51 498	192,1	7,9	43 832
		168,3	41,22	IEU-G105	6 5/8 FH	209,6	120,7	69 526 P	196,9	10,3	55 399	194,5	9,1	49 563
		168,3	41,22	IEU-S135	6 5/8 FH	215,9	108,0	88 144 P	203,2	13,5	71 471	199,2	11,5	61 339

NOTE 1 Tool joints of outside diameters (OD) listed in this table should be adequate for all service

NOTE 2 The use of outside diameters (OD) smaller than those listed in this table may be acceptable due to special service requirements.

NOTE 3 Tool joints with torsional strengths considerably below that of the drill pipe may be adequate for much drilling service.

NOTE 4 Any tool joint with an outside diameter less than API bevel diameter should be provided with a minimum 0,8 mm depth x 45 degree bevel on the outside and inside diameter of the box shoulder and outside diameter of the pin shoulder.

NOTE 5 P = Pin limit; B = Box limit.

[a] Labels are for information and assistance in ordering

[b] Tool joint with dimensions shown has lower torsional yield ratio than the 0,80 which is generally used.

[c] Tool joint diameters specified are required to retain torsional strength in the tool joint comparable to the torsional strength of the attached drill pipe.

[d] Recommended make-up torque is based on 496,4 MPa stress

[e] This thickness measurement shall be made in the plane of the face from the ID of the counterbore to the outside diameter of the box, disregarding the bevels.

[f] In calculation of torsional strengths of tool joints, both new and worn, the bevels of the tool joint shoulders should be disregarded

Table A.11 — – Buoyancy factors

Mud density	Pressure gradient	Buoyancy factor	Mud density	Pressure gradient	Buoyancy factor
g/cm^3	kPa/m	Kb	g/cm^3	kPa/m	Kb
1,01	9,87	0,872	1,73	16,92	0,780
1,03	10,11	0,869	1,75	17,16	0,777
1,05	10,34	0,866	1,77	17,39	0,774
1,08	10,58	0,862	1,80	17,63	0,771
1,10	10,81	0,859	1,82	17,86	0,768
1,13	11,05	0,856	1,85	18,10	0,765
1,15	11,28	0,853	1,87	18,33	0,762
1,17	11,52	0,850	1,89	18,57	0,759
1,20	11,75	0,847	1,92	18,80	0,756
1,22	11,99	0,844	1,94	19,04	0,752
1,25	12,22	0,841	1,97	19,27	0,749
1,27	12,46	0,838	1,99	19,51	0,746
1,29	12,69	0,835	2,01	19,74	0,743
1,32	12,93	0,832	2,04	19,98	0,740
1,34	13,16	0,829	2,06	20,21	0,737
1,37	13,40	0,826	2,08	20,45	0,734
1,39	13,63	0,823	2,11	20,68	0,731
1,41	13,87	0,820	2,13	20,92	0,728
1,44	14,10	0,817	2,16	21,15	0,725
1,46	14,34	0,814	2,18	21,39	0,722
1,49	14,57	0,811	2,20	21,62	0,719
1,51	14,81	0,807	2,23	21,86	0,716
1,53	15,04	0,804	2,25	22,09	0,713
1,56	15,28	0,801	2,28	22,33	0,710
1,58	15,51	0,798	2,30	22,56	0,707
1,61	15,75	0,795	2,32	22,80	0,704
1,63	15,98	0,792	2,35	23,03	0,700
1,65	16,22	0,789	2,37	23,27	0,697
1,68	16,45	0,786	2,40	23,50	0,694
1,70	16,69	0,783			

Table A.12 — – Rotary shouldered connection interchange list

Common name	Size	Pin base dia. (tapered) mm	Pitch mm	Taper mm/mm	Thread form[a]	Same as or interchanges with
Internal Flush (IF)	2 3/8	73,05	6,35	1/6	V-0.065 (V-0.038R)	2 7/8 Slim Hole NC26[b]
	2 7/8	86,13	6,35	1/6	V-0.065 (V-0.038R)	3 1/2 Slim Hole NC31[b]
	3 1/2	102,01	6,35	1/6	V-0.065 (V-0.038R)	4 1/2 Slim Hole NC38[b]
	4	122,78	6,35	1/6	V-0.065 (V-0.038R)	4 1/2 Extra Hole NC46[b]
	4 1/2	133,35	6,35	1/6	V-0.065 (V-0.038R)	5 Extra Hole NC50[b] 5 1/2 Double Streamline
Full Hole	4	108,71	6,35	1/6	V-0.065 (V-0.038R)	4 1/2 Double streamline NC40[b]
Extra Hole (X.H.) (E.H.)	2 7/8	84,51	6,35	1/6	V-0.065 (V-0.038R)	3 1/2 Double Streamline
	3 1/2	96,82	6,35	1/6	V-0.065 (V-0.038R)	4 Slim Hole 4 1/2 External Flush
	4 1/2	122,78	6,35	1/6	V-0.065 (V-0.038R)	4 Internal Flush NC46[b]
	5	133,35	6,35	1/6	V-0.065 (V-0.038R)	4 1/2 Internal Flush NC50[b] 5 1/2 Double Streamline
Slim Hole (S.H.)	2 7/8	73,05	6,35	1/6	V-0.065 (V-0.038R)	2 3/8 Internal Flush NC26[b]
	3 1/2	86,13	6,35	1/6	V-0.065 (V-0.038R)	2 7/8 Internal Flush NC31[b]
	4	96,82	6,35	1/6	V-0.065 (V-0.038R)	3 1/2 Extra Hole 4 1/2 External Flush
	4 1/2	102,01	6,35	1/6	V-0.065 (V-0.038R)	3 1/2 Internal Flush NC38[b]
Double Streamline (DSL)	3 1/2	84,51	6,35	1/6	V-0.065 (V-0.038R)	2 7/8 Extra Hole
	4 1/2	108,71	6,35	1/6	V-0.065 (V-0.038R)	4 Full Hole NC40[b]
	5 1/2	133,35	6,35	1/6	V-0.065 (V-0.038R)	4 1/2 Internal Flush 5 Extra Hole NC50[b]
External Flush (E.F.)	4 1/2	96,82	6,35	1/6	V-0.065 (V-0.038R)	4 Slim Hole 3 1/2 Extra Hole
Numbered Connection (NC)	26	73,05	6,35	1/6	V-038R	2 3/8 Internal Flush 2 7/8 Slim Hole
	31	86,13	6,35	1/6	V-038R	2 7/8 Internal Flush 3 1/2 Slim Hole
	38	102,01	6,35	1/6	V-038R	3 1/2 Internal Flush 4 1/2 Slim Hole
	40	108,71	6,35	1/6	V-038R	4 Full Hole 4 1/2 Double streamline
	46	122,78	6,35	1/6	V-038R	4 Internal Flush 4 1/2 Extra Hole
	50	133,35	6,35	1/6	V-038R	4 1/2 Internal Flush 5 Extra Hole

[a] Connections with two thread forms shown may be machined with either thread form without affecting gauging or interchangeability.

[b] Numbered connections (NC) should be machined only with the V - 0.038Radius thread form.

Table A.13 — – Drill collar weight (steel)a

kg/m

1	2	3	4	5	6	7	8	9	10	11	12	13	14
Drill collar OD	**Drill collar ID mm**												
mm	25,40	31,75	38,10	44,45	50,80	57,15	63,50	71,44	76,20	82,55	88,90	95,25	101,60
73,03	29	27	24										
76,20	32	30	27										
79,38	35	33	30										
82,55	38	36	33										
88,90	45	42	40										
95,25	52	50	47										
101,60	60	57	55										
104,78	64	61	59										
107,95	68	66	63										
114,30	76	74	72										
120,65	86	83	81										
127,00			90	87	83	79	75						
133,35			101	97	94	89	85	78					
139,70			111	108	104	100	95	89					
146,05			122	119	115	111	107	100	96	89			
152,40			134	131	127	123	118	112	107	101			
158,75			146	143	139	135	130	124	119	113	107		
165,10			159	156	152	148	143	136	132	126	119		
171,45			172	169	165	161	156	150	145	139	132		
177,80			186	183	179	175	170	163	159	153	146	139	131
184,15			200	197	193	189	184	177	173	167	160	153	145
190,50			215	211	208	203	199	192	188	182	175	168	160
196,85			230	226	223	219	214	207	203	197	190	183	175
203,20			245	242	238	234	229	223	219	212	206	198	191
209,55			262	258	255	250	246	239	235	228	222	215	207
215,90			278	275	271	267	262	256	251	245	238	231	224
222,25			295	292	288	284	279	273	268	262	256	248	241
228,60			313	310	306	302	297	290	286	280	273	266	258
234,95			331	328	324	320	315	309	304	298	291	284	276
241,30			350	346	343	338	334	327	323	317	310	303	295
247,65			369	366	362	358	353	346	342	336	329	322	314
254,00			388	385	381	377	373	366	362	355	349	341	334
279,40			472	469	465	461	456	449	445	439	432	425	417
304,80			563	560	556	552	547	541	536	530	524	516	509

See ISO 10424-1 or API Specification 7 for standard drill collar dimensions.

For special configurations of drill collars, consult manufacturer for reduction or increase in weight.

[a] Weight per foot is based on drill collar steel having a density of 7841,72 kg/m^3

Table A.14 — –Recommended make-up torque[a] for rotary shouldered drill collar connections

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
Connection Name		OD mm	Minimum make-up torque (N·m)[b]											
Labels			ID of drill collar (mm)											
1	2		25,4	31,75	38,10	44,45	50,80	57,15	63,50	71,44	76,20	82,55	88,90	95,25
NC23	API NC	76,20	*3 400	*3 400	*3 400									
		79,38	*4 515	*4 515	3 589									
		82,55	5 423	4 592	3 589									
2 3/8	API Regular	76,20		*3 038	*3 038	2 371								
		79,38		*4 105	3 490	2 371								
		82,55		4 454	3 490	2 371								
2 7/8	PAC[c]	76,20		*5 148	*5 148	3 967								
		79,38		*6 733	5 628	3 967								
		82,55		7 058	5 628	3 967								
2 3/8	API IF	85,73		*4 855	*4 855	*4 855								
NC 26	API NC	88,90		*6 245	*6 245	5 012								
2 7/8	Slim Hole	95,25		7 458	6 329	5 012								
2 7/8	API Regular	88,90		*5 204	*5 204	*5 204								
		95,25		7 818	6 713	5 426								
		98,43		7 818	6 713	5 426								
2 7/8	Extra Hole	95,25		*5 544	*5 544	*5 544								
3 1/2	DSL	98,43		*7 256	*7 256	*7 256								
2 7/8	Mod. Open	104,78		*10 927	*10 927	10 078								
2 7/8	API IF	98,43		*6 291	*6 291	*6 291	*6 291							
NC 31	API NC	104,78		*10 019	*10 019	*10 019	9 291							
3 1/2	API Regular	104,78		*8 767	*8 767	*8 767	*8 767	7 708						
		107,95		*10 692	*10 692	*10 692	9 647	7 708						
		114,30		14 197	12 899	11 381	9 647	7 708						
3 1/2	Slim Hole	107,95		*12 010	*12 010	11 065	9 291	7 309						
		114,30		13 946	12 619	11 065	9 291	7 309						
NC 35	API NC	114,30				*12 254	*12 254	*12 254	10 048					
		120,65				16 640	14 678	12 476	10 048					
		127,00				16 640	14 678	12 476	10 048					
3 1/2	Extra Hole	107,95				*6 997	*6 997	*6 997	*6 997					
4	Slim Hole	114,30				*11 496	*11 496	*11 496	11 268					
3 1/2	Mod. Open	120,65				*16 370	16 003	13 753	11 268					
		127,00				18 009	16 003	13 753	11 268					
		133,35				18 009	16 003	13 753	11 268					
3 1/2	API IF	120,65				*13 539	*13 539	*13 539	*13 539	11 274				
NC 38	API NC	127,00				*18 912	*18 912	17 500	14 883	11 274				
4 1/2	Slim Hole	133,35				21 974	19 853	17 500	14 883	11 274				
		139,70				21 974	19 853	17 500	14 883	11 274				
3 1/2	H-90[d]	120,65				*11 912	*11 912	*11 912	*11 912	*11 912				
		127,00				*17 346	*17 346	*17 346	*17 346	14 111				
		133,35				*23 176	22 953	20 523	17 830	14 111				
		139,70				25 112	22 953	20 523	17 830	14 111				
4	API Full Hole	127,00				*14 792	*14 792	*14 792	*14 792	*14 792				
NC 40	API NC	133,35				*20 730	*20 730	*20 730	20 295	16 439				
4	Mod. Open	139,70				*27 096	25 606	23 087	20 295	16 439				
4 1/2	DSL	146,05				27 847	25 606	23 087	20 295	16 439				
		152,40				27 847	25 606	23 087	20 295	16 439				
See end of table for notes														

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
Connection Name		**OD**	**Minimum make-up torque (N·m)[b]**											
Labels		**Mm**	**ID of drill collar (mm)**											
1	2		25,4	31,75	38,1	44,45	50,8	57,15	63,5	71,4375	76,2	82,55	88,9	95,25
4	H-90[d]	133,35				17070	17070	17070	17070	17070				
		139,70				23593	23593	23593	23593	22420				
		146,05				30548	30548	29440	26497	22420				
		152,40				34449	32094	29440	26497	22420				
		158,75				34449	32094	29440	26497	22420				
4 1/2	API Regular	139,70				21118	21118	21118	21118	21118				
		146,05				27942	27942	27942	26575	22546				
		152,40				34447	32114	29488	26575	22546				
		158,75				34447	32114	29488	26575	22546				
NC44	API NC	146,05				28330	28330	28330	28330	24623				
		152,40				35865	34587	31852	28821	24623				
		158,75				37014	34587	31852	28821	24623				
		165,10				37014	34587	31852	28821	24623				
4 1/2	API Full Hole	139,70					17589	17589	17589	17589	17589			
		146,05					24566	24566	24566	24566	24269			
		152,40					32004	32004	31222	27009	24269			
		158,75					37006	34264	31222	27009	24269			
		165,10					37006	34264	31222	27009	24269			
4 1/2	Extra Hole	146,05						24049	24049	24049	24049			
NC46	API NC	152,40						31756	31756	30406	27538			
4	API IF	158,75						37991	34812	30406	27538			
4 1/2	Semi If	165,10						37991	34812	30406	27538			
5	DSL	171,45						37991	34812	30406	27538			
4 1/2	Mod Open													
4 1/2	H-90[d]	146,05						24430	24430	24430	24430			
		152,40						32107	32107	31399	28541			
		158,75						38955	35790	31399	28541			
		165,10						38955	35790	31399	28541			
		171,45						38955	35790	31399	28541			
								9	10	11	12			
5	H-90[d]	158,75						34384	34384	34384	34384	32523		
		165,10						43244	43244	39861	36834	32523		
		171,45						47850	44505	39861	36834	32523		
		177,80						47850	44505	39861	36834	32523		
4 1/2	API IF	158,75						31189	31189	31189	31189	31189		
NC 50	API NC	165,10						40239	40239	40239	40239	36166		
5	Extra Hole	171,45						49815	48571	43762	40628	36166		
5	Mod Open	177,80						52035	48571	43762	40628	36166		
5 1/2	DSL	184,15						52035	48571	43762	40628	36166		
5	Semi IF	190,50						52035	48571	43762	40628	36166		
5 1/2	H-90[d]	171,45						46787	46787	46787	46290	41733		
		177,80						56935	54391	49489	46290	41733		
		184,15						57919	54391	49489	46290	41733		
		19vf0.50						57919	54391	49489	46290	41733		
5 1/2	API Regular	171,45						43306	43306	43306	43306	41346		
		177,80						53445	53445	49128	45919	41346		
		184,15						57597	54051	49128	45919	41346		
		190,50						57597	54051	49128	45919	41346		
See end of table for notes														

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
Connection name		**OD**	**Minimum make-up torque (N·m)[b]**											
Labels		**mm**	**ID of drill collar (mm)**											
1	2		25,40	31,75	38,10	44,45	50,80	57,15	63,50	71,44	76,20	82,55	88,90	95,25
5 1/2	API Full Hole	177,80						*44 419	*44 419	*44 419	*44 419	*44 419		
		184,15						*55 586	*55 586	*55 586	*55 586	*55 586		
		190,50						*67 331	*67 331	64 748	61 269	56 311		
		196,85						73 912	70 078	64 748	61 269	56 311		
NC 56	API NC	184,15							*54 908	*54 908	*54 908	*54 908		
		190,50							*66 516	65 379	61 934	57 023		
		196,85							70 658	65 379	61 934	57 023		
		203,20							70 658	65 379	61 934	57 023		
6 5/8	API Regular	190,50							*62 909	*62 909	*62 909	*62 909		
		196,85							*75 420	72 327	68 745	62 909		
		203,20							77 814	72 327	68 745	62 909		
		209,55							77 814	72 327	68 745	62 909		
6 5/8	H-90[d]	190,50							*63 058	*63 058	*63 058	*63 058		
		196,85							*75 530	*75 530	72 711	67 594		
		203,20							81 784	76 296	72 711	67 594		
		209,55							81 784	76 296	72 711	67 594		
NC 61	API NC	203,20							*74 748	*74 748	*74 748	*74 748		
		209,55							*88 722	*88 722	*88 722	83 551		
		215,90							98 527	92 735	88 951	83 551		
		222,25							98 527	92 735	88 951	83 551		
		228,60							98 527	92 735	88 951	83 551		
5 1/2	API IF	203,20							*76 795	*76 795	*76 795	*76 795	*76 795	
		209,55							*91 020	*91 020	*91 020	85 933	80 030	
		215,90							101 179	95 283	91 431	85 933	80 030	
		222,25							101 179	95 283	91 431	85 933	80 030	
		228,60							101 179	95 283	91 431	85 933	80 030	
		234,95							101 179	95 283	91 431	85 933	80 030	
6 5/8	API Full Hole	215,90							*91 910	*91 910	*91 910	*91 910	*91 910	91 089
		222,25							*107 847	*107 847	*107 847	103 999	97 757	91 089
		228,60							120 101	113 878	109 809	103 999	97 757	91 089
		234,95							120 101	113 878	109 809	103 999	97 757	91 089
		241,30							120 101	113 878	109 809	103 999	97 757	91 089
NC 70	API NC	228,60							*102 745	*102 745	*102 745	*102 745	*102 745	*102 745
		234,95							*120 399	*120 399	*120 399	*120 399	*120 399	*120 399
		241,30							*138 773	*138 773	*138 773	137 083	130 449	123 358
		247,65							154 170	147 569	143 252	137 083	130 449	123 358
		254,00							154 170	147 569	143 252	137 083	130 449	123 358
		260,35							154 170	147 569	143 252	137 083	130 449	123 358
NC 77	API NC	254,00							*146 691	*146 691	*146 691	*146 691	*146 691	*146 691
		260,35							*168 191	*168 191	*168 191	*168 191	*168 191	*168 191
		266,70							*190 480	*190 480	*190 480	190 476	183 197	175 409
		273,05							209 199	201 969	197 239	190 476	183 197	175 409
		279,40							209 199	201 969	197 239	190 476	183 197	175 409
7	H-90[d]	203,20							*72 474	*72 474	*72 474	*72 474	*72 474	*72 474
		209,55							*86 417	*86 417	*86 417	*86 417	82 666	76 444
		215,90							*100 979	97 708	93 911	88 490	82 666	76 444
7 5/8	API Regular	215,90							*81 894	*81 894	*81 894	*81 894	*81 894	*81 894
		222,25							*97 848	*97 848	*97 848	*97 848	*97 848	*97 848
See end of table for notes														

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
Connection Name		**OD mm**	**Minimum make-up torque (N·m)[b]**											
Labels			**ID of drill collar (mm)**											
1	2		25,40	31,75	38,10	44,45	50,80	57,15	63,50	71,44	76,20	82,55	88,90	95,25
7 5/8	API Regular	228,60							*114 488	*114 488	*114 488	114 188	107 836	101 048
		234,95							130 567	124 238	120 098	114 188	107 836	101 048
		241,30							130 567	124 238	120 098	114 188	107 836	101 048
7 5/8	H-90[d]	228,60							*98 998	*98 998	*98 998	*98 998	*98 998	*98 998
		234,95							*116 608	*116 608	*116 608	*116 608	*116 608	*116 608
		241,30							*134 915	*134 915	*134 915	*134 915	*134 915	130 545
8 5/8	API Regular	254,00							*148 252	*148 252	*148 252	*148 252	*148 252	*148 252
		260,35							*169 834	*169 834	*169 834	*169 834	*169 834	169 523
		266,70							*192 210	*192 210	191 352	184 589	177 310	169 523
8 5/8	H-90[d]	260,35							*153 861	*153 861	*153 861	*153 861	*153 861	*153 861
		266,70							*176 342	*176 342	*176 342	*176 342	*176 342	*176 342
7	H-90[d]	222,25								*92 278	*92 278	91 188	85 206	78 815
	(with low	228,60								100 649	96 753	91 188	85 206	78 815
	torque face)													
7 5/8	API Regular	234,95									*99 109	*99 109	*99 109	*99 109
	(with low	241,30									*117 228	*117 228	111 797	104 790
	torque face)	247,65									124 449	118 352	111 797	104 790
		254,00									124 449	118 352	111 797	104 790
7 5/8	H-90[d]	247,65								*124 284	*124 284	*124 284	*124 284	*124 284
	(with low	254,00								*144 069	*144 069	*144 069	141 237	133 960
	torque face)	260,35								158 783	154 361	148 039	141 237	133 960
		266,70								158 783	154 361	148 039	141 237	133 960
8 5/8	API Regular	273,05									*153 049	*153 049	*153 049	*153 049
	(with low	279,40									*177 167	*177 167	*177 167	*177 167
	torque face)	285,75									200 140	193 109	185 538	177 437
8 5/8	H-90[d]	273,05									*126 037	*126 037	*126 037	*126 037
	(with low	279,40									*150 199	*150 199	*150 199	*150 199
	torque face)	285,75									*175 176	*175 176	*175 176	*175 176

Torque figures preceded by an asterisk (*) indicate that the weaker member for the corresponding outside diameter (OD) and bore is the BOX; for all other torque values, the weaker member is the PIN.

In each connection size and type group, torque values apply to all connection types in the group, when used with the same drill collar outside diameter and bore, i.e., 2 3/8 API IF, NC 26 and 2 7/8 Slim Hole connections used with 88,9 mm x 31,75 mm drill collars all have the same minimum make-up torque of 6 245 N·m, and the Box is the weaker member.

Stress relief features are disregarded in determining make-up torque.

[a] Basis of calculations for recommended make-up torque assumed the use of a thread compound containing 40-60 % by weight of finely powdered metallic zinc or 60 % by weight finely powdered metallic lead, with not more than 0,3 % total active sulphur (reference the caution regarding the use of hazardous materials in Appendix G of API Specification 7) applied thoroughly to all threads and shoulders and using the modified Screw Jack formula in A.8 and a unit stress of 430,9 MPa in the box or pin, whichever is weaker.

[b] Normal torque range is tabulated value plus 10 %. Higher values may be used under extreme conditions.

[c] Make-up torque for PAC connections is based on 603,3 MPa and other factors listed in footnote 1.

[d] Make-up torque for H-90 connections is based on 387,5 MPa and other factors listed in footnote 1

Table A.15 — – Strength of kellys a

1	2	3	4	5	6	6	7	8	9	10	11
								Torsional yield		Yield in bending	Internal pressure at minimum yield
Kelly size	Kelly type	Bore	Label [b]	OD	Minimum Recommended casing OD[1]	Lower pin connection [c]	Drive section	Lowerpin connection	Drive section	Through drive section	Drive section
mm		mm		mm	mm	N	N	N·m	N·m	N·m	MPa
63,5	Square	31,8	NC 26 (2 3/8 IF)	85,7	114,3	1 850 460	1 976 790	13 084	16 677	17 626	205,5
76,2	Square	44,4	NC31 (2 7/8 IF)	104,8	139,7	2 379 799	2 591 089	19 592	26 438	30 235	175,8
88,9	Square	57,2	NC 38 (3 ½ IF)	120,6	168,3	3 220 513	3 225 851	30 777	38 370	46 369	153,1
108,0	Square	71,4	NC 46 (4 IF)	158,8	219,1	4 688 426	4 657 288	53 351	66 571	81 756	134,4
108,0	Square	71,4	NC 50 (4 1/2 IF)	161,9	219,1	6 117 195	4 657 288	75 668	66 571	81 756	134,4
133,4	Square	82,6	5 1/2 FH	177,8	244,5	7 157 189	7 577 101	98 907	134 768	158 631	142,0
76,2	Hexagonal	38,1	NC 26 (2 3/8 IF)	85,7	114,3	1 583 567	2 404 264	11 253	27 659	27 116	184,1
88,9	Hexagonal	47,6	NC31 (2 7/8 IF)	104,8	139,7	2 201 870	3 158 238	18 168	42 573	42 302	175,8
108,0	Hexagonal	57,2	NC 38 (3 1/2 IF)	120,6	168,3	3 220 513	4 655 509	30 777	76 739	75 926	172,4
133,4	Hexagonal	76,2	NC 46 (4 IF)	158,8	219,1	4 270 293	6 706 139	48 064	138 158	139 649	142,0
133,4	Hexagonal	82,6	NC 50 (4 1/2 IF)	161,9	219,1	5 168 834	6 214 611	63 384	129 481	134 633	142,0
152,4	Hexagonal	88,9	5 1/2 FH	177,8	244,5	6 507 749	8 609 534	89 958	203 102	206 762	125,5

NOTE: Clearance between protector rubber on kelly saver sub and casing inside diameter should also be checked.

[a] All values have a safety factor of 1,0 and are based on 758,4 MPa minimum tensile yield (quenched and tempered) for connections and 620,5 MPa minimum tensile yield (normalized and tempered) for the drive section. Fully quenched and tempered drive sections will have higher values than those shown. Shear strength is based on 57,7 % of the minimum tensile yield strength.

[b] Labels are for information and assistance in ordering.

[c] Tensile area calculated at root of thread 19,0 mm from pin shoulder.

Table A.16 — – Contact angle between kelly and bushing for development of maximum width wear pattern

1	2	3	4	5	6	7	8	9
	Hexagonal kelly				Square kelly			
Kelly size	For minimum clearance	Contact angle	For maximum clearance	Contact angle	For minimum clearance	Contact angle	For maximum clearance	Contact angle
mm	mm	degrees	Mm	degrees	mm	degrees	mm	degrees
63,50	-	-	-	-	0,38	6,17	2,72	16,48
76,20	0,38	5,68	1,52	11,37	0,38	5,65	2,72	15,08
88,90	0,38	5,27	1,52	10,53	0,38	5,23	2,72	14,03
107,95	0,38	4,80	1,52	9,57	0,38	4,75	3,12	13,60
133,35	0,38	4,32	1,52	8,62	0,38	4,28	3,12	12,27
152,40	0,38	4,03	1,52	8,07	-	-	-	-

Table A.17 — – Strength of remachined kellys a

1	2	3	4	5	6	7	8	9	10	11	12
Original kelly		Remachined kelly		Lower pin connection			Tensile yield		Torsional yield		
Size	Type	Size	Type	Bore	Label [b]	OD	Lower [c] pin connection	Drive section	Lower pin connection	Drive section	Yield in bending through drive section
mm		mm		Mm		mm	N	N	N·m	N·m	N·m
108,0	Square	101,6	Square	73,0	NC50	161,9	5 979 300	3 711 596	75 248	49 081	64 808
108,0	Square	101,6	Square	73,0	NC46	158,8	4 499 821	3 711 596	51 928	49 081	64 808
133,4	Square	127,0	Square	95,2	5 1/2 IF	187,3	8 559 714	5 416 155	125 684	88 128	122 295
133,4	Square	127,0	Square	95,2	5 1/2 FH	177,8	6 035 348	5 416 155	79 858	88 128	122 295
133,4	Hexagonal	123,0	Hexagonal	82,6	NC46	158,8	3 602 170	4 791 180	41 488	93 009	100 331
133,4	Hexagonal	127,0	Hexagonal	82,6	NC46	158,8	3 602 170	5 323 632	41 488	106 432	112 940
133,4	Hexagonal	127,0	Hexagonal	88,9	NC50	161,9	4 447 777	4 793 404	55 317	96 399	106 296
152,4	Hexagonal	146,1	Hexagonal	101,6	5 1/2 FH	177,8	5 291 160	6 420 564	69 553	147 920	162 563
152,4	Hexagonal	146,1	Hexagonal	104,8	5 1/2 IF	187,3	7 424 972	6 100 736	109 008	140 734	157 546

NOTE: Kelly bushings are normally available for remachined kellys in the above table.

[a] All values have a safety factor of 1,0 and are based on 758,4 MPa minimum tensile yield (quenched and tempered) for connections and 620,5 MPa minimum tensile yield (normalized and tempered) for the drive section. Fully quenched and tempered drive sections will have higher values than those shown. Shear strength is based on 57,7 % of the minimum tensile yield strength.

[b] Labels are for information and ordering.

[c] Tensile area calculated at root of thread 19,0 mm from pin shoulder.

Table A.18 — – Section modulus values for drill pipe

1	2	3	4	5
Label 1[a]	**Label 2[a]**	**Pipe OD** **mm**	**Pipe weight** **kg/m**	**I/C** **mm^3**
2 3/8	4,85	60,3	7,22	10 803
2 3/8	6,65	60,3	9,90	14 176
2 7/8	6,85	73,0	10,19	18 331
2 7/8	10,40	73,0	15,48	26 205
3 1/2	9,50	88,9	14,14	32 083
3 1/2	13,30	88,9	19,79	42 074
3 1/2	15,50	88,9	23,07	47 822
4	11,85	101,6	17,63	44 168
4	14,00	101,6	20,83	52 816
4	15,70	101,6	23,36	58 536
4 1/2	13,75	114,3	20,46	58 758
4 1/2	16,60	114,3	24,70	69 870
4 1/2	20,00	114,3	29,76	83 691
4 1/2	22,82	114,3	33,96	92 787
4 1/2	24,66	114,3	36,70	98 650
4 1/2	25,50	114,3	37,95	101 394
5	16,25	127,0	24,18	79 480
5	19,50	127,0	29,02	93 365
5	25,60	127,0	38,10	118 518
5 1/2	19,20	139,7	28,57	99 958
5 1/2	21,90	139,7	32,59	115 010
5 1/2	24,70	139,7	36,76	128 309
6 5/8	25,20	168,3	37,50	160 075
6 5/8	27,70	168,3	41,22	173 036
a Labels are for information and assistance in ordering.				

Table A.19 — – Effect of drilling fluid type on coefficient of friction

Drilling fluid	Typical coefficient of friction	Critical hole angle degrees
Water-base mud	0,35	71
Oil-base mud	0,25	76
Synthetic-base mud	0,17	80

Table A.20 — – Compensation factor for different mud densities

Mud density		Mud density	
g/cm^3	f_{MW}	g/cm^3	f_{MW}
0,96	1,04	1,68	0,98
1,08	1,93	1,80	0,97
1,20	1,02	1,92	0,96
1,32	1,01	2,04	0,95
1,44	1,00	2,16	0,94
1,56	0,99	2,28	0,93

Table A.21 — – Youngstown steel test results

Grade	Minimum yield strength	Yield strength maximum	Tensile strength minimum	Average tensile strength of test samples	Endurance limit Minimum test value	Endurance limit Median test value
	MPa	Mpa	MPa	MPa	MPa	MPa
E75	517	724	689	848	207	221
X95	655	862	724	910	221	241
G105	724	931	793	993	234	262
S135	931	1138	986	1151	248	276

Table A.22 —Hook-load at minimum yield strength for New, Premium class (used) and Class 2 (used) drill pipe

1	2	3	4	5	6	7	8	9	10	11	12
Labels [a]		**Nom. linear mass**	**ID**	**Class**	**OD** [b]	**Wall** [c] **thick-ness**	**Cross-sectional area**	**Yield strength - MPa**			
								517	655	724	931
								Hook load			
1	2	kg/m	mm		mm	mm	mm	N			
2 3/8	4.85	7,22	50,67	New	60,325	4,83	841,42	435 112	551 144	609 157	783 203
				Premium	58,395	3,86	661,42	342 037	433 248	478 851	615 665
				Class 2	57,429	3,38	573,61	296 634	375 737	415 286	533 942
2 3/8	6.65	9,90	46,10	New	60,325	7,11	1 188,96	614 807	778 759	860 731	1 106 655
				Premium	57,480	5,69	925,74	478 700	606 350	670 178	861 661
				Class 2	56,058	4,98	798,90	413 111	523 271	578 353	743 596
2 7/8	6.85	10,19	62,00	New	73,025	5,51	1 169,03	604 522	765 730	846 332	1 088 142
				Premium	70,820	4,42	920,00	475 720	602 578	666 010	856 296
				Class 2	69,718	3,86	798,32	412 799	522 884	577 922	743 044
2 7/8	10.40	15,48	54,64	New	73,025	9,19	1 843,80	953 450	1 207 706	1 334 831	1 716 213
				Premium	69,347	7,37	1 432,58	740 785	938 330	1 037 099	1 333 417
				Class 2	67,508	6,43	1 234,90	638 573	808 860	894 004	1 149 434
3 1/2	9.50	14,14	76,00	New	88,900	6,45	1 671,09	864 129	1 094 565	1 209 783	1 555 436
				Premium	86,319	5,16	1 315,93	680 485	861 950	952 680	1 224 876
				Class 2	85,029	4,52	1 142,32	590 693	748 209	826 969	1 063 245
3 1/2	13.30	19,79	70,21	New	88,900	9,35	2 336,06	1 207 999	1 530 135	1 691 201	2 174 402
				Premium	85,161	7,47	1 824,96	943 690	1 195 340	1 321 166	1 698 642
				Class 2	83,292	6,55	1 577,61	815 795	1 033 340	1 142 112	1 468 429
3 1/2	15.50	23,07	66,09	New	88,900	11,40	2 776,58	1 435 775	1 818 647	2 010 085	2 584 395
				Premium	84,338	9,12	2 155,87	1 114 813	1 412 097	1 560 739	2 006 660
				Class 2	82,057	7,98	1 857,80	960 669	1 216 847	1 344 938	1 729 206
4	11.85	17,63	88,29	New	101,600	6,65	1 984,96	1 026 449	1 300 171	1 437 029	1 847 613
				Premium	98,938	5,33	1 565,74	809 648	1 025 555	1 133 509	1 457 371
				Class 2	97,607	4,65	1 360,26	703 406	890 983	984 770	1 266 133
4	14.00	20,83	84,84	New	101,600	8,38	2 454,70	1 269 340	1 607 828	1 777 074	2 284 811
				Premium	98,247	6,71	1 928,45	997 211	1 263 130	1 396 092	1 794 978
				Class 2	96,571	5,87	1 671,93	864 570	1 095 121	1 210 397	1 556 224
4	15.70	23,36	82,30	New	101,600	9,65	2 788,12	1 441 749	1 826 218	2 018 447	2 595 150
				Premium	97,739	7,72	2 183,67	1129 186	1 430 299	1 580 858	2 032 530
				Class 2	95,809	6,76	1 890,19	977 443	1 238 096	1 368 420	1 759 387
4 1/2	13.75	20,46	100,53	New	114,300	6,88	2 322,90	1 201 171	1 521 483	1 681 637	2 162 107
				Premium	111,547	5,51	1 834,45	948 619	1 201 585	1 328 070	1 707 517
				Class 2	110,170	4,83	1 594,77	824 656	1 044 563	1 154 518	1 484 381
See end of table for notes											

1	2	3	4	5	6	7	8	9	10	11	12
Labels [a]		**Nom. linear mass**	**ID**	**Class**	**OD** [b]	**Wall thick-ness** [c]	**Cross-sectional area**	**Yield strength - MPa**			
								517	655	724	931
								Hook load			
1	2	kg/m	mm		mm	mm	mm^2	N			
4 1/2	16.60	24,70	97,18	New	114,300	8,56	2 843,48	1 470 395	1 862 502	2 058 553	2 646 710
				Premium	110,876	6,86	2 238,00	1 157 272	1 465 876	1 620 180	2 083 089
				Class 2	109,164	5,99	1 942,12	1 004 280	1 272 089	1 405 994	1 807 704
4 1/2	20.00	29,76	92,46	New	114,300	10,92	3 547,15	1 834 260	2 323 395	2 567 963	3 301 666
				Premium	109,931	8,74	2 777,74	1 436 402	1 819 438	2 010 961	2 585 520
				Class 2	107,747	7,64	2 404,32	1 243 287	1 574 826	1 740 598	2 237 914
4 1/2	22.82	33,96	88,90	New	114,300	12,70	4 053,67	2 096 176	2 655 157	2 934 648	3 773 115
				Premium	109,220	10,16	3 161,86	1 635 015	2 071 021	2 289 024	2 943 033
				Class 2	106,680	8,89	2 731,16	1 412 297	1 788 910	1 977 217	2 542 137
5	16.25	24,18	111,96	New	127,000	7,52	2 822,12	1 459 342	1 848 499	2 043 077	2 626 813
				Premium	123,993	6,02	2 229,29	1 152 779	1 460 187	1 613 891	2 075 002
				Class 2	122,489	5,26	1 938,19	1 002 256	1 269 523	1 403 156	1 804 056
5	19.50	29,02	108,61	New	127,000	9,19	3 402,96	1 759 694	2 228 946	2 463 572	3 167 445
				Premium	123,322	7,37	2 679,87	1 385 777	1 755 322	1 940 092	2 494 403
				Class 2	121,483	6,43	2 326,32	1 202 942	1 523 730	1 684 119	2 165 297
5	25.60	38,10	101,60	New	127,000	12,70	4 560,38	2 358 198	2 987 048	3 301 475	4 244 756
				Premium	121,920	10,16	3 567,22	1 844 633	2 336 535	2 582 486	3 320 344
				Class 2	119,380	8,89	3 085,86	1 595 715	2 021 236	2 233 999	2 872 284
5 1/2	19.20	28,57	124,26	New	139,700	7,72	3 201,54	1 655 544	2 097 021	2 317 759	2 979 975
				Premium	136,611	6,17	2 531,29	1 308 934	1 657 986	1 832 512	2 356 085
				Class 2	135,067	5,41	2 201,74	1 138 540	1 442 149	1 593 954	2 049 371
5 1/2	21.90	32,59	121,36	New	139,700	9,17	3 760,12	1 944 389	2 462 896	2 722 147	3 499 901
				Premium	136,032	7,34	2 965,86	1 533 658	1 942 632	2 147 121	2 760 584
				Class 2	134,198	6,43	2 576,64	1 332 389	1 687 695	1 865 344	2 398 303
5 1/2	24.70	36,76	118,62	New	139,700	10,54	4 277,15	2 211 754	2 801 552	3 096 456	3 981 154
				Premium	135,484	8,43	3 365,86	1 740 523	2 204 659	2 436 732	3 132 941
				Class 2	133,375	7,37	2 920,70	1 510 318	1 913 074	2 114 449	2 718 575
6 5/8	25.20	37,50	151,51	New	168,275	8,38	4 210,44	2 177 245	2 757 844	3 048 144	3 919 039
				Premium	164,922	6,71	3 333,03	1 723 535	2 183 143	2 412 947	3 102 359
				Class 2	163,246	5,87	2 900,96	1 500 101	1 900 129	2 100 144	2 700 186
6 5/8	27.70	41,22	149,89	New	168,275	9,19	4 595,28	2 376 236	3 009 898	3 326 732	4 277 223
				Premium	164,597	7,37	3 633,73	1 879 013	2 380 083	2 630 621	3 382 223
				Class 2	162,758	6,43	3 160,90	1 634 521	2 070 394	2 288 330	2 942 139

[a] Labels are for information and assistance in ordering

[b] OD for new pipe is the original nominal; OD for Premium class is with 20 % wall reduction; OD for Class 2 is with 30 % wall reduction. All wall reduction occurs from the OD.

[c] Wall thickness for new pipe is the original nominal; wall thickness for Premium class pipe is the minimum remaining wall allowed for this class and is based on 80 % of new; wall thickness for Class 2 pipe is the minimum remaining wall allowed for this class and is based on 70 % of new.

Table A.23 — – Hook-load at minimum yield strength for New, Premium class (used) and Class 2 (used) tubing work strings

1	2	3	4	5	6	7	8	9	10	11	12
Labels [a]		Nom. linear mass	ID	Class	OD [b]	Wall [c] Thick-ness	Cross-Sectional Area	Yield strength - MPa			
								379	517	552	724
								Hook load			
1	2	kg/m	mm		mm	mm	mm^2	N			
3/4	1.20	1,79	20,93	New	26,670	2,87	214,58	81 380	110 974	118 372	155 363
				Premium	25,522	2,29	167,55	63 534	86 638	92 412	121 290
				Class 2	24,948	2,01	144,77	54 904	74 872	79 863	104 818
3/4	1.50	2,23	18,85	New	26,670	3,91	279,68	106 054	144 621	154 260	202 465
				Premium	25,105	3,12	216,06	81 927	111 717	119 168	156 404
				Class 2	24,323	2,74	185,68	70 411	96 015	102 416	134 421
1	1.80	2,68	26,64	New	33,401	3,38	318,64	120 827	164 767	175 749	230 671
				Premium	32,050	2,69	249,16	94 489	128 847	137 437	180 384
				Class 2	31,374	2,36	215,48	81 723	111 441	118 874	156 021
1	2.25	3,35	24,31	New	33,401	4,55	412,13	156 288	213 123	227 331	298 373
				Premium	31,582	3,63	319,35	121 089	165 127	176 132	231 174
				Class 2	30,673	3,18	274,84	104 231	142 134	151 609	198 987
1 1/4	2.40	3,57	35,05	New	42,164	3,56	431,29	163 557	223 034	237 900	312 247
				Premium	40,742	2,84	338,71	128 434	175 140	186 816	245 195
				Class 2	40,030	2,49	293,55	111 326	151 809	161 929	212 532
1 1/4	3.02	4,49	32,46	New	42,164	4,85	568,71	215 654	294 072	313 675	411 701
				Premium	40,223	3,89	443,10	168 036	229 141	244 416	320 797
				Class 2	39,253	3,40	382,58	145 070	197 821	211 010	276 951
1 1/4	3.20	4,76	32,11	New	42,164	5,03	586,71	222 491	303 395	323 621	424 756
				Premium	40,152	4,01	456,64	173 169	236 143	251 885	330 601
				Class 2	39,146	3,53	394,00	149 416	203 751	217 331	285 247
1 1/2	2.90	4,32	40,89	New	48,260	3,68	515,81	195 588	266 711	284 495	373 397
				Premium	46,787	2,95	405,81	153 886	209 845	223 835	293 783
				Class 2	46,050	2,57	352,06	133 518	182 070	194 209	254 901
1 1/2	4.19	6,24	37,13	New	48,260	5,56	746,13	282 951	385 843	411 563	540 179
				Premium	46,035	4,45	581,35	220 463	300 629	320 672	420 882
				Class 2	44,922	3,89	501,87	190 326	259 532	276 835	363 349
2 1/16	3.25	4,84	44,48	New	52,400	3,96	602,97	228 652	311 798	332 585	436 517
				Premium	50,815	3,18	474,45	179 930	245 355	261 716	343 501
				Class 2	50,023	2,77	411,74	156 128	212 901	227 095	298 062
2 3/8	4.70	6,99	50,67	New	60,325	4,83	841,42	319 084	435 112	464 123	609 157
				Premium	58,395	3,86	661,42	250 826	342 037	364 839	478 851
				Class 2	57,429	3,38	573,61	217 531	296 634	316 411	415 286
2 3/8	5.30	7,89	49,25	New	60,325	5,54	953,09	361 414	492 836	525 695	689 973
				Premium	58,110	4,42	747,03	283 289	386 301	412 057	540 819
				Class 2	57,003	3,89	646,90	245 319	334 528	356 827	468 336
See end of table for notes											

1	2	3	4	5	6	7	8	9	10	11	12
Labels [a]		Nom. linear mass	ID	Class	OD [b]	Wall [c] thick-ness	Cross sectional area				
								379	517	552	724
								Hook load			
1	2	kg/m	mm		mm	mm	mm^2	N			
2 3/8	5.95	8,85	47,42	New	60,325	6,45	1 091,93	414 072	564 640	602 285	790 498
				Premium	57,744	5,16	852,64	323 323	440 894	470 288	617 253
				Class 2	56,454	4,52	736,90	279 437	381 048	406 452	533 466
2 7/8	6.50	9,67	62,00	New	73,025	5,51	1 169,03	443 314	604 522	644 823	846 332
				Premium	70,820	4,42	920,00	339 964	475 720	507 435	666 010
				Class 2	69,718	3,86	798,32	391 684	412 799	440 321	577 922
2 7/8	8.70	12,95	57,38	New	73,025	7,82	1 602,51	607 680	828 655	883 897	1 160 118
				Premium	69,896	6,25	1 251,22	474 478	647 016	690 150	905 822
				Class 2	68,331	5,49	1 081,35	410 064	559 182	596 458	782 851
2 7/8	9.50	14,14	55,75	New	73,025	8,64	1 746,90	662 456	903 345	963 569	1 264 687
				Premium	69,571	6,91	1 360,06	515 749	703 291	750 179	984 609
				Class 2	67,843	6,05	1 173,68	445 058	606 898	647 359	849 659
2 7/8	10.70	15,92	53,11	New	73,025	9,96	1 972,77	748 102	1 020 142	1 088 151	1 428 200
				Premium	69,042	7,98	1 528,38	579 586	790 342	843 036	1 106 482
				Class 2	67,051	6,96	1 315,55	498 872	680 280	725 630	952 391
2 7/8	11.00	16,37	52,45	New	73,025	10,29	2 027,54	768 866	1 048 455	1 118 350	1 467 838
				Premium	68,910	8,23	1 568,84	594 923	811 258	865 339	1 135 760
				Class 2	66,853	7,19	1 349,48	511 732	697 815	744 339	976 945
3 1/2	12.80	19,05	70,21	New	88,900	9,35	2 336,06	885 868	1 207 999	1 288 534	1 691 201
				Premium	85,161	7,47	1 824,96	692 041	943 690	1 006 602	1 321 166
				Class 2	83,292	6,55	1 577,61	598 250	815 795	870 179	1 142 112
3 1/2	12.95	19,27	69,85	New	88,900	9,53	2 375,22	900 698	1 228 230	1 310 108	1 719 518
				Premium	85,090	7,62	1 854,58	703 268	959 001	1 022 935	1 342 598
				Class 2	83,185	6,68	1 602,77	607 792	828 806	884 062	1 160 332
3 1/2	15.80	23,51	64,72	New	88,900	12,09	2 917,48	1 106 340	1 508 641	1 609 220	2 112 100
				Premium	84,064	9,68	2 260,51	857 208	1 168 917	1 246 846	1 636 488
				Class 2	81,646	8,46	1 945,80	737 867	1 006 179	1 073 258	1 408 654
3 1/2	16.70	24,85	62,99	New	88,900	12,95	3 090,70	1 172 035	1 598 228	1 704 777	2 237 522
				Premium	83,718	10,36	2 388,25	905 640	1 234 964	1 317 296	1 728 953
				Class 2	81,128	9,07	2 052,77	778 443	1 061 515	1 132 282	1 486 120
4 1/2	15.50	23,07	97,18	New	114,300	8,56	2 843,48	1 078 289	1 470 395	1 568 421	2 058 553
				Premium	110,876	6,86	2 238,00	848 667	1 157 272	1 234 422	1 620 180
				Class 2	109,164	5,99	1 942,12	736 470	1 004 280	1 071 234	1 405 994
4 1/2	19.20	28,57	92,46	New	114,300	10,92	3 547,15	1 345 124	1 834 260	1 956 542	2 567 963
				Premium	109,931	8,74	2 777,74	1 053 361	1 436 402	1 532 159	2 010 961
				Class 2	107,747	7,64	2 404,32	911 743	1 243 287	1 326 171	1 740 598

[a] Labels are for information and assistance in ordering

[b] OD for new pipe is the original nominal; OD for Premium class is with 20 % wall reduction; OD for Class 2 is with 30 % wall reduction. All wall reduction occurs from the OD.

[c] Wall thickness for new pipe is the original nominal; wall thickness for Premium class pipe is the minimum remaining wall allowed for this class and is based on 80 % of new; wall thickness for Class 2 pipe is the minimum remaining wall allowed for this class and is based on70 % of new.

Table A.24 — – Maximum stress at the root of the last engaged thread for the pin of an NC 50 axisymmetric model

Load condition	First condition [a]		Second condition [b]	
SRG width mm	Maximum equivalent Stress [c,d]	Maximum axial stress	Maximum equivalent Stress [c,d]	Maximum axial stress
19,0	84%	82 %	83 %	81 %
25,4	70 %	56 %	63 %	53 %
31,8	75 %	63 %	73 %	64 %
No SRG	100 %	100 %	100 %	100 %

[a] Make-up only at 2 500 kN axial force on shoulder

[b] 5 004 kN axial tension applied to connection which causes shoulder separation.

[c] Equivalent stress is equal to 0.707 x $[(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2]^{1/2}$ where σ_1, σ_2 and σ_3 are principle stresses.

[d] In each case shown, equivalent stress at the root of the last engaged thread has exceeded the yield strength because these finite element calculations have been made for linear elastic material behavior. The behavior of an actual pin is elastic-plastic.

Table A.25 — IADC roller bit classification chart

	Series	Formations	Type	Features		
				Standard roller bearings	Roller bearing, air cooled	Roller bearing gauge protected
				(1)	(2)	(3)
Milled tooth bits	1	Soft formations with low compressive strength and high drillability	1 2 3 4			
	2	Medium to medium hard formations with high compressive strength	1 2 3 4			
	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			
	5	Soft to medium formations with low compressive strength	1 2 3 4			
	6	Medium hard formations with high compressive strength	1 2 3 4			
	7	Hard semi-abrasive and abrasive formations	1 2 3 4			
	8	Extremely hard and abrasive formations	1 2 3 4			

Table A.25 — IADC roller bit classification chart (continued)

	Series	Formations	Type	Features			
				Sealed roller bearing	Sealed roller bearing gauge protected	Sealed friction bearing	Sealed friction bearing gauge protected
				(4)	(5)	(6)	(7)
Milled tooth bits	1	Soft formations with low compressive strength and high drillability	1 2 3 4				
	2	Medium to medium hard formations with high compressive strength	1 2 3 4				
	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				
	5	Soft to medium formations with low compressive strength	1 2 3 4				
	6	Medium hard formations with high compressive strength	1 2 3 4				
	7	Hard semi-abrasive and abrasive formations	1 2 3 4				
	8	Extremely hard and abrasive formations	1 2 3 4				

Table A.26 — IADC bit classification codes Fourth position

Code [a]	Code feature	Code [a]	Code feature
A	Air application [b]	N	
B		O	
C	Center jet	P	
D	Deviation control	Q	
E	Extended jets [c]	R	Reinforced welds [d]
F		S	Standard steel tooth model [e]
G	Extra gauge/body protection	T	
H		U	
I		V	
J	Jet deflection	W	
K		X	Chisel inserts
L		Y	Conical inserts
M		Z	Other insert shapes

[a] The above codes are used in the 4th position of the 4 – character IADC bit classification code to indicate additional design features

[b] Journal bearing bits with air circulation nozzles

[c] Full extension (welded tubes with nozzles). Partial extensions should be noted elsewhere.

[d] For percussion applications

[e] Milled tooth bits with none of the extra features listed in this table. As an example, a milled tooth bit designed for the softest series, softest type in that series, with standard gauge, and no extra features, are designated 1-1-1-S on the bit carton. The manufacture will also list this bit designation in this block of the form

Table A.27 — – Recommended make-up torque ranges for roller cone bits

Connection Label	Minimum make-up torque N·m	Maximum make-up torque N·m
2 3/8 Reg	4 100	4 700
2 7/8 Reg	6 100	7 500
3 1/2 Reg	9 500	12 200
4 1/2 Reg	16 300	21 700
6 5/8 Reg	38 000	43 400
7 5/8 Reg	46 100	54 200
8 5/8 Reg	54 200	81 300

NOTE 1 Basis of calculation for recommended make-up torque assumed the use of a thread compound containing 40 to 60 % by weight of finely powdered metallic zinc with no more than 0.3 % total active sulfur, applied thoroughly to all threads and shoulders (see the caution regarding the use of hazardous materials in Appendix G of API Specification 7).

NOTE 2 Due to the irregular geometry of the ID bore in roller cone bits, torque values are based on estimated cross-sectional areas and have been proven by field experience.

Table A.28 — – Recommended minimum make-up torque

For diamond drill bits

RSC Label [a]	Maximum pin ID mm	Bit sub OD mm	Minimum make-up torque [b, c] N·m
2 3/8 Reg	25,4	76,20 79,38 82,55	2 428 * 3 280 * 4 183 *
2 7/8 Reg	31,8	88,90 95,25 98,43	4 166 * 6 260 6 315
3 1/2 Reg	38,1	104,78 107,95 114,30	7 011 * 8 550 * 10 386
4 1/2 Reg	57,2	139,70 146,05 152,40 158,75	16 881 * 22 338 * 23 796 24 075
6 5/8 Reg	82,6	190,50 196,85 203,20 209,55	50 301 * 51 327 51 783 52 236
7 5/8 Reg	95,2	215,90 222,25 228,60 234,95 241,30	65 481 * 78 236 * 81 303 81 932 82 563

NOTE 1 Torque figures followed by an asterisk * indicate that the weaker member for the corresponding outside diameter (OD) and bore is the BOX. For all other torque values the weaker member is the PIN.

NOTE 2 Normal torque range is tabulated value plus 10 %. Higher torque values may be used under extreme conditions.

[a] Labels are for information and assistance in ordering

[b] Basis of calculation for recommended make-up torque assumed the use of a thread compound containing 40 to 60 % by weight of finely powdered metallic zinc, with no more than 0,3 % total active sulfur, applied thoroughly to all threads and shoulders (see the caution regarding the use of hazardous materials in Appendix G of API Specification 7).

[c] Calculations are done using the modified Screw Jack formula in 5.8.2.3 and a unit stress of 344,74 MPa in the pin or box, whichever is weaker.

Table A.29 — – Common roller bit sizes

Size of bit **mm**	**Size of bit** **mm**
95,25	241,30
98,43	25083
120,65	269,88
149,23	279,40
152,40	311,15
155,58	342,90
158,75	368,30
165,10	406,40
171,45	444,50
200,03	508,00
212,73	558,80
215,90	609,60
222,25	660,40

Table A.30 — – Common fixed cutter bit sizes

Size of bit mm	Size of bit mm
98,43	215,90
114,30	222,25
120,65	241,30
149,23	250,83
152,40	269,88
155,58	311,15
158,75	374,65
165,10	406,40
171,45	444,50
200,03	

Annex B
(normative)

Tables in US Customary units

Table B.1 — New drill pipe dimensional data

Labels[a]		Outside diameter D	Nominal linear mass	Plain end linear mass[b]	Wall thickness t	Inside diameter d	Section area body of pipe A_{DP}[c]	Polar sectional modulus Z[d]
1	2	in	lbm/ft	lbm/ft	in	in	in^2	in^3
1	2	3	4	5	6	7	8	9
2 3/8	4.85	2.375	4.85	4.44	0.190	1.995	1.3042	1.321
	6.65	2.375	6.65	6.27	0.280	1.815	1.8429	1.733
2 7/8	6.85	2.875	6.85	6.17	0.217	2.441	1.8120	2.241
	10.40	2.875	10.40	9.72	0.362	2.151	2.8579	3.204
3 1/2	9.50	3.500	9.50	8.81	0.254	2.992	2.5902	3.923
	13.30	3.500	13.30	12.32	0.368	2.764	3.6209	5.144
	15.50	3.500	15.50	14.64	0.449	2.602	4.3037	5.847
4	11.85	4.000	11.85	10.47	0.262	3.476	3.0767	5.400
	14.00	4.000	14.00	12.95	0.330	3.340	3.8048	6.458
	15.70	4.000	15.70	14.71	0.380	3.240	4.3216	7.157
4 1/2	13.75	4.500	13.75	12.25	0.271	3.958	3.6004	7.184
	16.60	4.500	16.60	15.00	0.337	3.826	4.4074	8.543
	20.00	4.500	20.00	18.71	0.430	3.640	5.4981	10.232
	22.82	4.500	22.82	21.38	0.500	3.500	6.2832	11.345
5	16.25	5.000	16.25	14.88	0.296	4.408	4.3743	9.718
	19.50	5.000	19.50	17.95	0.362	4.276	5.2746	11.415
	25.60	5.000	25.60	24.05	0.500	4.000	7.0686	14.491
5 1/2	19.20	5.500	19.20	16.89	0.304	4.892	4.9624	12.221
	21.90	5.500	21.90	19.83	0.361	4.778	5.8282	14.062
	24.70	5.500	24.70	22.56	0.415	4.670	6.6296	15.688
6 5/8	25.20	6.625	25.20	22.21	0.330	5.965	6.5262	19.572
	27.70	6.625	27.70	24.24	0.362	5.901	7.1227	21.156

[a] Labels are for information and assistance in ordering.

[b] lbm/ft = 10.69(D – t)t

[c] $A_{DP} = 0.7854\,(D^2 - d^2)$

[d] $Z = 0.19635\,(D^4 - d^4) / D$

Table B.2 — New drill pipe torsional and tensile data

1	2	3	4	5	6	7	8	9	10	11	12
Labels [a]		**Outside diameter D in**	**Nominal linear mass T&C lbm/ft**	**Torsional data** [b] **torsional yield strength ft-lbs**				**Tensile data based on minimum values load at the minimum yield strength** [c] **lbs**			
1	2			E75	X95	G105	S135	E75	X95	G105	S135
2 3/8	4.85	2.375	4.85	4 763	6 033	6 668	8 584	97 817	123 902	136 944	176 071
	6.65	2.375	6.65	6 250	7 917	8 751	11 251	138 214	175 072	193 500	248 786
2 7/8	6.85	2.875	6.85	8 083	10 238	11 316	14 549	135 902	172 143	190 263	244 624
	10.40	2.875	10.40	11 554	14 635	16 176	20 798	214 344	271 503	300 082	385 820
3 1/2	9.50	3.500	9.50	14146	17 918	19 805	25 463	194 264	246 068	271 970	349 676
	13.30	3.500	13.30	18 551	23 498	25 972	33 392	271 569	343 988	380 197	488 825
	15.50	3.500	15.50	21 086	26 708	29 520	37 954	322 775	408 848	451 885	580 995
4	11.85	4.000	11.85	19 474	24 668	27 264	35 054	230 755	292 290	323 057	415 360
	14.00	4.000	14.00	23 288	29 498	32 603	41 918	285 359	361 454	399 502	513 646
	15.70	4.000	15.70	25 810	32 692	36 134	46 458	324 118	410 550	453 765	583 413
4 1/2	13.75	4.500	13.75	25 907	32 816	36 270	46 633	270 034	342 043	378 047	486 061
	16.60	4.500	16.60	30 807	39 022	43 130	55 453	330 558	418 707	462 781	595 004
	20.00	4.500	20.00	36 901	46 741	51 661	66 421	412 358	522 320	577 301	742 244
	22.82	4.500	22.82	40 912	51 821	57 276	73 641	471 239	596 903	659 734	848 230
5	16.25	5.000	16.25	35 044	44 389	49 062	63 079	328 073	415 559	459 302	590 531
	19.50	5.000	19.50	41 167	52 144	57 633	74 100	395 595	501 087	553 833	712 070
	25.60	5.000	25.60	52 257	66 192	73 159	94 062	530 144	671 515	742 201	954 259
5 1/2	19.20	5.500	19.20	44 074	55 826	61 703	79 332	372 181	471 429	521 053	669 925
	21.90	5.500	21.90	50 710	64 233	70 994	91 278	437 116	553 681	611 963	786 809
	24.70	5.500	24.70	56 574	71 660	79 204	101 833	497 222	629 814	696 111	894 999
6 5/8	25.20	6.625	25.20	70 580	89 402	98 812	127 044	489 464	619 988	685 250	881 035
	27.70	6.625	27.70	76 295	96 640	106 813	137 330	534 199	676 651	747 877	961 556

[a] Labels are for information and assistance in ordering.

[b] Based on the shear strength equal to 57.7 % of minimum yield strength and nominal wall thickness. Minimum torsional yield strength calculated from equation E.15

[c] Minimum tensile strength calculated from equation E.13

Table B.3 — New drill pipe collapse and internal pressure data)

1	2	3	4	5	6	7	8	9	10	11	12
Labels [a]		Outside diameter D in	Nominal linear mass T&C lbm/ft	Collapse pressure based on minimum values[b] psi				Internal pressure at minimum yield strength[b] psi			
1	2			E75	X95	G105	S135	E75	X95	G105	S135
2 3/8	4.85	2.375	4.85	11 040	13 984	15 456	19 035	10 500	13 300	14 700	18 900
	6.65	2.375	6.65	15 599	19 759	21 839	28 079	15 474	19 600	21 663	27 853
2 7/8	6.85	2.875	6.85	10 467	12 940	14 020	17 034	9 907	12 548	13 869	17 832
	10.40	2.875	10.40	16 509	20 911	23 112	29 716	16 526	20 933	23 137	29 747
3 1/2	9.50	3.500	9.50	10 001	12 077	13 055	15 748	9 525	12 065	13 335	17 145
	13.30	3.500	13.30	14 113	17 877	19 758	25 404	13 800	17 480	19 320	24 840
	15.50	3.500	15.50	16 774	21 247	23 484	30 194	16 838	21 328	23 573	30 308
4	11.85	4.000	11.85	8 381	9 978	10 708	12 618	8 597	10 889	12 036	15 474
	14.00	4.000	14.00	11 354	14 382	15 896	20 141	10 828	13 716	15 159	19 491
	15.70	4.000	15.70	12 896	16 335	18 055	23 213	12 469	15 794	17 456	22 444
4 1/2	13.75	4.500	13.75	7 173	8 412	8 956	10 283	7 904	10 012	11 066	14 228
	16.60	4.500	16.60	10 392	12 765	13 825	16 773	9 829	12 450	13 761	17 693
	20.00	4.500	20.00	12 964	16 421	18 149	23 335	12 542	15 886	17 558	22 575
	22.82	4.500	22.82	14 815	18 765	20 741	26 667	14 583	18 472	20 417	26 250
5	16.25	5.000	16.25	6 938	8 108	8 616	9 831	7 770	9 842	10 878	13 986
	19.50	5.000	19.50	9 962	12 026	12 999	15 672	9 503	12 037	13 304	17 105
	25.60	5.000	25.60	13 500	17 100	18 900	24 300	13 125	16 625	18 375	23 625
5 1/2	19.20	5.500	19.20	6 039	6 942	7 313	8 093	7 255	9 189	10 156	13 058
	21.90	5.500	21.90	8 413	10 019	10 753	12 679	8 615	10 912	12 061	15 507
	24.70	5.500	24.70	10 464	12 933	14 013	17 023	9 903	12 544	13 865	17 826
6 5/8	25.20	6.625	25.20	4 788	5 321	5 500	6 036	6 538	8 281	9 153	11 768
	27.70	6.625	27.70	5 894	6 755	7 103	7 813	7 172	9 084	10 040	12 909

[a] Labels are for information and assistance in ordering.

[b] Calculations are based on formulas in API Bulletin 5C3.

Table B.4 — Used drill pipe torsional and tensile data API premium class

1	2	3	4	5	6	7	8	9	10	11	12
Labels [a]		Outside diameter D in	Nominal linear mass T&C lbm/ft	Torsional data [b] Torsional yield strength ft-lbs				Tensile data based on minimum values Load at the minimum yield strength [c] lbs			
1	2			E75	X95	G105	S135	E75	X95	G105	S135
2 3/8	4.85	2.375	4.85	3 725	4 719	5 215	6 705	76 893	97 398	107 650	138 407
	6.65	2.375	6.65	4 811	6 093	6 735	8 659	107 616	136 313	150 662	193 709
2 7/8	6.85	2.875	6.85	6 332	8 020	8 865	11 397	106 946	135 465	149 725	192 503
	10.40	2.875	10.40	8 858	11 220	12 401	15 945	166 535	210 945	233 149	299 764
3 1/2	9.50	3.500	9.50	11 094	14 052	15 531	19 968	152 979	193 774	214 171	275 363
	13.30	3.500	13.30	14 361	18 191	20 106	25 850	212 150	268 723	297 010	381 870
	15.50	3.500	15.50	16 146	20 452	22 605	29 063	250 620	317 452	350 868	451 115
4	11.85	4.000	11.85	15 310	19 392	21 433	27 557	182 016	230 554	254 823	327 630
	14.00	4.000	14.00	18 196	23 048	25 474	32 752	224 182	283 963	313 854	403 527
	15.70	4.000	15.70	20 067	25 418	28 094	36 120	253 851	321 544	355 391	456 931
4 1/2	13.75	4.500	13.75	20 403	25 844	28 564	36 725	213 258	270 127	298 561	383 864
	16.60	4.500	16.60	24 139	30 576	33 795	43 450	260 165	329 542	364 231	468 297
	20.00	4.500	20.00	28 683	36 332	40 157	51 630	322 916	409 026	452 082	581 248
	22.82	4.500	22.82	31 587	40 010	44 222	56 856	367 566	465 584	514 593	661 620
5	16.25	5.000	16.25	27 607	34 969	38 650	49 693	259 155	328 263	362 817	466 479
	19.50	5.000	19.50	32 285	40 895	45 199	58 113	311 535	394 612	436 150	560 764
	25.60	5.000	25.60	40 544	51 356	56 762	72 979	414 690	525 274	580 566	746 443
5 1/2	19.20	5.500	19.20	34 764	44 035	48 670	62 575	294 260	372 730	411 965	529 669
	21.90	5.500	21.90	39 863	50 494	55 809	71 754	344 780	436 721	482 692	620 604
	24.70	5.500	24.70	44 320	56 139	62 048	79 776	391 285	495 627	547 799	704 313
6 5/8	25.20	6.625	25.20	55 766	71 522	79 050	101 635	387 466	490 790	542 452	697 438
	27.70	6.625	27.70	60 192	77 312	85 450	109 864	422 419	535 064	591 387	760 354

[a] Labels are for information and assistance in ordering.

[b] Based on the shear strength equal to 57.7 % of minimum yield strength and nominal wall thickness. Minimum torsional yield strength calculated from equation E.15

[c] Minimum tensile strength calculated from equation E.13

Table B.5 — Used drill pipe collapse and internal pressure data API premium class

1	2	3	4	5	6	7	8	9	10	11	12
Labels [a]		Outside diameter D in	Nominal linear mass T&C lbm/ft	Collapse pressure based on minimum values[b] psi				Internal pressure at minimum yield strength[b] psi			
1	2			E75	X95	G105	S135	E75	X95	G105	S135
2 3/8	4.85	2.375	4.85	8 522	10 161	10 912	12 891	9 600	12 160	13 440	17 280
	6.65	2.375	6.65	13 378	16 945	18 729	24 080	14 147	17 920	19 806	25 465
2 7/8	6.85	2.875	6.85	7 640	9 017	9 633	11 186	9 057	11 473	12 680	16 303
	10.40	2.875	10.40	14 223	18 016	19 912	25 602	15 110	19 139	21 153	27 197
3 1/2	9.50	3.500	9.50	7 074	8 284	8 813	10 093	709	11 031	12 192	15 675
	13.30	3.500	13.30	12 015	15 218	16 820	21 626	12 617	15 982	17 664	22 711
	15.50	3.500	15.50	14 472	18 331	20 260	26 049	15 394	19 499	21 552	27 710
4	11.85	4.000	11.85	5 704	6 508	6 827	7 445	7 860	9 956	11 004	14 148
	14.00	4.000	14.00	9 012	10 795	11 622	13 836	9 900	12 540	13 860	17 820
	15.70	4.000	15.70	10 914	13 825	15 190	18 593	11 400	14 440	15 960	20 520
4 1/2	13.75	4.500	13.75	4 686	5 190	5 352	5 908	7 227	9 154	10 117	13 008
	16.60	4.500	16.60	7 525	8 868	9 467	10 964	8 987	11 383	12 581	16 176
	20.00	4.500	20.00	10 975	13 901	15 350	18 806	11 467	14 524	16 053	20 640
	22.82	4.500	22.82	12 655	16 030	17 718	22 780	13 333	16 889	18 667	24 000
5	16.25	5.000	16.25	4 490	4 935	5 067	5 661	7 104	8 998	9 946	12 787
	19.50	5.000	19.50	7 041	8 241	8 765	10 029	8 688	11 005	12 163	15 638
	25.60	5.000	25.60	11 458	14 514	16 042	20 510	12 000	15 200	16 800	21 600
5 1/2	19.20	5.500	19.20	3 736	4 130	4 336	4 714	6 633	8 401	9 286	11 939
	21.90	5.500	21.90	5 730	6 542	6 865	7 496	7 876	9 977	11 027	14 177
	24.70	5.500	24.70	7 635	9 011	9 626	11 177	9 055	11 469	12 676	16 298
6 5/8	25.20	6.625	25.20	2 931	3 252	3 353	3 429	5 977	7 571	8 368	10 759
	27.70	6.625	27.70	3 615	4 029	4 222	4 562	6 557	8 306	9 180	11 803

[a] Labels are for information and assistance in ordering.

[b] Calculations are based on formulas in API Bulletin 5C3.

Table B.6 — Used drill pipe torsional and tensile data API class 2

1	2	3	4	5	6	7	8	9	10	11	12
Labels [a]		Outside diameter D in	Nominal linear mass T&C lbm/ft	Torsional data [b] Torsional yield strength ft-lbs				Tensile data based on minimum values Load at the minimum yield strength [c] lbs			
1	2			E75	X95	G105	S135	E75	X95	G105	S135
2 3/8	4.85	2.375	4.85	3 224	4 083	4 513	5 802	66 686	84 469	93 360	120 035
	6.65	2.375	6.65	4 130	5 232	5 782	7 434	92 871	117 636	130 019	167 167
2 7/8	6.85	2.875	6.85	5 484	6 946	7 677	9 871	92 801	117 549	129 922	167 043
	10.40	2.875	10.40	7 591	9 615	10 627	13 663	143 557	181 839	200 980	258 403
3 1/2	9.50	3.500	9.50	9 612	12 176	13 457	17 302	132 793	168 204	185 910	239 027
	13.30	3.500	13.30	12 365	15 663	17 312	22 258	183 398	232 304	256 757	330 116
	15.50	3.500	15.50	13 828	17 515	19 359	24 890	215 967	273 558	302 354	388 741
4	11.85	4.000	11.85	13 281	16 823	18 594	23 907	158 132	200 301	221 385	284 638
	14.00	4.000	14.00	15 738	19 935	22 034	28 329	194 363	246 193	272108	349 852
	15.70	4.000	15.70	17 315	21 932	24 241	31 168	219 738	278 335	307 633	395 528
4 1/2	13.75	4.500	13.75	17 715	22 439	24 801	31 887	185 389	234 827	259 545	333 701
	16.60	4.500	16.60	20 908	26 483	29 271	37 634	225 771	285 977	316 080	406 388
	20.00	4.500	20.00	24 747	31 346	34 645	44 544	279 502	354 035	391 302	503 103
	22.82	4.500	22.82	27 161	34 404	38 026	48 890	317 497	402 163	444 496	571 495
5	16.25	5.000	16.25	23 974	30 368	33 564	43 154	225 316	285 400	315 442	405 568
	19.50	5.000	19.50	27 976	35 436	39 166	50 356	270 432	342 548	378 605	486 778
	25.60	5.000	25.60	34 947	44 267	48 926	62 905	358 731	454 392	502 223	645 715
5 1/2	19.20	5.500	19.20	30 208	38 263	42 291	54 374	255 954	324 208	358 335	460 717
	21.90	5.500	21.90	34 582	43 804	48 414	62 247	299 533	379 409	419 346	539 160
	24.70	5.500	24.70	38 383	48 619	53 737	69 090	339 533	430 076	475 347	611 160
6 5/8	25.20	6.625	25.20	48497	61 430	67 896	87 295	337 236	427 166	472 131	607 026
	27.70	6.625	27.70	52 308	66 257	73 231	94 155	367 455	465 443	514 437	661 419

[a] Labels are for information and assistance in ordering.

[b] Based on the shear strength equal to 57.7 % of minimum yield strength and nominal wall thickness. Minimum torsional yield strength calculated from equation E.15

[c] Minimum tensile strength calculated from equation E.13

Table B.7 — Used drill pipe collapse and internal pressure data API class 2

1	2	3	4	5	6	7	8	9	10	11	12
Labels [a]		Outside diameter D in	Nominal linear mass T&C lbm/ft	Collapse pressure based on minimum values[b] psi				Internal pressure at minimum yield strength[b] psi			
1	2			E75	X95	G105	S135	E75	X95	G105	S135
2 3/8	4.85	2.375	4.85	6 852	7 996	8 491	9 664	8 400	10 640	11 760	15 120
	6.65	2.375	6.65	12 138	15 375	16 993	21 849	12 379	15 680	17 331	22 282
2 7/8	6.85	2.875	6.85	6 055	6 963	7 335	8 123	7 925	10 039	11 095	14 265
	10.40	2.875	10.40	12 938	16 388	18 113	23 288	13 221	16 746	18 509	23 798
3 1/2	9.50	3.500	9.50	5 544	6 301	6 596	7 137	7 620	9 652	10 668	13 716
	13.30	3.500	13.30	10 858	13 753	15 042	18 396	11 040	13 984	15 456	19 872
	15.50	3.500	15.50	13 174	16 686	18 443	23 712	13 470	17 062	18 858	24 246
4	11.85	4.000	11.85	4 311	4 702	4 876	5 436	6 878	8 712	9 629	12 380
	14.00	4.000	14.00	7 295	8 570	9 134	10 520	8 663	10 973	12 128	15 593
	15.70	4.000	15.70	9 531	11 468	12 374	14 840	9 975	12 635	13 965	17 955
4 1/2	13.75	4.500	13.75	3 397	3 845	4 016	4 287	6 323	8 010	8 853	11 382
	16.60	4.500	16.60	5 951	6 828	7 185	7 923	7 863	9 960	11 009	14 154
	20.00	4.500	20.00	9 631	11 598	12 520	15 033	10 033	12 709	14 047	18 060
	22.82	4.500	22.82	11 458	14 514	16 042	20 510	11 667	14 779	16 333	21 000
5	16.25	5.000	16.25	3 275	3 696	3 850	4 065	6 216	7 874	8 702	11 189
	19.50	5.000	19.50	5 514	6 262	6 552	7 079	7 602	9 629	10 643	13 684
	25.60	5.000	25.60	10 338	12 640	13 685	16 587	10 500	13 300	14 700	18 900
5 1/2	19.20	5.500	19.20	2 835	3 128	3 215	3 265	5 804	7 351	8 125	10 447
	21.90	5.500	21.90	4 334	4 733	4 899	5 465	6 892	8 730	9 649	12 405
	24.70	5.500	24.70	6 050	6 957	7 329	8 115	7 923	10 035	11 092	14 261
6 5/8	25.20	6.625	25.20	2 227	2 343	2 346	2 346	5 230	6 625	7 322	9 414
	27.70	6.625	27.70	2 765	3 037	3 113	3 148	5 737	7 267	8 032	10 327

[a] Labels are for information and assistance in ordering.

[b] Calculations are based on formulas in API Bulletin 5C3.

Table B.8 — Mechanical properties of new tool joints and New Grade E75 drill pipe

1	2	3	4	5	6	7	8	9	10	11	12	13	14
										Mechanical properties			
Drill pipe data						Tool joint data				Tensile yield		Torsional yield	
Labels		Nominal size in	Nominal weight	Approx. weight[a]	Type upset	Conn.	OD	ID	Drift dia.[b]	Pipe[c]	Tool Joint[d]	Pipe[e]	Tool Joint[f]
1	2	in	lbm/ft	lbm/ft			in	in	in	lb	lb	ft-lb	ft-lb
2 3/8	4.85	2.375	4.85	5.26	EU	NC 26	3 3/8	1 3/4	1.625	97 817	313 681	4 763	6 875 b
		2.375	4.85	4.95	EU	OH	3 1/8	2	1.807	97 817	206 416	4 763	4 526 p
		2.375	4.85	5.05	EU	SLH90	3 1/4	2	1.850	97 817	202 670	4 763	5 075 p
		2.375	4.85	5.15	EU	WO	3 3/8	2	1.807	97 817	195 677	4 763	4 235 p
	6.65	2.375	6.65	6.99	EU	NC 26	3 3/8	1 3/4	1.625	138 214	313 681	6 250	6 875 b
		2.375	6.65	6.89	EU	OH	3 1/4	1 3/4	1.625	138 214	294 620	6 250	6 305 b
		2.375	6.65	6.71	IU	PAC	2 7/8	1 3/8	1.250	138 214	238 504	6 250	4 672 p
		2.375	6.65	6.78	EU	SLH90	3 1/4	2	1.670	138 214	202 850	6 250	5 075 p
2 7/8	6.85	2.875	6.85	7.50	EU	NC31	4 1/8	2 1/8	2.000	135 902	447 130	8 083	11 790 p
		2.875	6.85	6.93	EU	OH	3 3/4	2 7/16	2.253	135 902	223 937	8 083	5 464 P
		2.875	6.85	7.05	EU	SLH90	3 7/8	2 7/16	2.296	135 902	260 783	8 083	7 513 p
		2.875	6.85	7.31	EU	WO	4 1/8	2 7/16	2.253	135 902	277 553	8 083	7 015 p
	10.40	2.875	10.40	10.87	EU	NC31	4 1/8	2 1/8	1.963	214 344	447 130	11 554	11 790 p
		2.875	10.40	10.59	EU	OH	3 7/8	2 5/32	1.963	214 344	345 566	11 554	8 659 P
		2.875	10.40	10.27	IU	PAC	3 1/8	1 1/2	1.375	214 344	272 938	11554	5 706 P
		2.875	10.40	10.59	EU	SLH90	3 7/8	2 5/32	2.006	214 344	382 765	11 554	11 227 p
		2.875	10.40	11.19	IU	XH	4 1/4	1 7/8	1.750	214 344	505 054	11 554	13 088 p
		2.875	10.40	10.35	IU	NC26	3 3/8	1 3/4	1.625	214 344	313 681	11 554	6 875 B
3 1/2	9.50	3.500	9.50	10.58	EU	NC38	4 3/4	2 11/16	2.563	194 264	587 308	14 146	18 071 p
		3.500	9.50	9.84	EU	OH	4 1/2	3	2.804	194 264	392 071	14 146	11 803 p
		3.500	9.50	9.99	EU	SLH90	4 5/8	3	2.847	194 264	366 705	14 146	12 458 p
		3.500	9.50	10.14	EU	WO	4 3/4	3	2.804	194 264	419 797	14 146	12 723 p
	13.30	3.500	13.30	14.37	EU	H90	5 1/4	2 3/4	2.619	271 569	664 050	18 551	23 443 p
		3.500	13.30	13.93	EU	NC38	4 3/4	2 11/16	2.457	271 569	587 308	18 551	18 071 p
		3.500	13.30	13.75	EU	OH	4 3/4	2 11/16	2.414	271 569	559 582	18 551	17 167 p
		3.500	13.30	13.40	IU	NC31	4 1/8	2 1/8	2.000	271 569	447 130	18 551	11 790 P
		3.500	13.30	13.91	EU	XH	4 3/4	2 7/16	2.313	271 569	570 939	18 551	16 867 p
	15.50	3.500	15.50	16.54	EU	NC38	5	2 9/16	2.414	322 775	649 158	21 086	20 095 p
See end of table for notes.													

1	2	3	4	5	6	7	8	9	10	11	12	13	14
										Mechanical properties			
Drill pipe data						Tool joint data				Tensile yield		Torsional yield	
Labels		Nominal size in	Nominal weight	Approx. weight [a]	Type upset	Conn.	OD	ID	Drift dia.[b]	Pipe[c]	Tool Joint[d]	Pipe[e]	Tool Joint[f]
1	2	in	lbm/ft	lbm/ft			in	in	in	lb	lb	ft-lb	ft-lb
4	11.85	4.000	11.85	13.00	IU	H90	5 1/2	2 13/16	2.688	230 755	913 708	19 474	35 308 p
		4.000	11.85	13.52	EU	NC46	6	3 1/4	3.125	230 755	901 164	19 474	33 228 p
		4.000	11.85	12.10	EU	OH	5 1/4	3 15/32	3.287	230 755	621357	19 474	21 903 p
		4.000	11.85	12.91	EU	WO	5 3/4	3 7/16	3.313	230 755	782 987	19 474	28 643 p
	14.00	4.000	14.00	15.04	IU	NC40	5 1/4	2 13/16	2.688	285 359	711 611	23 288	23 279 p
		4.000	14.00	15.43	IU	H90	5 1/2	2 13/16	2.688	285 359	913 708	23 288	35 308 p
		4.000	14.00	15.85	EU	NC46	6	3 1/4	3.125	285 359	901 164	23 288	33 228 p
		4.000	14.00	15.02	EU	OH	5 1/2	3 1/4	3.125	285 359	759 875	23 288	27 060 p
		4.000	14.00	14.35	IU	SH	4 5/8	2 9/16	2.438	285 359	512 035	23 288	15 026 p
	15.70	4.000	15.70	16.80	IU	NC40	5 1/4	2 11/16	2.563	324 118	776 406	25 810	25 531p
		4.000	15.70	17.09	IU	H90	5 1/2	2 13/16	2.688	324 118	913 708	25 810	35 308 p
		4.000	15.70	17.54	EU	NC46	6	3 1/4	3.095	324 118	901 164	25 810	33 228 p
4 1/2	13.75	4.500	13.75	15.23	IU	H90	6	3 1/4	3.125	270 034	938 403	25 907	38 544 p
		4.500	13.75	15.36	EU	NC50	6 5/8	3 3/4	3.625	270 034	939 096	25 907	37 269 p
		4.500	13.75	14.04	EU	OH	5 3/4	3 31/32	3.770	270 034	554 844	25 907	20 678 p
		4.500	13.75	14.77	EU	WO	6 1/8	3 7/8	3.750	270 034	848 619	25 907	33 492 p
	16.60	4.500	16.60	18.14	IEU	FH	6	3	2.875	330 558	976 156	30 807	34 367 p
		4.500	16.60	17.92	IEU	H90	6	3 1/4	3.125	330 558	938 403	30 807	38 544 p
		4.500	16.60	17.95	EU	NC50	6 5/8	3 3/4	3.625	330 558	939 096	30 807	37 269 p
		4.500	16.60	17.07	EU	OH	5 7/8	3 3/4	3.625	330 558	713 979	30 807	26 936 p
		4.500	16.60	16.79	IEU	NC38	5	2 11/16	2.563	330 558	587 308	30 807	18 071 p
		4.500	16.60	18.37	IEU	NC46	6 1/4	3 1/4	3.125	330 558	901 164	30 807	33 228 p
	20.00	4.500	20.00	21.64	IEU	FH	6	3	2.875	412 358	976 156	36 901	34 367 p
		4.500	20.00	21.64	IEU	H90	6	3	2.875	412 358	1 085 665	36 901	44 948 p
		4.500	20.00	21.59	EU	NC50	6 5/8	3 5/8	3.452	412 358	1 025 980	36 901	40 915 p
		4.500	20.00	22.09	IEU	NC46	6 1/4	3	2.875	412358	1 048 426	36 901	38 998 p
	22.82	4.500	22.82	24.11	EU	NC50	6 5/8	3 5/8	3.452	471 239	1 025 980	40 912	40 915 p
		4.500	22.82	24.56	IEU	NC46	6 1/4	3	2.875	471 239	1 048 426	40 912	38 998 p
See end of table for notes.													

1	2	3	4	5	6	7	8	9	10	11	12	13	14
										Mechanical properties			
Drill pipe data						**Tool joint data**				**Tensile yield**		**Torsional yield**	
Labels		**Nominal size**	**Nominal weight**	**Approx. weight** [a]	**Type upset**	**Conn.**	**OD**	**ID**	**Drift dia.** [b]	**Pipe** [c]	**Tool Joint** [d]	**Pipe** [e]	**Tool Joint** [f]
1	2	in	lbm/ft	lbm/ft			in	in	in	lb	lb	ft-lb	ft-lb
5	19.50	5.000	19.50	22.28	IEU	5 1/2 FH	7	3 3/4	3.625	395 595	1 448 407	41 167	62 903 b
		5.000	19.50	20.85	IEU	NC50	6 5/8	3 3/4	3.625	395 595	939 095	41 167	37 269 p
	25.60	5.000	25.60	28.27	IEU	5 1/2 FH	7	3 1/2	3.375	530 144	1 619 231	52 257	62 903 p
		5.000	25.60	26.85	IEU	NC50	6 5/8	3 1/2	3.375	530 144	1 109 920	52 257	44 456 p
5 1/2	21.90	5.500	21.90	23.78	IEU	FH	7	4	3.875	437 116	1 265 802	50 710	55 687 p
	24.70	5.500	24.70	26.30	IEU	FH	7	4	3.875	497 222	1 265 802	56 574	55 687 p
6 5/8	25.20	6.625	25.20	27.28	IEU	FH	8	5	4.875	489 464	1 447 697	70 580	73 224 p
	27.70	6.625	27.70	29.06	IEU	FH	8	5	4.875	534 198	1 447 697	76 295	73 224 p

[a] Tool joint plus drill pipe. for range 2 steel pipe (See appendix A for method of calculation).

[b] See subclause 4.4.

[c] The tensile yield strength of Grade E drill pipe is based on 75 000 psi minimum yield strength.

[d] The tensile strength of the tool joint pin is based on 120 000 psi minimum yield and the cross sectional area at the root of the thread 15.875 mm (5/8 inch) from the shoulder.

[e] The torsional yield strength is based on a shear strength of 57.7 % of the minimum yield strength.

[f] p = pin limited yield; b = box limited yield; P or B indicates that tool joint could not meet 80 % of tube torsional yield.

Table B.9 — Mechanical properties of new tool joints and new high strength drill pipe

1	2	3	4	5	6	7	8	9	10	11	12	13	14
Drill pipe data						Tool joint data				Mechanical properties			
Labels [a]		Nominal size	Nominal linear mass	Approx. weight [b]	Type upset & Pipe grade	Conn.	OD	ID	Drift Diameter [c]	Tensile yield		Torsional yield	
										Pipe [d]	Tool Joint [e]	Pipe [d]	Tool Joint [f]
1	2	in.	lbm/ft	lbm/ft			in.	in.	in.	lb	lb	ft-lbf	ft-lbf
2 3/8	6.65	2.375	6.65	7.11	EU-X95	NC26	3 3/8	1 3/4	1.625	175 072	313 681	7 917	6 875 b
		2.375	6.65	6.99	EU-X95	SLH90	3 1/4	1 13/16	1.670	175 072	270 223	7 917	6 862 p
		2.375	6.65	7.11	EU-G105	NC26	3 3/8	1 3/4	1.625	193 500	313 681	8 751	6 875 b
		2.375	6.65	6.99	EU-G105	SLH90	3 1/4	1 13/16	1.670	193 500	270 223	8 751	6 862 P
2 7/8	10.40	2.875	10.40	11.09	EU-X95	NC31	4 1/8	2	1.875	271 503	495 726	14 635	13 158 p
		2.875	10.40	10.95	EU-X95	SLH90	4	2	1.875	271 503	443 971	14 635	13 119 p
		2.875	10.40	11.09	EU-G105	NC31	4 1/8	2	1.875	300 082	495 726	16 176	13 158 p
		2.875	10.40	10.95	EU-G105	SLH90	4	2	1.875	300 082	443 971	16 176	13 119 p
		2.875	10.40	11.55	EU-S135	NC31	4 3/8	1 5/8	1.500	385 820	623 844	20 798	16 809 p
		2.875	10.40	11.26	EU-S135	SLH90	4 1/8	1 5/8	1.500	385 820	572 089	20 798	17 130 p
3 1/2	13.30	3.500	13.30	14.60	EU-X95	H90	5 1/4	2 3/4	2.619	343 988	664 050	23 498	23 443 p
		3.500	13.30	14.62	EU-X95	NC38	5	2 9/16	2.438	343 988	649 158	23 498	20 095 p
		3.500	13.30	14.06	EU-X95	SLH90	4 3/4	2 9/16	2.438	343 988	596 066	23 498	20 709 p
		3.500	13.30	14.71	EU-G105	NC38	5	2 7/16	2.313	380 197	708 063	25 972	23 035 p
		3.500	13.30	14.06	EU-G105	SLH90	4 3/4	2 9/16	2.438	380 197	596 066	25 972	20 709 p
		3.500	13.30	14.92	EU-S135	NC38	5	2 1/8	2.000	488 825	842 440	33 392	26 503 p
		3.500	13.30	14.65	EU-S135	SLH90	5	2 1/8	2.000	488 825	789 348	33 392	27 809 p
		3.500	13.30	15.13	EU-S135	NC40	5 3/8	2 7/16	2.313	488 825	897 161	33 392	29 764 p
	15.50	3.500	15.50	16.82	EU-X95	NC38	5	2 7/16	2.313	408 848	708 063	26 708	22 035 p
		3.500	15.50	17.03	EU-G105	NC38	5	2 1/8	2.000	451 885	842 440	29 520	26 503 p
		3.500	15.50	16.97	EU-G105	NC40	5 1/4	2 9/16	2.438	451 885	838 257	29 520	27 693 p
		3.500	15.50	17.57	EU-S135	NC40	5 1/2	2 1/4	2.125	580 995	979 996	37 954	32 693 p
4	14.00	4.000	14.00	15.34	IU-X95	NC40	5 1/4	2 11/16	2.563	361 454	776 406	29 498	25 531 p
		4.000	14.00	15.63	IU-X95	H90	5 1/2	2 13/16	2.688	361 454	913 708	29 498	35 308 p
		4.000	14.00	16.19	EU-X95	NC46	6	3 1/4	3.125	361 454	901 164	29 498	33 228 p
		4.000	14.00	15.91	IU-G105	NC40	5 1/2	2 7/16	2.313	399 502	897 161	32 603	29 764 p
		4.000	14.00	15.63	IU-G105	H90	5 1/2	2 13/16	2.688	399 502	913 708	32 603	35 308 p
		4.000	14.00	16.19	EU-G105	NC46	6	3 1/4	3.125	399 502	901 164	32 603	33 228 p
See end of table for notes.													

1	2	3	4	5	6	7	8	9	10	11	12	13	14
Drill pipe data						Tool joint data				Mechanical properties			
Labels [a]		Nominal size	Nominal linear mass	Approx. weight [b]	Type upset & Pipe grade	Conn.	OD	ID	Drift Diameter [c]	Tensile yield		Torsional yield	
										Pipe [d]	Tool Joint [e]	Pipe [d]	Tool Joint [f]
1	2	in.	lbm/ft	lbm/ft			in.	in.	in.	lb	lb	ft-lbf	ft-lbf
4	14.00	4.000	14.00	16.19	IU-S135	NC40	5 1/2	2	1.875	513 646	1 080 135	41 918	36 262 p
		4.000	14.00	15.63	IU-S135	H90	5 1/2	2 13/16	2.688	513 646	913 708	41 918	35 308 p
		4.000	14.00	16.42	EU-S135	NC46	6	3	2.875	513 646	1 048 426	41 918	38 998 p
	15.70	4.000	15.70	17.52	IU-X95	NC40	5 1/2	2 7/16	2.313	410 550	897 161	32 692	29 764 p
		4.000	15.70	17.23	IU-X95	H90	5 1/2	2 13/16	2.688	410 550	913 708	32 692	35 308 p
		4.000	15.70	17.80	EU-X95	NC46	6	3 1/4	3.125	410 550	901 164	32 692	33 228 p
		4.000	15.70	17.52	IU-G105	NC40	5 1/2	2 7/16	2.313	453 765	897 161	36 134	29 764 p
		4.000	15.70	17.23	IU-G105	H90	5 1/2	2 13/16	2.688	453 765	913 708	36 134	35 308 p
		4.000	15.70	17.80	EU-G105	NC46	6	3 1/4	3.125	453 765	901 164	36 134	33 228 p
		4.000	15.70	18.02	EU-S135	NC46	6	3	2.875	583 413	1 048 426	46 458	38 998 p
4 1/2	16.60	4.500	16.60	18.33	IEU-X95	FH	6	3	2.875	418 707	976 156	39 022	34 367 p
		4.500	16.60	18.11	IEU-X95	H90	6	3 1/4	3.125	418 707	938 403	39 022	38 544 p
		4.500	16.60	18.36	EU-X95	NC50	6 5/8	3 3/4	3.625	418 707	939 095	39 022	37 269 p
		4.500	16.60	18.79	IEU-X95	NC46	6 1/4	3	2.875	418 707	1 048 426	39 022	38 998 p
		4.500	16.60	18.33	IEU-G105	FH	6	3	2.625	462 781	976 156	43 130	34 367 p
		4.500	16.60	18.33	IEU-G105	H90	6	3	3.125	462 781	1 085 665	43 130	44 948 p
		4.500	16.60	18.36	EU-G105	NC50	6 5/8	3 3/4	3.625	462 781	939 095	43 130	37 269 p
		4.500	16.60	18.79	IEU-G105	NC46	6 1/4	3	2.875	462 781	1 048 426	43 130	38 998 p
		4.500	16.60	19.19	IEU-S135	FH	6 1/4	2 1/2	2.375	595 004	1 235 337	55 453	44 214 p
		4.500	16.60	18.33	IEU-S135	H90	6	3	2.875	595 004	1 085 665	55 453	44 948 p
		4.500	16.60	18.62	EU-S135	NC50	6 5/8	3 1/2	3.375	595 004	1 109 920	55 453	44 456 p
		4.500	16.60	19.00	IEU-S135	NC46	6 1/4	2 3/4	2.625	595 004	1 183 908	55 453	44 359 p
	20.00	4.500	20.00	22.39	IEU-X95	FH	6	2 1/2	2.375	522 320	1 235 337	46 741	44 214 p
		4.500	20.00	21.78	IEU-X95	H90	6	3 1/4	3.125	522 320	938 403	46 741	38 544 p
		4.500	20.00	22.08	EU-X95	NC50	6 5/8	3 1/2	3.375	522 320	1 109 920	46 741	44 456 p
		4.500	20.00	22.67	IEU-X95	NC46	6 1/4	2 3/4	2.625	522 320	1 183 908	46 741	44 359 p
		4.500	20.00	22.39	IEU-G105	FH	6	2 1/2	2.375	577 301	1 235 337	51 661	44 214 p
		4.500	20.00	22.00	IEU-G105	H90	6	3	2.875	577 301	1 085 665	51 661	44 948 p
		4.500	20.00	22.08	EU-G105	NC50	6 5/8	3 1/2	3.375	577 301	1 109 920	51 661	44 456 p
		4.500	20.00	22.86	IEU-G105	NC46	6 1/4	2 1/2	2.375	577 301	1 307 608	51 661	49 297 p
		4.500	20.00	23.03	EU-S135	NC50	6 5/8	3	2.875	742 244	1 416 225	66 421	57 534 p
		4.500	20.00	23.03	IEU-S135	NC46	6 1/4	2 1/4	2.125	742 244	1 419 527	66 421	53 800 p

See end of table for notes.

1	2	3	4	5	6	7	8	9	10	11	12	13	14
Drill pipe data						Tool joint data				Mechanical properties			
Labels [a]		Nominal size	Nominal linear mass	Approx. weight [b]	Type upset & Pipe grade	Conn.	OD	ID	Drift Diameter [c]	Tensile yield		Torsional yield	
										Pipe [d]	Tool Joint [e]	Pipe [d]	Tool Joint [f]
1	2	in.	lbm/ft	lbm/ft			in.	in.	in.	lb	lb	ft-lbf	ft-lbf
4 ½	22.82	4.500	22.82	25.13	IEU-X95	FH	6 1/4	2 1/4	2.125	596 903	1 347 256	51 821	48 522 p
		4.500	22.82	24.24	EU-X95	NC50	6 5/8	3 1/2	3.375	596 903	1 109 920	51 821	44 456 p
		4.500	22.82	24.77	IEU-X95	NC46	6 1/4	2 3/4	2.625	596 903	1 183 908	51 821	44 359 p
		4.500	22.82	24.72	EU-G105	NC50	6 5/8	3 1/4	3.125	659 735	1268 963	57 276	51 217 p
		4.500	22.82	24.96	IEU-G105	NC46	6 1/4	2 1/2	2.375	659 735	1 307 608	57 276	49 297 p
		4.500	22.82	25.41	EU-S135	NC50	6 5/8	2 3/4	2.625	848 230	1 551 706	73 641	63 393 p
5	19.50	5.000	19.50	22.62	IEU-X95	5 1/2 FH	7	3 3/4	3.625	501 087	1 448 407	52 144	62 903 b
		5.000	19.50	21.93	IEU-X95	H90	6 1/2	3 1/4	3.125	501 087	1 176 265	52 144	51 220 p
		5.000	19.50	21.45	IEU-X95	NC50	6 5/8	3 1/2	3.375	501 087	1 109 920	52 144	44 456 p
		5.000	19.50	22.62	IEU-G105	5 1/2 FH	7	3 3/4	3.625	553 833	1 448 407	57 633	62 903 b
		5.000	19.50	22.15	IEU-G105	H90	6 1/2	3	2.875	553 833	1 323 527	57 633	58 008 p
		5.000	19.50	21.93	IEU-G105	NC50	6 5/8	3 1/4	3.125	553 833	1 268 963	57 633	51 217 p
		5.000	19.50	23.48	IEU-S135	5 1/2 FH	7 1/4	3 1/2	3.375	712 070	1 619 231	74 100	72 213 p
		5.000	19.50	22.61	IEU-S135	NC50	6 5/8	2 3/4	2.625	712 070	1 551 706	74 100	63 393 p
	25.60	5.000	25.60	28.59	IEU-X95	5 1/2 FH	7	3 1/2	3.375	671 515	1 619 231	66 192	62 903 b
		5.000	25.60	27.87	IEU-X95	NC50	6 5/8	3	2.875	671 515	1 416 225	66 192	57 534 b
		5.000	25.60	29.16	IEU-G105	5 1/2 FH	7 1/4	3 1/2	3.375	742 201	1 619 231	73 159	72 213 p
		5.000	25.60	28.32	IEU-G105	NC50	6 5/8	2 3/4	2.625	742 201	1 551 706	73 159	63 393 b
		5.000	25.60	29.43	IEU-S135	5 1/2 FH	7 1/4	3 1/4	3.125	954 259	1 778 274	94 062	78 716 b
5 1/2	21.90	5.500	21.90	24.53	IEU-X95	FH	7	3 3/4	3.625	553 681	1 448 407	64 233	62903 b
		5.500	21.90	24.80	IEU-X95	H90	7	3 1/2	3.125	553 681	1 268 877	64 233	58 033 p
		5.500	21.90	25.38	IEU-G105	FH	7 1/4	3 1/2	3.375	611 963	1 619 231	70 994	72 213 p
		5.500	21.90	26.50	IEU-S135	FH	7 1/2	3	2.875	786 809	1 925 536	91 278	86 765 p
	24.70	5.500	24.70	27.85	IEU-X95	FH	7 1/4	3 1/2	3.375	629 814	1 619 231	71 660	72 213 p
		5.500	24.70	27.85	IEU-G105	FH	7 1/4	3 1/2	3.375	696 111	1 619 231	79 204	72 213 p
		5.500	24.70	27.77	IEU-S135	FH	7 1/2	3	2.875	894 999	1 925 536	101 833	86 765 p
See end of table for notes.													

1	2	3	4	5	6	7	8	9	10	11	12	13	14
Drill pipe data						**Tool joint data**				**Mechanical properties**			
Labels [a]		**Nominal size**	**Nominal linear mass**	**Approx. weight** [b]	**Type upset & Pipe grade**	**Conn.**	**OD**	**ID**	**Drift Diameter** [c]	**Tensile yield**		**Torsional yield**	
										Pipe [d]	**Tool Joint** [e]	**Pipe** [d]	**Tool Joint** [f]
1	2	in.	lbm/ft	lbm/ft			In.	In.	In.	lbf	lbf	ft-lbf	ft-lbf
6 5/8	25.20	6.625	25.20	27.15	IEU-X95	FH	8	5	4.875	619 988	1 448 416	89 402	73 224 p
		6.625	25.20	28.20	IEU-G105	FH	8 1/4	4 3/4	4.625	685 250	1 678 145	98 812	85 467 p
		6.625	25.20	29.63	IEU-S135	FH	8 1/2	4 1/4	4.125	881 035	2 102 260	127 044	108 353 p
	27.70	6.625	27.70	30.11	IEU-X95	FH	8 1/4	4 3/4	4.625	676 651	1 678 145	96 640	85 467 p
		6.625	27.70	30.11	IEU-G105	FH	8 1/4	4 3/4	4.625	747 250	1 678 145	106 813	85 467 p
		6.625	27.70	31.54	IEU-S135	FH	8 1/2	4 1/4	4.125	961 556	2 102 260	137 330	108 353 p

[a] Labels are for information and assistance in ordering.

[b] Tool joint plus drill pipe. for Range 2 steel pipe (see appendix E for method of calculation).

[c] See subclause 4.4.

[d] The torsional yield strength is based on a shear strength of 57.7 % of the minimum yield strength.

[e] The tensile strength of the tool joint pin is based on 120.000 psi yield and the cross sectional area at the root of the thread 5/8 inch from the shoulder.

[f] p = pin limited yield; b = box limited yield; P or B indicates that tool joint could not meet 80 % of tube torsional yield.

Table B.10 — Recommended minimum OD and make-up torque of weld-on type tool joints Based on torsional strength of box and drill pipe

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
		Drill pipe data			New tool joint data				Premium class			Class 2		
Labels[a]		Nom OD	Nom linear mass	Type upset and pipe grade	Connection	New OD	New ID	Make-up torque[d]	Min OD tool joint[c,f]	Min box shoulder with eccentric wear[e]	Make-up torque for min OD tool joint[d]	Min OD tool joint[c,f]	Min box shoulder with eccentric wear[e]	Make-up torque for min OD tool joint[d]
1	2	in	lbm/ft			in	in	ft-lbf	in	in	ft-lbf	in	in	ft-lbf
2 3/8	4.85	2 3/8	4.85	EU-E75	NC26	3 3/8	1 3/4	4.125 B	3 1/8	3/64	1.945	3 3/32	1/32	1.689
		2 3/8	4.85	EU-E75	W.O.	3 3/8	2	2.541 P	3 1/16	1/16	1.994	3 1/32	3/64	1.746
		2 3/8	4.85	EU-E75	2 3/8 OHLW	3 1/8	2	2.716 P	3	1/16	1.723	2 31/32	3/64	1.481
		2 3/8	4.85	EU-E75	2 3/8 SL-H90	3 1/4	2	3.042 P	2 31/32	1/16	1.996	2 15/16	3/64	1.726
	6.65	2 3/8	6.65	IU-E75	2 3/8 PAC[b]	2 7/8	1 3/8	2.803 P	2 25/32	9/64	2.455	2 23/32	7/64	2.055
		2 3/8	6.65	EU-E75	NC26	3 3/8	1 3/4	4.125 B	3 3/16	5/64	2.467	3 5/32	1/16	2.204
		2 3/8	6.65	EU-E75	2 3/8 SL-H90	3 1/4	2	3.042 P	3 1/32	3/32	2.549	2 31/32	1/16	1.996
		2 3/8	6.65	EU-E75	2 3/8 OHSW	3 1/4	1 3/4	3.783 B	3 1/16	3/32	2.216	3 1/32	5/64	1.967
		2 3/8	6.65	EU-X95	NC26	3 3/8	1 3/4	4.125 B	3 1/4	7/64	3.005	3 7/32	3/32	2.734
		2 3/8	6.65	EU-G105	NC26[b]	3 3/8	1 3/4	4.125 B	3 9/32	1/8	3.279	3 1/4	7/64	3.005
2 7/8	6.85	2 7/8	6.85	EU-E75	NC31	4 1/8	2 1/8	7.074 P	3 11/16	5/64	3.154	3 21/32	1/16	2.804
		2 7/8	6.85	EU-E75	2 7/8 WO	4 1/8	2 7/16	4.209 P	3 5/8	5/64	3.216	3 19/32	1/16	2.876
		2 7/8	6.85	EU-E75	2 7/8 OHLW[b]	3 3/4	2 7/16	3.290 P	3 1/2	7/64	3.290	3 7/16	5/64	2.804
		2 7/8	6.85	EU-E75	2 7/8 SL-H90	3 7/8	2 7/16	4.504 P	3 1/2	3/32	3.397	3 7/16	1/16	2.666
	10.40	2 7/8	10.40	EU-E75	NC31	4 1/8	2 1/8	7.074 P	3 13/16	9/64	4.597	3 3/4	7/64	3.867
		2 7/8	10.40	IU-E75	2 7/8 XH	4 1/4	1 7/8	7.853 P	3 23/32	9/64	4.357	3 21/32	7/64	3.664
		2 7/8	10.40	IU-E75	NC26[b]	3 3/8	1 3/4	4.125 B	3 3/8	11/64	4.125	3 11/32	5/32	3.839
		2 7/8	10.40	EU-E75	2 7/8 OHSW[b]	3 7/8	2 5/32	5.194 P	3 19/32	5/32	4.411	3 9/16	7/64	4.079
		2 7/8	10.40	EU-E75	2 7/8 SL-H90	3 7/8	2 5/32	6.732 P	3 19/32	9/64	4.529	3 17/32	7/64	3.770
		2 7/8	10.40	IU-E75	2 7/8 PAC[b]	3 1/8	1 1/2	3.424 P	3 1/8	15/64	3.424	3 1/8	15/64	3.424
		2 7/8	10.40	EU-X95	NC31	4 1/8	2	7.895 P	3 29/32	3/16	5.726	3 27/32	5/32	4.969
		2 7/8	10.40	EU-X95	2 7/8 SL-H90[b]	3 7/8	2 5/32	6.732 P	3 11/16	3/16	5.702	3 5/8	5/32	4.915
		2 7/8	10.40	EU-G105	NC31	4 1/8	2	7.895 P	3 15/16	13/64	6.110	3 7/8	11`/64	5.345
		2 7/8	10.40	EU-S135	NC31	4 3/8	1 5/8	10.086 P	4 1/16	17/64	7.694	4	15/64	6.893
3 1/2	9.50	3 1/2	9.50	EU-E75	NC38	4 3/4	3	7.595 P	4 13/32	1/8	5.773	4 11/32	3/32	4.797
		3 1/2	9.50	EU-E75	NC38	4 3/4	2 11/16	10.843 P	4 13/32	1/8	5.773	4 11/32	3/32	4.797
		3 1/2	9.50	EU-E75	3 1/2 OHLW	4 3/4	3	7.082 P	4 9/32	1/8	5.340	4 1/4	7/64	4.868
		3 1/2	9.50	EU-E75	3 1/2 SL-H90	4 5/8	3	7.469 P	4 3/16	7/64	5.521	4 5/32	3/32	5.003
	13.30	3 1/2	13.30	EU-E75	NC38	4 3/4	2 11/16	10.843 P	4 1/2	11/64	7.274	4 7/16	9/64	6.268
		3 1/2	13.30	IU-E75	NC31[b]	4 1/8	2 1/8	7.074 P	4	15/64	6.893	3 15/16	13/64	6.110
		3 1/2	13.30	EU-E75	3 1/2 OHSW	4 3/4	2 11/16	10.300 P	4 13/32	3/16	7.278	4 11/32	5/32	6.299
		3 1/2	13.30	EU-E75	31/2 H90	5 1/4	2 3/4	14.043 P	4 17/32	1/8	7.064	4 1/2	7/64	6.487
		3 1/2	13.30	EU-X95	NC38	5	2 9/16	12.057 P	4 19/32	7/32	8.822	4 17/32	3/16	7.785
		3 1/2	13.30	EU-X95	3 1/2 SL-H90[b]	4 5/8	2 11/16	11.073 P	4 3/8	13/64	8.742	4 5/16	11/64	7.647
		3 1/2	13.30	EU-X95	3 1/2 H90	5 1/4	2 3/4	14.043 P	4 5/8	11/64	8.826	4 9/16	9/64	7.646

See end of table for notes

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
		Drill pipe data			New tool joint data				Premium class			Class 2		
Labels[a]		Nom OD	Nom linear mass	Type upset and pipe grade	Connection	New OD	New ID	Make-up torque[d]	Min OD tool joint[c.f]	Min box shoulder with eccentric wear[e]	Make-up torque for min OD tool joint[d]	Min OD tool joint[c.f]	Min box shoulder with eccentric wear[e]	Make-up torque for min OD tool joint[d]
1	2	in	lbm/ft			in	in	ft-lbf	in	in	ft-lbf	in	in	ft-lbf
		3.500	13.30	EU-G105	NC38	5	2 7/16	13.221 P	4 21/32	1/4	9.879	4 19/32	7/32	8.822
	13.30	3.500	13.30	EU-S135	NC40	5 3/8	2 7/16	17.858 P	5	9/32	12.569	4 29/32	15/64	10.768
		3.500	13.30	EU-S135	NC38	5	2 1/8	15.902 P	4 13/16	21/64	12.614	4 23/32	9/32	10.957
3 1/2		3.500	15.50	EU-E75	NC38	5	2 9/16	12.057 P	4 17/32	3/16	7.785	4 15/32	5/32	6.769
		3.500	15.50	EU-X95	NC38	5	2 7/16	13.221 P	4 21/32	1/4	9.879	4 19/32	7/32	8.822
	15.50	3.500	15.50	EU-G105	NC38	5	2 1/8	15.902 P	4 23/32	9/32	10.957	4 5/8	15/64	9.348
		3.500	15.50	EU-G105	NC40	5 1/4	2 9/16	16.616 P	4 15/16	1/4	11.363	4 27/32	13/64	9.595
		3.500	15.50	EU-S135	NC40	5 1/2	2 1/4	19.616 P	5 3/32	21/64	14.419	4 31/32	17/64	11.963
		4.000	11.85	EU-E75	NC46	6	3 1/4	19.937 P	5 7/32	7/64	7.843	5 5/32	5/64	6.476
	11.85	4.000	11.85	EU-E75	4 WO	5 3/4	3 7/16	17.186 P	5 7/32	7/64	7.843	5 5/32	5/64	6.476
		4.000	11.85	EU-E75	4 OHLW	5 1/4	3 15/32	13.186 P	5	9/64	7.866	4 15/16	7/64	6.593
		4.000	11.85	IU-E75	4 H90	5 1/2	2 13/16	21.185 P	4 7/8	7/64	7.630	4 27/32	3/32	6.962
		4.000	14.00	IU-E75	NC40	5 1/4	2 13/16	13.968 P	4 13/16	3/16	9.017	4 3/4	5/32	7.877
		4.000	14.00	EU-E75	NC46	6	3 1/4	19.937 P	5 9/32	9/64	9.233	5 7/32	7/64	7.843
		4.000	14.00	IU-E75	4 SH[b]	4 5/8	2 9/16	9.016 P	4 7/16	15/64	8.782	4 3/8	13/64	7.817
		4.000	14.00	EU-E75	4 OHSW	5 1/2	3 1/4	16.236 P	5 1/16	11/64	9.131	5	9/64	7.839
		4.000	14.00	IU-E75	4 H90	5 1/2	2 13/16	21.185 P	4 15/16	9/64	8.986	4 7/8	7/64	7.630
		4.000	14.00	IU-X95	NC40	5 1/4	2 11/16	15.319 P	4 15/16	1/4	11.363	4 27/32	13/64	9.595
	14.00	4.000	14.00	EU-X95	NC46	6	3 1/4	19.937 P	5 3/8	3/16	11.363	5 5/16	5/32	9.937
		4.000	14.00	IU-X95	4 H90	5 1/2	2 13/16	21.185 P	5 1/32	3/16	11.065	4 31/32	5/32	9.673
4		4.000	14.00	IU-G105	NC40	5 1/2	2 7/16	17.858 P	5	9/32	12.569	4 29/32	15/64	10.768
		4.000	14.00	EU-G105	NC46	6	3 1/4	19.937 P	5 7/16	7/32	12.813	5 11/32	11/64	10.647
		4.000	14.00	IU-G105	4 H90	5 1/2	2 13/16	21.185 P	5 3/32	7/32	12.481	5 1/32	3/16	11.065
		4.000	14.00	EU-S135	NC46	6	3	23.399 P	5 9/16	9/32	15.787	5 1/2	1/4	14.288
		4.000	15.70	IU-E75	NC40	5 1/4	2 11/16	15.319 P	4 7/8	7/32	10.179	4 25/32	11/64	8.444
		4.000	15.70	EU-E75	NC46	6	3 1/4	19.937 P	5 5/16	5/32	9.937	5 1/4	1/8	8.535
		4.000	15.70	IU-E75	4 H90	5 1/2	2 13/16	21.185 P	4 31/32	5/32	9.673	4 29/32	1/8	8.305
		4.000	15.70	IU-X95	NC40	5 1/2	2 7/16	17.858 P	5	9/32	12.569	4 29/32	15/64	10.768
	15.70	4.000	15.70	EU-X95	NC46	6	3	23.399 P	5 7/16	7/32	12.813	5 11/32	11/64	10.647
		4.000	15.70	IU-X95	4 H90	5 1/2	2 13/16	21.185 P	5 3/32	7/32	12.481	5 1/32	3/16	11.065
		4.000	15.70	EU-G105	NC46	6	3	23.399 P	5 15/32	15/64	13.547	5 13/32	13/64	12.085
		4.000	15.70	IU-G105	4 H90	5 1/2	2 13/16	21.185 P	5 5/32	1/4	13.922	5 1/16	13/64	11.770
		4.000	15.70	IU-S135	NC46	6	2 5/8	26.982 B	5 21/32	21/64	18.083	5 17/32	17/64	15.035
		4.000	15.70	EU-S135	NC46	6	2 7/8	25.038 P	5 21/32	21/64	18.083	5 17/32	17/64	15.035

See end of table for notes

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
		Drill pipe data			New tool joint data				Premium class			Class 2		
Labels[a]		Nom OD	Nom linear mass	Type upset and pipe grade	Connection	New OD	New ID	Make-up torque[d]	Min OD tool joint[c,f]	Min box shoulder with eccentric wear[e]	Make-up torque for min OD tool joint[d]	Min OD tool joint[c,f]	Min box shoulder with eccentric wear[e]	Make-up torque for min OD tool joint[d]
1	2	in	lbm/ft			in	in	ft-lbf	in	in	ft-lbf	in	in	ft-lbf
		4.500	16.60	IEU-E75	4 1/2 FH	6	3	20 620 P	5 3/8	13/64	12 125	5 9/32	5/32	10 072
		4.500	16.60	IEU-E75	NC46	6 1/4	3 1/4	19 937 P	5 13/32	13/64	12 085	5 11/32	11/64	10 647
		4.500	16.60	IEU-E75	4 1/2 OHSW	5 7/8	3 3/4	16 162 P	5 7/16	13/64	11 862	5 3/8	11/64	10 375
		4.500	16.60	EU-E75	NC50	6 5/8	3 3/4	22 361 P	5 23/32	5/32	11 590	5 11/16	9/64	10 773
		4.500	16.60	IEU-E75	4 1/2 H-90	6	3 1/4	23 126 P	5 11/32	3/16	12 215	5 9/32	5/32	10 642
		4.500	16.60	IEU-X95	41/2 FH	6	2 3/4	23 695 P	5 1/2	17/64	14 945	5 13/32	7/32	12 821
		4.500	16.60	IEU-X95	NC46	6 1/4	3 1/4	19 937 P	5 17/32	17/64	15 035	5 7/16	7/32	12 813
	16.60	4.500	16.60	EU-X95	NC50	6 5/8	3 3/4	22 361 P	5 27/32	7/32	14 926	5 25/32	3/16	13 245
		4.500	16.60	IEU-X95	4 1/2 H-90	6	3	26 969 P	5 15/32	1/4	15 441	5 3/8	13/64	13 013
		4.500	16.60	IEU-G105	4 1/2 FH	6	2 3/4	23 695 P	5 9/16	19/64	16 391	5 15/32	1/4	14 231
		4.500	16.60	IEU-G105	NC46	6 1/4	3	23 399 P	5 19/32	19/64	16 546	5 1/2	1/4	14 288
		4.500	16.60	EU-G105	NC50	6 5/8	3 3/4	22 361 P	5 29/32	1/4	16 633	5 13/16	13/64	14 082
4 1/2		4.500	16.60	IEU-G105	4 1/2 H-90	6	3	26 969 P	5 1/2	17/64	16 264	5 7/16	15/64	14 625
		4.500	16.60	IEU-S135	NC46	6 1/4	2 3/4	26 615 P	5 25/32	25/64	21 230	5 21/32	21/64	18 083
		4.500	16.60	EU-S135	NC50	6 5/8	3 1/2	26 674 P	6 1/16	21/64	21 017	5 31/32	9/32	18 367
		4.500	20.00	IEU-E75	4 1/2 FH	6	3	20 620 P	5 15/32	1/4	14 231	5 3/8	13/64	12 125
		4.500	20.00	IEU-E75	NC46	6 1/4	3	23 399 P	5 1/2	1/4	14 288	5 13/32	13/64	12 085
		4.500	20.00	EU-E75	NC50	6 5/8	3 5/8	24 549 P	5 13/16	13/64	14 082	5 3/4	3/16	12 415
		4.500	20.00	IEU-E75	4 1/2 H-90	6	3	26 969 P	5 13/32	7/32	13 815	5 11/32	3/16	12 215
		4.500	20.00	IEU-X95	4 1/2 FH	6	2 1/2	26 528 P	5 5/8	21/64	17 861	5 17/32	9/32	15 665
	20.00	4.500	20.00	IEU-X95	NC46	6 1/4	2 3/4	26 615 P	5 21/32	21/64	18 083	5 9/16	9/32	15 787
		4.500	20.00	EU-X95	NC50	6 5/8	3 1/2	26 674 P	5 15/16	17/64	17 497	5 7/8	15/64	15 776
		4.500	20.00	IEU-X95	4 1/2 H-90	6	3	26 969 P	5 9/16	19/64	17 929	5 15/32	1/4	15 441
		4.500	20.00	IEU-G105	NC46	6 1/4	2 1/2	29 578 P	5 23/32	23/64	19 644	5 5/8	5/16	17 311
		4.500	20.00	EU-G105	NC50	6 5/8	3 1/2	26 674 P	6 1/32	5/16	20 127	5 29/32	1/4	16 633
		4.500	20.00	EU-S135	NC50	6 5/8	3	34 520 P	6 7/32	13/32	25 569	6 3/32	11/32	21 914
		5.000	19.50	IEU-E75	NC50	6 5/8	3 3/4	22 361 P	5 7/8	15/64	15 776	5 13/16	13/64	14 082
		5.000	19.50	IEU-X95	NC50	6 5/8	3 1/2	26 674 P	6 1/32	5/16	20 127	5 15/16	17/64	17 497
		5.000	19.50	IEU-X95	5 H-90	6 1/2	3 1/4	30 732 P	5 27/32	19/64	19 862	5 3/4	1/4	17 116
5	19.50	5.000	19.50	IEU-G105	NC50	6 5/8	3 1/4	30 730 P	6 3/32	11/32	21 914	6	19/64	19 244
		5.000	19.50	IEU-G105	5 H-90	6 1/2	3	34 805 P	5 29/32	21/64	21 727	5 13/16	9/32	18 940
		5.000	19.50	IEU-S135	NC50	6 5/8	2 3/4	38 036 P	6 5/16	29/64	28 381	6 3/16	25/64	24 645
		5.000	19.50	IEU-S135	5 1/2 FH	7 1/4	3 1/2	43 328 P	6 3/4	3/8	28 737	6 5/8	5/16	24 412

See end of table for notes

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
		Drill pipe data			**New tool joint data**				**Premium class**			**Class 2**		
Labels[a]		**Nom OD**	**Nom linear mass**	**Type upset and pipe grade**	**Connection**	**New OD**	**New ID**	**Make-up torque**[d]	**Min OD Tool Joint**[c,f]	**Min box shoulder with eccentric wear**[e]	**Make-up torque for min OD tool joint**[d]	**Min OD tool joint**[c,f]	**Min box shoulder with eccentric wear**[e]	**Make-up torque for min OD tool joint**[d]
1	2	in	lbm/ft			in	in	ft-lbf	in	in	ft-lbf	in	in	ft-lbf
5	25.60	5.000	25.60	IEU-E75	NC50	6 5/8	3 1/2	26.674 P	6 1/32	5/16	20.127	5 15/16	17/64	17.497
		5.000	25.60	IEU-E75	5 1/2 FH	7	3 1/2	37.742 B	6 1/2	1/4	20.205	6 13/32	13/64	17.127
		5.000	25.60	IEU-X95	NC50	6 5/8	3	34.520 P	6 7/32	13/32	25.569	6 3/32	11/32	21.914
		5.000	25.60	IEU-X95	5 1/2 FH	7	3 1/2	37.742 B	6 21/32	21/64	25.483	6 9/16	9/32	22.294
		5.000	25.60	IEU-G105	NC50	6 5/8	2 3/4	38.036 P	6 9/32	7/16	27.437	6 5/32	3/8	23.728
		5.000	25.60	IEU-G105	5 1/2 FH	7 1/4	3 1/2	43.328 P	6 23/32	23/64	27.645	6 5/8	5/16	24.412
		5.000	25.60	IEU-S135	5 1/2 FH	7 1/4	3 1/4	47.230 B	6 15/16	15/32	35.446	6 13/16	13/32	30.943
5 1/2	21.90	5.500	21.90	IEU-E75	5 1/2 FH	7	4	33.412 P	6 15/32	15/64	19.172	6 13/32	13/64	17.127
		5.500	21.90	IEU-X95	5 1/2 FH	7	3 3/4	37.742 B	6 5/8	5/16	24.412	6 17/32	17/64	21.246
		5.500	21.90	IEU-X95	5 1/2 H-90	7	3 1/2	34.820 P	6 3/16	21/64	24.414	6 3/32	9/32	21.349
		5.500	21.90	IEU-G105	5 1/2 FH	7 1/4	3 1/2	43.328 P	6 23/32	23/64	27.645	6 19/32	19/64	23.350
		5.500	21.90	IEU-S135	5 1/2 FH	7 1/2	3	52.059 P	6 15/16	15/32	35.446	6 13/16	13/32	30.943
	24.70	5.500	24.70	IEU-E75	5 1/2 FH	7	4	33.412 P	6 9/16	9/32	22.294	6 15/32	15/64	19.172
		5.500	24.70	IEU-X95	5 1/2 FH	7 1/4	3 1/2	43.328 P	6 23/32	23/64	27.645	6 19/32	19/64	23.350
		5.500	24.70	IEU-G105	5 1/2 FH	7 1/4	3 1/2	43.328 P	6 25/32	25/64	29.836	6 11/16	11/32	26.560
		5.500	24.70	IEU-S135	5 1/2 FH	7 1/2	3	52.059 P	7 1/32	33/64	38.901	6 7/8	7/16	33.180
6 5/8	25.20	6.625	25.20	IEU-E75	6 5/8 FH	8	5	43.934 P	7 7/16	1/4	26.810	7 3/8	7/32	24.100
		6.625	25.20	IEU-X95	6 5/8 FH	8	5	43.934 P	7 5/8	11/32	35.139	7 1/2	9/32	29.552
		6.625	25.20	IEU-G105	6 5/8 FH	8 1/4	4 3/4	51.280 P	7 11/16	5/8	37.983	7 19/32	21/64	33.730
		6.625	25.20	IEU-S135	6 5/8 FH	8 1/2	4 1/4	65.012 P	7 29/32	31/64	48.204	7 25/32	27/64	42.312
	27.70	6.625	27.70	IEU-E75	6 5/8 FH	8	5	43.934 P	7 1/2	9/32	29.552	7 13/32	15/64	25.451
		6.625	27.70	IEU-X95	6 5/8 FH	8 1/4	4 3/4	51.280 P	7 11/16	3/8	37.983	7 9/16	5/16	32.329
		6.625	27.70	IEU-G105	6 5/8 FH	8 1/4	4 3/4	51.280 P	7 3/4	13/32	40.860	7 21/32	23/64	36.556
		6.625	27.70	IEU-S135	6 5/8 FH	8 1/2	4 1/4	65.012 P	8	17/32	52.714	7 27/32	29/64	45.241

NOTE 1 Tool joints of outside diameters (OD) listed in this table should be adequate for all service

NOTE 2 The use of outside diameters (OD) smaller than those listed in this table may be acceptable due to special service requirements.

NOTE 3 Tool joints with torsional strengths considerably below that of the drill pipe may be adequate for much drilling service.

NOTE 4 Any tool joint with an outside diameter less than API bevel diameter should be provided with a minimum 1/32 inch depth x 45 degree bevel on the outside and inside diameter of the box shoulder and outside diameter of the pin shoulder.

NOTE 5 P = Pin limit; B = Box limit.

[a] Labels are for information and assistance in ordering

[b] Tool joint with dimensions shown has lower torsional yield ratio than the 0.80 which is generally used.

[c] Tool joint diameters specified are required to retain torsional strength in the tool joint comparable to the torsional strength of the attached drill pipe.

[d] Recommended make-up torque is based on 72 000 psi stress

[e] This thickness measurement shall be made in the plane of the face from the ID of the counterbore to the outside diameter of the box, disregarding the bevels.

[f] In calculation of torsional strengths of tool joints, both new and worn, the bevels of the tool joint shoulders should be disregarded

Table B.11 — Buoyancy factors

Mud density		Buoyancy factor	Mud density		Buoyancy factor
Lbs/gallon	Lbs/ft^3	Kb	Lbs/gallon	Lbs/ft^3	Kb
8.4	62.84	0.872	14.4	107.72	0.780
8.6	64.33	0.869	14.6	109.22	0.777
8.8	65.83	0.866	14.8	110.71	0.774
9	67.32	0.862	15	112.21	0.771
9.2	68.82	0.859	15.2	113.70	0.768
9.4	70.32	0.856	15.4	115.20	0.765
9.6	71.81	0.853	15.6	116.70	0.762
9.8	73.31	0.850	15.8	118.19	0.759
10	74.81	0.847	16	119.69	0.756
10.2	76.30	0.844	16.2	121.18	0.752
10.4	77.80	0.841	16.4	122.68	0.749
10.6	79.29	0.838	16.6	124.18	0.746
10.8	80.79	0.835	16.8	125.67	0.743
11	82.29	0.832	17	127.17	0.740
11.2	83.78	0.829	17.2	128.66	0.737
11.4	85.28	0.826	17.4	130.16	0.734
11.6	86.77	0.823	17.6	131.66	0.731
11.8	88.27	0.820	17.8	133.15	0.728
12	89.77	0.817	18	134.65	0.725
12.2	91.26	0.814	18.2	136.15	0.722
12.4	92.76	0.811	18.4	137.64	0.719
12.6	94.25	0.807	18.6	139.14	0.716
12.8	95.75	0.804	18.8	140.63	0.713
13	97.25	0.801	19	142.13	0.710
13.2	98.74	0.798	19.2	143.63	0.707
13.4	100.24	0.795	19.4	145.12	0.704
13.6	101.74	0.792	19.6	146.62	0.700
13.8	103.23	0.789	19.8	148.11	0.697
14	104.73	0.786	20	149.61	0.694
14.2	106.22	0.783			

Table B.12 — Rotary shouldered connection interchange list

Common Name	Size	Pin base dia. (tapered) inch	Threads per inch	Taper inches/ft	Thread Form[a]	Same as or interchanges with
Internal Flush (IF)	2 3/8	2.876	4	2	V-0.065 (V-0.038R)	2 7/8 Slim Hole NC26[b]
	2 7/8	3.391	4	2	V-0.065 (V-0.038R)	3 1/2 Slim Hole NC31[b]
	3 1/2	4.016	4	2	V-0.065 (V-0.038R)	4 1/2 Slim Hole NC38[b]
	4	4.834	4	2	V-0.065 (V-0.038R)	4 1/2 Extra Hole NC46[b]
	4 1/2	5.250	4	2	V-0.065 (V-0.038R)	5 Extra Hole NC50[b] 5 1/2 Double Streamline
Full Hole	4	4.280	4	2	V-0.065 (V-0.038R)	4 1/2 Double streamline NC40[b]
Extra Hole (X.H.) (E.H.)	2 7/8	3.327	4	2	V-0.065 (V-0.038R)	3 1/2 Double Streamline
	3 1/2	3.812	4	2	V-0.065 (V-0.038R)	4 Slim Hole 4 1/2 External Flush
	4 1/2	4.834	4	2	V-0.065 (V-0.038R)	4 Internal Flush NC46[b]
	5	5.250	4	2	V-0.065 (V-0.038R)	4 1/2 Internal Flush NC50[b] 5 1/2 Double Streamline
Slim Hole (S.H.)	2 7/8	2.876	4	2	V-0.065 (V-0.038R)	2 3/8 Internal Flush NC26[b]
	3 1/2	3.391	4	2	V-0.065 (V-0.038R)	2 7/8 Internal Flush NC31[b]
	4	3.812	4	2	V-0.065 (V-0.038R)	3 1/2 Extra Hole 4 1/2 External Flush
	4 1/2	4.016	4	2	V-0.065 (V-0.038R)	3 1/2 Internal Flush NC38[b]
Double Streamline (DSL)	3 1/2	3.327	4	2	V-0.065 (V-0.038R)	2 7/8 Extra Hole
	4 1/2	4.280	4	2	V-0.065 (V-0.038R)	4 Full Hole NC40[b]
	5 1/2	5.250	4	2	V-0.065 (V-0.038R)	4 1/2 Internal Flush 5 Extra Hole NC50[b]
External Flush (E.F.)	4 1/2	3.812	4	2	V-0.065 (V-0.038R)	4 Slim Hole 3 1/2 Extra Hole
Numbered Connection (NC)	26	2.876	4	2	V-038R	2 3/8 Internal Flush 2 7/8 Slim Hole
	31	3.391	4	2	V-038R	2 7/8 Internal Flush 3 1/2 Slim Hole
	38	4.016	4	2	V-038R	3 1/2 Internal Flush 4 1/2 Slim Hole
	40	4.280	4	2	V-038R	4 Full Hole 4 1/2 Double streamline
	46	4.834	4	2	V-038R	4 Internal Flush 4 1/2 Extra Hole
	50	5.250	4	2	V-038R	4 1/2 Internal Flush 5 Extra Hole

[a] Connections with two thread forms shown may be machined with either thread form without affecting gauging or interchangeability.
[b] Numbered connections (NC) should be machined only with the V - 0.038Radius thread form.

Table B.13 — Drill collar weight (steel)a

lbm/ft

1	2	3	4	5	6	7	8	9	10	11	12	13	14
Drill collar OD	**Drill collar ID inches**												
inches	1	1 1/4	1 1/2	1 3/4	2	2 1/4	2 1/2	2 13/16	3	3 1/4	3 1/2	3 3/4	4
2 7/8	19	18	16										
3	21	20	18										
3 1/8	23	22	20										
3 1/4	26	24	22										
3 1/2	30	29	27										
3 3/4	35	33	32										
4	40	39	37										
4 1/8	43	41	39										
4 1/4	46	44	42										
4 1/2	51	50	48										
4 3/4	58	56	54										
5			61	59	56	53	50						
5 1/4			68	65	63	60	57	52					
5 1/2			75	73	70	67	64	60					
5 3/4			82	80	78	75	72	67	64	60			
6			90	88	85	83	79	75	72	68			
6 1/4			98	96	94	91	88	83	80	76	72		
6 1/2			107	105	102	99	96	92	89	85	80		
6 3/4			116	113	111	108	105	101	98	93	89		
7			125	123	120	117	114	110	107	103	98	93	88
7 1/4			134	132	130	127	124	119	116	112	108	103	98
7 1/2			144	142	140	137	134	129	126	122	117	113	107
7 3/4			154	152	150	147	144	139	136	132	128	123	118
8			165	163	160	157	154	150	147	143	138	133	128
8 1/4			176	174	171	168	165	161	158	154	149	144	139
8 1/2			187	185	182	179	176	172	169	165	160	155	150
8 3/4			198	196	194	191	188	183	180	176	172	167	162
9			210	208	206	203	200	195	192	188	184	179	174
9 1/4			222	220	218	215	212	207	204	200	196	191	186
9 1/2			235	233	230	227	224	220	217	213	208	203	198
9 3/4			248	246	243	240	237	233	230	226	221	216	211
10			261	259	256	253	250	246	243	239	234	229	224
11			317	315	312	310	306	302	299	295	290	286	280
12			378	376	374	371	368	363	360	356	352	347	342

See ISO 10424-1 or API Specification 7 for standard drill collar dimensions.

For special configurations of drill collars, consult manufacturer for reduction or increase in weight.

[a] Weight per foot is based on drill collar steel having a density of 489.54 lbm/ft^3

Table B.14 — Recommended make-up torque[a] for rotary shouldered drill collar connections

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
Connection Name		OD inch	Minimum make-up torque (ft-lbs)[b]											
			ID of drill collar (inches)											
Label 1	Label 2		1	1 1/4	1 1/2	1 3/4	2	2 1/4	2 1/2	2 13/16	3	3 1/4	3 1/2	3 3/4
NC23	API NC	3	*2 508	*2 508	*2 508									
		3 1/8	*3 330	*3 330	2 647									
		3 1/4	4 000	3 387	2 647									
2 3/8	API Regular	3		*2 241	*2 241	1 749								
		3 1/8		*3 028	2 574	1 749								
		3 1/4		3 285	2 574	1 749								
2 7/8	PAC[c]	3		*3 797	*3 797	2 926								
		3 1/8		*4 966	4 151	2 926								
		3 1/4		5 206	4 151	2 926								
2 3/8	API IF	3 3/8		*3 581	*3 581	*3 581								
NC 26	API NC	3 1/2		*4 606	*4 606	3 697								
2 7/8	Slim Hole	3 3/4		5 501	4 668	3 697								
2 7/8	API Regular	3 1/2		*3 838	*3 838	*3 838								
		3 3/4		5 766	4 951	4 002								
		3 7/8		5 766	4 951	4 002								
2 7/8	Extra Hole	3 3/4		*4 089	*4 089	*4 089								
3 1/2	DSL	3 7/8		*5 352	*5 352	*5 352								
2 7/8	Mod. Open	4 1/8		8 059	8 059	7 433								
2 7/8	API IF	3 7/8		*4 640	*4 640	*4 640	*4 640							
NC 31	API NC	4 1/8		*7 390	*7 390	*7 390	6 853							
3 1/2	API Regular	4 1/8		*6 466	*6 466	*6 466	*6 466	5 685						
		4 1/4		*7 886	*7 886	*7 886	7 115	5 685						
		4 1/2		10 471	9 514	8 394	7 115	5 685						
3 1/2	Slim Hole	4 1/4		*8 858	*8 858	8 161	6 853	5 391						
		4 1/2		10 286	9 307	8 161	6 853	5 391						
NC 35	API NC	4 1/2				*9 038	*9 038	*9 038	7 411					
		4 3/4				12 273	10 826	9 202	7 411					
		5				12 273	10 826	9 202	7 411					
3 1/2	Extra Hole	4 1/4				*5 161	*5 161	*5 161	*5 161					
4	Slim Hole	4 1/2				*8 479	*8 479	*8 479	8 311					
3 1/2	Mod. Open	4 3/4				*12 074	11 803	10 144	8 311					
		5				13 283	11 803	10 144	8 311					
		5 1/4				13 283	11 803	10 144	8 311					
3 1/2	API IF	4 3/4				*9 986	*9 986	*9 986	*9 986	8 315				
NC 38	API NC	5				*13 949	*13 949	12 907	10 977	8 315				
4 1/2	Slim Hole	5 1/4				16 207	14 643	12 907	10 977	8 315				
		5 1/2				16 207	14 643	12 907	10 977	8 315				
3 1/2	H-90[d]	4 3/4				*8 786	*8 786	*8 786	*8 786	*8 786				
		5				*12 794	*12 794	*12 794	*12 794	10 408				
		5 1/4				*17 094	16 929	15 137	13 151	10 408				
		5 1/2				18 522	16 929	15 137	13 151	10 408				
4	API Full Hole	5				*10 910	*10 910	*10 910	*10 910	*10 910				
NC 40	API NC	5 1/4				*15 290	*15 290	*15 290	14 969	12 125				
4	Mod. Open	5 1/2				*19 985	18 886	17 028	14 969	12 125				
4 1/2	DSL	5 3/4				20 539	18 886	17 028	14 969	12 125				
		6				20 539	18 886	17 028	14 969	12 125				
See end of table for notes														

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
Connection Name		OD	Minimum make-up torque (ft-lbs)[b]											
		inch	ID of drill collar (inches)											
Label 1	Label 2		1	1 1/4	1 1/2	1 3/4	2	2 1/4	2 1/2	2 13/16	3	3 1/4	3 1/2	3 3/4
4	H-90[d]	5 1/4				*12 590	*12 590	*12 590	*12 590	*12 590				
		5 1/2				*17 401	*17 401	*17 401	*17 401	16 536				
		5 3/4				*22 531	*22 531	21 714	19 543	16 536				
		6				25 408	23 671	21 714	19 543	16 536				
		6 1/4				25 408	23 671	21 714	19 543	16 536				
4 1/2	API Regular	5 1/2				*15 576	*15 576	*15 576	*15 576	*15 576				
		5 3/4				*20 609	*20 609	*20 609	19 601	16 629				
		6				25 407	23 686	21 749	19 601	16 629				
		6 1/4				25 407	23 686	21 749	19 601	16 629				
NC44	API NC	5 3/4				*20 895	*20 895	*20 895	*20 895	18 161				
		6				*26 453	25 510	23 493	21 257	18 161				
		6 1/4				27 300	25 510	23 493	21 257	18 161				
		6 1/2				27 300	25 510	23 493	21 257	18 161				
4 1/2	API Full Hole	5 1/2					*12 973	*12 973	*12 973	*12 973	*12 973			
		5 3/4					*18 119	*18 119	*18 119	*18 119	17 900			
		6					*23 605	*23 605	23 028	19 921	17 900			
		6 1/4					27 294	25 272	23 028	19 921	17 900			
		6 1/2					27 294	25 272	23 028	19 921	17 900			
4 1/2	Extra Hole	5 3/4						*17 738	*17 738	*17 738	*17 738			
NC46	API NC	6						*23 422	*23 422	22 426	20 311			
4	API IF	6 1/4						28 021	25 676	22 426	20 311			
4 1/2	Semi If	6 1/2						28 021	25 676	22 426	20 311			
5	DSL	6 3/4						28 021	25 676	22 426	20 311			
4 1/2	Mod Open													
4 1/2	H-90[d]	5 3/4						*18 019	*18 019	*18 019	*18 019			
		6						*23 681	*23 681	23 159	21 051			
		6 1/4						28 732	26 397	23 159	21 051			
		6 1/2						28 732	26 397	23 159	21 051			
		6 3/4						28 732	26 397	23 159	21 051			
5	H-90[d]	6 1/4						*25 360	*25 360	*25 360	*25 360	23 988		
		6 1/2						*31 895	*31 895	29 400	27 167	23 988		
		6 3/4						35 292	32 825	29 400	27 167	23 988		
		7						35 292	32 825	29 400	27 167	23 988		
4 1/2	API IF	6 1/4						*23 004	*23 004	*23 004	*23 004	*23 004		
NC 50	API NC	6 1/2						*29 679	*29 679	*29 679	*29 679	26 675		
5	Extra Hole	6 3/4						*36 742	35 824	32 277	29 966	26 675		
5	Mod Open	7						38 379	35 824	32 277	29 966	26 675		
5 1/2	DSL	7 1/4						38 379	35 824	32 277	29 966	26 675		
5	Semi IF	7 1/2						38 379	35 824	32 277	29 966	26 675		
5 1/2	H-90[d]	6 3/4						*34 508	*34 508	*34 508	34 142	30 781		
		7						*41 993	40 117	36 501	34 142	30 781		
		7 1/4						42 719	40 117	36 501	34 142	30 781		
		7 1/2						42 719	40 117	36 501	34 142	30 781		
5 1/2	API Regular	6 3/4						*31 941	*31 941	*31 941	*31 941	30 495		
		7						*39 419	*39 419	36 235	33 868	30 495		
		7 1/4						42 481	39 866	36 235	33 868	30 495		
		7 1/2						42 481	39 866	36 235	33 868	30 495		
See end of table for notes														

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
Connection Name		OD	Minimum make-up torque (ft-lbs)[b]											
		inch	ID of drill collar (inches)											
Label 1	Label 2		1	1 1/4	1 1/2	1 3/4	2	2 1/4	2 1/2	2 13/16	3	3 1/4	3 1/2	3 3/4
5 1/2	API Full Hole	7						*32 762	*32 762	*32 762	*32 762	*32 762		
		7 1/4						*40 998	*40 998	*40 998	*40 998	*40 998		
		7 1/2						*49 661	*49 661	47 756	45 190	41 533		
		7 3/4						54 515	51 687	47 756	45 190	41 533		
NC 56	API NC	7 1/4							*40 498	*40 498	*40 498	*40 498		
		7 1/2							*49 060	48 221	45 680	42 058		
		7 3/4							52 115	48 221	45 680	42 058		
		8							52 115	48 221	45 680	42 058		
6 5/8	API Regular	7 1/2							*46 399	*46 399	*46 399	*46 399		
		7 3/4							*55 627	53 346	50 704	46 399		
		8							57 393	53 346	50 704	46 399		
		8 1/4							57 393	53 346	50 704	46 399		
6 5/8	H-90[d]	7 1/2							*46 509	*46 509	*46 509	*46 509		
		7 3/4							*55 708	*55 708	53 629	49 855		
		8							60 321	56 273	53 629	49 855		
		8 1/4							60 321	56 273	53 629	49 855		
NC 61	API NC	8							*55 131	*55 131	*55 131	*55 131		
		8 1/4							*65 438	*65 438	*65 438	61 624		
		8 1/2							72 670	68 398	65 607	61 624		
		8 3/4							72 670	68 398	65 607	61 624		
		9							72 670	68 398	65 607	61 624		
5 1/2	API IF	8							*56 641	*56 641	*56 641	*56 641	*56 641	
		8 1/4							*67 133	*67 133	*67 133	63 381	59 027	
		8 1/2							74 626	70 277	67 436	63 381	59 027	
		8 3/4							74 626	70 277	67 436	63 381	59 027	
		9							74 626	70 277	67 436	63 381	59 027	
		9 1/4							74 626	70 277	67 436	63 381	59 027	
6 5/8	API Full Hole	8 1/2							*67 789	*67 789	*67 789	*67 789	*67 789	67 184
		8 3/4							*79 544	*79 544	*79 544	76 706	72 102	67 184
		9							88 582	83 992	80 991	76 706	72 102	67 184
		9 1/4							88 582	83 992	80 991	76 706	72 102	67 184
		9 1/2							88 582	83 992	80 991	76 706	72 102	67 184
NC 70	API NC	9							*75 781	*75 781	*75 781	*75 781	*75 781	*75 781
		9 1/4							*88 802	*88 802	*88 802	*88 802	*88 802	*88 802
		9 1/2							*102 354	*102 354	*102 354	101 107	96 214	90 984
		9 3/4							113 710	108 841	105 657	101 107	96 214	90 984
		10							113 710	108 841	105 657	101 107	96 214	90 984
		10 1/4							113 710	108 841	105 657	101 107	96 214	90 984
NC 77	API NC	10							*108 194	*108 194	*108 194	*108 194	*108 194	*108 194
		10 1/4							*124 051	*124 051	*124 051	*124 051	*124 051	*124 051
		10 1/2							*140 491	*140 491	*140 491	140 488	135 119	129 375
		10 3/4							154 297	148 965	145 476	140 488	135 119	129 375
		11							154 297	148 965	145 476	140 488	135 119	129 375
7	H-90[d]	8							*53 454	*53 454	*53 454	*53 454	*53 454	*53 454
		8 1/4							*63 738	*63 738	*63 738	*63 738	60 971	56 382
		8 1/2							*74 478	72 066	69 265	65 267	60 971	56 382
7 5/8	API Regular	8 1/2							*60 402	*60 402	*60 402	*60 402	*60 402	*60 402
		8 3/4							*72 169	*72 169	*72 169	*72 169	*72 169	*72 169

See end of table for notes

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
Connection Name		**OD**	**Minimum make-up torque (ft-lbs)[b]**											
		inch	**ID of drill collar (inches)**											
Label 1	Label 2		1	1 1/4	1 1/2	1 3/4	2	2 1/4	2 1/2	2 13/16	3	3 1/4	3 1/2	3 3/4
7 5/8	API Regular	9							*84 442	*84 442	*84 442	84 221	79 536	74 529
		9 1/4							96 301	91 633	88 580	84 221	79 536	74 529
		9 1/2							96 301	91 633	88 580	84 221	79 536	74 529
7 5/8	H-90[d]	9							*73 017	*73 017	*73 017	*73 017	*73 017	*73 017
		9 1/4							*86 006	*86 006	*86 006	*86 006	*86 006	*86 006
		9 1/2							*99 508	*99 508	*99 508	*99 508	*99 508	96 285
8 5/8	API Regular	10							*109 345	*109 345	*109 345	*109 345	*109 345	*109 345
		10 1/4							*125 263	*125 263	*125 263	*125 263	*125 263	125 034
		10 1/2							*141 767	*141 767	141 134	136 146	130 777	125 034
8 5/8	H-90[d]	10 1/4							*113 482	*113 482	*113 482	*113 482	*113 482	*113 482
		10 1/2							*130 063	*130 063	*130 063	*130 063	*130 063	*130 063
7	H-90[d]	8 3/4								*68 061	*68 061	67 257	62 845	58 131
	(with low	9								74 235	71 361	67 257	62 845	58 131
	torque face)													
7 5/8	API Regular	9 1/4									*73 099	*73 099	*73 099	*73 099
	(with low	9 1/2									*86 463	*86 463	82 457	77 289
	torque face)	9 3/4									91 789	87 292	82 457	77 289
		10									91 789	87 292	82 457	77 289
7 5/8	H-90[d]	9 3/4								*91 667	*91 667	*91 667	*91 667	*91 667
	(with low	10								*106 260	*106 260	*106 260	104 171	98 804
	torque face)	10 1/4								117 112	113 851	109 188	104 171	98 804
		10 1/2								117 112	113 851	109 188	104 171	98 804
8 5/8	API Regular	10 3/4									*112 883	*112 883	*112 883	*112 883
	(with low	11									*130 672	*130 672	*130 672	*130 672
	torque face)	11 1/4									147 616	142 430	136 846	130 871
8 5/8	H-90[d]	10 3/4									*92 960	*92 960	*92 960	*92 960
	(with low	11									*110 781	*110 781	*110 781	*110 781
	torque face)	11 1/4									*129 203	*129 203	*129 203	*129 203

Torque figures preceded by an asterisk (*) indicate that the weaker member for the corresponding outside diameter (OD) and bore is the BOX; for all other torque values, the weaker member is the PIN.

In each connection size and type group, torque values apply to all connection types in the group, when used with the same drill collar outside diameter and bore, i.e., 2 7/8 API IF, NC 26 and 2 7/8 Slim Hole connections used with 3 ½ x 1 ¼ drill collars all have the same minimum make-up torque of 4600 ft-lbf, and the Box is the weaker member.

Stress relief features are disregarded in determining make-up torque.

[a] Basis of calculations for recommended make-up torque assumed the use of a thread compound containing 40-60 % by weight of finely powdered metallic zinc or 60 % by weight finely powdered metallic lead, with not more than 0.3 % total active sulphur (reference the caution regarding the use of hazardous materials in Appendix G of API Specification 7) applied thoroughly to all threads and shoulders and using the modified Screw Jack formula in E.8 and a unit stress of 62.500 psi in the box or pin, whichever is weaker.

[b] Normal torque range is tabulated value plus 10 %. Higher values may be used under extreme conditions.

[c] Make-up torque for PAC connections is based on 87.500 psi and other factors listed in footnote 1.

[d] Make-up torque for H-90 connections is based on 56.200 psi and other factors listed in footnote 1

Table B.15 — Strength of kellys a

1	2	3	4	5	6	7	8	9	10	11	12
Kelly date			**Lower pin connection**			**Tensile yield**		**Torsional yield**		**Yield in bending**	**Internal pressure at minimum yield**
Kelly size	**Kelly type**	**Kelly bore**	**Label [b]**	**OD**	**Minimum Recommended casing OD [1]**	**Lower pin connection [c]**	**Drive section**	**Lower pin connection**	**Drive section**	**Through drive section**	**Drive section**
in		in		in	in	lbf	lbf	Ft-lbf	Ft-lbf	Ft-lbf	psi
2 1/2	Square	1 1/4	NC 26 (2 3/8 IF)	3 3/8	4 1/2	416 000	444 400	9 650	12 300	13 000	29 800
3	Square	1 3/4	NC31 (2 7/8 IF)	4 1/8	5 1/2	535 000	582 500	14 450	19 500	22 300	25 500
3 1/2	Square	2 1/4	NC 38 (3 ½ IF)	4 3/4	6 5/8	724 000	725 200	22 700	28 300	34 200	22 200
4 1/4	Square	2 13/16	NC 46 (4 IF)	6 1/4	8 5/8	1 054 000	1 047 000	39 350	49 100	60 300	19 500
4 1/4	Square	2 13/16	NC 50 (4 1/2 IF)	6 3/8	8 5/8	1 375 200	1 047 000	55 810	49 100	60 300	19 500
5 1/4	Square	3 1/4	5 1/2 FH	7	9 5/8	1 609 000	1 703 400	72 950	99 400	117 000	20 600
3	Hexagonal	1 1/2	NC 26 (2 3/8 IF)	3 3/8	4 1/2	356 000	540 500	8 300	20 400	20 000	26 700
3 1/2	Hexagonal	1 7/8	NC31 (2 7/8 IF)	4 1/8	5 1/2	495 000	710 000	13 400	31 400	31 200	25 500
4 1/4	Hexagonal	2 1/4	NC 38 (3 1/2 IF)	4 3/4	6 5/8	724 000	1 046 600	22 700	56 600	56 000	25 000
5 1/4	Hexagonal	3	NC 46 (4 IF)	6 1/4	8 5/8	960 000	1 507 600	35 450	101 900	103 000	20 600
5 1/4	Hexagonal	3 1/4	NC 50 (4 1/2 IF)	6 3/8	8 5/8	1 162 000	1 397 100	46 750	95 500	99 300	20 600
6	Hexagonal	3 1/2	5 1/2 FH	7	9 5/8	1 463 000	1 935 500	66 350	149 800	152 500	18 200

NOTE: Clearance between protector rubber on kelly saver sub and casing inside diameter should also be checked.

[a] All values have a safety factor of 1.0 and are based on 110 000 psi minimum tensile yield (quenched and tempered) for connections and 90 000 psi minimum tensile yield (normalized and tempered) for the drive section. Fully quenched and tempered drive sections will have higher values than those shown. Shear strength is based on 57.7 % of the minimum tensile yield strength.

[b] Labels are for information and assistance in ordering.

[c] Tensile area calculated at root of thread 3/4 inch from pin shoulder.

Table B.16 — Contact angle between kelly and bushing for development of maximum width wear pattern

1	2	3	4	5	6	7	8	9
	Hexagonal kelly				Square kelly			
Kelly size	For minimum clearance	Contact angle	For maximum clearance	Contact angle	For minimum clearance	Contact angle	For maximum clearance	Contact angle
in	in	degree	in	degree	in	degree	in	degree
2 1/2	-	-	-	-	0.015	6.17	0.107	16.48
3	0.015	5.68	0.060	11.37	0.015	5.65	0.107	15.08
3 1/2	0.015	5.27	0.060	10.53	0.015	5.23	0.107	14.03
4 1/4	0.015	4.80	0.060	9.57	0.015	4.75	0.123	13.60
5 1/4	0.015	4.32	0.060	8.62	0.015	4.28	0.123	12.27
6	0.015	4.03	0.060	8.07	-	-	-	-

Table B.17 — Strength of remachined kellys a

1	2	3	4	5	6	7	8	9	10	11	12
Original kelly		Remachined kelly		Lower pin connection			Tensile yield		Torsional yield		Yield in bending through drive section
Size	Type	Size	Type	Bore	Label [b]	OD	Lower pin connection	Drive section	Lower pin connection	Drive section	
in		in		in		in	lbf	lbf	ft-lbf	ft-lbf	ft-lbf
4 1/4	Square	4	Square	2 7/8	NC50	6 3/8	1 344 200	834 400	55 500	36 200	47 800
4 1/4	Square	4	Square	2 7/8	NC46	6 1/4	1 011 600	834 400	38 300	36 200	47 800
5 1/4	Square	5	Square	3 3/4	5 1/2 IF	7 3/8	1 924 300	1 217 600	92 700	65 000	90 200
5 1/4	Square	5	Square	3 3/4	5 1/2 FH	7	1 356 800	1 217 600	58 900	65 000	90 200
5 1/4	Hexagonal	4 27/32	Hexagonal	3 1/4	NC46	6 1/4	809 800	1 077 100	30 600	68 600	74 000
5 1/4	Hexagonal	5	Hexagonal	3 1/4	NC46	6 1/4	809 800	1 196 800	30 600	78 500	83 300
5 1/4	Hexagonal	5	Hexagonal	3 1/2	NC50	6 3/8	999 900	1 077 600	40 800	71 100	78 400
6	Hexagonal	5 3/4	Hexagonal	4	5 1/2 FH	7	1 189 500	1 443 400	51 300	109 100	119 900
6	Hexagonal	5 3/4	Hexagonal	4 1/8	5 1/2 IF	7 3/8	1 669 200	1 371 500	80 400	103 800	116 200

NOTE: Kelly bushings are normally available for remachined kellys in the above table.

[a] All values have a safety factor of 1.0 and are based on 110 000 psi minimum tensile yield (quenched and tempered) for connections and 90 000 psi minimum tensile yield (normalized and tempered) for the drive section. Fully quenched and tempered drive sections will have higher values than those shown. Shear strength is based on 57.7 % of the minimum tensile yield strength.

[b] Labels are for information and assistance in ordering.

[c] Tensile area calculated at root of thread 3/4 inch from pin shoulder.

Table B.18 — Section modulus values for drill pipe

1	2	3	4	5
Label 1[a]	Label 2[a]	Pipe OD in.	Pipe weight lbm/ft	I/C cu. in.
2 3/8	4.85	2 3/8	4.85	0.66
2 3/8	6.65	2 3/8	6.65	0.86
2 7/8	6.85	2 7/8	6.85	1.12
2 7/8	10.40	2 7/8	10.40	1.60
3 1/2	9.50	3 1/2	9.50	1.96
3 1/2	13.30	3 1/2	13.30	2.57
3 1/2	15.50	3 1/2	15.50	2.92
4	11.85	4	11.85	2.70
4	14.00	4	14.00	3.22
4	15.70	4	15.70	3.57
4 1/2	13.75	4 1/2	13.75	3.59
4 1/2	16.60	4 1/2	16.60	4.26
4 1/2	20.00	4 1/2	20.00	5.11
4 1/2	22.82	4 1/2	22.82	5.66
4 1/2	24.66	4 1/2	24.66	6.02
4 1/2	25.50	4 1/2	25.50	6.19
5	16.25	5	16.25	4.85
5	19.50	5	19.50	5.70
5	25.60	5	25.60	7.23
5 1/2	19.20	5 1/2	19.20	6.10
5 1/2	21.90	5 1/2	21.90	7.02
5 1/2	24.70	5 1/2	24.70	7.83
6 5/8	25.20	6 5/8	25.20	9.77
6 5/8	27.70	6 5/8	27.70	10.56

[a] Labels are for information and assistance in ordering.

Table B.19 — Effect of drilling fluid type on coefficient of friction

Drilling fluid	Typical coefficientof friction	Critical hole angledegrees
Water-base mud	0.35	71
Oil-base mud	0.25	76
Synthetic-base mud	0.17	80

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Table B.20 — Compensation factor for different mud densities

Mud density		Mud density	
lbm/gal	f_{MW}	lbm/gal	f_{MW}
8.0	1.04	14.0	0.98
9.0	1.93	15.0	0.97
10.0	1.02	16.0	0.96
11.0	1.01	17.0	0.95
12.0	1.00	18.0	0.94
13.0	0.99	19.0	0.93

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Table B.21 — Youngstown steel test results

Grade	Minimum yield strength	Yield strength maximum	Tensile strength minimum	Average tensile strength of test samples	Endurance limit Minimum test value	Endurance limit Median test value
	ksi	ksi	ksi	ksi	ksi	ksi
E75	75	105	100	123	30	32
X95	95	125	105	132	32	35
G105	105	135	115	144	34	38
S135	135	165	143	167	36	40

3-10-04

Table B.22 — Hook-load at minimum yield strength for New, Premium class (used) and Class 2 (used) drill pipe -26

1	2	3	4	5	6	7	8	9	10	11	12
Labels [a]		Nom.linearmass	ID	Class	OD [b]	Wall [c] thick-ness	Cross-sectional area	Yield strength - psi			
								75 000	95 000	105 000	135 000
								Hook load			
1	2	lbm/ft	in.		in.	in.	sq. in.	lbf			
2 3/8	4.85	4.85	1.995	New	2.375 0	0.190	1.304 2	97 817	123 902	136 944	176 071
				Premium	2.299 0	0.152	1.025 2	76 893	97 398	107 650	138 407
				Class 2	2.261 0	0.133	0.889 1	66 686	84 469	93 360	120 035
2 3/8	6.65	6.65	1.815	New	2.375 0	0.280	1.842 9	138 214	175 072	193 500	248 786
				Premium	2.263 0	0.224	1.434 9	107 616	136 313	150 662	193 709
				Class 2	2.207 0	0.196	1.238 3	92 871	117 636	130 019	167 167
2 7/8	6.85	6.85	2.441	New	2.875 0	0.217	1.812 0	135 902	172 143	190 263	244 624
				Premium	2.788 2	0.174	1.426 0	106 946	135 465	149 725	192 503
				Class 2	2.744 8	0.152	1.237 4	92 801	117 549	129 922	167 043
2 7/8	10.40	10.40	2.151	New	2.875 0	0.362	2.857 9	214 344	271 503	300 082	385 820
				Premium	2.730 2	0.290	2.220 5	166 535	210 945	233 149	299 764
				Class 2	2.657 8	0.253	1.914 1	143 557	181 839	200 980	258 403
3 1/2	9.50	9.50	2.992	New	3.500 0	0.254	2.590 2	194 264	246 068	271 970	349 676
				Premium	3.398 4	0.203	2.039 7	152 979	193 774	214 171	275 363
				Class 2	3.347 6	0.178	1.770 6	132 793	168 204	185 910	239 027
3 1/2	13.30	13.30	2.764	New	3.500 0	0.368	3.620 9	271 569	343 988	380 197	488 825
				Premium	3.352 8	0.294	2.828 7	212 150	268 723	297 010	381 870
				Class 2	3.279 2	0.258	2.445 3	183 398	232 304	256 757	330 116
3 1/2	15.50	15.50	2.602	New	3.500 0	0.449	4.303 7	322 775	408 848	451 885	580 995
				Premium	3.320 4	0.359	3.341 6	250 620	317 452	350 868	451 115
				Class 2	3.230 6	0.314	2.879 6	215 967	273 558	302 354	388 741
4	11.85	11.85	3.476	New	4.000 0	0.262	3.076 7	230 755	292 290	323 057	415 360
				Premium	3.895 2	0.210	2.426 9	182 016	230 554	254 823	327 630
				Class 2	3.842 8	0.183	2.108 4	158 132	200 301	221 385	284 638
4	14.00	14.00	3.340	New	4.000 0	0.330	3.804 8	285 359	361 454	399 502	513 646
				Premium	3.868 0	0.264	2.989 1	224 182	283 963	313 854	403 527
				Class 2	3.802 0	0.231	2.591 5	194 363	246 193	272 108	349 853
4	15.70	15.70	3.240	New	4.000 0	0.380	4.321 6	324 118	410 550	453 765	583 413
				Premium	3.848 0	0.304	3.384 7	253 851	321 544	355 391	456 931
				Class 2	3.772 0	0.266	2.929 8	219 738	278 335	307 633	395 526
4 1/2	13.75	13.75	3.958	New	4.500 0	0.271	3.600 5	270 034	342 043	378 047	486 061
				Premium	4.391 6	0.217	2.843 4	213 258	270 127	298 562	383 865
				Class 2	4.337 4	0.190	2.471 9	185 390	234 827	259 546	333 702
See end of table for notes											

1	2	3	4	5	6	7	8	9	10	11	12
Labels [a]		Nom. linear mass	ID	Class	OD [b]	Wall thick-ness [c]	Cross-sectional area				
								75 000	95 000	105 000	135 000
								Hook load			
1	2	lbm/ft	in.		in.	in.	sq. in.	lbf			
4 1/2	16.60	16.60	3.826	New	4.500 0	0.337	4.407 4	330 558	418 707	462 781	595 004
				Premium	4.365 2	0.270	3.468 9	260 165	329 542	364 231	468 297
				Class 2	4.297 8	0.236	3.010 3	225 771	285 977	316 080	406 388
4 1/2	20.00	20.00	3.640	New	4.500 0	0.430	5.498 1	412 358	522 320	577 301	742 244
				Premium	4.328 0	0.344	4.305 5	322 916	409 026	452 082	581 248
				Class 2	4.242 0	0.301	3.726 7	279 502	354 035	391 302	503 103
4 1/2	22.82	22.82	3.500	New	4.500 0	0.500	6.283 2	471 239	596 903	659 735	848 230
				Premium	4.300 0	0.400	4.900 9	367 566	465 584	514 593	661 620
				Class 2	4.200 0	0.350	4.233 3	317 497	402 163	444 496	571 495
5	16.25	16.25	4.408	New	5.000 0	0.296	4.374 3	328 073	415 559	459 302	590 531
				Premium	4.881 6	0.237	3.455 4	259 155	328 263	362 817	466 479
				Class 2	4.822 4	0.207	3.004 2	225 316	285 400	315 442	405 568
5	19.50	19.50	4.276	New	5.000 0	0.362	5.274 6	395 595	501 087	553 833	712 070
				Premium	4.855 2	0.290	4.153 8	311 535	394 612	436 150	560 764
				Class 2	4.782 8	0.253	3.605 8	270 432	342 548	378 605	486 778
5	25.60	25.60	4.000	New	5.000 0	0.500	7.068 6	530 144	671 515	742 201	954 259
				Premium	4.800 0	0.400	5.529 2	414 690	525 274	580 566	746 443
				Class 2	4.700 0	0.350	4.783 1	358 731	454 392	502 223	645 715
5 1/2	19.20	19.20	4.892	New	5.500 0	0.304	4.962 4	372 181	471 429	521 053	669 925
				Premium	5.378 4	0.243	3.923 5	294 260	372 730	411 965	529 669
				Class 2	5.317 6	0.213	3.412 7	255 954	324 208	358 335	460 717
5 1/2	21.90	21.90	4.778	New	5.500 0	0.361	5.828 2	437 116	553 681	611 963	786 809
				Premium	5.355 6	0.289	4.597 1	344 780	436 721	482 692	620 604
				Class 2	5.283 4	0.253	3.993 8	299 533	379 409	419 346	539 160
5 1/2	24.70	24.70	4.670	New	5.500 0	0.415	6.629 6	497 222	629 814	696 111	894 999
				Premium	5.334 0	0.332	5.217 1	391 285	495 627	547 799	704 313
				Class 2	5.251 0	0.290	4.527 1	339 533	430 076	475 347	611 160
6 5/8	25.20	25.20	5.965	New	6.625 0	0.330	6.526 2	489 464	619 988	685 250	881 035
				Premium	6.493 0	0.264	5.166 2	387 466	490 790	542 452	697 438
				Class 2	6.427 0	0.231	4.496 5	337 236	427 166	472 131	607 026
6 5/8	27.70	27.70	5.901	New	6.625 0	0.362	7.122 7	534 199	676 652	747 879	961 558
				Premium	6.480 2	0.290	5.632 3	422 419	535 064	591 387	760 354
				Class 2	6.407 8	0.253	4.899 4	367 455	465 443	514 437	661 419

a Labels are for information and assistance in ordering

b OD for new pipe is the original nominal; OD for Premium class is with 20 % wall reduction; OD for Class 2 is with 30 % wall reduction. All wall reduction occurs from the OD.

c Wall thickness for new pipe is the original nominal; wall thickness for Premium class pipe is the minimum remaining wall allowed for this class and is based on 80 % of new; wall thickness for Class 2 pipe is the minimum remaining wall allowed for this class and is based on 70 % of new.

Table B.23 — Hook-load at minimum yield strength for New, Premium class (used) and Class 2 (used) tubing work strings

1	2	3	4	5	6	7	8	9	10	11	12
Labels [a]		**Nom. linear mass**	**ID**	**Class**	**OD** [b]	**Wall thick-ness** [c]	**Cross-sectional area**	**Yield strength - psi**			
								55 000	75 000	80 000	105 000
								Hook load			
1	2	lbm/ft	in.		in.	in.	sq. in.	lbf			
3/4	1.20	1.20	0.824	New	1.050 0	0.113	0.332 6	18 295	24 948	26 611	34 927
				Premium	1.004 8	0.090	0.259 7	14 283	19 477	20 775	27 267
				Class 2	0.982 2	0.079	0.224 4	12 343	16 832	17 954	23 564
3/4	1.50	1.50	0.742	New	1.050 0	0.154	0.433 5	23 842	32 512	34 679	45 516
				Premium	0.988 4	0.123	0.334 9	18 418	25 115	26 790	35 161
				Class 2	0.957 6	0.108	0.287 8	15 829	21 585	23 024	30 219
1	1.80	1.80	1.049	New	1.315 0	0.133	0.493 9	27 163	37 041	39 510	51 857
				Premium	1.261 8	0.106	0.386 2	21 242	28 966	30 897	40 552
				Class 2	1.235 2	0.093	0.334 0	18 372	25 053	26 724	35 075
1	2.25	2.25	0.957	New	1.315 0	0.179	0.638 8	35 135	47 912	51 106	67 077
				Premium	1.243 4	0.143	0.495 0	27 222	37 122	39 596	51 970
				Class 2	1.207 6	0.125	0.426 0	23 432	31 953	34 083	44 734
1 1/4	2.40	2.40	1.380	New	1.660 0	0.140	0.668 5	36 769	50 140	53 482	70 196
				Premium	1.604 0	0.112	0.525 0	28 873	39 373	41 998	55 122
				Class 2	1.576 0	0.098	0.455 0	25 027	34 128	36 403	47 779
1 1/4	3.02	3.02	1.278	New	1.660 0	0.191	0.881 5	48 481	66 110	70 517	92 554
				Premium	1.583 6	0.153	0.686 8	37 776	51 513	54 947	72 118
				Class 2	1.545 4	0.134	0.593 0	32 613	44 472	47 437	62 261
1 1/4	3.20	3.20	1.264	New	1.660 0	0.198	0.909 4	50 018	68 206	72 753	95 489
				Premium	1.580 8	0.158	0.707 8	38 930	53 087	56 626	74 322
				Class 2	1.541 2	0.139	0.610 7	33 590	45 805	48 858	64 126
1 1/2	2.90	2.90	1.610	New	1.900 0	0.145	0.799 5	43 970	59 959	63 957	83 943
				Premium	1.842 0	0.116	0.629 0	34 595	47 175	50 320	66 045
				Class 2	1.813 0	0.101	0.545 7	30 016	40 931	43 660	57 304
1 1/2	4.19	4.19	1.462	New	1.900 0	0.219	1.156 5	63 610	86 741	92 523	121 437
				Premium	1.812 4	0.175	0.901 1	49 562	67 584	72 090	94 618
				Class 2	1.768 6	0.153	0.777 9	42 787	58 345	62 235	81 684
2 1/16	3.25	3.25	1.751	New	2.063 0	0.156	0.934 6	51 403	70 095	74 768	98 133
				Premium	2.000 6	0.125	0.735 4	40 450	55 158	58 836	77 222
				Class 2	1.969 4	0.109	0.638 2	35 099	47 862	51 053	67 007
2 3/8	4.70	4.70	1.995	New	2.375 0	0.190	1.304 2	71 733	97 817	104 339	136 944
				Premium	2.299 0	0.152	1.025 2	56 388	76 893	82 019	107 650
				Class 2	2.261 0	0.133	0.889 1	48 903	66 686	71 132	93 360
2 3/8	5.30	5.30	1.939	New	2.375 0	0.218	1.477 3	81 249	110 794	118 181	155 112
				Premium	2.287 8	0.174	1.157 9	63 686	86 844	92 634	121 581
				Class 2	2.244 2	0.153	1.002 7	55 150	75 205	80 218	105 286
See end of table for notes											

1	2	3	4	5	6	7	8	9	10	11	12
Labels [a]		Nom. linear mass	ID	Class	OD [b]	Wall [c] thick-ness	Cross-sectional area	55 000	75 000	80 000	105 000
								Hook load			
1	2	lbm/ft	in.		in.	in.	sq. in.	lbf			
2 3/8	5.95	5.95	1.867	New	2.375 0	0.254	1.692 5	93 087	126 936	135 399	177 711
				Premium	2.273 4	0.203	1.321 6	72 686	99 117	105 725	138 764
				Class 2	2.222 6	0.178	1.142 2	62 820	85 663	91 374	119 928
2 7/8	6.50	6.50	2.441	New	2.875 0	0.217	1.812 0	99 661	135 902	144 962	190 263
				Premium	2.788 2	0.174	1.426 0	76 427	106 946	114 076	149 725
				Class 2	2.744 8	0.152	1.237 4	88 054	92 801	98 988	129 922
2 7/8	8.70	8.70	2.259	New	2.875 0	0.308	2.483 9	136 612	186 289	198 708	260 805
				Premium	2.751 8	0.246	1.939 4	106 667	145 455	155 152	203 637
				Class 2	2.690 2	0.216	1.676 1	92 186	125 709	134 089	175 992
2 7/8	9.50	9.50	2.195	New	2.875 0	0.340	2.707 7	148 926	203 080	216 619	284 313
				Premium	2.739 0	0.272	2.108 1	115 945	158 106	168 647	221 349
				Class 2	2.671 0	0.238	1.819 2	100 053	136 436	145 532	191 011
2 7/8	10.70	10.70	2.091	New	2.875 0	0.392	3.057 8	168 180	229 337	244 626	321 072
				Premium	2.718 2	0.314	2.369 0	130 296	177 676	189 522	248 747
				Class 2	2.639 8	0.274	2.039 1	112 151	152 933	163 128	214 106
2 7/8	11.00	11.00	2.065	New	2.875 0	0.405	3.142 7	172 848	235 702	251 415	329 983
				Premium	2.713 0	0.324	2.431 7	133 744	182 378	194 536	255 329
				Class 2	2.632 0	0.283	2.091 7	115 042	156 875	167 334	219 626
3 1/2	12.80	12.80	2.764	New	3.500 0	0.368	3.620 9	199 151	271 569	289 674	380 197
				Premium	3.352 8	0.294	2.828 7	155 577	212 150	226 293	297 010
				Class 2	3.279 2	0.258	2.445 3	134 492	183 398	195 624	256 757
3 1/2	12.95	12.95	2.750	New	3.500 0	0.375	3.681 6	202 485	276 117	294 524	386 563
				Premium	3.350 0	0.300	2.874 6	158 101	215 592	229 965	301 828
				Class 2	3.275 0	0.263	2.484 3	136 637	186 323	198 745	260 853
3 1/2	15.80	15.80	2.548	New	3.500 0	0.476	4.522 1	248 715	339 156	361 767	474 819
				Premium	3.309 6	0.381	3.503 8	192 708	262 783	280 302	367 897
				Class 2	3.214 4	0.333	3.016 0	165 879	226 198	241 278	316 678
3 1/2	16.70	16.70	2.480	New	3.500 0	0.510	4.790 6	263 484	359 296	383 249	503 015
				Premium	3.296 0	0.408	3.701 8	203 596	277 631	296 140	388 684
				Class 2	3.194 0	0.357	3.181 8	175 001	238 638	254 547	334 093
4 1/2	15.50	15.50	3.826	New	4.500 0	0.337	4.407 4	242 409	330 558	352 595	462 781
				Premium	4.365 2	0.270	3.468 9	190 788	260 165	277 509	364 231
				Class 2	4.297 8	0.236	3.010 3	165 565	225 771	240 823	316 080
4 1/2	19.20	19.20	3.640	New	4.500 0	0.430	5.498 1	302 396	412 358	439 848	577 301
				Premium	4.328 0	0.344	4.305 5	236 805	322 916	344 443	452 082
				Class 2	4.242 0	0.301	3.726 7	204 968	279 502	298 135	391 302

a Labels are for information and assistance in ordering

b OD for new pipe is the original nominal; OD for Premium class is with 20 % wall reduction; OD for Class 2 is with 30 % wall reduction. All wall reduction occurs from the OD.

c Wall thickness for new pipe is the original nominal; wall thickness for Premium class pipe is the minimum remaining wall allowed for this class and is based on 80 % of new; wall thickness for Class 2 pipe is the minimum remaining wall allowed for this class and is based on 70 % of new.

Table B.24 — Maximum stress at the root of the last engaged thread for the pin of an NC 50 axisymmetric model

Load condition	First condition a		Second condition b	
SRG width inches	Maximum equivalent Stress c,d	Maximum axial stress	Maximum equivalent Stress c,d	Maximum axial stress
3/4	84%	82 %	83 %	81 %
1	70 %	56 %	63 %	53 %
1 1/4	75 %	63 %	73 %	64 %
No SRG	100 %	100 %	100 %	100 %

a Make-up only at 562 000 pounds axial force on shoulder

b 1 125 000 pounds axial tension applied to connection which causes shoulder separation.

c Equivalent stress is equal to 0.707 x [(σ1 – σ2)2 + (σ2 – σ3)2 + (σ3 – σ1)2]1/2 where σ1, σ2 and σ3 are principle stresses.

d In each case shown, equivalent stress at the root of the last engaged thread has exceeded the yield strength because these finite element calculations have been made for linear elastic material behavior. The behavior of an actual pin is elastic-plastic.

Table B.25 — IADC roller bit classification chart

	Series	Formations	Type	Features: Standard roller bearings	Roller bearing, air cooled	Roller bearing gauge protected
				(1)	(2)	(3)
Milled tooth bits	1	Soft formations with low compressive strength and high drillability	1 2 3 4			
	2	Medium to medium hard formations with high compressive strength	1 2 3 4			
	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			
	5	Soft to medium formations with low compressive strength	1 2 3 4			
	6	Medium hard formations with high compressive strength	1 2 3 4			
	7	Hard semi-abrasive and abrasive formations	1 2 3 4			
	8	Extremely hard and abrasive formations	1 2 3 4			

Table B.25 — IADC roller bit classification chart (continued)

	Series	Formations	Type	Features			
				Sealed roller bearing	Sealed roller bearing gauge protected	Sealed friction bearing	Sealed friction bearing gauge protected
				(4)	(5)	(6)	(7)
Milled tooth bits	1	Soft formations with low compressive strength and high drillability	1 2 3 4				
	2	Medium to medium hard formations with high compressive strength	1 2 3 4				
	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				
	5	Soft to medium formations with low compressive strength	1 2 3 4				
	6	Medium hard formations with high compressive strength	1 2 3 4				
	7	Hard semi-abrasive and abrasive formations	1 2 3 4				
	8	Extremely hard and abrasive formations	1 2 3 4				

Table B.26 — IADC bit classification codes

Fourth position

Code [a]	Code feature	Code [a]	Code feature
A	Air application [b]	N	
B		O	
C	Center jet	P	
D	Deviation control	Q	
E	Extended jets [c]	R	Reinforced welds [d]
F		S	Standard steel tooth model [e]
G	Extra gauge/body protection	T	
H		U	
I		V	
J	Jet deflection	W	
K		X	Chisel inserts
L		Y	Conical inserts
M		Z	Other insert shapes

[a] The above codes are used in the 4th position of the 4 – character IADC bit classification code to indicate additional design features

[b] Journal bearing bits with air circulation nozzles

[c] Full extension (welded tubes with nozzles). Partial extensions should be noted elsewhere.

[d] For percussion applications

[e] Milled tooth bits with none of the extra features listed in this table. As an example, a milled tooth bit designed for the softest series, softest type in that series, with standard gauge, and no extra features, are designated 1-1-1-S on the bit carton. The manufacture will also list this bit designation in this block of the form

Table B.27 — Recommended make-up torque ranges

for roller cone bits

Connection Label	Minimum make-up torque Ft-lbs	Maximum make-up torque Ft-lbs
2 3/8 Reg	3 000	3 500
2 7/8 Reg	4 500	5 500
3 1/2 Reg	7 000	9 000
4 1/2 Reg	12 000	16 000
6 5/8 Reg	28 000	32 000
7 5/8 Reg	34 000	40 000
8 5/8 Reg	40 000	60 000

NOTE 1 Basis of calculation for recommended make-up torque assumed the use of a thread compound containing 40 to 60 % by weight of finely powdered metallic zinc with no more than 0.3 % total active sulfur, applied thoroughly to all threads and shoulders (see the caution regarding the use of hazardous materials in Appendix G of API Specification 7).

NOTE 2 Due to the irregular geometry of the ID bore in roller cone bits, torque values are based on estimated cross-sectional areas and have been proven by field experience.

Table B.28 — Recommended minimum make-up torque

For diamond drill bits

RSC Label [a]	Maximum pin ID inches	Bit sub OD inches	Minimum make-up torque [b, c] Ft-lbs
2 3/8 Reg	1	3 3 1/8 3 1/4	1 791 * 2 419 * 3 085 *
2 7/8 Reg	1 1/4	3 1/2 3 3/4 3 7/8	3 073 * 4 617 4 658
3 1/2 Reg	1 1/2	4 1/8 4 1/4 4 1/2	5 171 * 6 306 * 7 660
4 1/2 Reg	2 1/4	5 1/2 5 3/4 6 6 1/4	12 451 * 16 476 * 17 551 17 757
6 5/8 Reg	3 1/4	7 1/2 7 3/4 8 8 1/4	37 100 * 37 857 38 193 38 527
7 5/8 Reg	3 3/4	8 1/2 8 3/4 9 9 1/4 9 1/2	48 296 * 57 704 * 59 966 60 430 60 895

NOTE 1 Torque figures followed by an asterisk * indicate that the weaker member for the corresponding outside diameter (OD) and bore is the BOX. For all other torque values the weaker member is the PIN.

NOTE 2 Normal torque range is tabulated value plus 10 %. Higher torque values may be used under extreme conditions.

[a] Labels are for information and assistance in ordering

[b] Basis of calculation for recommended make-up torque assumed the use of a thread compound containing 40 to 60 % by weight of finely powdered metallic zinc, with no more than 0,3 % total active sulfur, applied thoroughly to all threads and shoulders (see the caution regarding the use of hazardous materials in Appendix G of API Specification 7).

[c] Calculations are done using the modified Screw Jack formula in 5.8.2.3 and a unit stress of

50 000 psi in the pin or box, whichever is weaker.

Table B.29 — Common roller bit sizes

Size of bit inches	Size of bit inches
3 3/4	9 1/2
3 7/8	9 7/8
4 3/4	10 5/8
5 7/8	11
6	12 1/4
6 1/8	13 1/2
6 1/4	14 1/2
6 1/2	16
6 3/4	17 1/2
7 7/8	20
8 3/8	22
8 1/2	24
8 3/4	26

Table B.30 — Common fixed cutter bit sizes

Size of bit inches	Size of bit inches
3 7/8	8 1/2
4 1/2	8 3/4
4 3/4	9 1/2
5 7/8	9 7/8
6	10 5/8
6 1/8	12 1/4
6 1/4	14 3/4
6 1/2	16
6 3/4	17 1/2
7 7/8	

Table B.31 — Maximum stress at the root of the last engaged thread for the pin of an NC 50 axisymmetric model

Load condition	First condition [a]		Second condition [b]	
SRG width mm	Maximum equivalent Stress [c,d]	Maximum axial stress	Maximum equivalent Stress [c,d]	Maximum axial stress
19.0	84%	82 %	83 %	81 %
25.4	70 %	56 %	63 %	53 %
31.8	75 %	63 %	73 %	64 %
No SRG	100 %	100 %	100 %	100 %

[a] Make-up only at 2 500 kN axial force on shoulder

[b] 5 004 kN axial tension applied to connection which causes shoulder separation.

[c] Equivalent stress is equal to 0.707 x $[(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2]^{1/2}$ where σ_1, σ_2 and σ_3 are principle stresses.

[d] In each case shown, equivalent stress at the root of the last engaged thread has exceeded the yield strength because these finite element calculations have been made for linear elastic material behavior. The behavior of an actual pin is elastic-plastic.

Table B.32 — S_H/S_A, The slip crushing constant

Net slip die contact length, inches	**Coefficient** of friction μ*	**Transverse** load factor "K"	**Drill pipe size, inches**								
			2.38	2.88	3.5	4	4.5	5	5.5	5.88	6.63
			S_H/S_T, the Slip Crushing Constant								
12.00	0.06	4.3676	1.27	1.34	1.43	1.50	1.58	1.65	1.73	1.79	1.91
	0.08	4.0000	1.25	1.31	1.39	1.45	1.52	1.59	1.66	1.71	1.82
	0.10	3.6875	1.22	1.28	1.35	1.41	1.47	1.54	1.60	1.65	1.75
	0.12	3.4186	1.21	1.26	1.32	1.38	1.43	1.49	1.55	1.59	1.68
	0.14	3.1848	1.19	1.24	1.30	1.35	1.40	1.45	1.50	1.55	1.63
13.75	0.06	4.3676	1.23	1.29	1.37	1.43	1.49	1.56	1.62	1.67	1.78
	0.08	4.0000	1.21	1.26	1.33	1.39	1.44	1.50	1.56	1.61	1.70
	0.10	3.6875	1.19	1.24	1.30	1.35	1.40	1.46	1.51	1.55	1.64
	0.12	3.4186	1.18	1.22	1.27	1.32	1.37	1.42	1.47	1.50	1.58
	0.14	3.1848	1.16	1.20	1.25	1.30	1.34	1.38	1.43	1.46	1.53
16.00	0.06	4.3676	1.20	1.24	1.31	1.36	1.41	1.47	1.52	1.56	1.65
	0.08	4.0000	1.18	1.22	1.28	1.32	1.37	1.42	1.47	1.51	1.59
	0.10	3.6875	1.16	1.20	1.25	1.29	1.34	1.38	1.43	1.46	1.53
	0.12	3.4186	1.15	1.18	1.23	1.27	1.31	1.35	1.39	1.42	1.49
	0.14	3.1848	1.14	1.17	1.21	1.25	1.28	1.32	1.36	1.39	1.45
19.06	0.06	4.3676	1.16	1.20	1.25	1.29	1.33	1.38	1.42	1.46	1.53
	0.08	4.0000	1.15	1.18	1.23	1.26	1.30	1.34	1.38	1.41	1.48
	0.10	3.6875	1.13	1.16	1.21	1.24	1.27	1.31	1.35	1.38	1.43
	0.12	3.4186	1.12	1.15	1.19	1.22	1.25	1.28	1.32	1.34	1.40
	0.14	3.1848	1.11	1.14	1.17	1.20	1.23	1.26	1.29	1.32	1.36
19.25	0.06	4.3676	1.16	1.20	1.25	1.29	1.33	1.37	1.42	1.45	1.52
	0.08	4.0000	1.14	1.18	1.22	1.26	1.30	1.34	1.38	1.41	1.47
	0.10	3.6875	1.13	1.16	1.20	1.24	1.27	1.31	1.34	1.37	1.43
	0.12	3.4186	1.12	1.15	1.19	1.22	1.25	1.28	1.31	1.34	1.39
	0.14	3.1848	1.11	1.14	1.17	1.20	1.23	1.26	1.29	1.31	1.36
22.00	0.06	4.3676	1.14	1.17	1.21	1.25	1.28	1.32	1.36	1.39	1.45
	0.08	4.0000	1.12	1.15	1.19	1.22	1.26	1.29	1.32	1.35	1.40
	0.10	3.6875	1.11	1.14	1.17	1.20	1.23	1.26	1.29	1.32	1.37
	0.12	3.4186	1.10	1.13	1.16	1.19	1.21	1.24	1.27	1.29	1.33
	0.14	3.1848	1.10	1.12	1.15	1.17	1.20	1.22	1.25	1.27	1.31
24.56	0.06	4.3676	1.12	1.15	1.19	1.22	1.25	1.28	1.31	1.34	1.39
	0.08	4.0000	1.11	1.14	1.17	1.20	1.23	1.25	1.28	1.31	1.35
	0.10	3.6875	1.10	1.12	1.15	1.18	1.20	1.23	1.26	1.28	1.32
	0.12	3.4186	1.09	1.11	1.14	1.16	1.19	1.21	1.24	1.26	1.29
	0.14	3.1848	1.09	1.11	1.13	1.15	1.17	1.20	1.22	1.24	1.27

* A coefficient of friction of 0.08 is typical for the backs of slips lubricated with thread compound.

Table B.33 — Buoyancy Factors – SI units

Based on a specific gravity of 7.842 for steel

Specific gravity of mud	buoyancy factor Kb	Specific gravity of mud	buoyancy factor Kb	Specific gravity of mud	buoyancy factor Kb	Specific gravity of mud	buoyancy factor Kb
0.950	0.879	1.300	0.834	1.650	0.790	2.000	0.745
0.975	0.876	1.325	0.831	1.675	0.786	2.025	0.742
1.000	0.872	1.350	0.828	1.700	0.783	2.050	0.739
1.025	0.869	1.375	0.825	1.725	0.780	2.075	0.735
1.050	0.866	1.400	0.821	1.750	0.777	2.100	0.732
1.075	0.863	1.425	0.818	1.775	0.774	2.125	0.729
1.100	0.860	1.450	0.815	1.800	0.770	2.150	0.726
1.125	0.857	1.475	0.812	1.825	0.767	2.175	0.723
1.150	0.853	1.500	0.809	1.850	0.764	2.200	0.719
1.175	0.850	1.525	0.806	1.875	0.761	2.225	0.716
1.200	0.847	1.550	0.802	1.900	0.758	2.250	0.713
1.225	0.844	1.575	0.799	1.925	0.755	2.275	0.710
1.250	0.841	1.600	0.796	1.950	0.751	2.300	0.707
1.275	0.837	1.625	0.793	1.975	0.748	2.325	0.704

Table B.34 — Buoyancy Factors

Based on a steel density of 0.2836 lbs/cubic foot

Mud Density Lb/gal	Mud Density lb/cu ft	Buoyancy Factor Kb	Mud Density lb/gal	Mud Density lb/cu ft	Buoyancy Factor Kb	Mud Density lb/gal	Mud Density lb/cu ft	Buoyancy Factor Kb
1	2	3	1	2	3	1	2	3
8.2	61.34	0.875	12.0	89.77	0.817	16.0	119.69	0.756
8.4	62.84	0.872	12.2	91.26	0.814	16.2	121.18	0.753
8.6	64.33	0.869	12.4	92.76	0.811	16.4	122.68	0.750
8.8	65.83	0.866	12.6	94.25	0.807	16.6	124.18	0.747
9.0	67.32	0.863	12.8	95.75	0.804	16.8	125.67	0.744
9.2	68.82	0.860	13.0	97.25	0.801	17.0	127.17	0.740
9.4	70.32	0.857	13.2	98.74	0.798	17.2	128.66	0.737
9.6	71.81	0.853	13.4	100.24	0.795	17.4	130.16	0.734
9.625	72.00	0.853	13.6	101.74	0.792	17.6	131.66	0.731
9.8	73.31	0.850	13.8	103.23	0.789	17.8	133.15	0.728
10.0	74.81	0.847	14.0	104.73	0.786	18.0	134.65	0.725
10.2	76.30	0.844	14.2	106.22	0.783	18.2	136.15	0.722
10.4	77.80	0.841	14.4	107.72	0.780	18.250	136.52	0.721
10.6	79.29	0.838	14.6	109.22	0.777	18.4	137.64	0.719
10.8	80.79	0.835	14.8	110.71	0.774	18.6	139.14	0.716
11.0	82.29	0.832	15.0	112.21	0.771	18.8	140.63	0.713
11.2	83.78	0.829	15.2	113.70	0.768	19.0	142.13	0.710
11.4	85.28	0.826	15.4	115.20	0.765	19.2	143.63	0.707
11.6	86.77	0.823	15.6	116.70	0.762	19.4	145.12	0.704
11.8	88.27	0.820	15.8	118.19	0.759	19.6	146.62	0.700

Table B.35 — Modulus of Elasticity (E)

Material	Used in	psi
Alloy Steel	Drill pipe and Drill Collars	30.0×10^6
Aluminum	Drill pipe and Drill Collars	10.5×10^6
Monel	Non-magnetic Collars	26.0×10^6
Stainless Steel	Non-magnetic Collars	28.0×10^6
Tungsten Carbide	Bit Inserts	87.0×10^6
Tungsten	Collars	51.5×10^6

Table B.36 — Maximum drill collar size that can be caught with an overshot or washed over with washpipe

Hole size, in.	Overshot		Washpipe		Max. Fish od to catch and/or wash over, in
	Size, in	Max. Catch, in	Size, in	Max fish od, in.	
6 1/8	5 3/4	5 1/8	5 1/2	4 3/4	4 3/4
6 1/4	5 3/4	5 1/8	5 3/4	4 7/8	4 7/8
6 3/4	6 3/8	5 1/4	6	5 1/8	5 1/8
7 7/8	7 3/8	6 1/4	7	6 1/8	6 1/8
8 3/8	7 7/8	6 3/4	7 3/8	6 1/2	6 1/2
8 1/2	8	6 7/8	7 5/8	6 3/4	6 3/4
8 3/4	8 1/4	7 1/8	8 1/8	7 1/8	7 1/8
9 1/2	9	7 7/8	8 5/8	7 5/8	7 5/8
9 7/8	9 1/8	8	9	8	8
10 5/8	9 3/4	8 5/8	9 5/8	8 1/2	8 1/2
11	10 1/2	8 7/8	10 3/4	9 5/8	8 7/8
12 1/4	11 3/4	10 1/8	11 3/4	10 1/2	10 1/8
13 3/4	12 3/4	11 1/4	12 3/4	11 1/2	11 1/4
14 3/4	13 3/4	12	13 3/8	12	12
17 1/2	15 1/8	13 3/8	16	14 1/2	13 3/8
20	16 3/4	14 3/4	18 5/8	17 3/8	14 3/4
24	20 1/4	16 3/4	21	19 1/2	16 3/4
26	24 3/4	22	21	19 1/2	19 1/2

* from “How to Select Bottom-hole Drilling Assemblies” by Gerald E. Wilson

Table B.37 — Youngstown steel test results*

Grade	API Minimum Yield Strength	API Yield Strength Maximum	API Tensile Strength Minimum	Average Tensile Strength of Test Samples	Endurance Limit Minimum Test Value	Endurance Limit Median Test Value
	ksi	ksi	Ksi	ksi	ksi	Ksi
E75	75	105	100	123	30	32
X95	95	125	105	132	32	35
G105	105	135	115	144	34	38
S135	135	165	145	167	36	40

*Youngstown Sheet and Tube Company, 1969 ASME Conference, Tulsa, Oklahoma.

Table B.38 — Fatigue Endurance Limits Compressively Loaded Drill Pipe

Grade	API Minimum Yield Strength	API Minimum Tensile Strength	Calculated Fatigue Endurance Limit
	ksi	ksi	ksi
E75	75	100	22.0
X95	95	105	23.1
G105	105	115	25.3
S135	135	145	31.9

Table B.39 — Results of Fatigue Tests by Morgan and Roblin[11]

					Endurance limit, kpsi		Endurance ratio (Endurance limit to Tensile strength)	
Grade	OD, in	WT, in.	No. pipe samples	Tensile strength, kpsi	Median	Minimum	Median	Minimum
E75	4..5	0.271 and 0.337	4	123.0	32	30	0.260	0.244
X95	4.5	0.337	2	132.4	35	32	0.264	0.242
G105	4.5	0.337	3	144.0	38	34	0.264	0.236
S135	4.5	0.337	5	167.0	40	36	0.240	0.216

Table B.40 — Recommended constants to determine k-factor (ka) for surface finish[16]

Surface Finish	p (kpsi)	q
Ground	1.34	-0.085
Machined or cold drawn	2.7	-0.265
Hot Rolled	14.4	-0.718
As forged	39.9	-0.995

Table B.41 — Size factor (kb) for API drill pipe

Drillpipe OD, in	k_b
2 3/8	0.8437
2 7/8	0.8103
3 1/2	0.7687
4	0.7353
4 1/2	0.7020
5	0.6687
5 1/2	0.6353
5 7/8	0.6103
6 5/8	0.5603

Table B.42 — Suggested endurance limits and S-N curve fit constants for API drill pipe at zero mean stress subjected to rotating (dogleg) bending at temperatures below 400 °F

			k-factors						S-N curve fit constants#		
									a, kpsi		
Drill pipe OD, in	Grade	Min. Tensile Strength, kpsi	k_a (Surface finish: hot rolled)	k_b (Size factor)	k_c (Load factor for rotating bending)	k_d (Temp. factor)	k_e (*Res. stress)*	Endurance limit at zero mean stress*, S_{eo}, kpsi	Non-corrosive environ.	Corrosive environ.	*b* (Corrosive and non-corrsove environ.)
2 3/8	E75	100	0.528	0.844	1.000	1.000	1.000	22.4	361.0	216.6	-0.201
	X95	105	0.510	0.844	1.000	1.000	1.000	22.7	392.6	235.5	-0.206
	G105	115	0.477	0.844	1.000	1.000	1.000	23.3	459.0	275.4	-0.216
	S135	145	0.404	0.844	1.000	1.000	1.000	24.9	683.5	410.1	-0.240
2 7/8	E75	100	0.528	0.810	1.000	1.000	1.000	21.6	375.9	225.5	-0.207
	X95	105	0.510	0.810	1.000	1.000	1.000	21.8	408.7	245.2	-0.212
	G105	115	0.477	0.810	1.000	1.000	1.000	22.4	477.9	286.7	-0.221
	S135	145	0.404	0.810	1.000	1.000	1.000	23.9	711.6	427.0	-0.246
3 1/2	E75	100	0.528	0.769	1.000	1.000	1.000	20.4	396.2	237.7	-0.215
	X95	105	0.510	0.769	1.000	1.000	1.000	20.7	430.9	258.5	-0.220
	G105	115	0.477	0.769	1.000	1.000	1.000	21.3	503.8	302.3	-0.229
	S135	145	0.404	0.769	1.000	1.000	1.000	22.7	750.2	450.1	-0.253
4	E75	100	0.528	0.735	1.000	1.000	1.000	19.6	414.2	248.5	-0.221
	X95	105	0.510	0.735	1.000	1.000	1.000	19.8	450.4	270.2	-0.226
	G105	115	0.477	0.735	1.000	1.000	1.000	20.3	526.6	316.0	-0.236
	S135	145	0.404	0.735	1.000	1.000	1.000	21.7	784.2	470.5	-0.260
4 1/2	E75	100	0.528	0.702	1.000	1.000	1.000	18.7	433.9	260.3	-0.228
	X95	105	0.510	0.702	1.000	1.000	1.000	18.9	471.8	283.1	-0.233
	G105	115	0.477	0.702	1.000	1.000	1.000	19.4	551.6	331.0	-0.242
	S135	145	0.404	0.702	1.000	1.000	1.000	20.7	821.5	492.9	-0.266
5	E75	100	0.528	0.669	1.000	1.000	1.000	17.8	455.5	273.3	-0.235
	X95	105	0.510	0.669	1.000	1.000	1.000	18.0	495.3	297.2	-0.240
	G105	115	0.477	0.669	1.000	1.000	1.000	18.5	579.1	347.5	-0.249
	S135	145	0.404	0.669	1.000	1.000	1.000	19.7	862.4	517.4	-0.273
5 1/2	E75	100	0.528	0.635	1.000	1.000	1.000	16.9	479.4	287.6	-0.242
	X95	105	0.510	0.635	1.000	1.000	1.000	17.1	521.3	312.8	-0.247
	G105	115	0.477	0.635	1.000	1.000	1.000	17.6	609.5	365.7	-0.257
	S135	145	0.404	0.635	1.000	1.000	1.000	18.8	907.7	544.6	-0.281
5 7/8	E75	100	0.528	0.610	1.000	1.000	1.000	16.2	499.0	299.4	-0.248
	X95	105	0.510	0.610	1.000	1.000	1.000	16.5	542.7	325.6	-0.253
	G105	115	0.477	0.610	1.000	1.000	1.000	16.9	634.5	380.7	-0.262
	S135	145	0.404	0.610	1.000	1.000	1.000	18.0	944.8	566.9	-0.287
6 5/8	E75	100	0.528	0.560	1.000	1.000	1.000	14.9	543.6	326.1	-0.260
	X95	105	0.510	0.560	1.000	1.000	1.000	15.1	591.1	354.7	-0.265
	G105	115	0.477	0.560	1.000	1.000	1.000	15.5	691.1	414.6	-0.275
	S135	145	0.404	0.560	1.000	1.000	1.000	16.5	1029.1	617.5	-0.299

* Endurance limit exists only for operation in a non-corrosive environment # S-N curve constants for Equation 2 or 15

Table B.43 — Values Used in Preparing Figure 77

Grade	Typical yield strength	Expected ultimate strength and fatigue stress limit for one revolution	Minimum fatigue stress limit for 1 000 000 revolutions
	ksi	ksi	Ksi
E75	87.5	121.5	26.7
X95	103.0	131.5	28.9
G105	124.0	149.5	32.9
S135	150.0	159.0	35.0

Table B.44 — Specifications of Various Tongs

Tong Model	Max.Torque (ft.-lbs.)	Arm Length (ft.)
SDD	100.000	5.000
DB	65.000	4.710
C	35.000	3.927
F	25.000	2.670
LF	16.000	2.250

Table B.45 — Drill Pipe Section Modulus

Pipe OD (inches)	Nominal Weight (lb/ft)	Section Modulus (cu.in.)
2 3/8	4.85	0.66
	6.65	0.87
2 7/8	6.85	1.12
	10.40	1.60
3-1/2	9.50	1.96
	13.30	2.57
	15.50	2.92
4	11.85	2.70
	14.00	3.22
	15.70	3.58
4-1/2	13.75	3.59
	16.60	4.27
	20.00	5.17
	22.82	5.68
	24.66	6.03
	25.50	6.19
5	16.25	4.86
	19.50	5.71
	25.60	7.25
5-1/2	19.20	6.11
		7.03
6-5/8	25.20	9.79

Table B.46 — Grant Prideco GPDS™ Connection Information

Connection	OD (in)	ID (in)	Tensile Yield (lb)	Torsional Yield (ft-lb)	Make-up Torque (ft-lb)
GPDS26[1]	3 ½	1 ¾	313,700	8,800	4,300-5,300
	3 ½	1 11/16	333,900	9,700	4,600-5,800
	3 ½	1 5/8	353,400	10,500	4,800-6,300
GPDS31[1]	4 1/8	2	495,700	17,200	7,900-10,300
	4 1/8	1 7/8	541,400	18,200	7,900-10,900
GPDS38[1]	5	2 7/16	708,100	29,200	13,300-17,500
	5	2 9/16	649,200	25,800	12,200-15,500
	4 7/8	2 7/16	708,100	28,600	12,900-17,100
	4 7/8	2 9/16	649,200	25,700	12,100-15,400
	4 ¾	2 1/8	842,400	28,100	10,700-16,900
	4 ¾	2 1/2	679,000	24,100	10700-14500
GPDS40[1]	5 ¼	2 7/16	897,200	38,100	16,700-22,800
	5 1/4	2 9/16	838,300	36,400	16,600-21,900
	5 ¼	2 11/16	776,400	32,700	15,400-19,600
	5	2 7/16	897,200	29,700	11,700-17,800
	5	2 9/16	838,300	28,100	11,700-16,900
GPDS46[1]	6 ½	3 ½	742,100	32,500	16,900-18,800
	6	2 ¾	1,183,900	60,700	25,800-36,400
	6	3	1,048,400	52,900	23,500-31,800
	6	3 1/4	901,200	42,900	20,100-25,700
GPDS50[1]	6 5/8	2 ¾	1,552,000	90,600	36,800-54,400
	6 5/8	3	1,416,500	82,900	34,600-49,700
	6 5/8	3 ¼	1,269,200	72,200	31,000-43,300
	6 ½	3 ¼	1,269,200	71,900	30,800-43,100
	6 ½	3 ½	1,110,200	60,200	26,900-36,100
	6 1/2	3 3/4	939,400	47,500	22,700-28,500
	6 3/8	3 ¼	1,269,200	68,700	28,900-41,200
GPDS55[1]	7 ¼	3 ¾	1,475,100	89,600	39,500-53,800
	7 ¼	4 1/8	1,196,700	66,800	32,000-40,100
	7 1/8	3 3/4	1,475,100	89,300	39,300-53,600
	7	3 ½	1,645,900	89,900	36,100-53,900
	7	3 ¾	1,475,100	83,900	36,100-50,300
	7	4	1,292,500	74,200	34,200-44,500
GPDS65[1]	8 ¼	4 ¾	1,709,800	119,000	52,700-71,400
	8	4 15/16	1,538,600	102,000	46,900-61,200
	8	4 7/8	1,596,400	107,500	48,700-64,500

NOTES:

[1] Trademark of Grant Prideco. Performance data reprinted according to information provided by Grant Prideco.

[2] This Table is based on information supplied by the manufacturer during preparation of this document. The user should contact manufacturer to insure data is current.

Table B.47 — Grant Prideco HI TORQUE® Connection Information

Connection	OD (in)	ID (in)	Tensile yield (lb)	Torsional yield (ft-lb)	Make-upTorque (ft-lb)
2 3/8 HTSLH90[1]	3 3/8	1.815	269,400	10,600	4,200-6,300
	3 1/8	1 7/16	385,100	11,400	3,200-6,800
	3 1/8	1 3/4	291,200	9,400	3,200-5,600
	3 1/8	1.975	212,200	7,600	3,200-4,600
2-7/8 HTPAC[1]	3 ¼	1 ½	273,000	8,600	3,500-5,200
	3 3/16	1 ½	273,000	8,600	3,500-5,100
	3 1/8	1 ½	273,000	8,500	3,400-5,100
HT26[1]	3 5/8	1 ¼	455,100	15,300	6,100-9,200
	3 5/8	1 3/8	424,100	14,500	5,900-8,700
	3 5/8	1 ½	390,300	13,100	5,400-7,900
	3 ½	1 ½	390,300	12,100	4,800-7,300
	3 3/8	1 ¾	313,700	8,700	3,700-5,200
HT31[1]	4 ¼	1 ¾	584,100	23,400	9,500-14,000
	4 1/8	1 7/8	541,400	19,900	7,900-11,900
	4 1/8	2	495,700	18,900	7,900-11,300
	4 1/8	2 1/8	447,100	16,600	7,200-10,000
	4 1/8	2 5/32	434,500	16,000	7,000-9,600
	4	2 5/32	434,500	14,900	6,300-8,900
HT38[1]	5	2 5/16	764,000	36,200	14,500-21,700
	5	2 7/16	708,100	33,000	13,400-19,800
	5	2 9/16	649,200	29,600	12,300-17,700
	4 15/16	2 9/16	649,200	29,500	12,300-17,700
	4 7/8	2 7/16	708,100	32,100	12,900-19,300
	4 7/8	2 9/16	649,200	29,400	12,200-17,700
	4 ¾	2 7/16	708,100	28,400	10,700-17,000
	4 ¾	2 9/16	649,200	26,900	10,700-16,100
	4 ¾	2 11/16	587,300	25,300	10,700-15,200
HT40[1]	5 ¼	2 13/16	711,600	31,900	14,200-19,200
	5 1/8	2 9/16	838,300	35,200	14,100-21,100
	5 1/8	2 11/16	776,400	33,600	14,100-20,100
	5	2 7/16	897,200	32,600	11,600-19,600
	5	2 9/16	838,300	31,100	11,600-18,600
HT46[1]	6 ¼	3	1,048,400	57,700	23,900-34,600
	6 ¼	3 ¼	901,200	47,600	20,500-285,00
HT50[1]	6 5/8	3	1,416,500	88,800	34,900-53,300
	6 5/8	3 1/4	1,269,200	78,000	31,200-46,800
	6 5/8	3 ½	1,110,200	66,200	27,200-39,700
	6 5/8	3 3/4	939,400	53,300	23,000-32,000
	6 ½	3 ½	1,110,200	66,000	27,100-39,600

Connection	OD (in)	ID (in)	Tensile yield (lb)	Torsional yield (ft-lb)	Make-upTorque (ft-lb)
HT55[1]	7 3/8	4	1,265,800	78,000	34,100-46,800
	7 1/4	3	1,925,500	120,000	45,400-72,000
	7 ¼	3 ¼	1,778,300	115,100	45,400-69,000
	7 1/4	3 1/2	1,619,200	106,500	43,600-63,900
	7 ¼	3 ¾	1,448,400	92,700	38,900-55,600
	7 ¼	3 7/8	1,358,600	85,400	36,500-51,200
	7 ¼	4	1,265,800	77,800	33,900-46,700
	7 1/8	3 ¼	1,778,300	107,000	40,600-64,200
	7 1/8	3 7/8	1,358,600	85,100	36,300-51,100
	7 1/8	4	1,265,800	77,500	33,800-46,500
	7	3	1,925,500	104,100	35,900-62,500
	7	3 ¼	1,778,300	99,200	35,900-59,500
	7	3 1/2	1,619,200	93,800	35,900-56,300
	7	3 ¾	1,448,400	87,700	35,900-52,600
	7	4	1,265,800	77,200	33,600-46,300
HT65[1]	8	5	1,448,400	99,700	44,200-59,800

NOTES:

1 Trademark of Grant Prideco. Performance data reprinted according to information provided by Grant Prideco.

2 This Table is based on information supplied by the manufacturer during preparation of this document. The user should contact manufacturer to insure data is current

Table B.48 — Grant Prideco eXtreme™ Torque (XT™) Connection Information

Connection	OD (in)	ID (in)	Tensile yield (lb)	Torsional yield (ft-lb)	Make-upTorque (ft-lb)
XT24[1]	3 1/8	1 ½	261,500	9,500	3,500-5,700
XT26[1]	3 1/2	1 1/4	432,200	16,400	5,600-9,900
	3 ½	1 ½	367,400	14,800	5,300-8,900
	3 1/2	1 3/4	290,900	11,500	4,200-6,900
XT29[1]	3 ¾	2	351,700	15,100	5,500-9,000
XT31[1]	4 1/8	1 7/8	509,400	23,400	8,500-14,000
	4 1/8	2	463,700	21,100	7,700-12,700
	4	2	463,700	20,400	7,300-12,200
	4	2 1/8	415,100	18,600	6,900-11,200
	4	2 5/32	402,500	18,000	6,700-10,800
	4	2 3/8	309,100	13,200	5,100-7,900
XT38[1]	4 7/8	2 9/16	599,600	31,500	11,600-18,900
	4 13/16	2 9/16	599,600	31,400	11,600-18,800
	4 3/4	2 7/16	658,500	34,200	12,300-20,500
	4 ¾	2 9/16	599,600	31,300	11,500-18,800
	4 3/4	2 11/16	537,800	27,700	10,300-16,600
XT39[1]	5	2 9/16	729,700	40,800	14,700-24,500
	5	2 11/16	667,800	37,000	13,400-22,200
	5	2 13/16	603,000	33,100	12,100-19,800
	5	2 7/8	569,500	31,000	11,400-18,600
	4 15/16	2 9/16	729,700	39,000	13,600-23,400
	4 15/16	2 13/16	603,000	33,000	12,100-19,800
	4 29/32	2 ¾	635,800	35,000	12,700-21,000
	4 29/32	2 13/16	603,000	33,000	12,100-19,800
	4 7/8	2 9/16	729,700	37,000	12,400-22,200
	4 7/8	2 11/16	667,800	35,300	12,400-21,200
	4 7/8	2 ¾	635,800	34,400	12,400-20,700
	4 7/8	2 13/16	603,000	32,900	12,100-19,800
XT40[1]	5 ¼	2 11/16	816,400	48,100	17,200-28,800
	5 ¼	2 13/16	751,600	44,000	15,800-26,400
	5 1/4	3	648,900	37,400	13,600-22,400
XT46[1]	6 ¼	3 ¼	1,069,300	70,200	26,000-42,100
XT50[1]	6 5/8	3 1/2	1,256,300	90,700	32,600-54,400
	6 5/8	3 ¾	1,085,500	77,300	28,100-46,400
	6 ½	2 ¾	1,698,100	102,600	31,000-61,600
	6 ½	3 ½	1,256,300	88,000	31,000-52,800
	6 ½	3 ¾	1,085,500	77,000	28,000-46,200
	6 3/8	3 ½	1,256,300	81,200	26,900-48,700
	6 3/8	3 3/4	1,085,500	75,200	26,900-45,100

Connection	OD (in)	ID (in)	Tensile yield (lb)	Torsional yield (ft-lb)	Make-upTorque (ft-lb)
XT54[1]	6 ¾	4	1,155,100	86,600	31,100-52,000
	6 5/8	4	1,155,100	83,200	29,000-49,900
XT55[1]	7 3/8	3 5/8	1,542,100	120,300	43,100-72,200
	7	4	1,272,600	96,900	35,100-58,100
XT57[1]	7 3/8	3 ¼	1,915,600	150,000	51,300-90,000
	7 ¼	3 3/16	1,953,500	142,800	46,200-85,700
	7 1/4	3 1/4	1,915,600	141,500	46,200-84,900
	7 ¼	3 ½	1,756,500	135,800	46,200-81,500
	7 1/4	3 3/4	1,585,700	127,100	44,800-76,300
	7 ¼	4	1,403,100	111,600	39,600-66,900
	7 ¼	4 ¼	1,208,700	94,800	34,100-56,900
	7 1/8	3 ¼	1,915,600	133,200	41,200-79,900
	7 1/8	4 ¼	1,208,700	94,600	33,900-56,800
	7	3 ½	1,756,500	119,400	36,400-71,600
	7	3 3/4	1,585,700	113,100	36,400-67,900
	7	4	1,403,100	106,200	36,400-63,700
	7	4 ¼	1,208,700	94,300	33,800-56,600
XT65[1]	8	5	1,543,700	135,300	49,000-81,200
XT69[1]	8 ½	5 ¼	1,770,100	167,200	59,400-100,300

NOTES:

[1] Trademark of Grant Prideco. Performance data reprinted according to information provided by Grant Prideco.

[2] This Table is based on information supplied by the manufacturer during preparation of this document. The user should contact manufacturer to insure data is current.

Table B.49 — Grant Prideco eXtreme™ Torque Metal Seal (XT-M™) Connection Information

Connection	OD (in)	ID (in)	Tensile yield (lb)	Torsional yield (ft-lb)	Make-upTorque (ft-lb)
XT-M26[1]	3 3/8	1 3/4	290,900	9,700	4,850-5,800
XT-M31[1]	4	2	463,700	17,900	8,950-10,800
	4	2 1/8	415,100	16,100	8,050-9,700
	3 7/8	2 1/8	415,100	14,200	7,100-8,500
XT-M34[1]	4 ¼	2 9/16	399,200	15,900	7,950-9,500
XT-M38[1]	4 ¾	2 9/16	599,600	27,900	13,950-16,700
	4 ¾	2 11/16	537,800	24,200	12,100-14,500
XT-M39[1]	5	2 7/16	788,600	38,800	19,400-23,300
	5	2 9/16	729,700	37,200	18,600-22,300
	5	2 13/16	603,000	29,400	14,700-17,700
	4 7/8	2 11/16	667,800	31,500	15,750-18,900
XT-M40[1]	5 ¼	2 11/16	816,400	43,600	21,800-26,200
XT-M43[1]	5 ¼	3	764,500	39,200	19,600-23,500
	5 ¼	3 ¼	617,300	32,500	16,250-19,500
XT-M46[1]	6 ¼	3	1,216,600	75,200	37,600-45,100
XT-M50[1]	6 5/8	3 ½	1,256,300	83,900	41,950-50,300
	6 ½	3 ¾	1,085,500	70,200	35,100-42,100
	6 ¼	3 ½	1,256,300	67,800	33,900-40,700
	6 ¼	3 5/8	1,172,400	64,900	32,450-38,900
XT-M57[1]	7 ¼	4 ¼	1,208,700	86,500	43,250-5,1900
	7	4 1/8	1,307,400	94,200	47,100-56,500
	7	4 1/4	1,208,700	86,000	43,000-51,600
XT-M65[1]	8	5	1,543,700	124,400	62,200-74,600
XT-M69[1]	8 ½	5 ¼	1,770,100	154,800	77,400-92,900

NOTES:

[1] Trademark of Grant Prideco. Performance data reprinted according to information provided by Grant Prideco.

[2] This Table is based on information supplied by the manufacturer during preparation of this document. The user should contact manufacturer to insure data is current.

Table B.50 — Omsco Tuff-Torque™ Connection Information

Connection	OD (in)	ID (in)	Tensile yield (lb)	Torsional yield (ft-lb)	Make-upTorque (ft-lb)
2 7/8 TT PAC[1]	3 1/8	1 1/2	273,024	8,963	5,378
TT26[1]	3 3/8	1 5/8	353,442	9,729	5,837
	3 3/8	1 3/4	313,681	9,002	5,401
	3 1/2	1 1/2	390,257	12,353	7,412
	3 5/8	1 1/2	390,257	12,472	7,483
TT31[1]	4 1/8	2	495,726	18,459	11,076
TT38[1]	4 3/4	2 9/16	649,158	26,540	15,924
	5	2 7/16	708,063	30,874	18,524
	5	2 9/16	649,158	27,461	16,477
TT40[1]	5 1/4	2 9/16	838,256	38,425	23,055
	5 1/4	2 11/16	776,406	34,638	20,783
	5 1/4	2 13/16	711,611	30,655	18,393
TT46[1]	6 1/4	3	1,048,426	55,374	33,224
	6 1/4	3 1/4	901,164	45,177	27,106
TT50[1]	6 3/8	3 1/2	1,109,920	63,060	37,836
	6 5/8	3	1,416,225	85,930	51,558
	6 5/8	3 1/4	1,268,963	74,980	44,988
	6 5/8	3 1/2	1,109,920	63,060	37,836
5 1/2 TT[1]	7	3 3/4	1,448,406	90,129	54,077
	7	4	1,265,801	76,318	45,791
6 5/8 TT[1]	8	5	1,448,416	98,861	59,317
	8 1/4	4 3/4	1,678,145	120,998	72,599

NOTES:

[1] Trademark of Omsco. Performance data reprinted according to information provided by Omsco.

[2] This Table is based on information supplied by the manufacturer during preparation of this document. The user should contact manufacturer to insure data is current

Table B.51 — Summary performance properties for alternative drill pipe sizes and weights (New Class)

Label 1	Label 2	Label 3	Torsion yield[5]			Tension			Burst			Collapse		
Nominal Size	Nominal Mass lb/ft	Nominal Wall (t) in	S-135[2] ft-lb	Z-140[3] ft-lb	V-150[4] ft-lb	S-135[2] lb	Z-140[3] lb	V-150[4] lb	S-135[2] psi	Z-140[3] psi	V-150[4] psi	S-135[2] psi	Z-140[3] psi	V-150[4] psi
5		0.750	121,100	125,600	134,500	1,351,900	1,401,900	1,502,100	35,438	36,750	39,375	34,425	35,700	38,250
5 1/2		0.750	152,700	158,400	169,700	1,510,900	1,566,900	1,678,800	32,216	33,409	35,795	31,798	32,975	35,331
5 1/2		0.813	159,900	165,800	177,600	1,616,100	1,676,000	1,795,700	34,922	36,215	38,802	34,011	35,271	37,790
5 7/8	23.40	0.361	105,500	109,400	117,200	844,200	875,500	938,000	14,517	15,054	16,130	10,825	11,023	11,376
5 7/8	26.30	0.415	117,900	122,300	131,000	961,000	996,600	1,067,800	16,690	17,306	18,543	14,890	15,266	15,976
5 7/8		0.500	135,900	141,000	151,000	1,139,800	1,182,000	1,266,500	20,106	20,851	22,340	21,023	21,802	23,216
5 7/8		0.625	159,200	165,100	176,900	1,391,600	1,443,200	1,546,300	25,133	26,064	27,926	25,668	26,618	28,520
5 7/8		0.750	179,000	185,600	198,900	1,630,200	1,690,600	1,811,300	30,160	31,277	33,511	30,068	31,182	33,409
5 7/8		0.813	187,700	194,700	208,600	1,745,400	1,810,100	1,939,300	32,693	33,904	36,326	32,193	33,385	35,770
6 5/8		0.500	178,000	184,600	197,800	1,298,900	1,347,000	1,443,200	17,830	18,491	19,811	17,031	17,497	18,395
6 5/8		0.522	184,000	190,800	204,400	1,351,100	1,401,200	1,501,300	18,615	19,304	20,683	18,500	19,030	20,057
6 5/8		0.625	210,000	217,800	233,400	1,590,400	1,649,300	1,767,100	22,288	23,113	24,764	23,069	23,923	25,632
6 5/8		0.640	213,600	221,500	237,300	1,624,500	1,684,700	1,805,000	22,823	23,668	25,358	23,563	24,436	26,181
6 5/8		0.750	237,900	246,700	264,300	1,868,800	1,938,000	2,076,400	26,745	27,736	29,717	27,106	28,110	30,117
6 5/8		0.813	250,500	259,700	278,300	2,004,000	2,078,200	2,226,700	28,992	30,066	32,213	29,068	30,144	32,297
6 5/8		1.000	250,500	259,700	278,300	2,004,000	2,078,200	2,226,700	28,992	30,066	32,213	29,068	30,144	32,297

NOTES:

1 Where mass is not given, use Label 3 (wall thickness) for designation.

2 Values for S-135 based on material minimum yield strength of 135,000 psi and nominal dimensions

3 Values for Z-140 based on material minimum yield strength of 140,000 psi and nominal dimensions

4 Values for V-150 based on material minimum yield strength of 150,000 psi and nominal dimensions

5 Torsional Yield based on shear strength equal to 57.7 percent of minimum yield strength.

Table B.52 — Summary performance properties for alternative drill pipe sizes and weights (Premium Class)

Label 1	Label 2	Label 3	Torsion Yield[5]			Tension			Burst			Collapse		
Nominal Size	Nominal Mass lb/ft	Nominal Wall (t) in	S-135[2] ft-lb	Z-140[3] ft-lb	V-150[4] ft-lb	S-135[2] lb	Z-140[3] lb	V-150[4] lb	S-135[2] psi	Z-140[3] psi	V-150[4] psi	S-135[2] psi	Z-140[3] psi	V-150[4] psi
5		0.750	91,600	95,000	101,800	1,043,300	1,082,000	1,159,200	34,468	35,745	38,298	30,068	31,182	33,409
5 1/2		0.750	116,500	120,800	129,400	1,170,600	1,213,900	1,300,600	31,154	32,308	34,615	27,559	28,580	30,621
5 1/2		0.813	121,100	125,600	134,600	1,248,000	1,294,300	1,386,700	33,935	35,192	37,706	29,670	30,769	32,967
5 7/8	23.40	0.361	83,000	86,100	92,300	666,500	691,200	740,600	13,607	14,111	15,119	6,204	6,296	6,450
5 7/8	26.30	0.415	92,500	96,000	102,800	757,100	785,200	841,200	15,702	16,283	17,446	9,368	9,503	9,728
5 7/8		0.500	106,100	110,000	117,900	894,900	928,000	994,300	19,031	19,736	21,145	14,824	15,195	15,899
5 7/8		0.625	123,200	127,700	136,900	1,086,800	1,127,000	1,207,500	24,000	24,889	26,667	21,867	22,677	24,296
5 7/8		0.750	137,100	142,200	152,300	1,266,000	1,312,900	1,406,600	29,058	30,135	32,287	25,931	26,891	28,812
5 7/8		0.813	143,000	148,300	158,900	1,351,500	1,401,500	1,501,600	31,642	32,814	35,158	27,934	28,969	31,038
6 5/8		0.500	139,400	144,600	154,900	1,022,100	1,060,000	1,135,700	16,809	17,432	18,677	11,183	11,397	11,781
6 5/8		0.522	143,900	149,300	159,900	1,062,400	1,101,800	1,180,500	17,573	18,224	19,526	12,435	12,702	13,197
6 5/8		0.625	163,300	169,400	181,500	1,245,800	1,292,000	1,384,300	21,176	21,961	23,529	18,341	18,863	19,876
6 5/8		0.640	166,000	172,100	184,400	1,271,800	1,318,900	1,413,100	21,705	22,509	24,117	19,207	19,767	20,856
6 5/8		0.750	183,500	190,300	203,900	1,456,800	1,510,800	1,618,700	25,613	26,561	28,458	23,183	24,042	25,759
6 5/8		0.813	192,300	199,400	213,700	1,558,400	1,616,100	1,731,500	27,875	28,908	30,972	24,997	25,923	27,775
6 5/8		1.000	192,300	199,400	213,700	1,558,400	1,616,100	1,731,500	27,875	28,908	30,972	24,997	25,923	27,775

NOTES:

1 Where mass is not given, use Label 3 (wall thickness) for designation.

2 Values for S-135 based on material minimum yield strength of 135,000 psi and nominal dimensions

3 Values for Z-140 based on material minimum yield strength of 140,000 psi and nominal dimensions

4 Values for V-150 based on material minimum yield strength of 150,000 psi and nominal dimensions

5 Torsional Yield based on shear strength equal to 57.7 percent of minimum yield strength.

Table B.53 — Conventional Heavy Weight Drill Pipe Properties (New)

Assembly				Tool joint					Tube			
Nominal Size	Nominal ID	Approx Mass lb/ft	RSC type	Nominal OD in	Nominal ID in	Tensile Yield [2] lb	Torsional Yield [2] ft-lb	Make-up Torque [3] ft-lb	Nominal OD in	Nominal ID in	Tensile Yield [4] lb	Torsional Yield [5] ft-lb
2 7/8 [1]	1 1/2	17.26	NC26	3 3/8	1 1/2	357,700	6,300	3,800	2 7/8	1 1/2	519,714	22,851
3 1/2	2 1/16	25.65	NC38	4 3/4	2 1/8	842,400	19,200	11,500	3 1/2	2 1/16	345,407	19,579
3 1/2	2 1/4	23.48	NC38	4 3/4	2 1/4	790,900	19,200	11,500	3 1/2	2 1/4	310,478	18,461
3 1/2	2 1/4	23.96	HT38	4 7/8	2 1/4	790,900	34,200	20,500	3 1/2	2 1/4	310,478	18,461
3 1/2	2 1/4	23.96	XT39	4 7/8	2 1/4	871,400	40,700	24,400	3 1/2	2 1/4	310,478	18,461
4	2 9/16	29.92	NC40	5 1/4	2 9/16	838,300	27,800	14,600	4	2 9/16	407,503	27,636
4	2 9/16	28.40	XT39	4 7/8	2 9/16	729,700	37,000	22,200	4	2 9/16	407,503	27,636
4 1/2	2 3/4	41.45	NC46	6 1/4	2 13/16	1,151,100	43,600	22,500	4 1/2	2 3/4	548,062	40,718
5	3	50.38	NC50	6 5/8	3	1,416,200	57,800	30,000	5	3	691,152	56,496
5	3	50.38	HT50	6 5/8	3	1,416,200	88,800	53,300	5	3	691,152	56,496
5 1/2	3 1/4	61.63	5 1/2 FH	7 1/4	3 1/4	1,778,300	78,700	41,200	5 1/2	3 1/4	850,441	75,859
5 1/2	3 1/4	61.63	HT55	7 1/4	3 1/4	1,778,300	115,100	69,000	5 1/2	3 1/4	850,441	75,859
5 7/8	4	57.42	XT57	7	4	1,403,100	106,200	63,700	5 7/8	4	799,819	82,669
6 5/8	4 1/2	71.43	6 5/8 FH	8	4 1/2	1,896,100	87,900	50,500	6 5/8	4 1/2	1,021,204	118,849

NOTES:

1 2 7/8" HWDP is manufactured integral only of 110,000 psi material (non-welded).

2 Tensile yield and tool joint torsional yield based on minimum material yield strength of

110,000 for tool joints for Nominal Size 2 7/8.

120,000 for tool joints for Nominal Size 3 1/2 to 6 5/8.

Pin stress relief groove is ignored.

3 Make-up torque based on 72,000 psi stress in the weaker of the two members (pin or box) or T3, whichever is less.

4 Tensile yield of tube based on 55,000 psi minimum material yield strength and nominal dimensions.

5 Torsional yield of tube based on shear strength equal to 57.7 percent of 55,000 psi minimum material yield strength and nominal dimensions.

Assembly				Tool joint					Tube			
Nominal Size	Nominal ID	Approx Mass lb/ft	RSC type	Nominal OD in	Nominal ID in	Tensile Yield [2] lb	Torsional Yield [2] ft-lb	Make-up Torque [3] ft-lb	Nominal OD in	Nominal ID in	Tensile Yield [4] lb	Torsional Yield [5] ft-lb

6 All dimensions are nominal. They should be considered reference only. Tolerances are at manufacturer's option.

7 The configuration of Heavy Weight Drill Pipe is shown on Figure 3. Other configurations, nominal dimensions and material properties are available and will have other properties than shown above.

Notes:

1. Bevel diameter same as for drill pipe
2. Nominal tool joint ID taken as largest used for common heavy weight
3. Tensile yield and tool joint torsional yield based on minimum material yield strength of
   - 110 000 psi for tool joints for Label 1 3 1/2 to 5
   - 100 000 psi for tool joints for Label 1 5 1/2 to 6 5/8
   - Pin stress relief groove is ignored
4. Make-up torque based on 62 500 psi target stress level.
5. Tensile yield of tube based on 55 000 psi minimum material yield strength and nominal dimensions.
6. Torsional yield of tube based on shear strength equal to 57.7 percent of 55 000 psi minimum material yield strength and nominal dimensions.
7. All dimensions are nominal. They should be considered reference only. Tolerances are at manufacturer's option.
8. The configuration of Heavy Weight is shown on Figure 3. Other configurations, nominal dimensions and material properties are available and will have other properties than shown above.

Annex C
(normative)

Figures in SI units

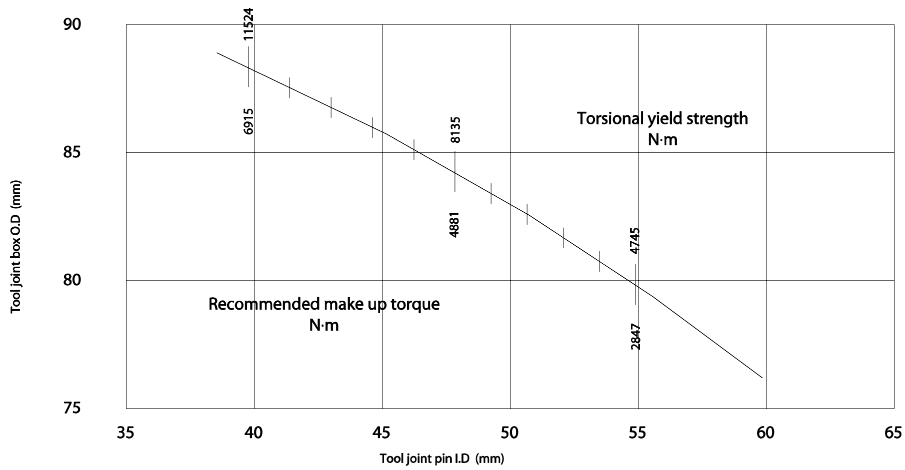


Figure C.1 — NC26 Torsional yield and make-up

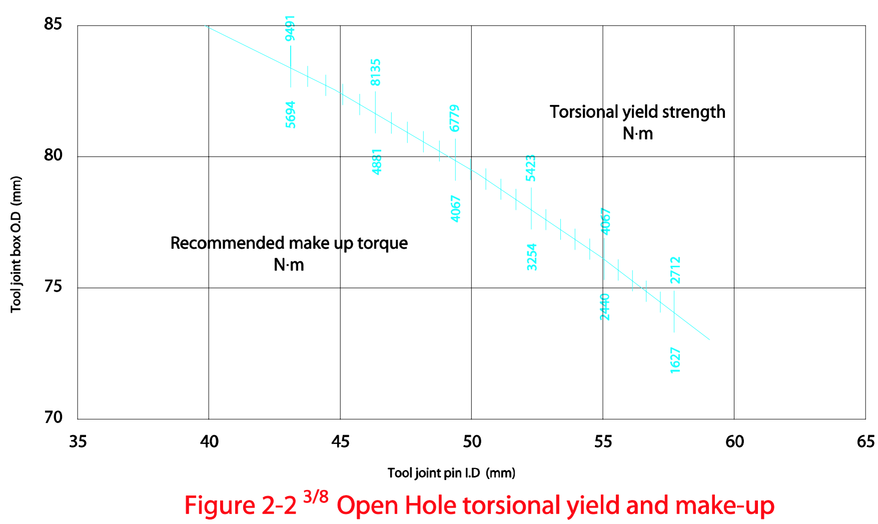


Figure C.2 — 2 3/8 Open hole torsional yield and make-up

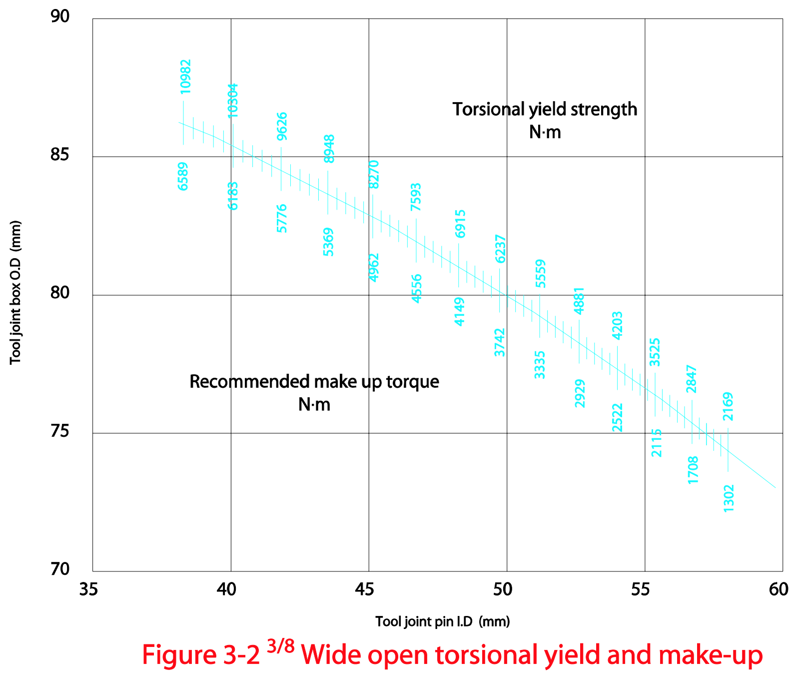


Figure C.3 — 2 3/8 Wide open torsional yield and make-up

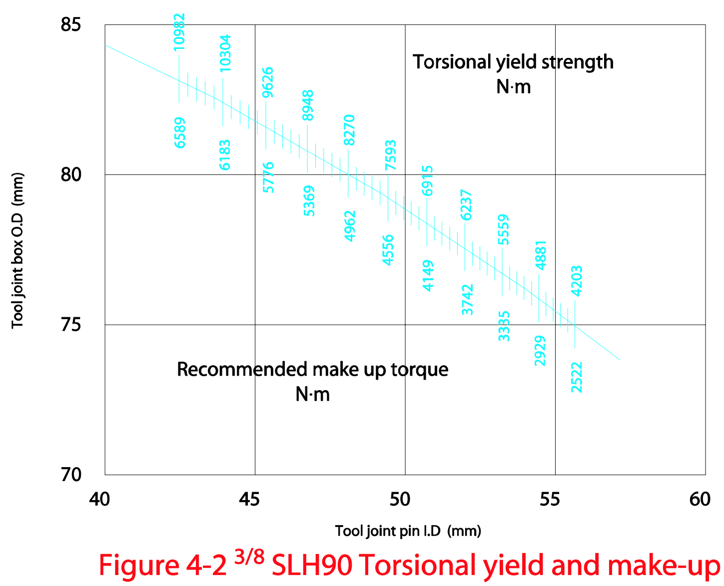


Figure C.4 — 2 3/8 SLH90 Torsional yield and make-up

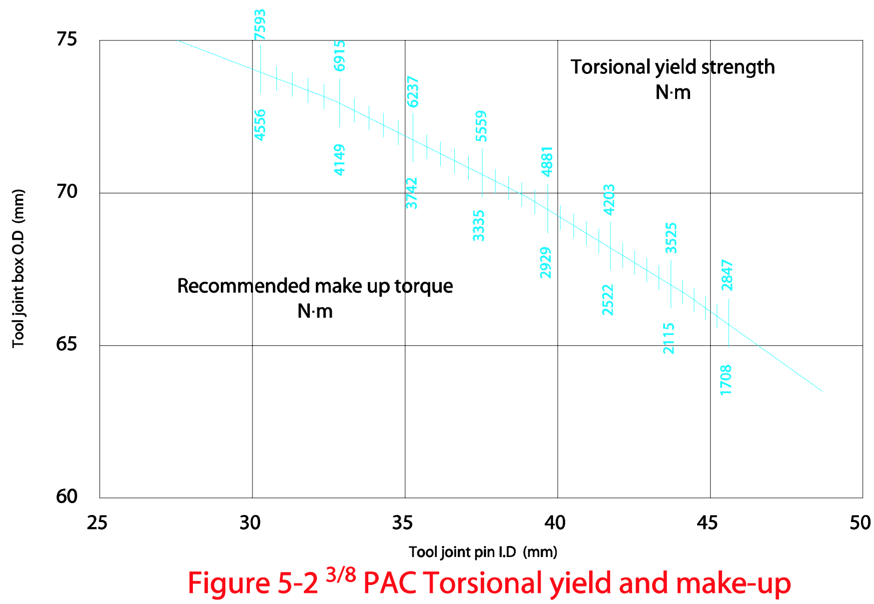


Figure C.5 — 2 3/8 PAC Torsional yield and make-up

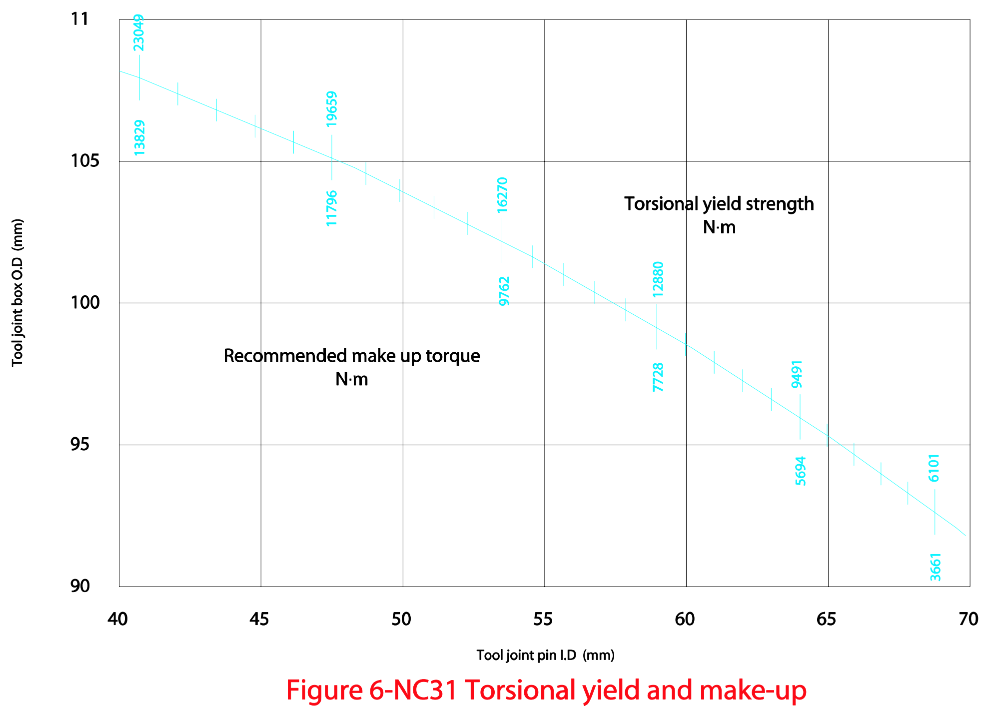


Figure C.6 — NC31 Torsional yield and make-up

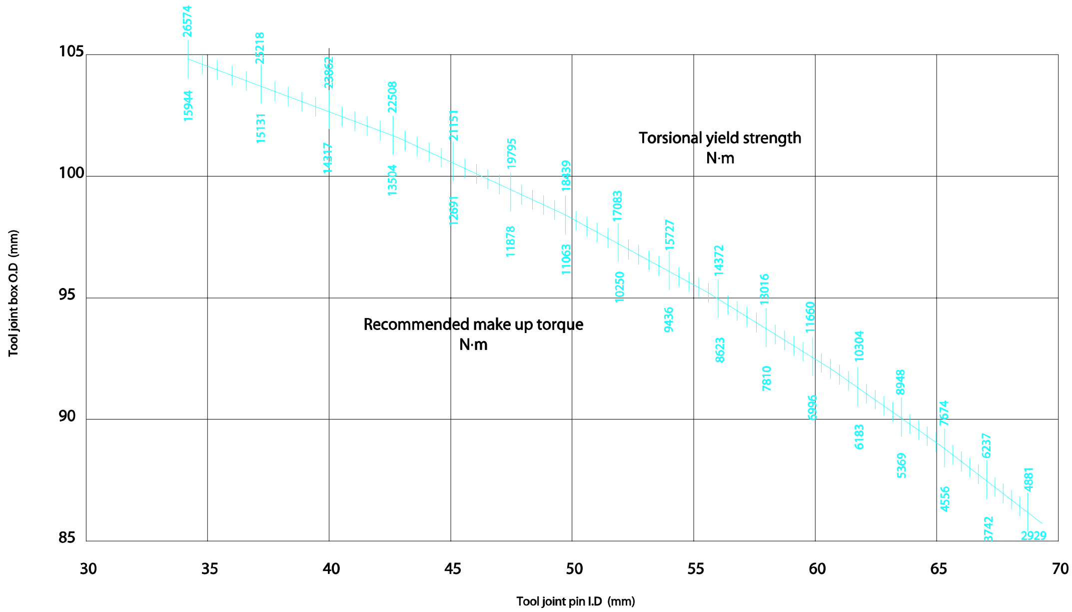


Figure 7-2 7/8 SLH90 Torsional yield and make-up

Figure C.7 — 2 7/8 SLH90 Torsional yield and make-up

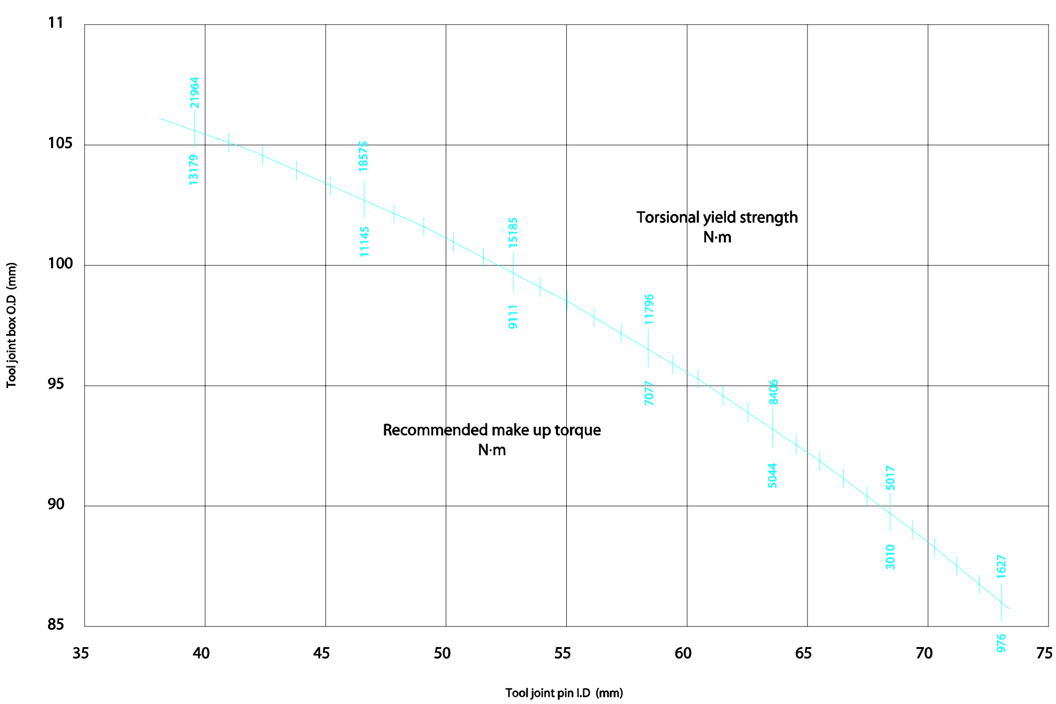


Figure 8-2 7/8 Wide open torsional yield and make-up

Figure C.8 — 2 7/8 Wide open torsional yield and make-up

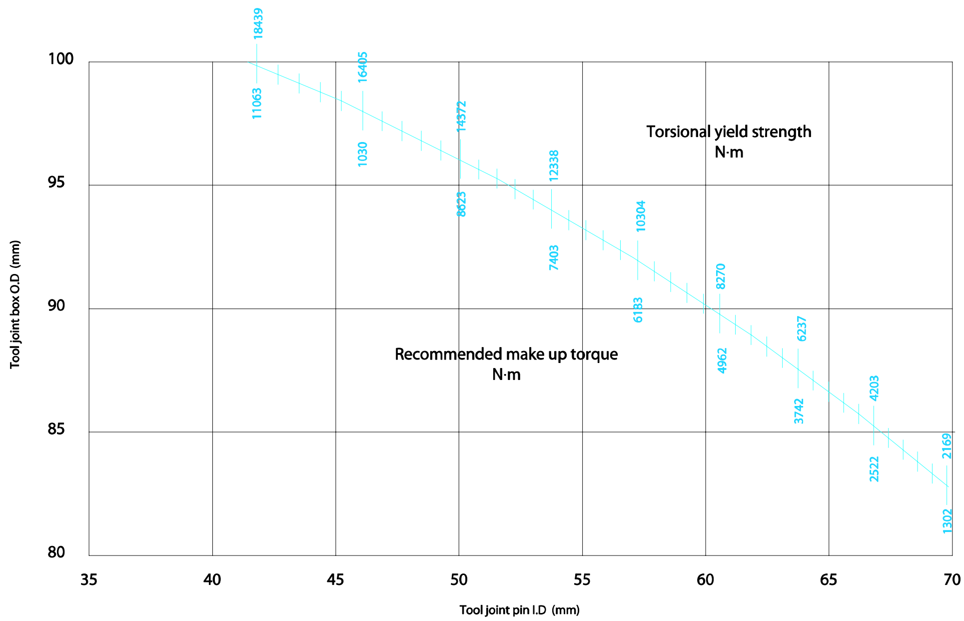


Figure 9-2 7/8 Open hole torsional yield and make-up

Figure C.9 — 2 7/8 Open hole torsional yield and make-up

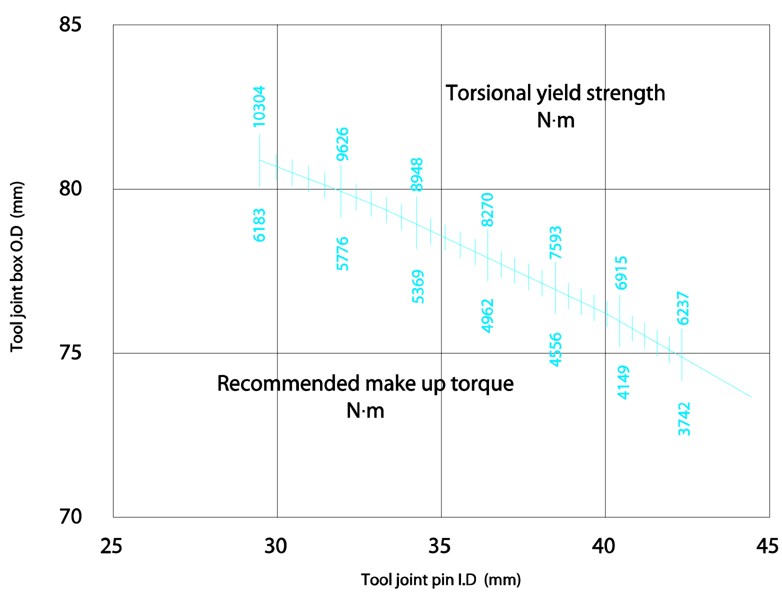


Figure 10-2 7/8 PAC Torsional yield and make-up

Figure C.10 — 2 7/8 PAC torsional yield and make-up

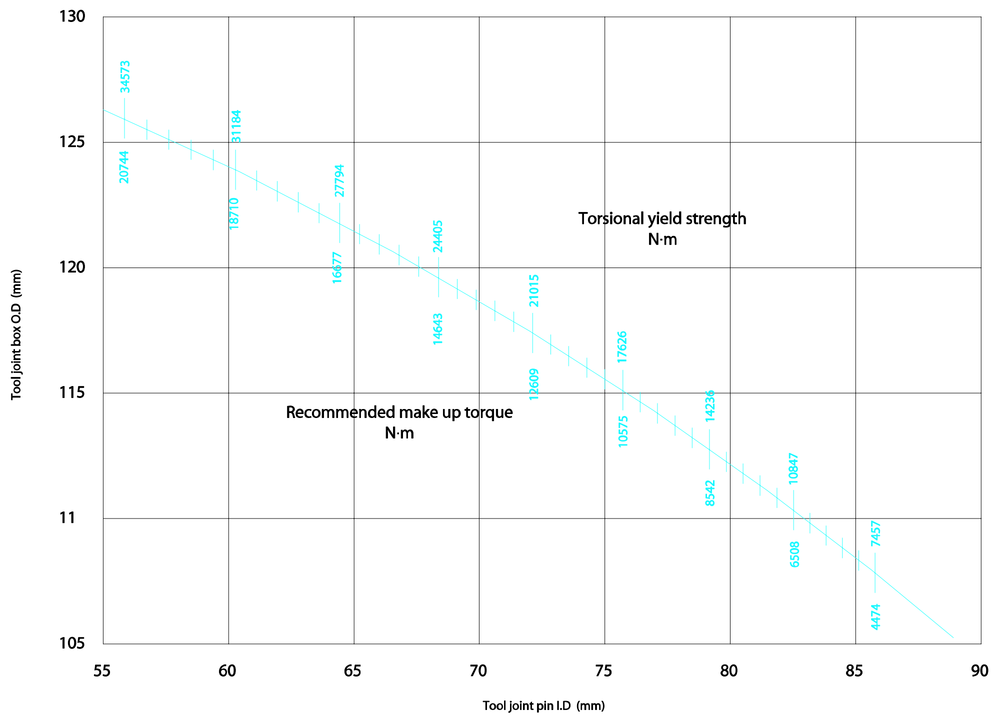


Figure 11-NC38 Torsional yield and make-up

Figure C.11 — NC38 torsional yield and make-up

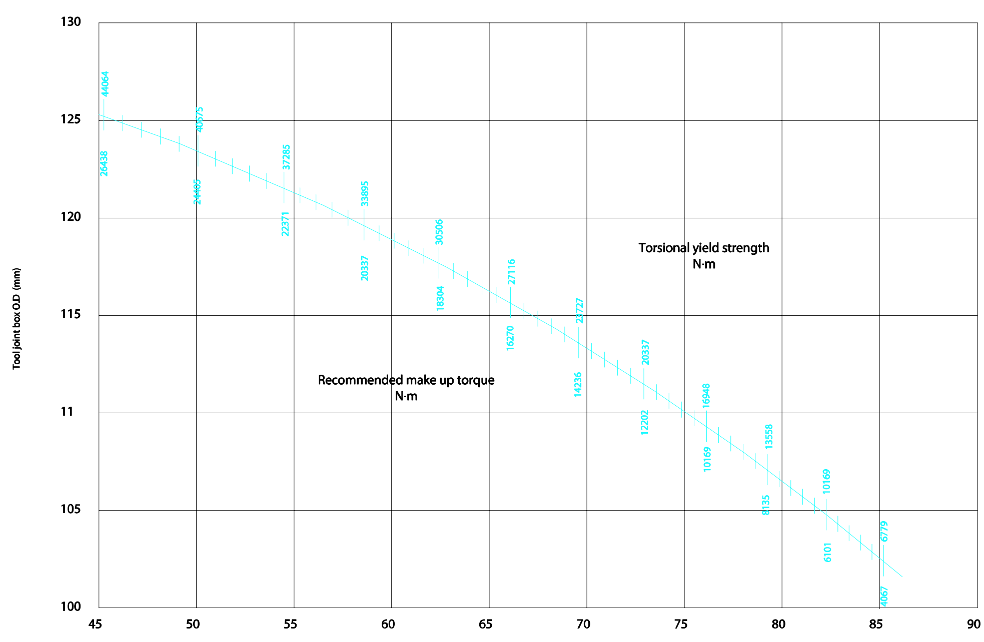


Figure 12-3 1/2 SLH90 Torsional yield and make-up

Figure C.12 — 3 ½ SLH90 torsional yield and make-up

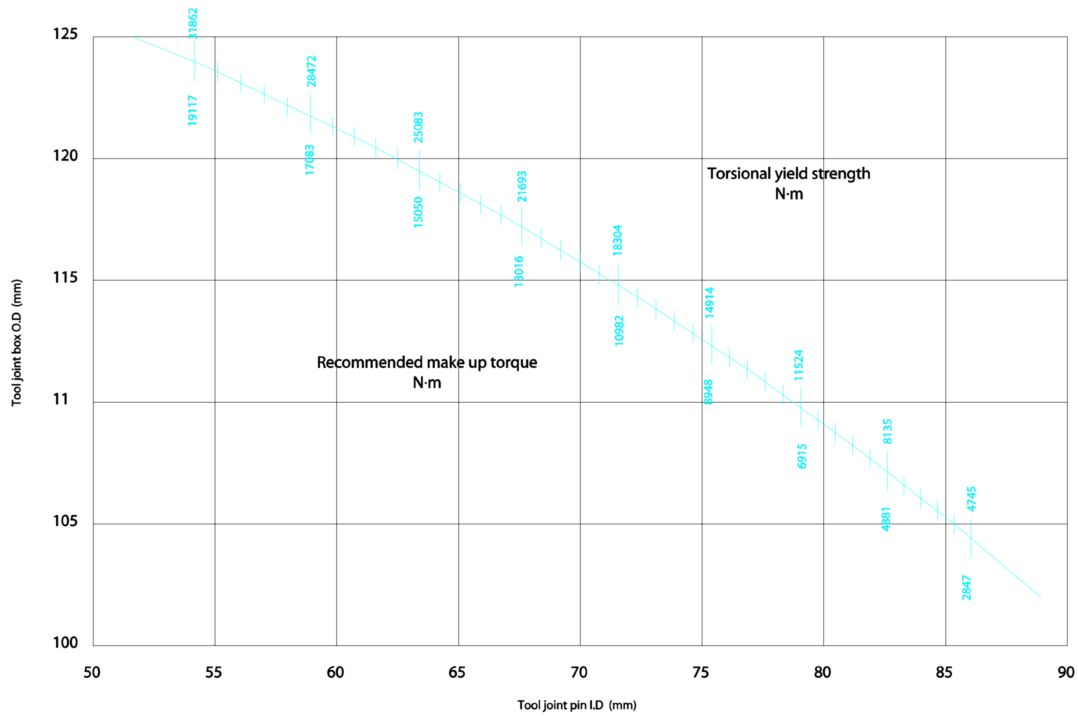


Figure 13-3 1/2 FH Torsional yield and make-up

Figure C.13 — 3 ½ FH torsional yield and make-up

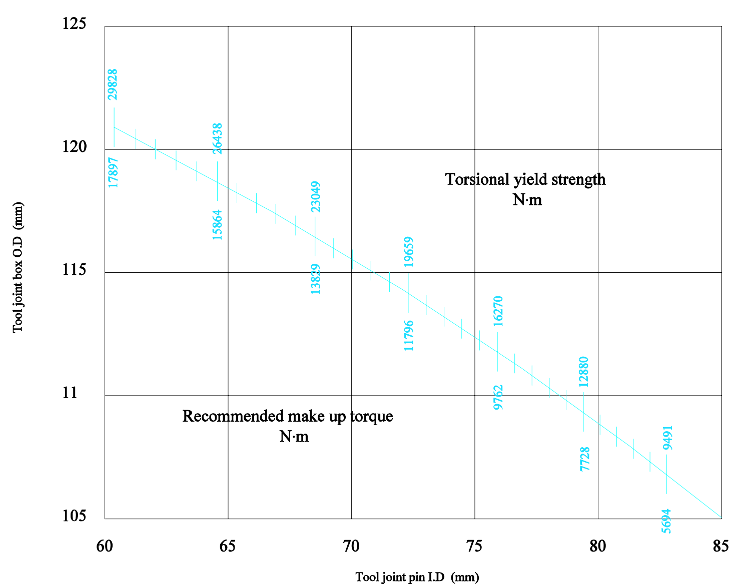


Figure 14-3 1/2 Open hole torsional yield and make-up

Figure C.14 — 3 ½ Open hole torsional yield and make-up

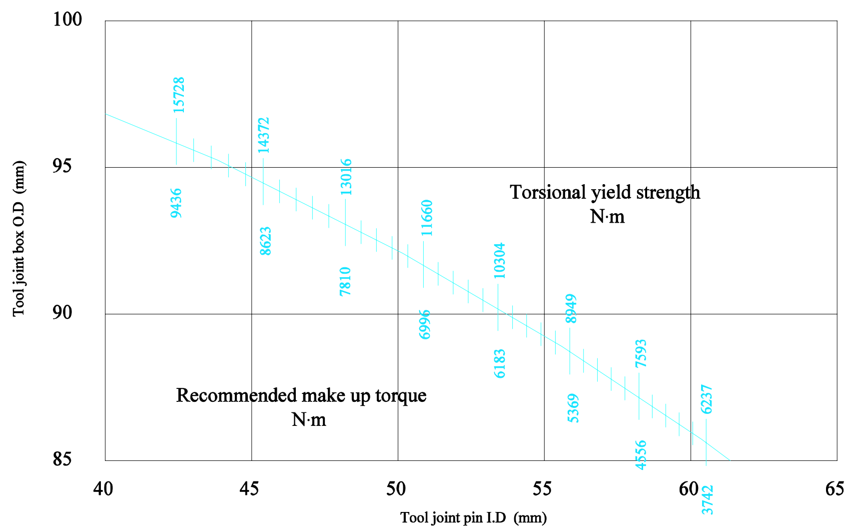


Figure 15-3 1/2 PAC Torsional yield and make-up

Figure C.15 — 3 ½ PAC torsional yield and make-up

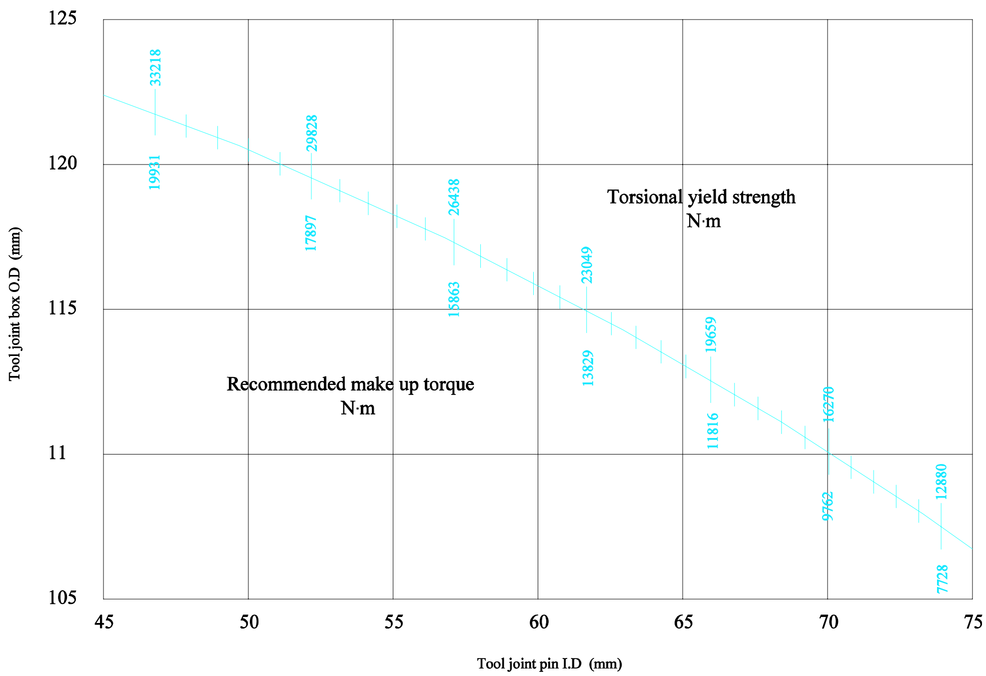


Figure 16-3 1/2 XH Torsional yield and make-up

Figure C.16 — 3 ½ XH torsional yield and make-up

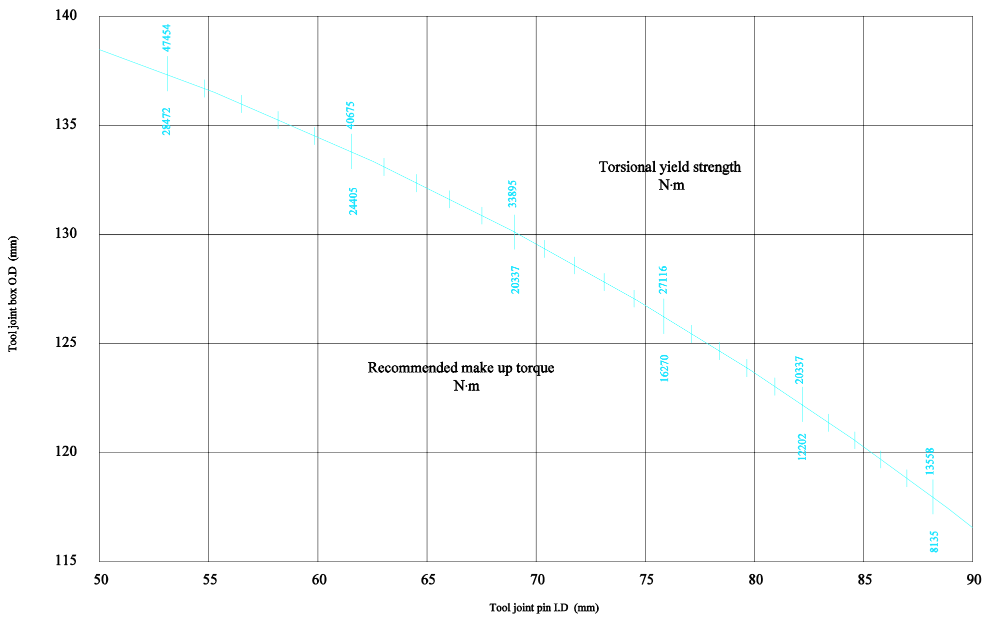


Figure 17-NC40 Torsional yield and make-up

Figure C.17 — NC40 torsional yield and make-up

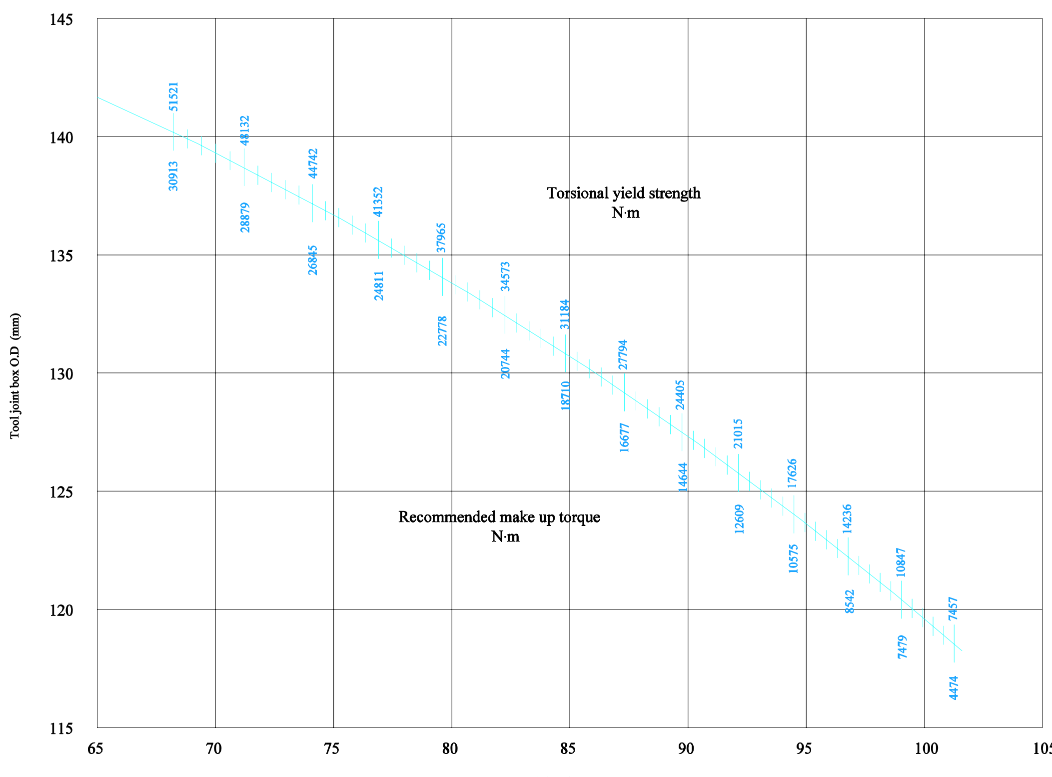


Figure 18-4-Inch H90 Torsional yield and make-up

Figure C.18 — 4-inch H90 torsional yield and make-up

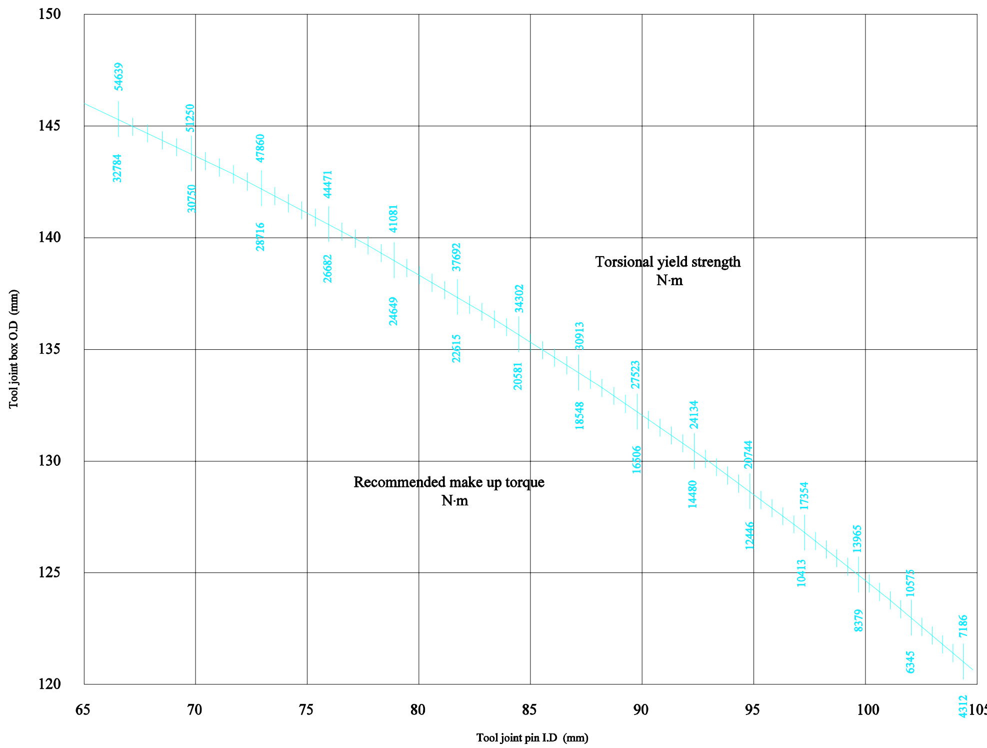


Figure 19-4-Inch open hole torsional yield and make-up

Figure C.19 — 4-inch open hole torsional yield and make-up

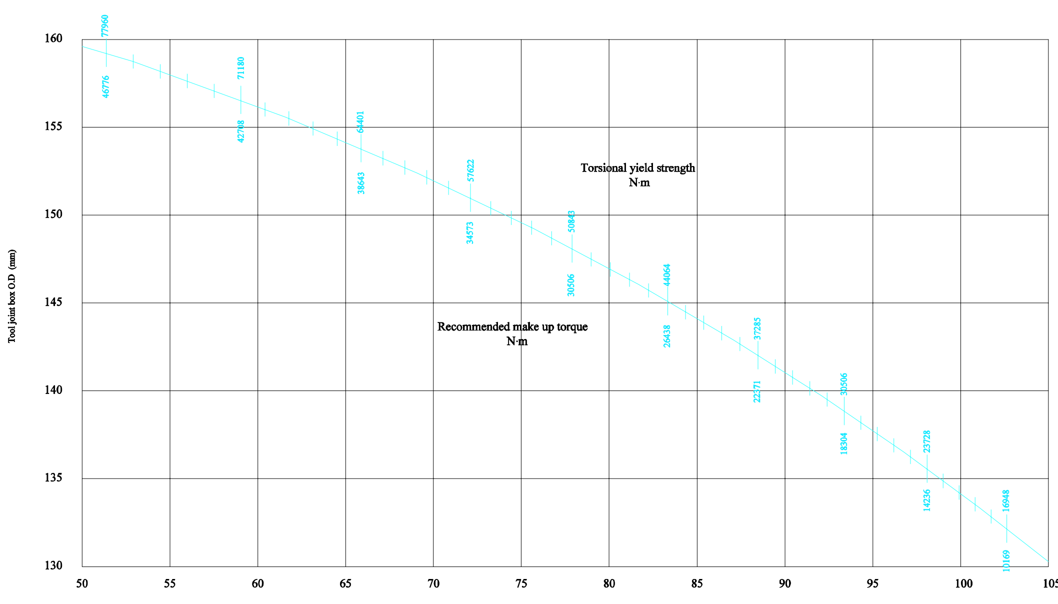


Figure 20-NC46 Torsional yield and make-up

Figure C.20 — NC46 torsional yield and make-up

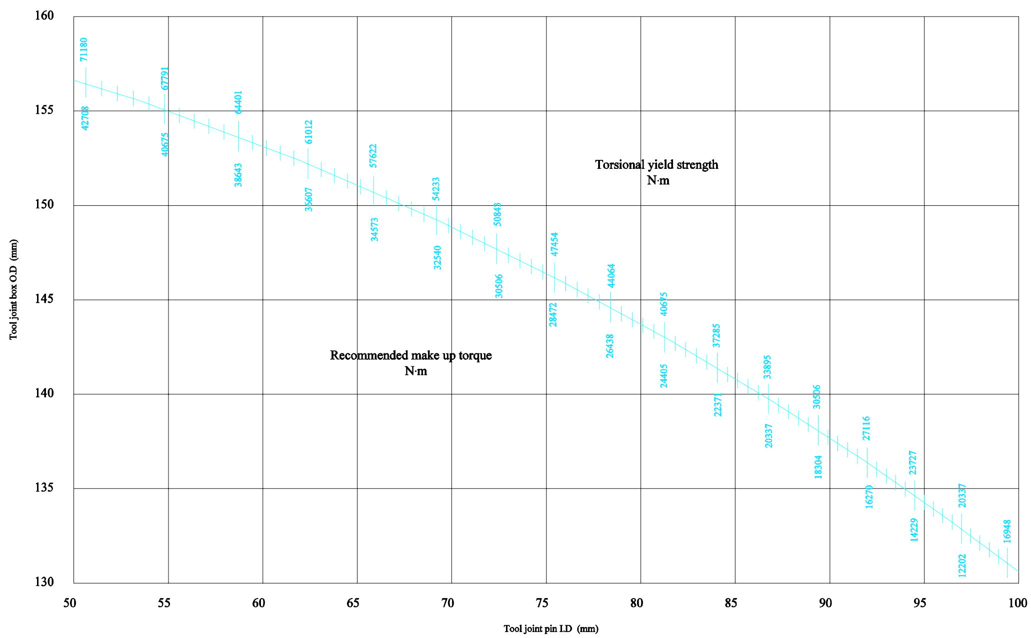


Figure 21-4 1/2 FH Torsional yield and make-up

Figure C.21 — 4 ½ FH torsional yield and make-up

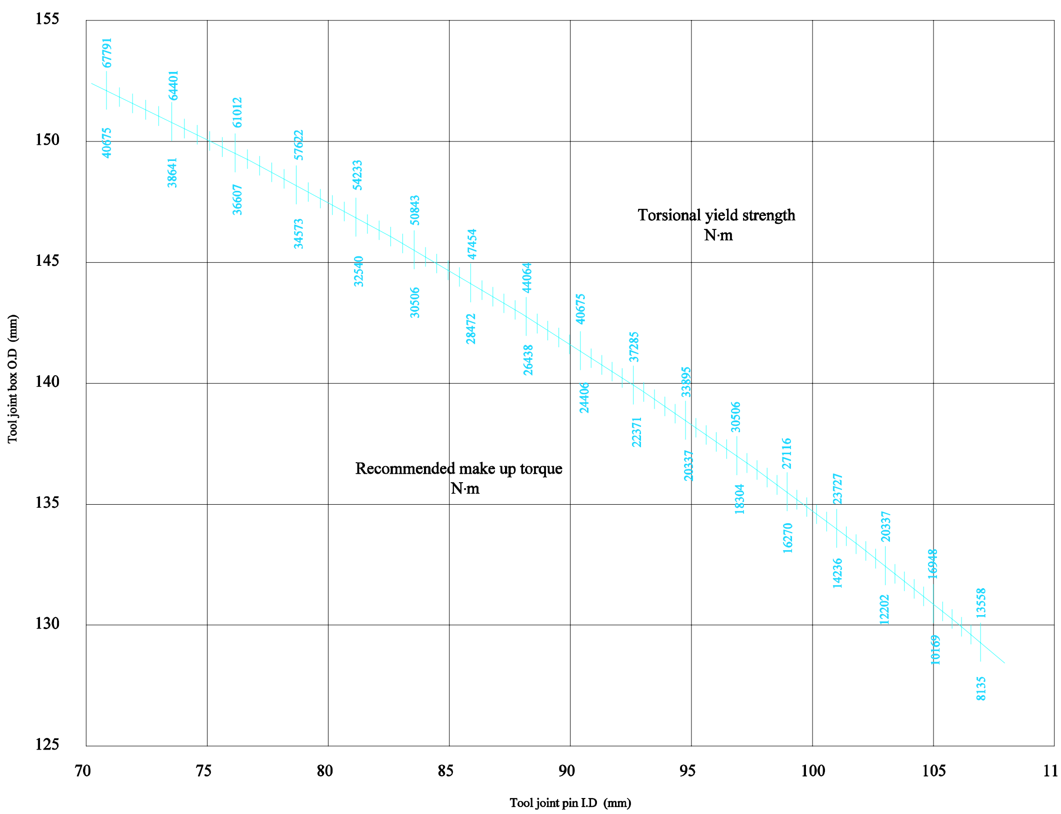


Figure 22-4 1/2 H90 Torsional yield and make-up

Figure C.22 — 4 ½ H90 torsional yield and make-up

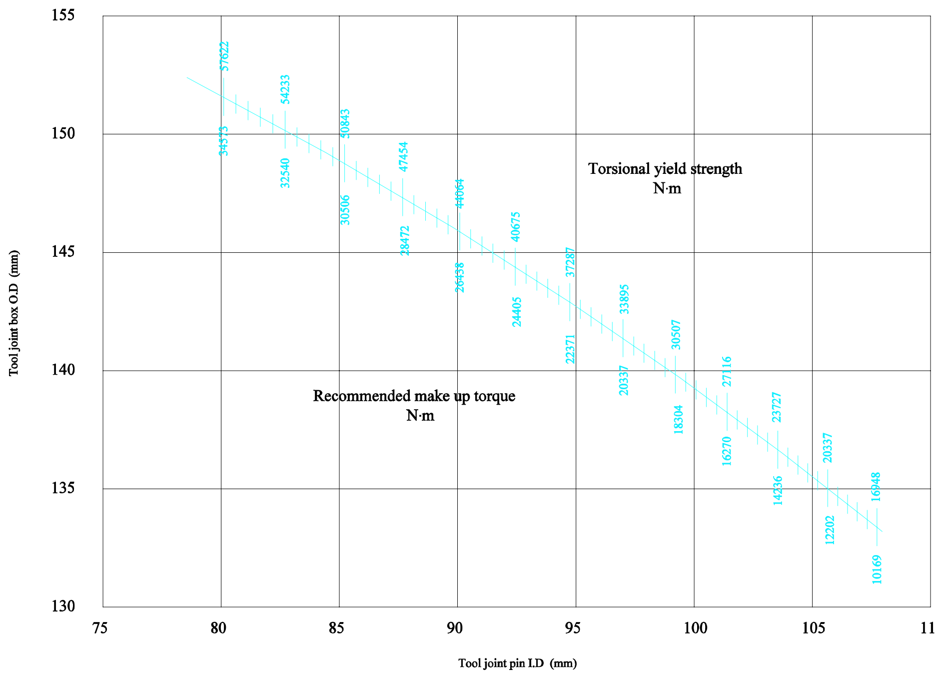


Figure 23-4 1/2 Open hole (standard weight) torsional yield and make-up

Figure C.23 — 4 ½ Open hole (standard weight) torsional yield and make-up

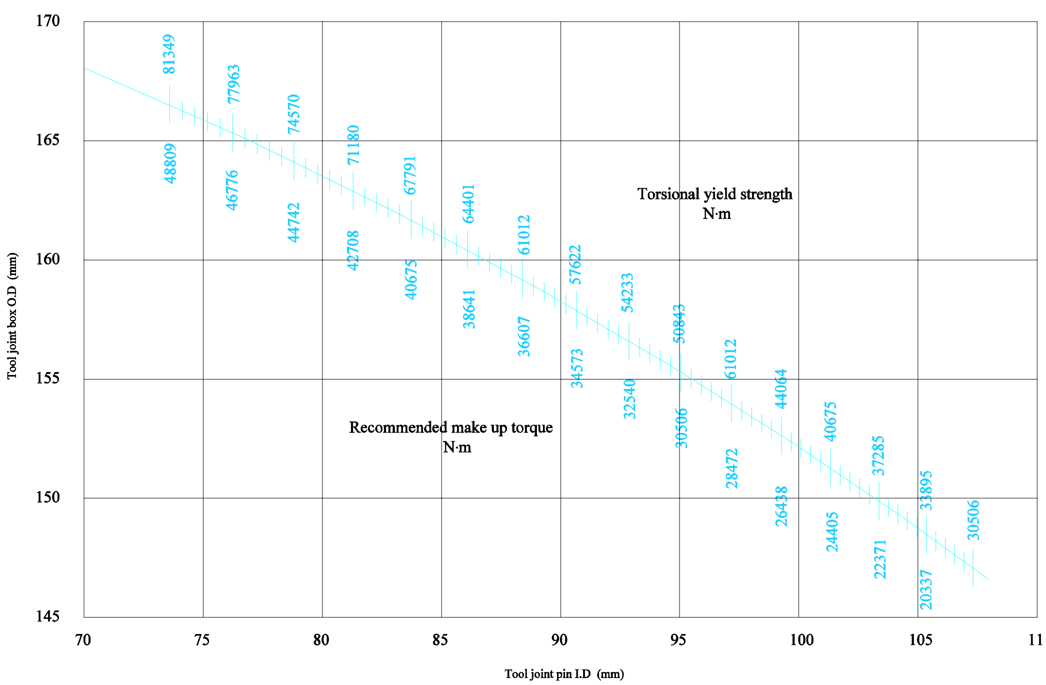


Figure 24-NC50 Torsional yield and make-up

Figure C.24 — NC50 torsional yield and make-up

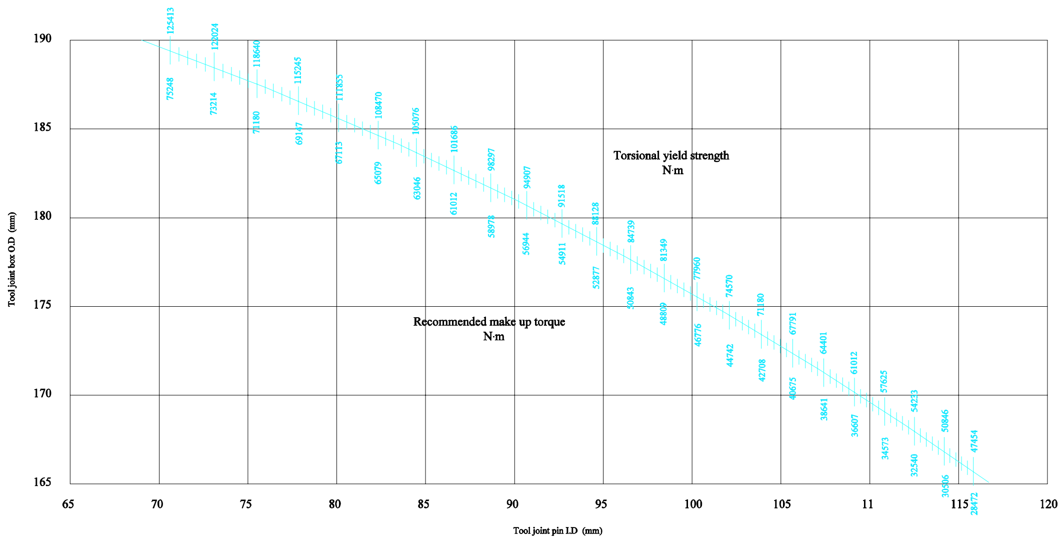


Figure 25-5 1/2 FH Torsional yield and make-up

Figure C.25 — 5 ½ FH torsional yield and make-up

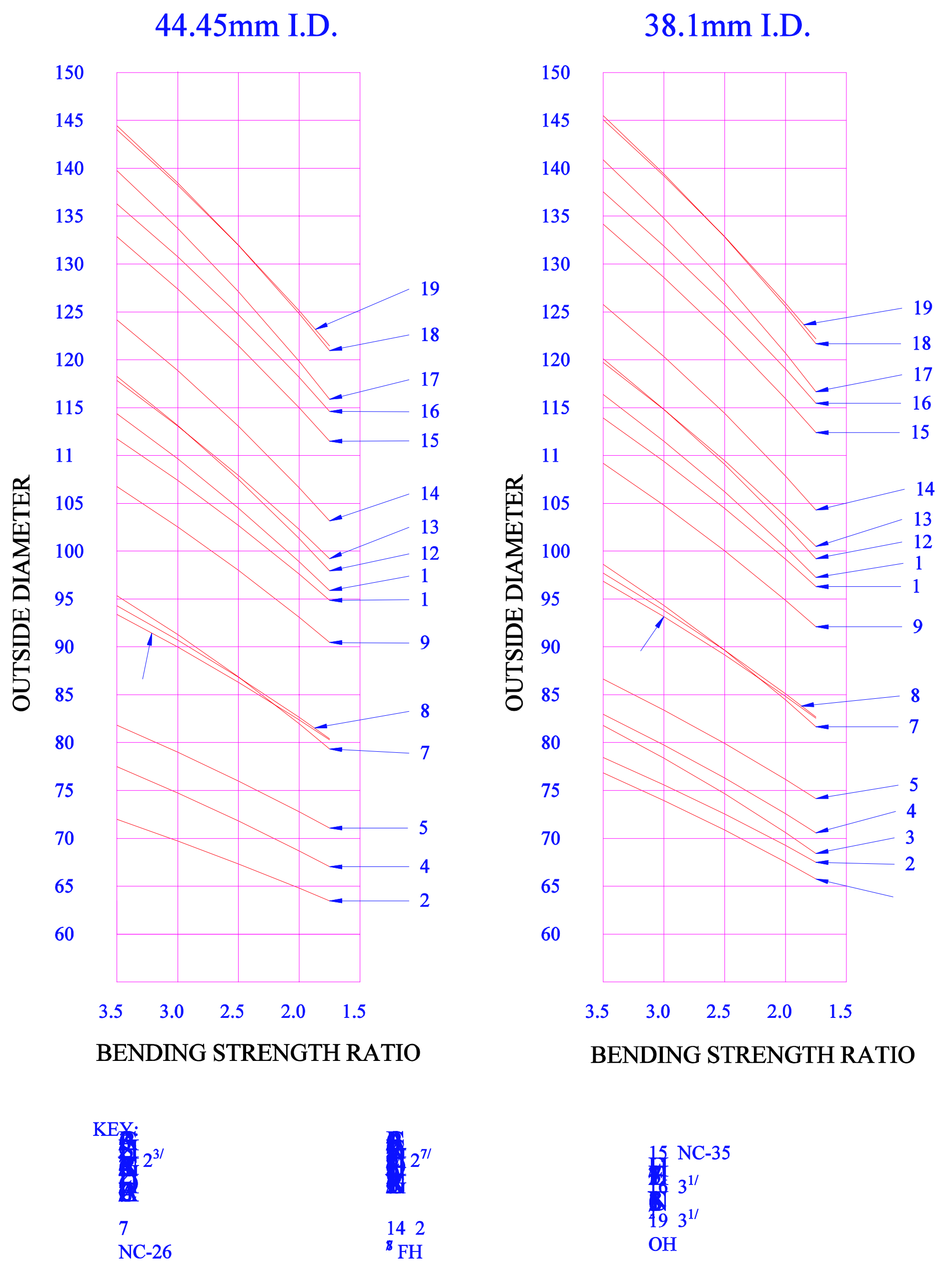


Figure 26-Drill collar bending strength ratios, 44.45mm and 38.1mm ID

Figure C.26 — Drill collar bending strength ratios, 44.45 mm and 38.1 mm ID

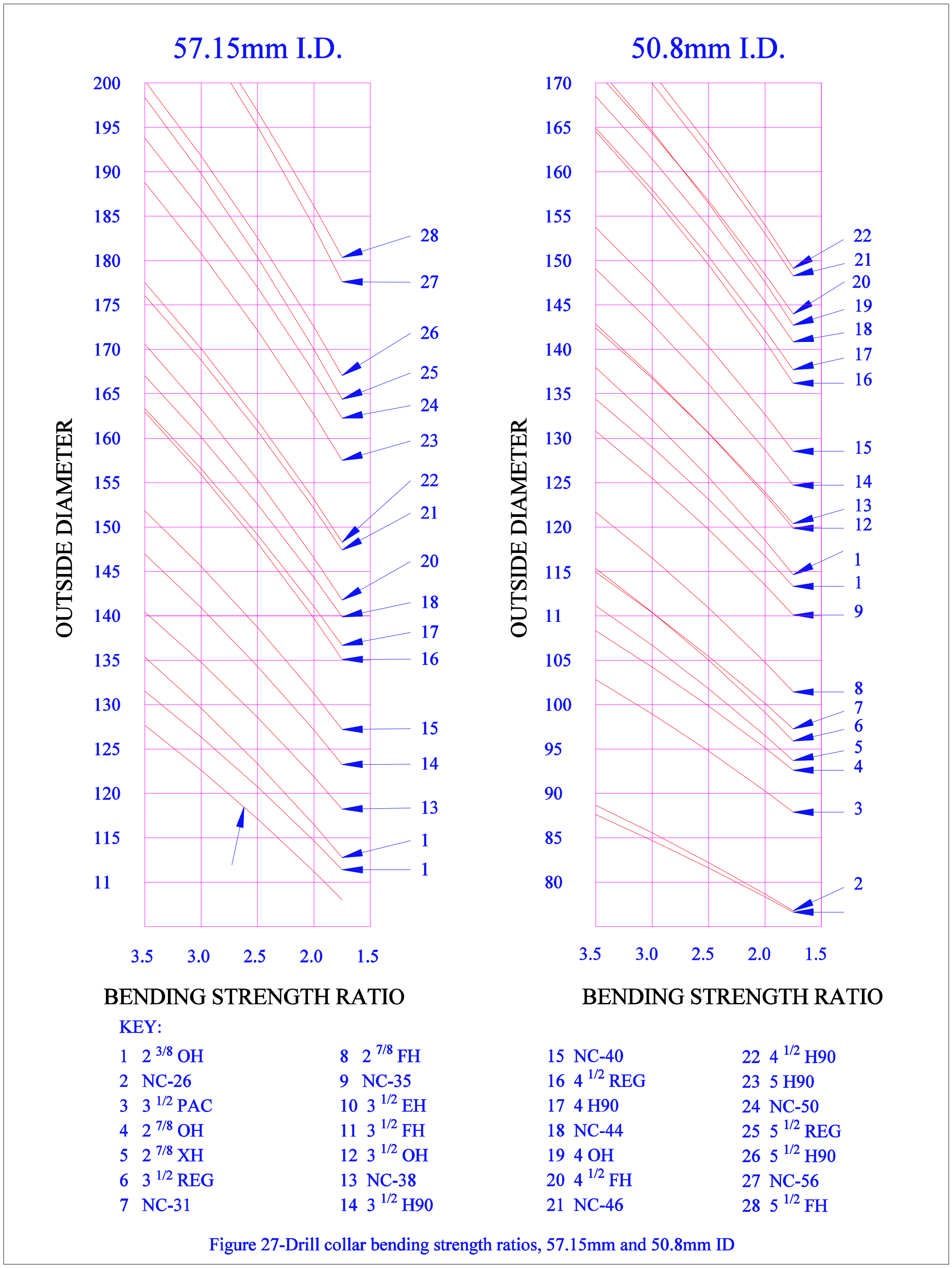
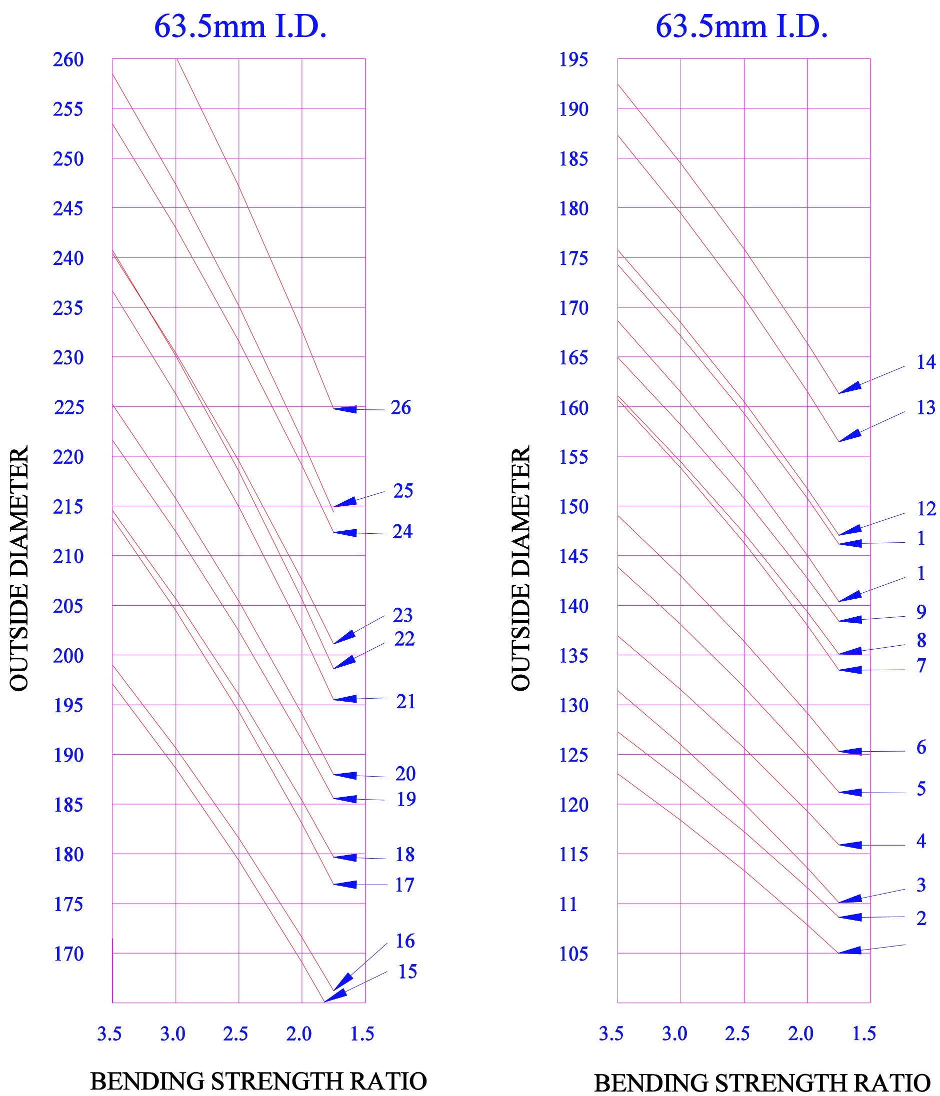


Figure C.27 — Drill collar bending strength ratios, 57.15 mm and 50.8 mm ID

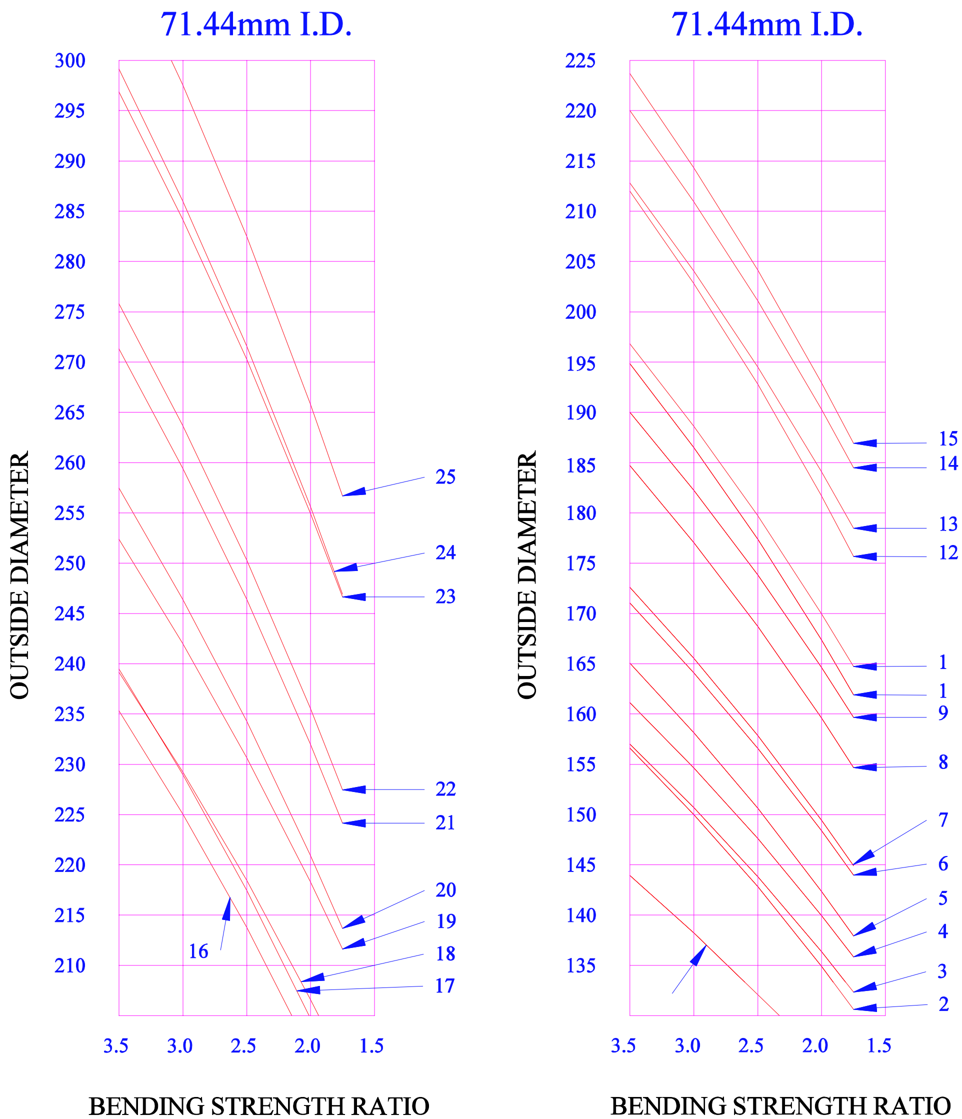


KEY:

1 NC-35	8 4 H90	15 5 1/2 REG	22 7 H90
2 3 1/2 EH	9 NC-44	16 5 1/2 H90	23 F5 1/
3 3 1/2 FH	10 4 1/2 FH	17 NC-56	24 FH6 5/
4 NC-38	11 NC-46	18 5 1/2 FH	25 REG7 5/
5 3 1/2 H90	12 4 1/2 H90	19 6 5/8 REG	26 NC-70
6 NC-40	13 5 H90	20 6 5/8 H90	
7 4 1/2 REG	14 NC-50	21 NC-61	

Figure 28-Drill collar bending strength ratios, 63.5mm ID

Figure C.28 — Drill collar bending strength ratios, 63.5 mm ID



KEY:

1 NC-40	8 5 H90	15 6 5/8 H90	22 7 5/8 H90
2 4 1/2 REG	9 NC-50	16 NC-61	23 8 5/8 REG
3 4 H90	10 5 1/2 REG	17 7 H90	24 NC-77
4 NC-44	11 5 1/2 H90	18 5 1/2 IF	25 8 5/8 H90
5 4 1/2 FH	12 NC-56	19 6 5/8 FH	
6 NC-46	13 5 1/2 FH	20 7 5/8 REG	
7 4 1/2 H90	14 6 5/8 REG	21 NC-70	

Figure 29-Drill collar bending strength ratios, 71.44mm ID

Figure C.29 — Drill collar bending strength ratios, 71.44 mm ID

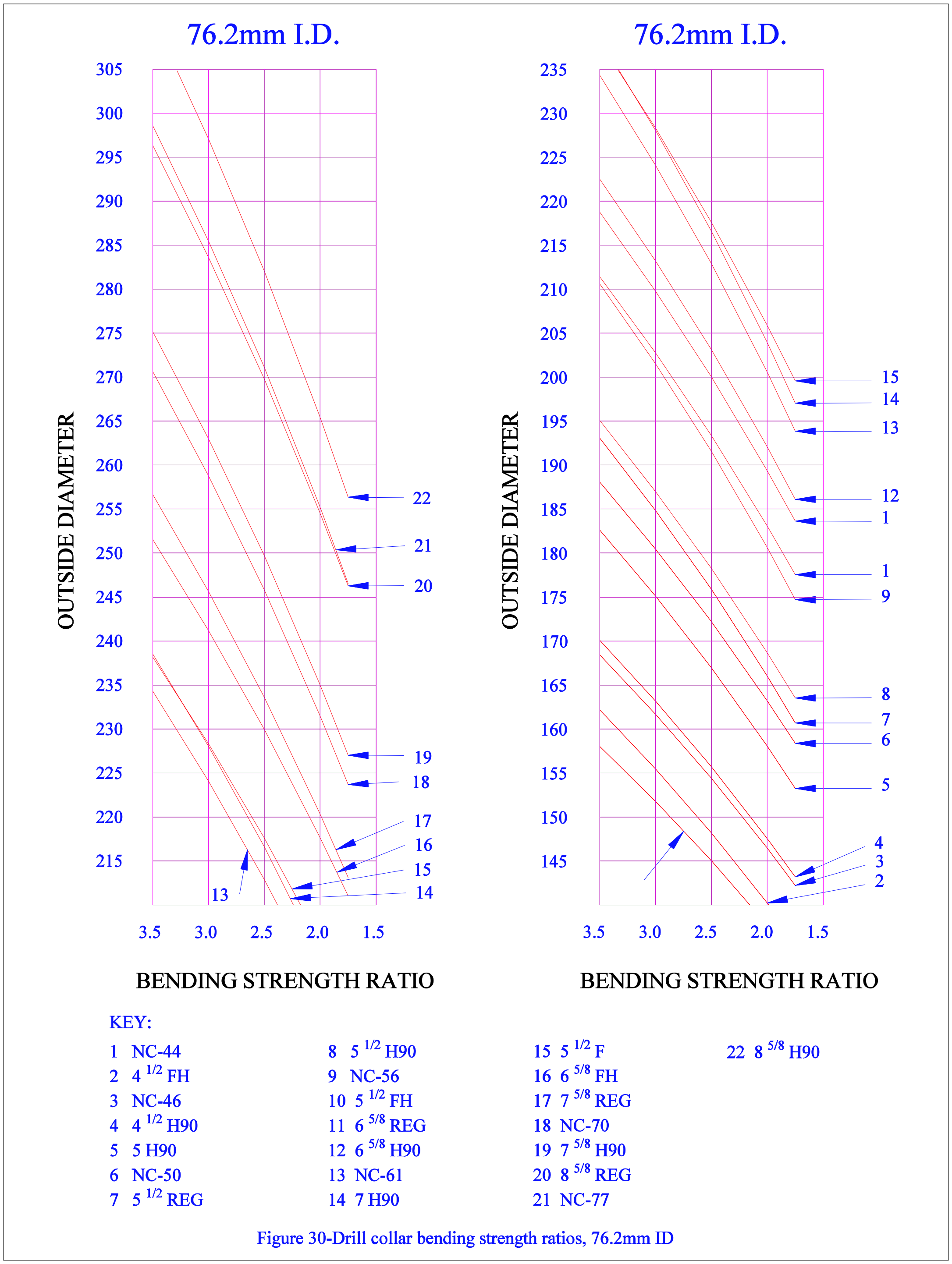
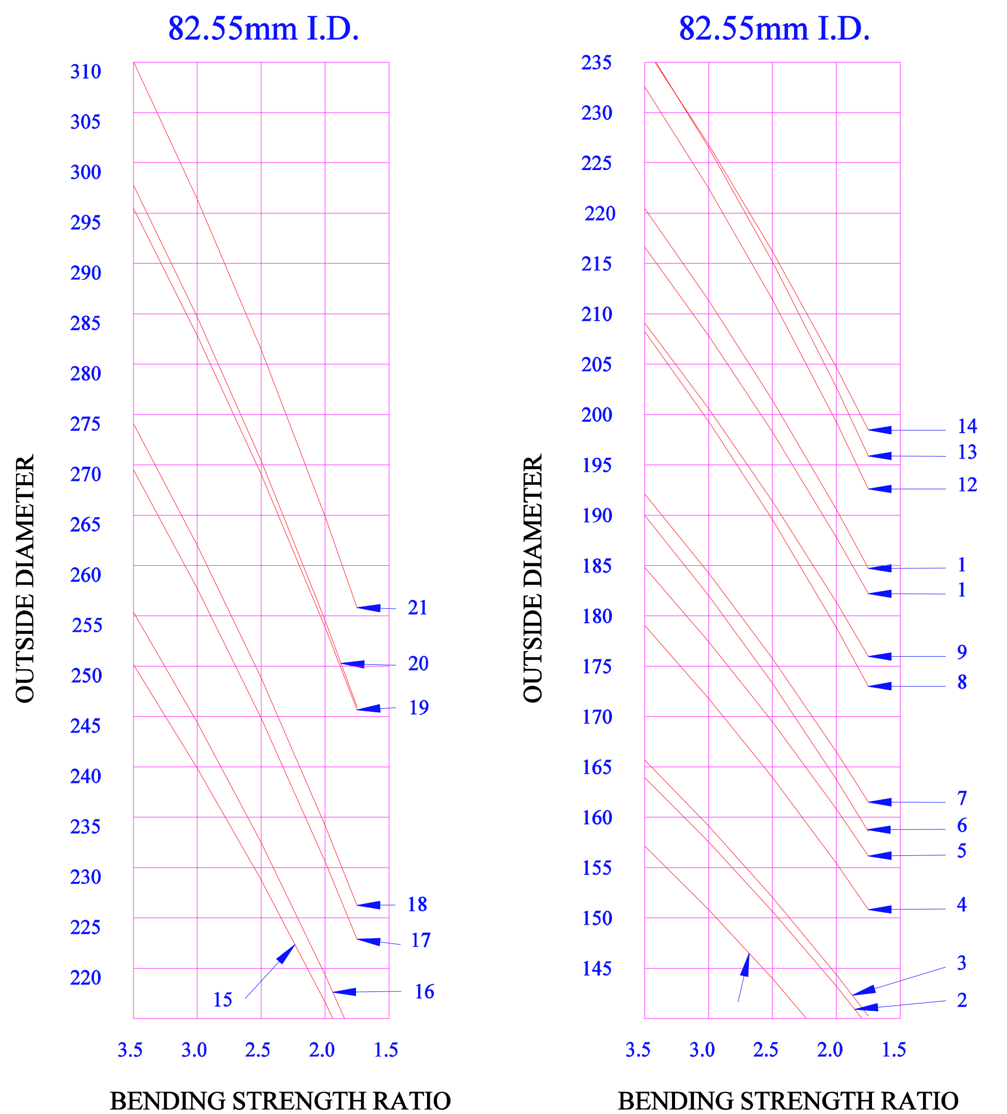


Figure C.30 — Drill collar bending strength ratios, 76.2 mm ID

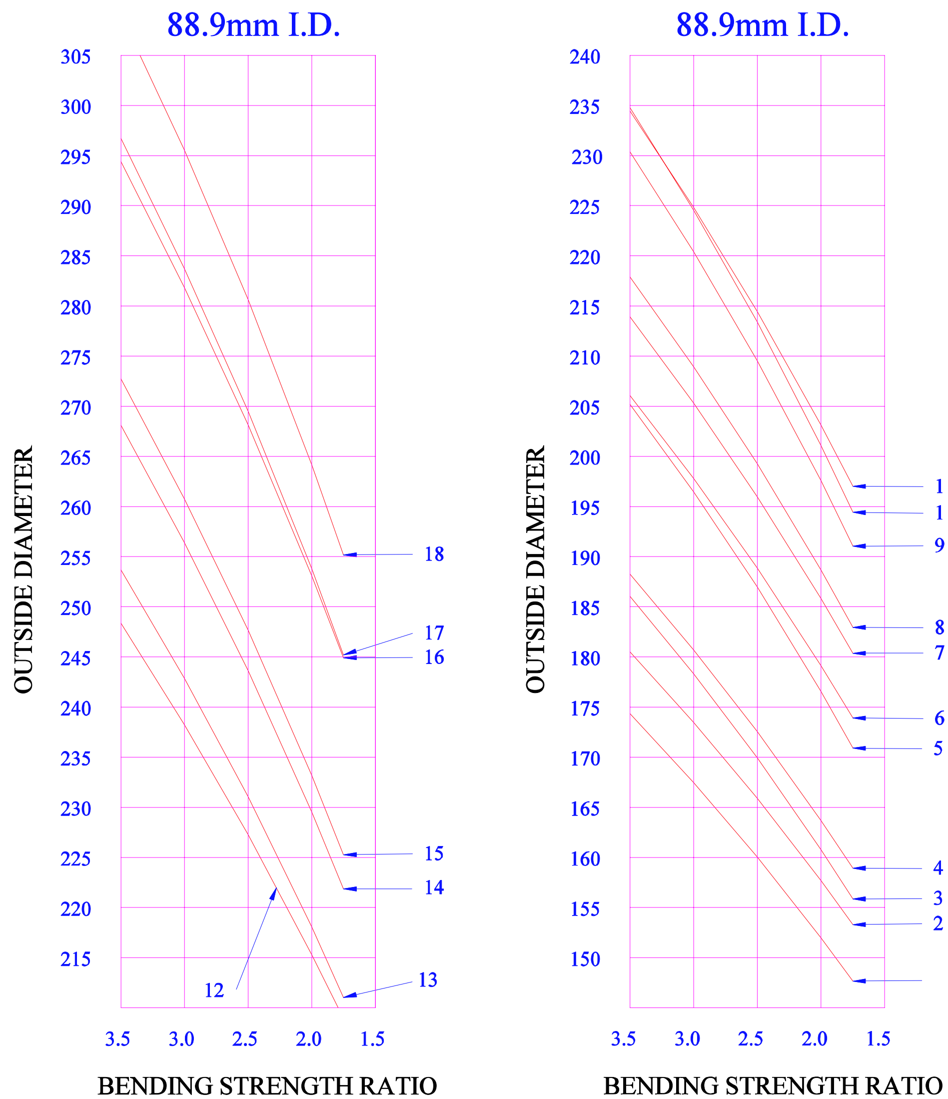


KEY:

1 4 1/2 FH	8 NC-56	15 6 5/8 FH
2 NC-46	9 5 1/2 FH	16 7 5/8 REG
3 4 1/2 H90	10 6 5/8 REG	17 NC-70
4 5 H90	11 6 5/8 H90	18 7 5/8 H90
5 NC-50	12 NC-61	19 8 5/8 REG
6 5 1/2 REG	13 7 H90	20 NC-77
7 5 1/2 H90	14 5 1/2 IF	21 8 5/8 H90

Figure 31-Drill collar bending strength ratios, 82.55mm ID

Figure C.31 — Drill collar bending strength ratios, 82.55 mm ID



KEY:

1 5 H90
2 NC-50
3 5 1/2 REG
4 5 1/2 H90
5 NC-56
6 5 1/2 FH
7 6 5/8 REG
8 6 5/8 H90
9 NC-61
10 7 H90
11 5 1/2 F
12 6 5/8 FH
13 7 5/8 REG
14 NC-70
15 7 5/8 H90
16 8 5/8 REG
17 NC-77
18 8 5/8 H90

Figure 32-Drill collar bending strength ratios, 88.9mm ID

Figure C.32 — Drill collar bending strength ratios, 88.9 mm ID

Figure C.33 — Delayed-failure characteristics of unnotched specimens of an SAE 4340 steel during cathodic charging with hydrogen under standardized conditions

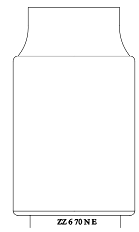


Figure C.34 — Figure 82

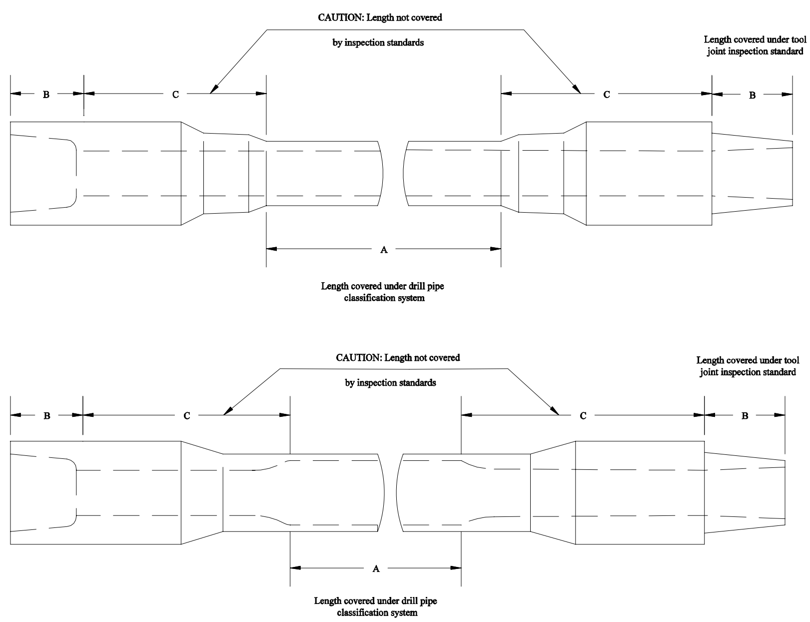


Figure 84-Identification of lengths covered by inspection standards

Figure C.35 — Figure 84 Identification of lengths covered by inspection standards

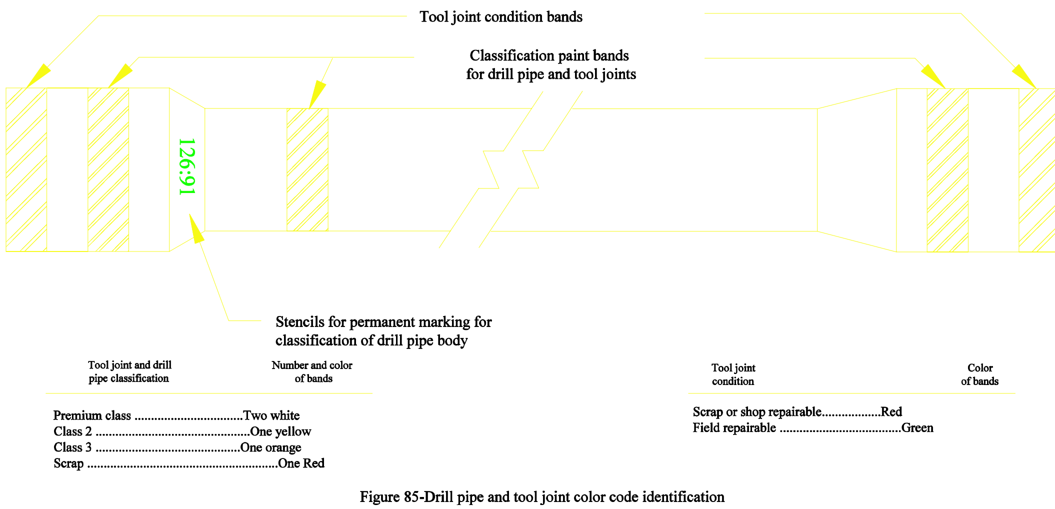


Figure C.36 — Figure 85 Drill pipe and tool joint color code identification

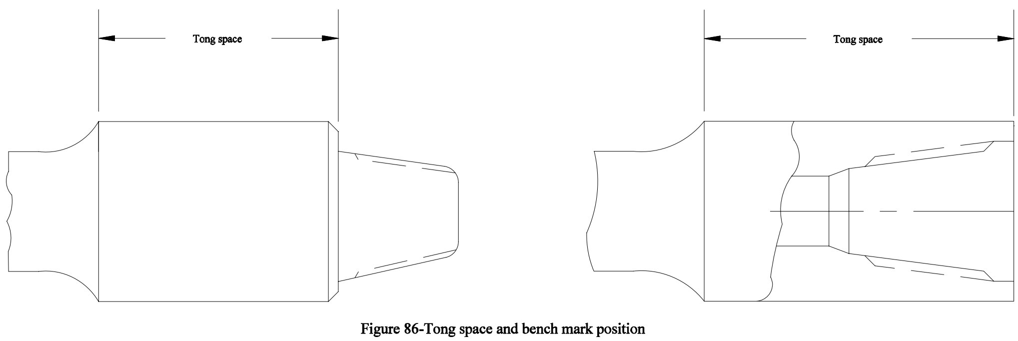


Figure C.37 — Figure 86   Tong space and bench mark position

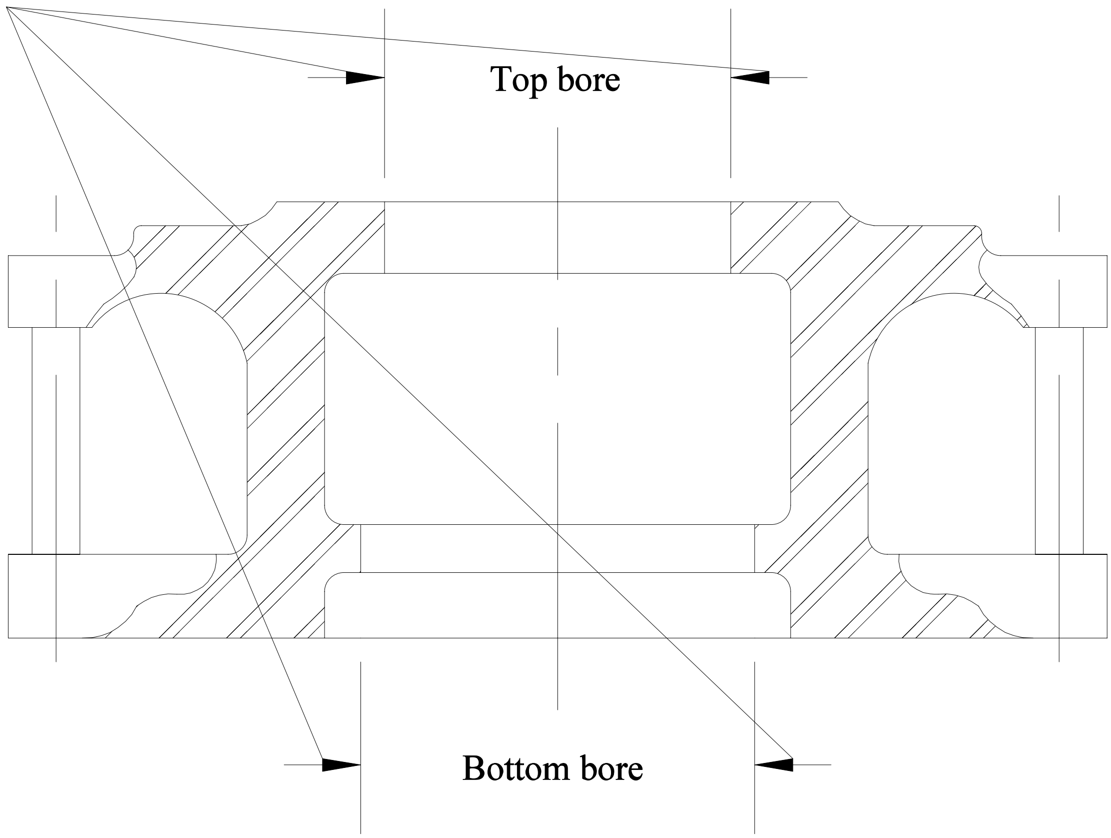


Figure 87-Drill collar elevator

Figure C.38 — Figure 87   Drill collar elevator

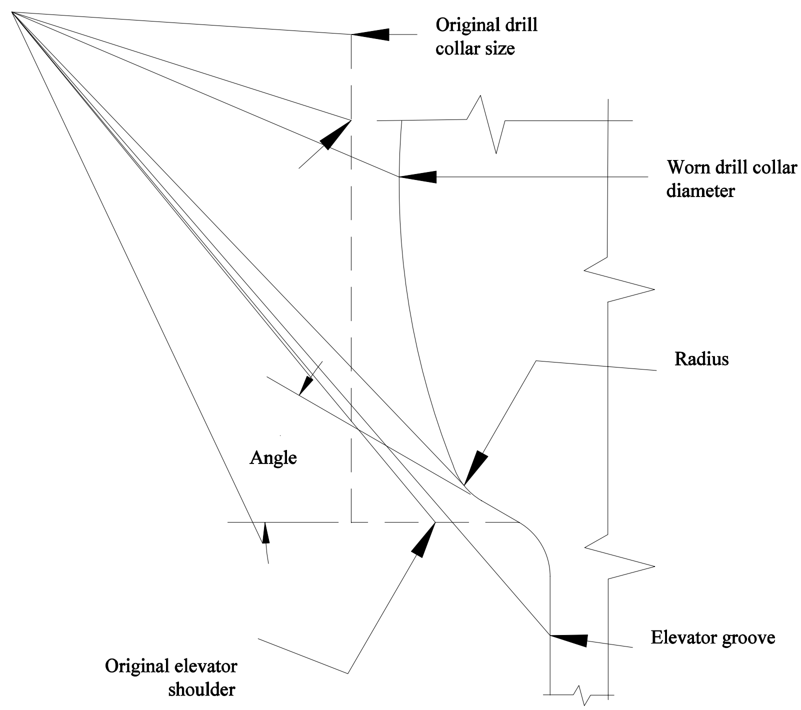


Figure 89-Drill collar wear

Figure C.39 — Figure 89 Drill collar wear

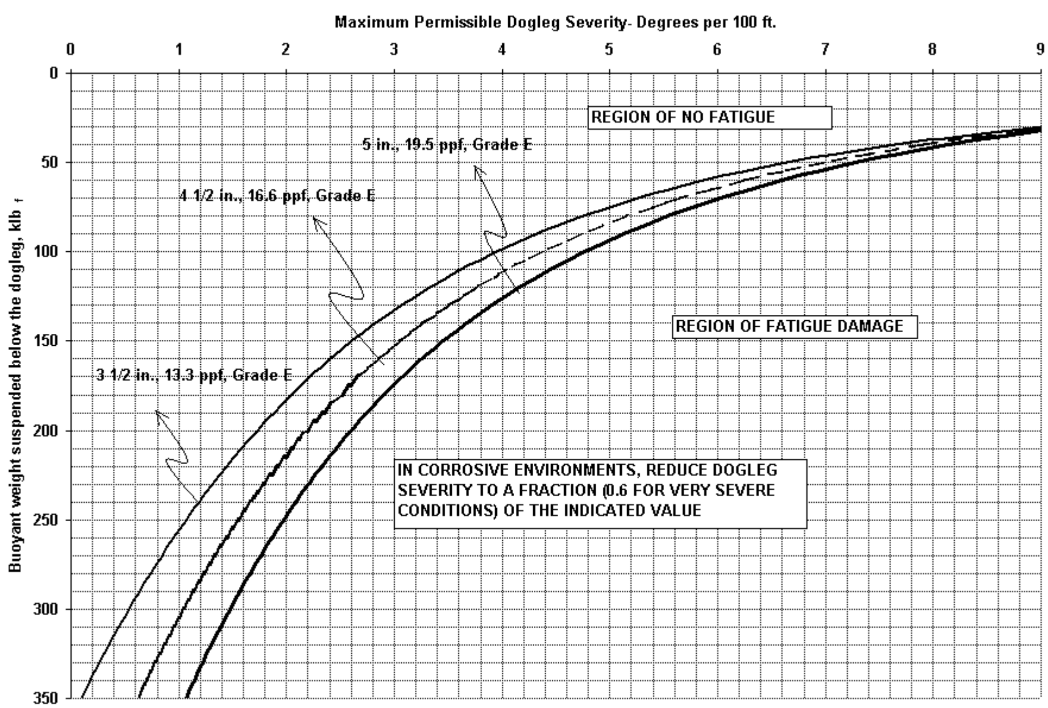


Figure C.40 — Dogleg Severity Limits for Fatigue of Grade E drill Pipe

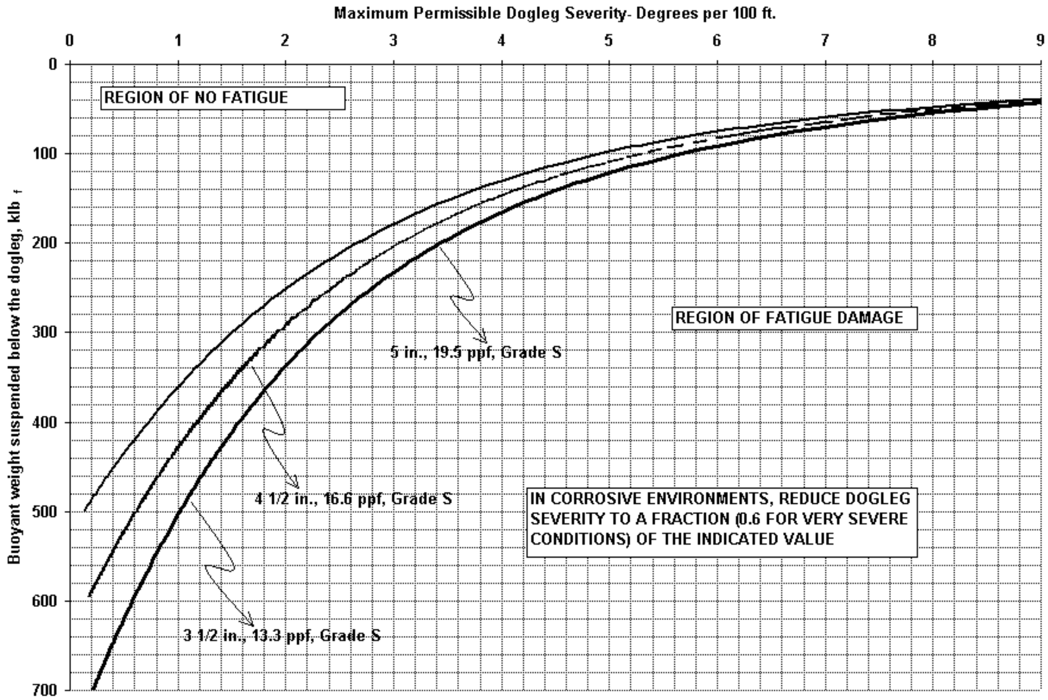


Figure C.41 — Dogleg Severity Limits for Fatigue of S-135 drill Pipe

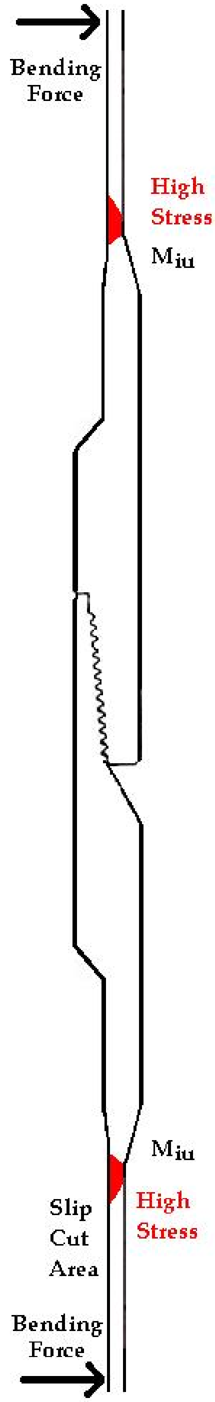


Figure C.42 — Figure 8 DP Cross-Section

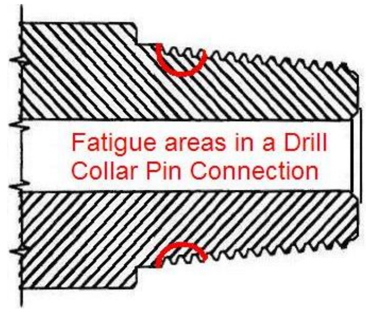


Figure C.43 — Figure 9 DC Pin Connection

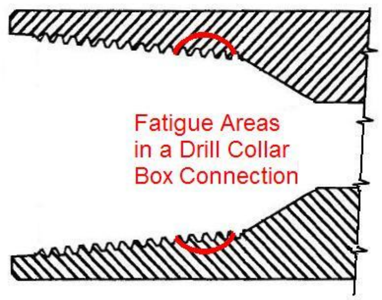


Figure C.44 — Figure 10 DC Box Connection

Figure C.45 — Figure 11 Wash out in the slip area

Figure C.46 — Figure 12 – CO_2 corrosion on drill pipe OD

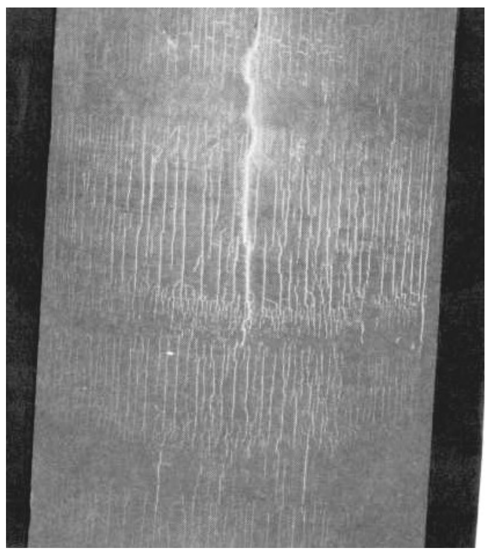
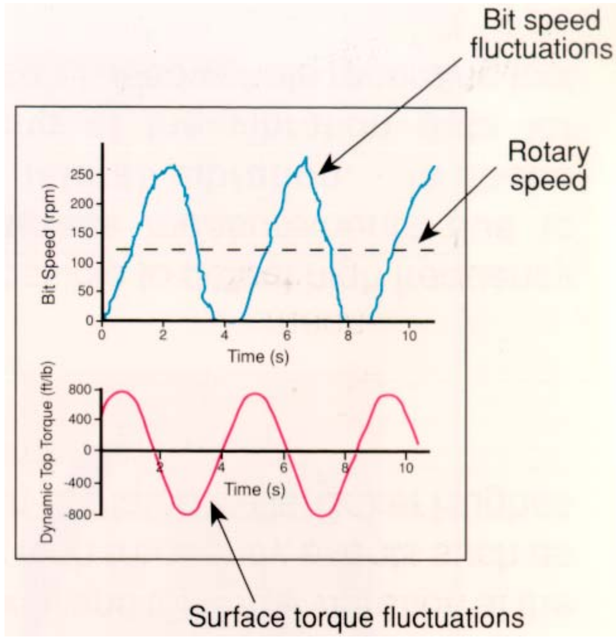


Figure C.47 — Figure 13 - Heat Checking on Box Tool Joint Shown Under Black Light

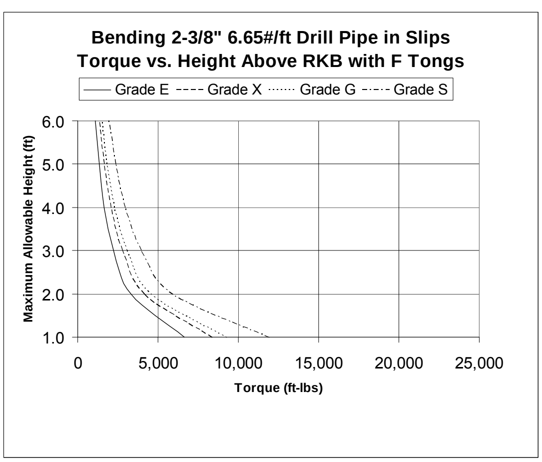


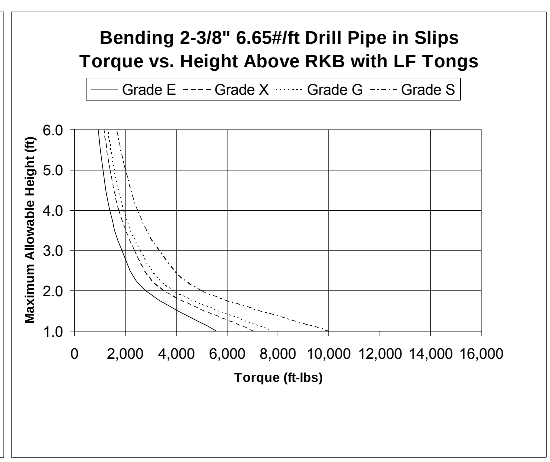
1. Bit speed fluctuations
2. Rotary speed
3. Surface torque fluctuations

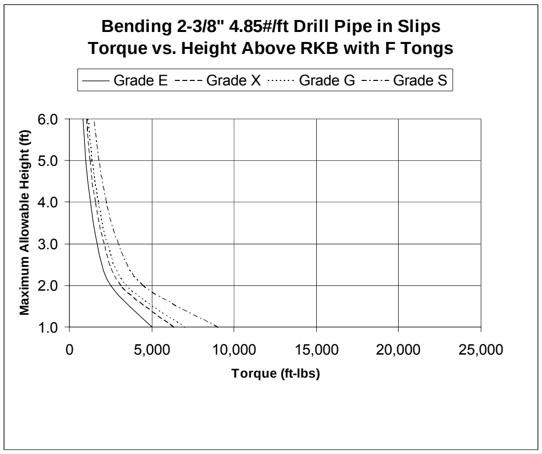
Figure C.48 — Stick-Slip Torsional Vibration

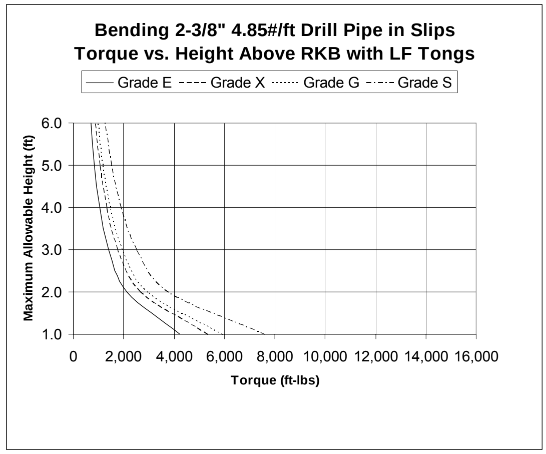
NOTE 1 These graphs are applicable for tong arms which are situated 180º from each other. Multiply the height by 1.4 to get the maximum allowable height if tong arms are 90o apart.

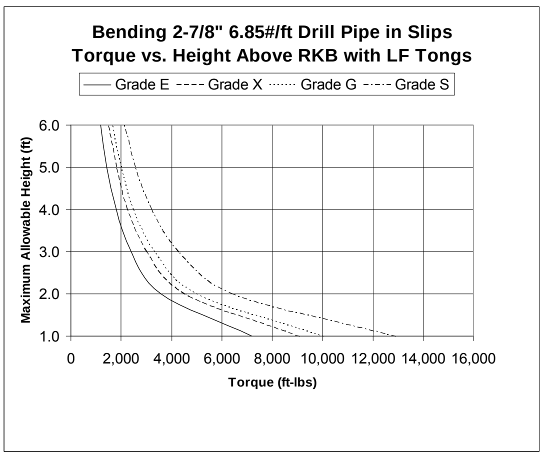
NOTE 2 These values are based on a tong arm lengths designated in Table B.44.

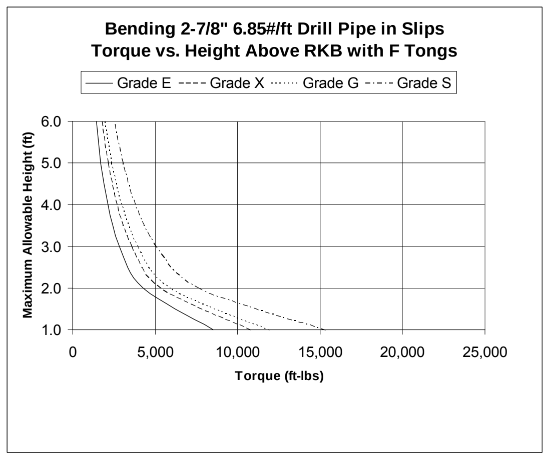


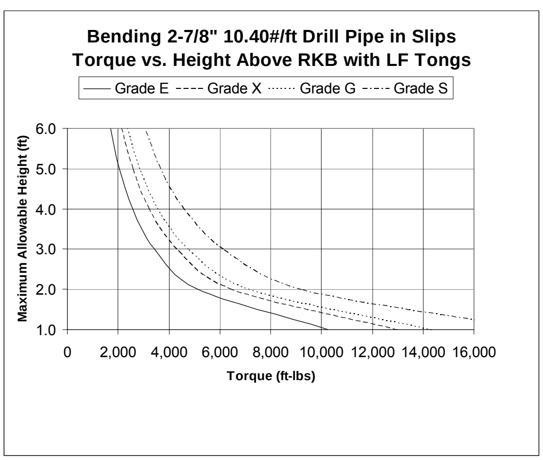


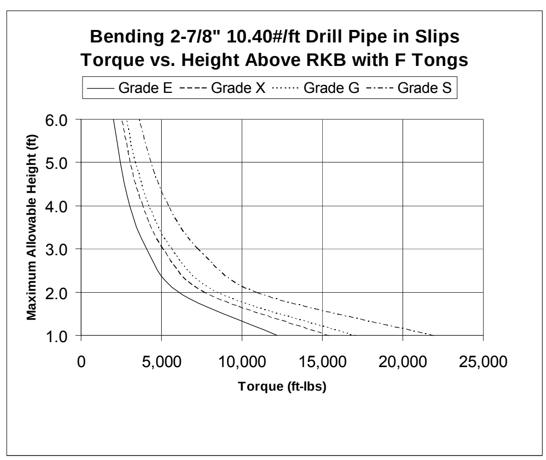

Bending 2-3/8" 4.85#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with F Tongs
Grade E
Grade X
Grade G
Grade S
Maximum Allowable Height (ft)
6.0
5.0
4.0
3.0
2.0
1.0
0
5,000
10,000
15,000
20,000
25,000
Torque (ft-lbs)

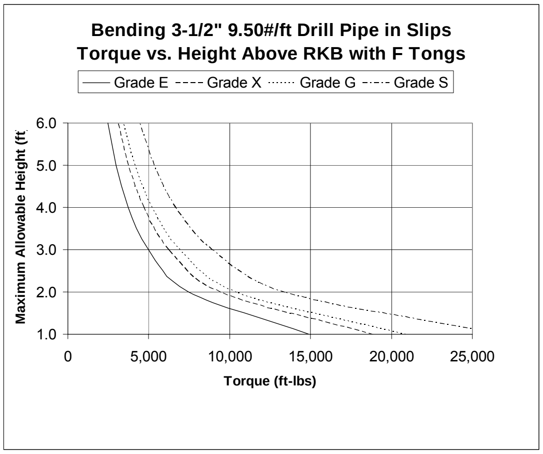

Bending 2-3/8" 4.85#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with LF Tongs
Grade E
Grade X
Grade G
Grade S
Maximum Allowable Height (ft)
6.0
5.0
4.0
3.0
2.0
1.0
0
2,000
4,000
6,000
8,000
10,000
12,000
14,000
16,000
Torque (ft-lbs)

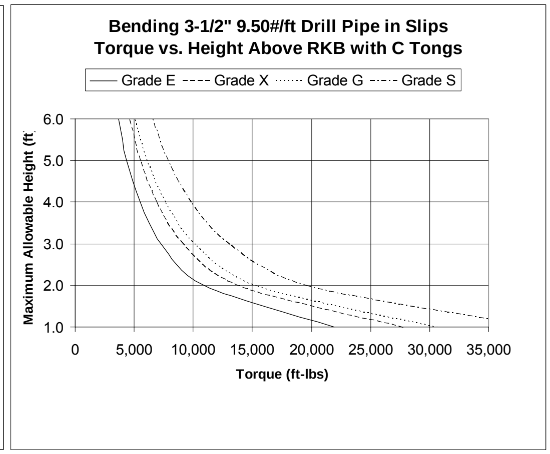

Bending 2-7/8" 6.85#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with LF Tongs
Grade E
Grade X
Grade G
Grade S
Maximum Allowable Height (ft)
6.0
5.0
4.0
3.0
2.0
1.0
0
2,000
4,000
6,000
8,000
10,000
12,000
14,000
16,000
Torque (ft-lbs)

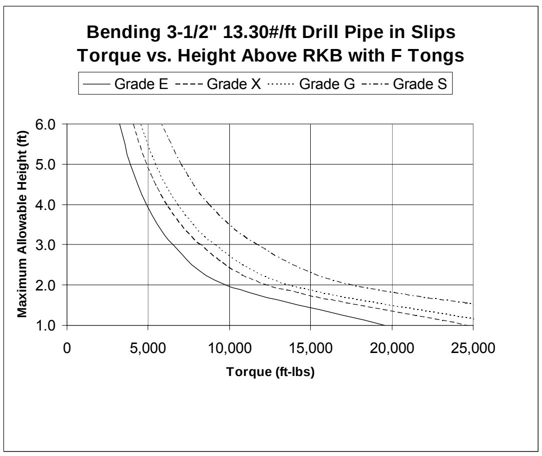

Bending 2-7/8" 6.85#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with F Tongs
Grade E
Grade X
Grade G
Grade S
Maximum Allowable Height (ft)
6.0
5.0
4.0
3.0
2.0
1.0
0
5,000
10,000
15,000
20,000
25,000
Torque (ft-lbs)

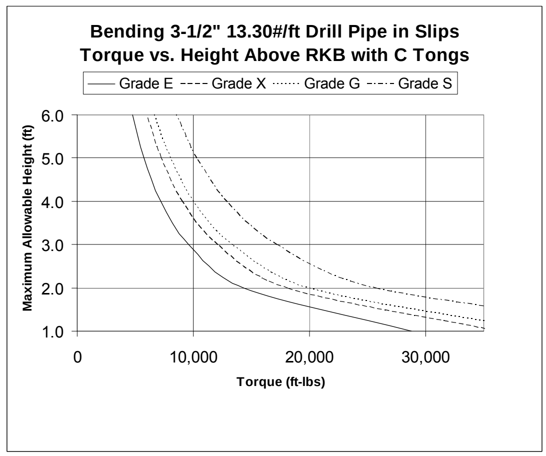

Bending 2-7/8" 10.40#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with LF Tongs
Grade E
Grade X
Grade G
Grade S
Maximum Allowable Height (ft)
6.0
5.0
4.0
3.0
2.0
1.0
0
2,000
4,000
6,000
8,000
10,000
12,000
14,000
16,000
Torque (ft-lbs)

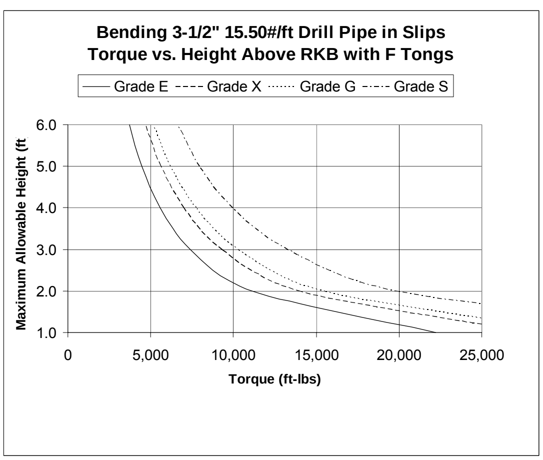

Bending 2-7/8" 10.40#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with F Tongs
Grade E
Grade X
Grade G
Grade S
Maximum Allowable Height (ft)
6.0
5.0
4.0
3.0
2.0
1.0
0
5,000
10,000
15,000
20,000
25,000
Torque (ft-lbs)

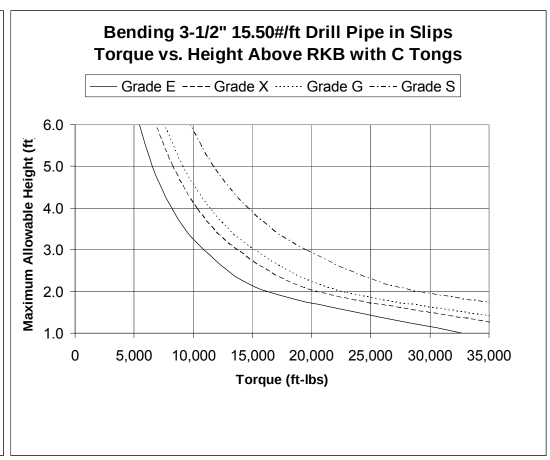

Bending 3-1/2" 9.50#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with F Tongs
Grade E
Grade X
Grade G
Grade S
Maximum Allowable Height (ft
6.0
5.0
4.0
3.0
2.0
1.0
0
5,000
10,000
15,000
20,000
25,000
Torque (ft-lbs)

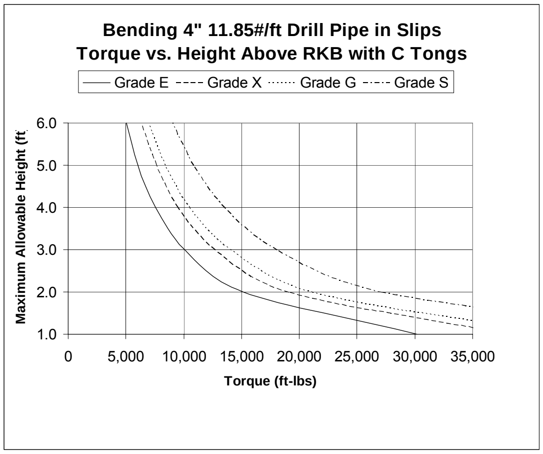

Bending 3-1/2" 9.50#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with C Tongs
Grade E
Grade X
Grade G
Grade S
Maximum Allowable Height (ft
6.0
5.0
4.0
3.0
2.0
1.0
0
5,000
10,000
15,000
20,000
25,000
30,000
35,000
Torque (ft-lbs)

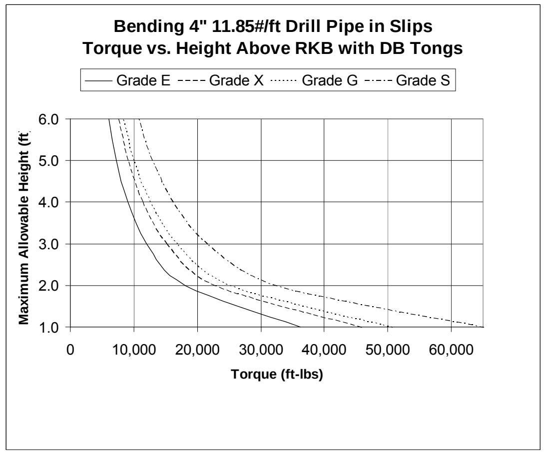

Bending 3-1/2" 13.30#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with F Tongs
Grade E
Grade X
Grade G
Grade S
Maximum Allowable Height (ft)
6.0
5.0
4.0
3.0
2.0
1.0
0
5,000
10,000
15,000
20,000
25,000
Torque (ft-lbs)

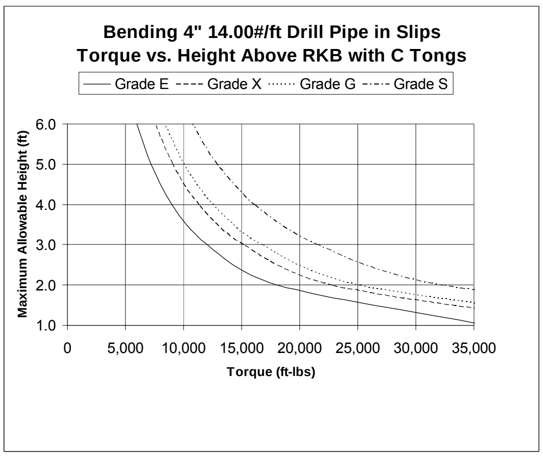

Bending 3-1/2" 13.30#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with C Tongs
Grade E
Grade X
Grade G
Grade S
Maximum Allowable Height (ft)
6.0
5.0
4.0
3.0
2.0
1.0
0
10,000
20,000
30,000
Torque (ft-lbs)

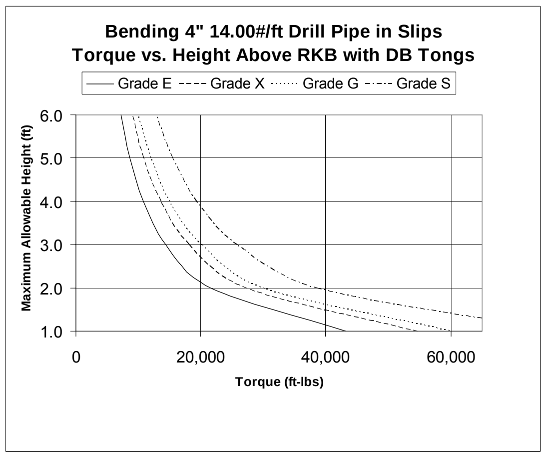

Bending 3-1/2" 15.50#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with F Tongs
Grade E
Grade X
Grade G
Grade S
Maximum Allowable Height (ft
6.0
5.0
4.0
3.0
2.0
1.0
0
5,000
10,000
15,000
20,000
25,000
Torque (ft-lbs)

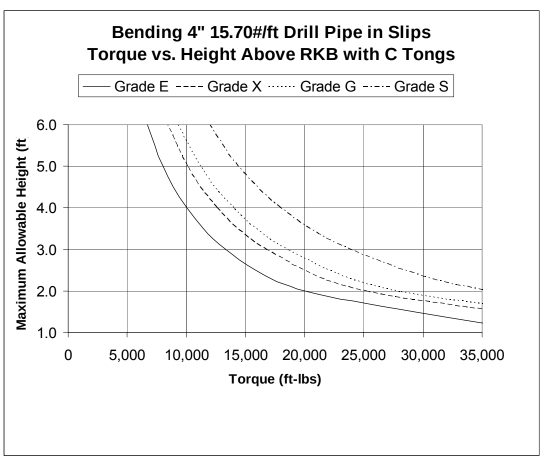

Bending 3-1/2" 15.50#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with C Tongs
Grade E
Grade X
Grade G
Grade S
Maximum Allowable Height (ft
6.0
5.0
4.0
3.0
2.0
1.0
0
5,000
10,000
15,000
20,000
25,000
30,000
35,000
Torque (ft-lbs)

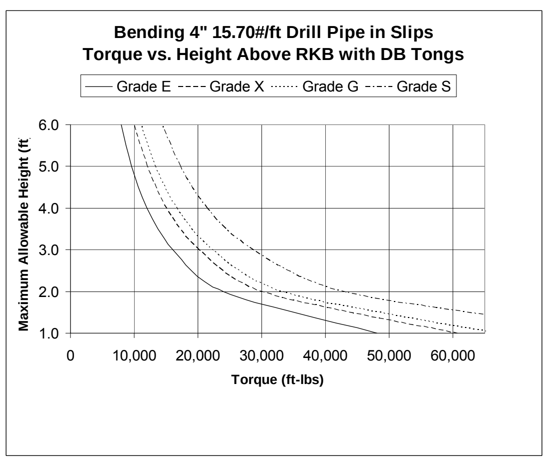

Bending 4" 11.85#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with C Tongs
Grade E
Grade X
Grade G
Grade S
Maximum Allowable Height (ft
Torque (ft-lbs)

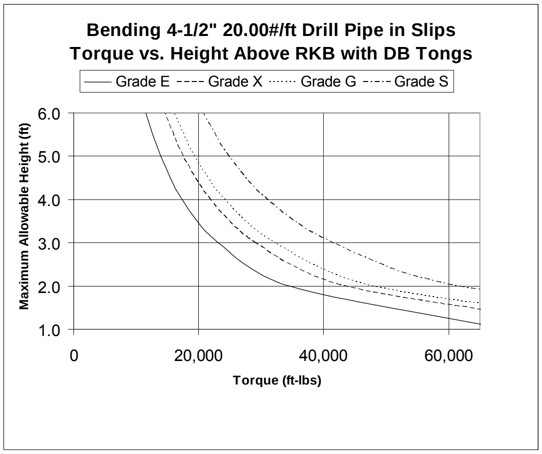

Bending 4" 11.85#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with DB Tongs
Grade E
Grade X
Grade G
Grade S
Maximum Allowable Height (ft
Torque (ft-lbs)

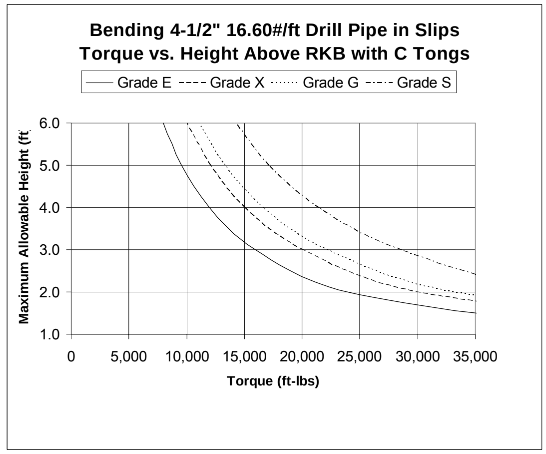

Bending 4" 14.00#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with C Tongs
Grade E
Grade X
Grade G
Grade S
Maximum Allowable Height (ft)
Torque (ft-lbs)

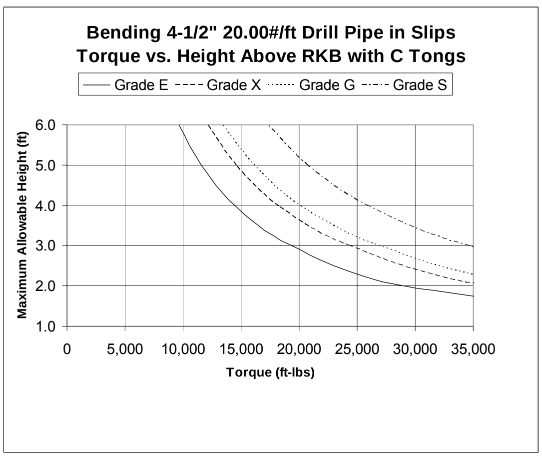

Bending 4" 14.00#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with DB Tongs
Grade E
Grade X
Grade G
Grade S
Maximum Allowable Height (ft)
Torque (ft-lbs)

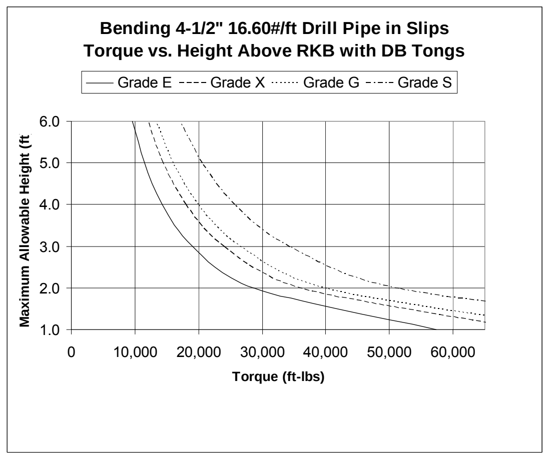

Bending 4" 15.70#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with C Tongs
Grade E
Grade X
Grade G
Grade S
Maximum Allowable Height (ft
Torque (ft-lbs)

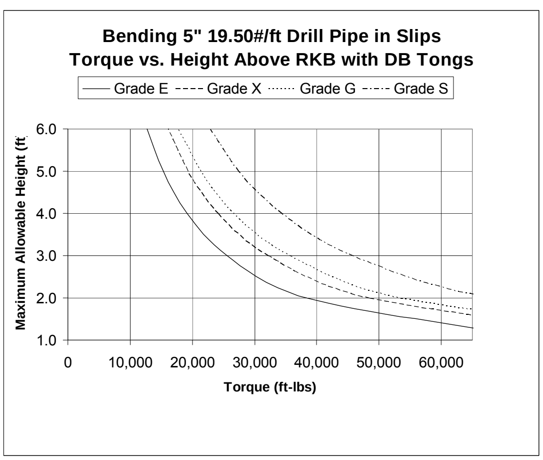

Bending 4" 15.70#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with DB Tongs
Grade E
Grade X
Grade G
Grade S
Maximum Allowable Height (ft
Torque (ft-lbs)

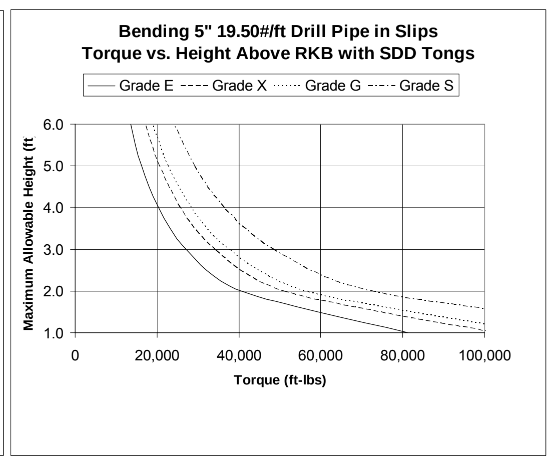

Bending 4-1/2" 20.00#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with DB Tongs
Grade E
Grade X
Grade G
Grade S
Maximum Allowable Height (ft)
6.0
5.0
4.0
3.0
2.0
1.0
0
20,000
40,000
60,000
Torque (ft-lbs)

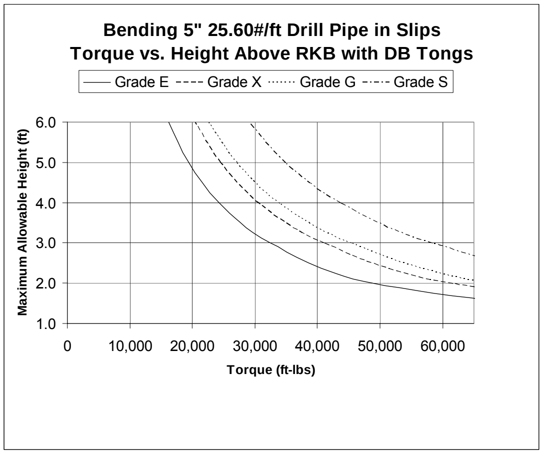

Bending 4-1/2" 16.60#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with C Tongs
Grade E
Grade X
Grade G
Grade S
Maximum Allowable Height (ft
6.0
5.0
4.0
3.0
2.0
1.0
0
5,000
10,000
15,000
20,000
25,000
30,000
35,000
Torque (ft-lbs)

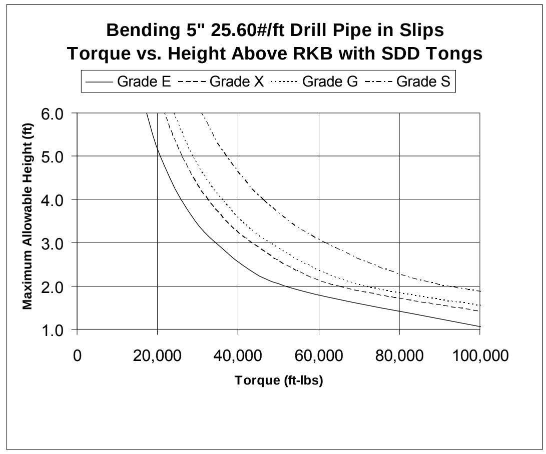

Bending 4-1/2" 20.00#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with C Tongs
Grade E
Grade X
Grade G
Grade S
Maximum Allowable Height (ft)
6.0
5.0
4.0
3.0
2.0
1.0
0
5,000
10,000
15,000
20,000
25,000
30,000
35,000
Torque (ft-lbs)

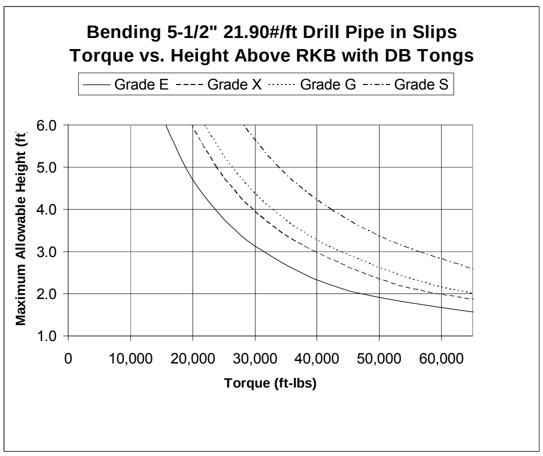

Bending 4-1/2" 16.60#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with DB Tongs
Grade E
Grade X
Grade G
Grade S
Maximum Allowable Height (ft
6.0
5.0
4.0
3.0
2.0
1.0
0
10,000
20,000
30,000
40,000
50,000
60,000
Torque (ft-lbs)

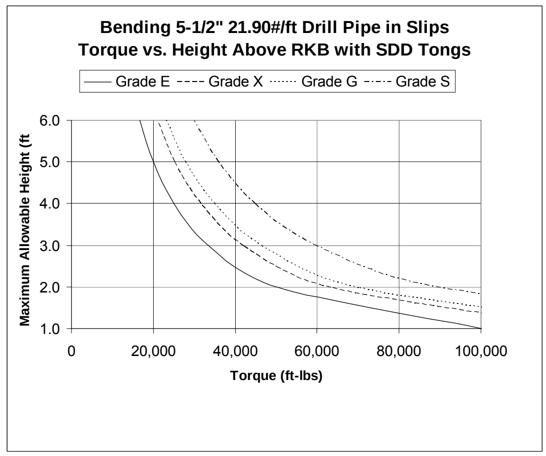

Bending 5" 19.50#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with DB Tongs
Grade E
Grade X
Grade G
Grade S
Maximum Allowable Height (ft
6.0
5.0
4.0
3.0
2.0
1.0
0
10,000
20,000
30,000
40,000
50,000
60,000
Torque (ft-lbs)

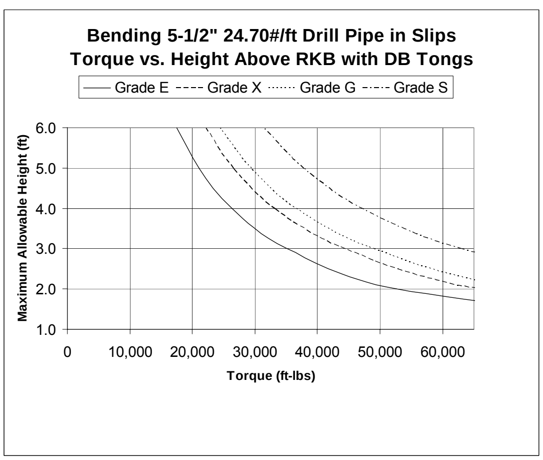

Bending 5" 19.50#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with SDD Tongs
Grade E
Grade X
Grade G
Grade S
Maximum Allowable Height (ft
6.0
5.0
4.0
3.0
2.0
1.0
0
20,000
40,000
60,000
80,000
100,000
Torque (ft-lbs)

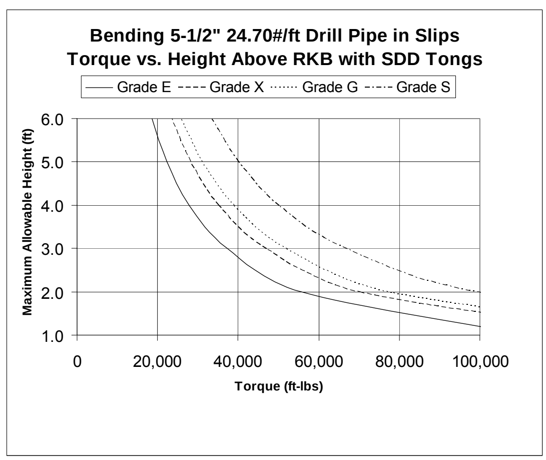

Bending 5" 25.60#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with DB Tongs
Grade E
Grade X
Grade G
Grade S
Maximum Allowable Height (ft)
6.0
5.0
4.0
3.0
2.0
1.0
0
10,000
20,000
30,000
40,000
50,000
60,000
Torque (ft-lbs)

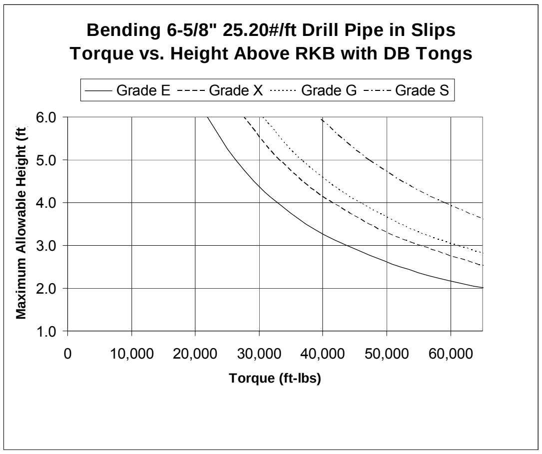

Bending 5" 25.60#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with SDD Tongs
Grade E
Grade X
Grade G
Grade S
Maximum Allowable Height (ft)
6.0
5.0
4.0
3.0
2.0
1.0
0
20,000
40,000
60,000
80,000
100,000
Torque (ft-lbs)

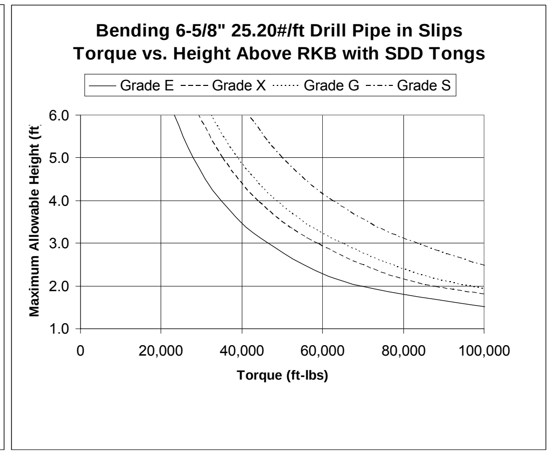

Bending 5-1/2" 21.90#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with DB Tongs
Grade E
Grade X
Grade G
Grade S
Maximum Allowable Height (ft
6.0
5.0
4.0
3.0
2.0
1.0
0
10,000
20,000
30,000
40,000
50,000
60,000
Torque (ft-lbs)


Bending 5-1/2" 21.90#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with SDD Tongs
Grade E
Grade X
Grade G
Grade S
Maximum Allowable Height (ft
6.0
5.0
4.0
3.0
2.0
1.0
0
20,000
40,000
60,000
80,000
100,000
Torque (ft-lbs)


Bending 5-1/2" 24.70#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with DB Tongs
Grade E
Grade X
Grade G
Grade S
Maximum Allowable Height (ft)
6.0
5.0
4.0
3.0
2.0
1.0
0
10,000
20,000
30,000
40,000
50,000
60,000
Torque (ft-lbs)


Bending 5-1/2" 24.70#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with SDD Tongs
Grade E
Grade X
Grade G
Grade S
Maximum Allowable Height (ft)
6.0
5.0
4.0
3.0
2.0
1.0
0
20,000
40,000
60,000
80,000
100,000
Torque (ft-lbs)


Bending 6-5/8" 25.20#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with DB Tongs
Grade E
Grade X
Grade G
Grade S
Maximum Allowable Height (ft
6.0
5.0
4.0
3.0
2.0
1.0
0
10,000
20,000
30,000
40,000
50,000
60,000
Torque (ft-lbs)


Bending 6-5/8" 25.20#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with SDD Tongs
Grade E
Grade X
Grade G
Grade S
Maximum Allowable Height (ft
6.0
5.0
4.0
3.0
2.0
1.0
0
20,000
40,000
60,000
80,000
100,000
Torque (ft-lbs)

Annex D
(normative)

Figures in US Customary units

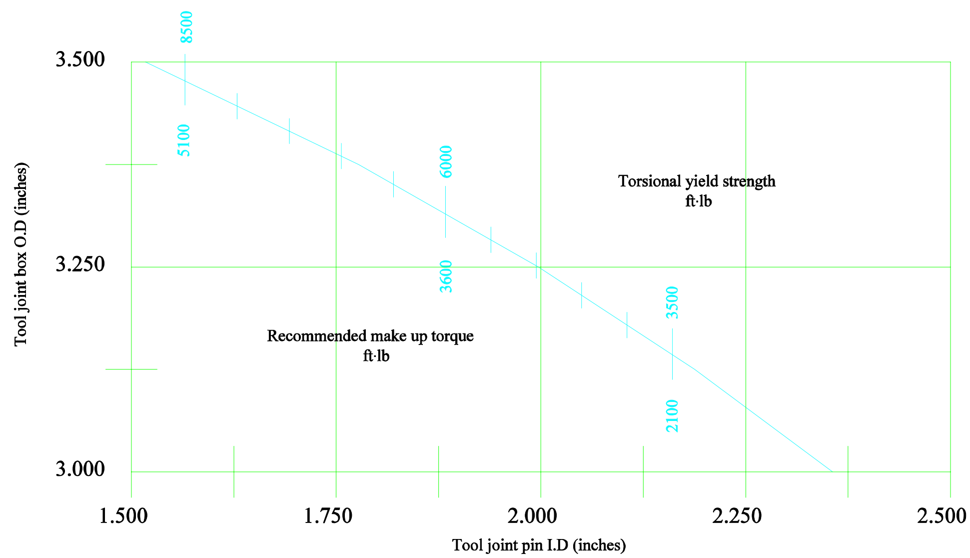


Figure 1-NC26 Torsional yield and make-up

Figure D.1 — NC26 Torsional yield and make-up

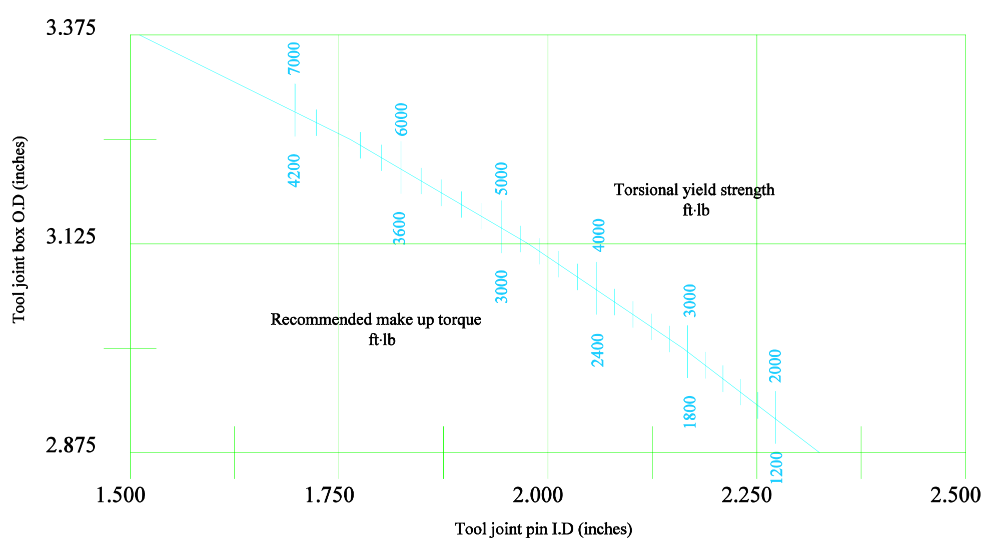


Figure 2-2 3/8 Open hole torsional yield and make-up

Figure D.2 — 2 3/8 Open hole torsional yield and make-up

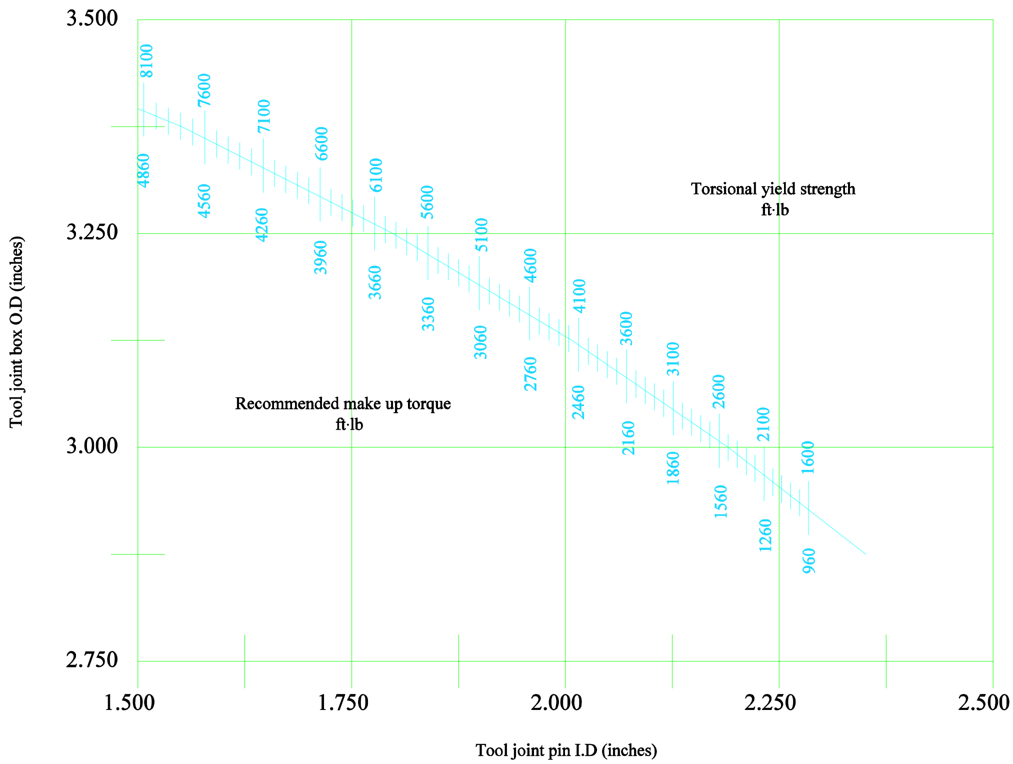


Figure 3-2 3/8 Wide open torsional yield and make-up

Figure D.3 — 2 3/8 Wide open torsional yield and make-up

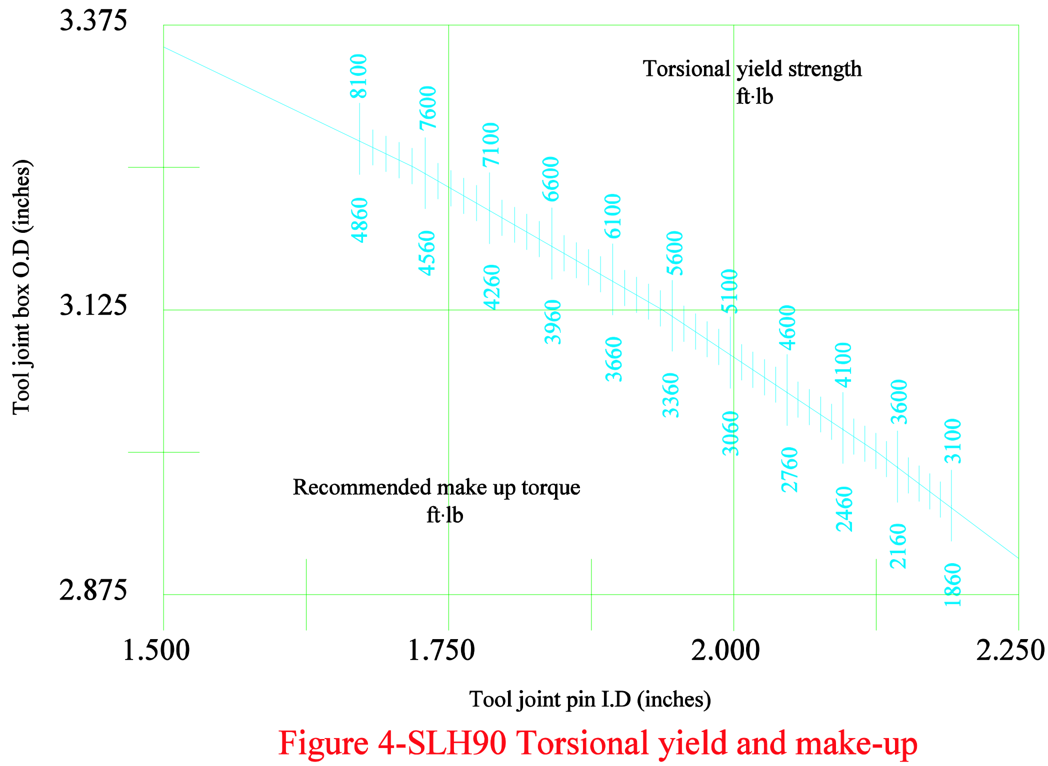


Figure D.4 — 2 3/8 SLH90 Torsional yield and make-up

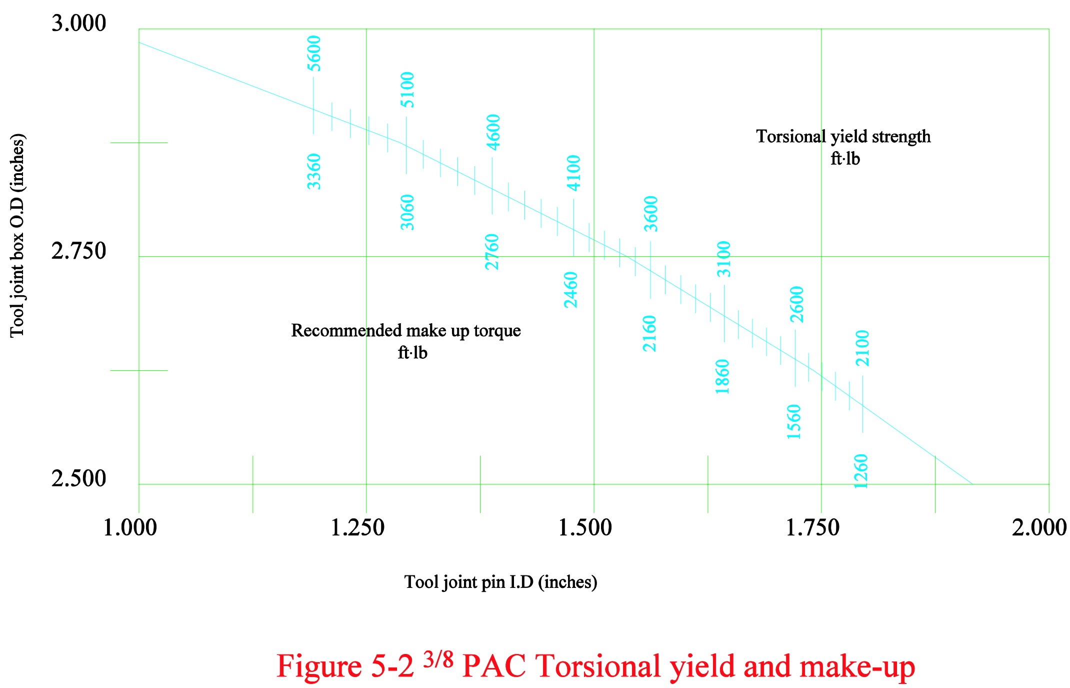


Figure D.5 — 2 3/8 PAC Torsional yield and make-up

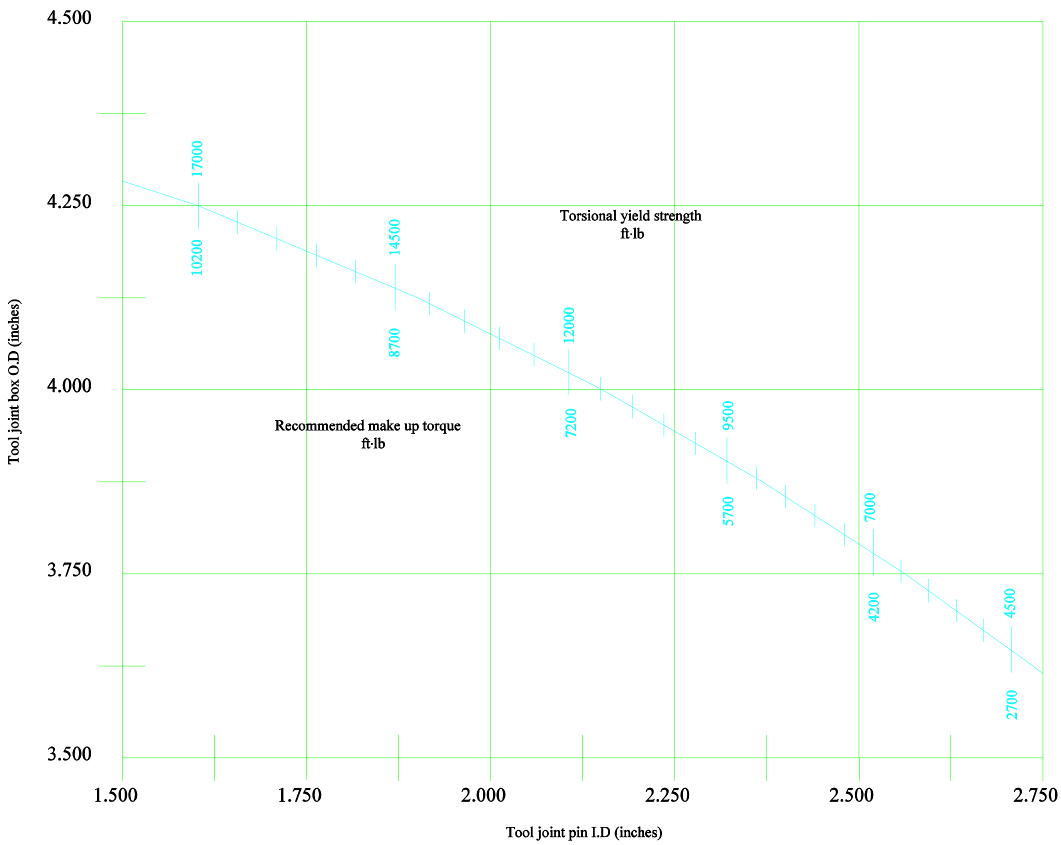


Figure 6-NC31 Torsional yield and make-up

Figure D.6 — NC31 Torsional yield and make-up

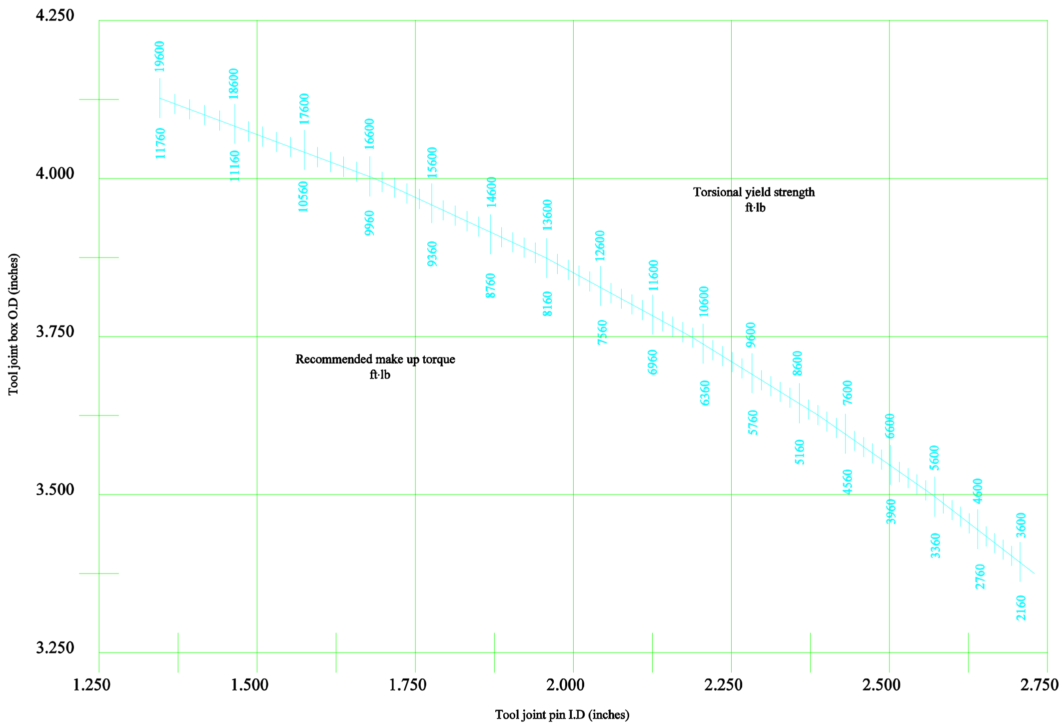


Figure 7-2 7/8 SLH90 Torsional yield and make-up

Figure D.7 — 2 7/8 SLH90 Torsional yield and make-up

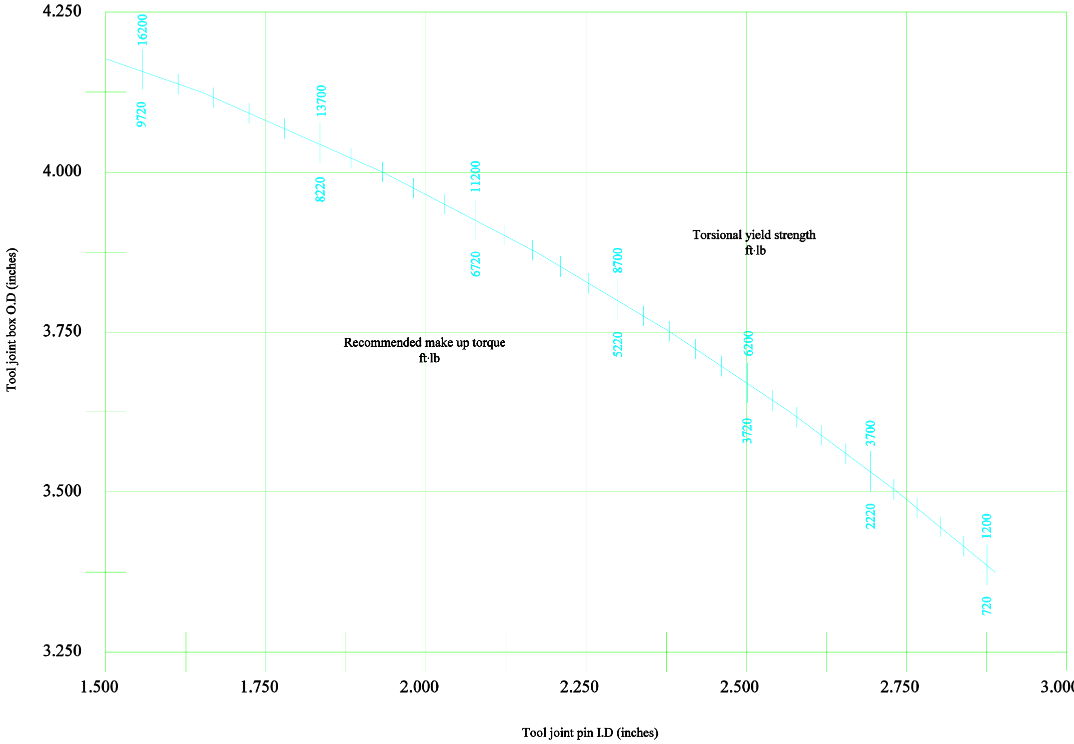


Figure 8-2 7/8 Wide open torsional yield and make-up

Figure D.8 — 2 7/8 Wide open torsional yield and make-up

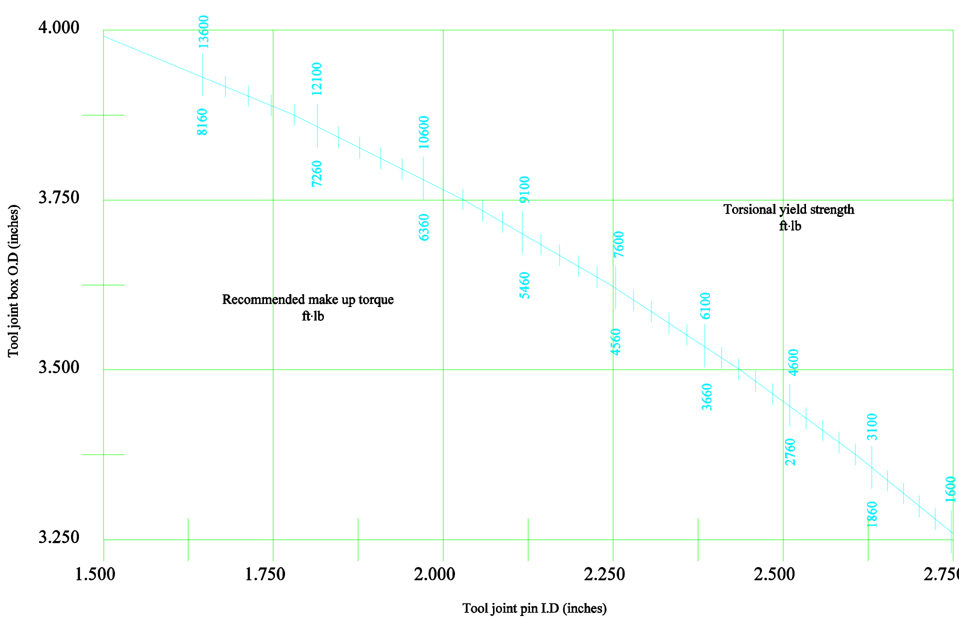


Figure 9-2 7/8 Open hole torsional yield and make-up

Figure D.9 — 2 7/8 Open hole torsional yield and make-up

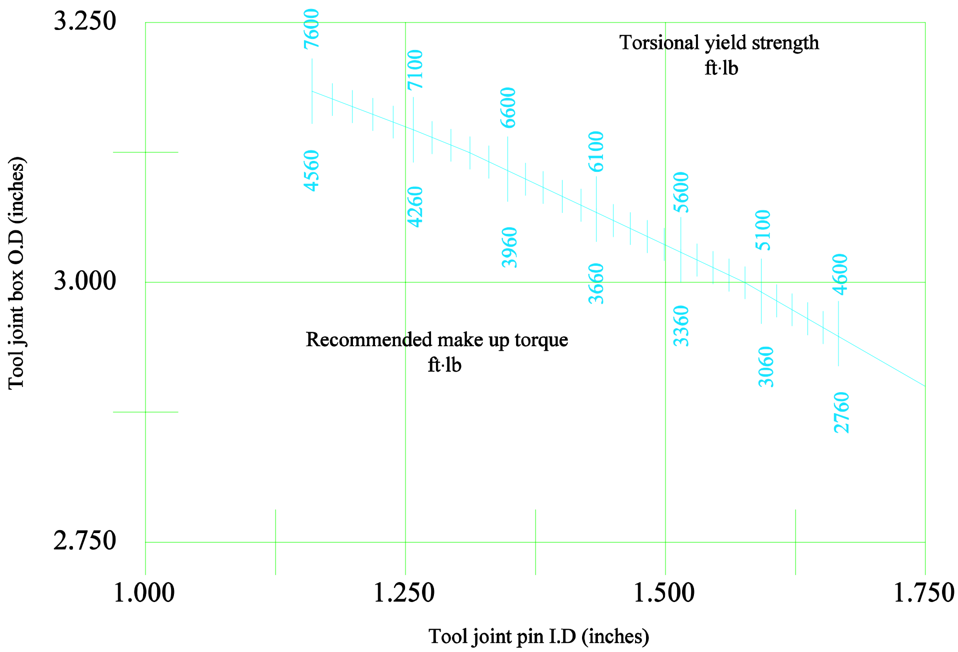


Figure 10-2 7/8 PAC torsional yield and make-up

Figure D.10 — 2 7/8 PAC torsional yield and make-up

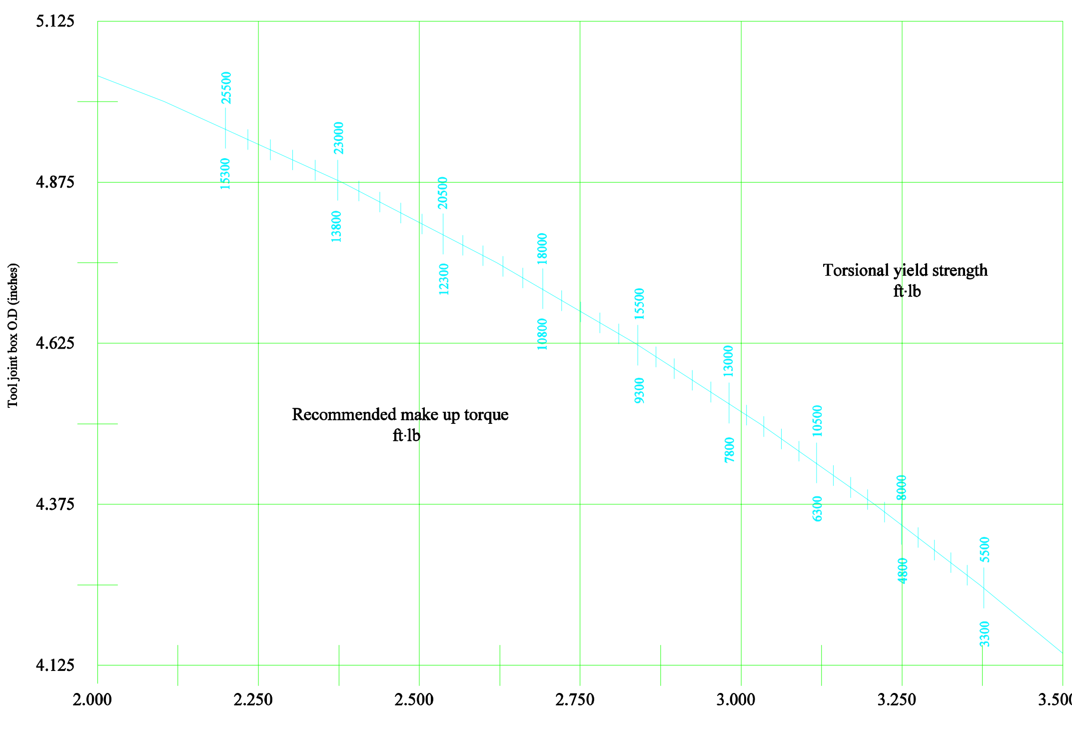


Figure 11-NC38 Torsional yield and make-up

Figure D.11 — NC38 torsional yield and make-up

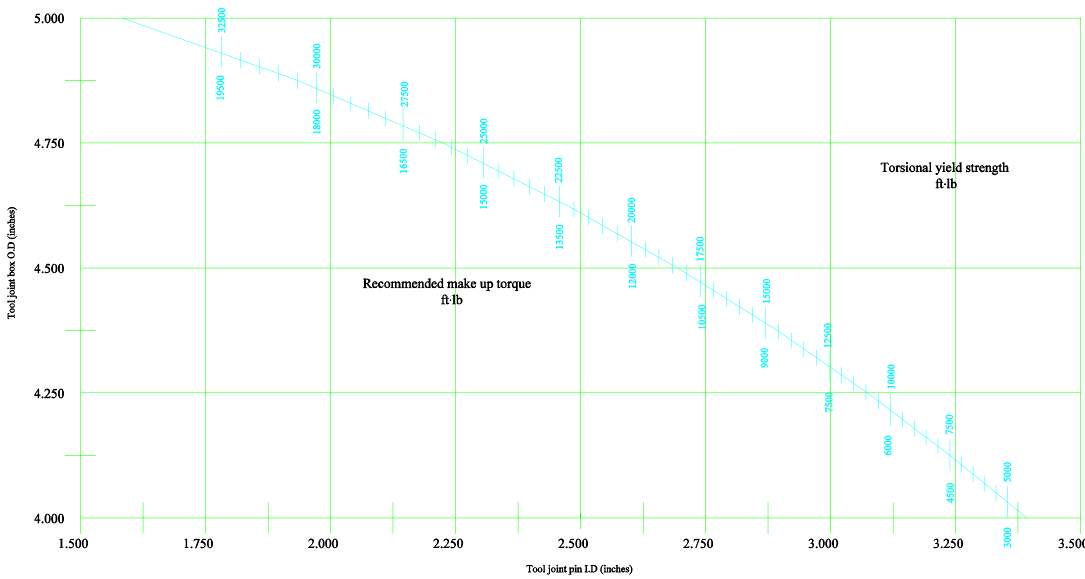


Figure 12-3 1/2 SLH90 Torsional yield and make-up

Figure D.12 — 3 ½ SLH90 torsional yield and make-up

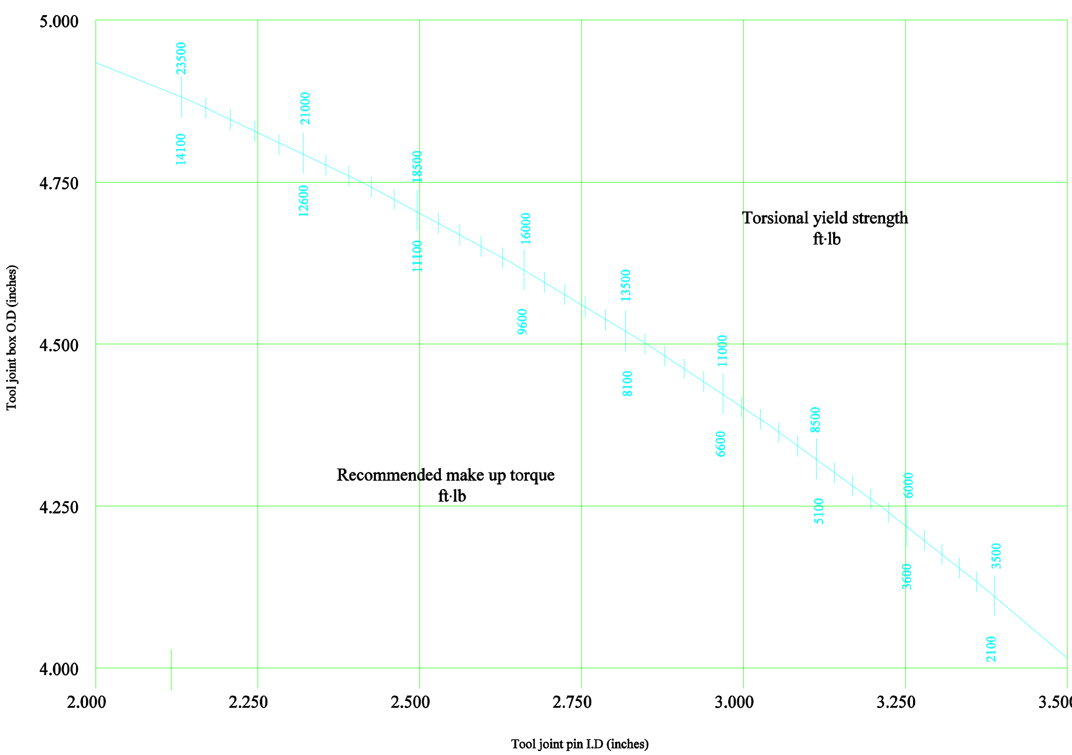


Figure 13-3 1/2 FH Torsional yield and make-up

Figure D.13 — 3 ½ FH torsional yield and make-up

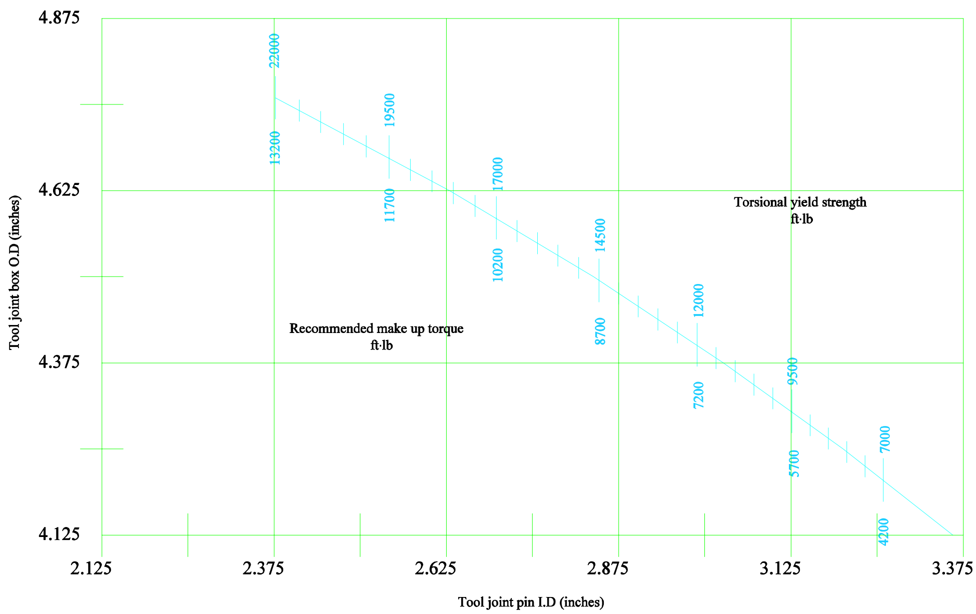


Figure 14-3 1/2 Open hole torsional yield and make-up

Figure D.14 — 3 ½ Open hole torsional yield and make-up

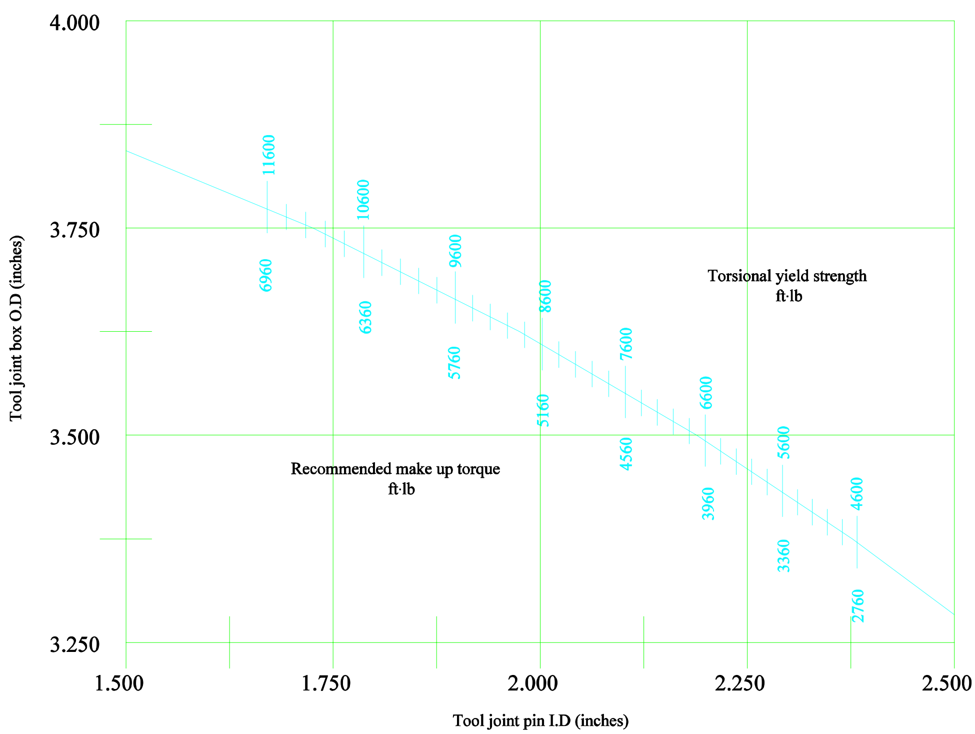


Figure 15-3 1/2 PAC Torsional yield and make-up

Figure D.15 — 3 ½ PAC torsional yield and make-up

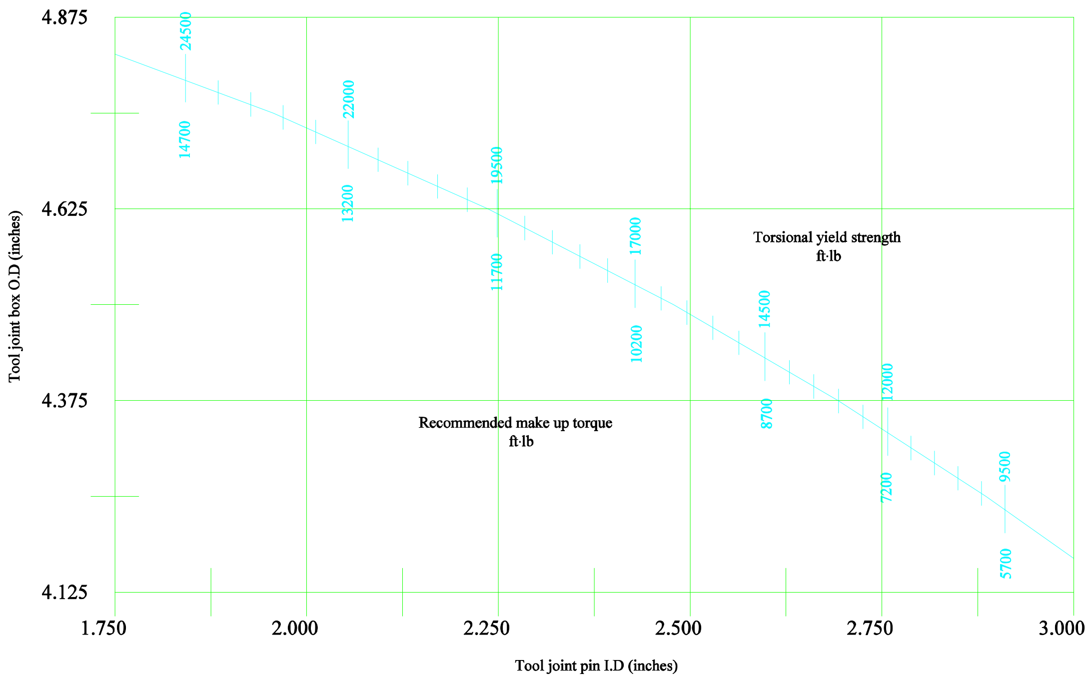


Figure 16-3 1/2 XH Torsional yield and make-up

Figure D.16 — 3 ½ XH torsional yield and make-up

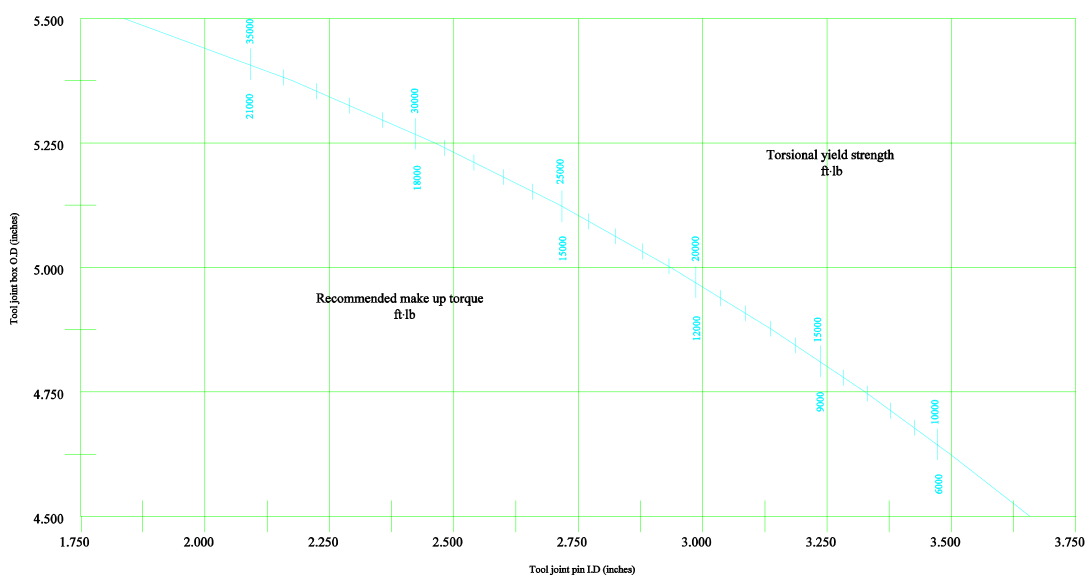


Figure 17-NC40 Torsional yield and make-up

Figure D.17 — NC40 torsional yield and make-up

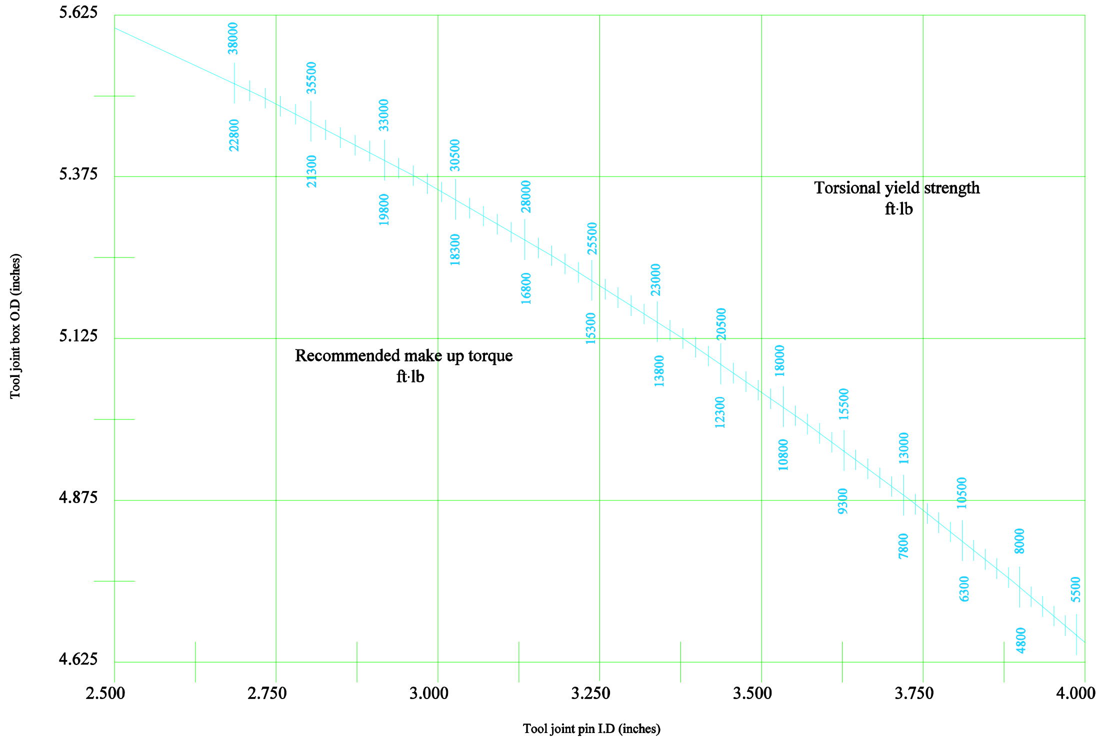


Figure 18-4-inch H90 torsional yield and make-up

Figure D.18 — 4-inch H90 torsional yield and make-up

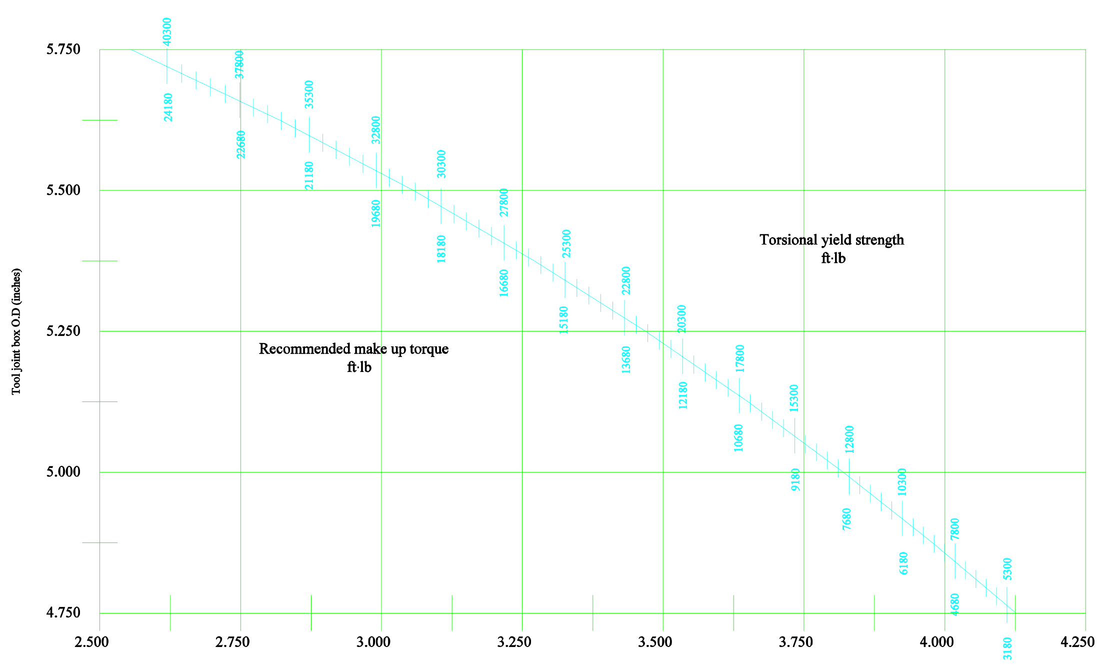


Figure 19-4-Inch open hole torsional yield and make-up

Figure D.19 — 4-inch open hole torsional yield and make-up

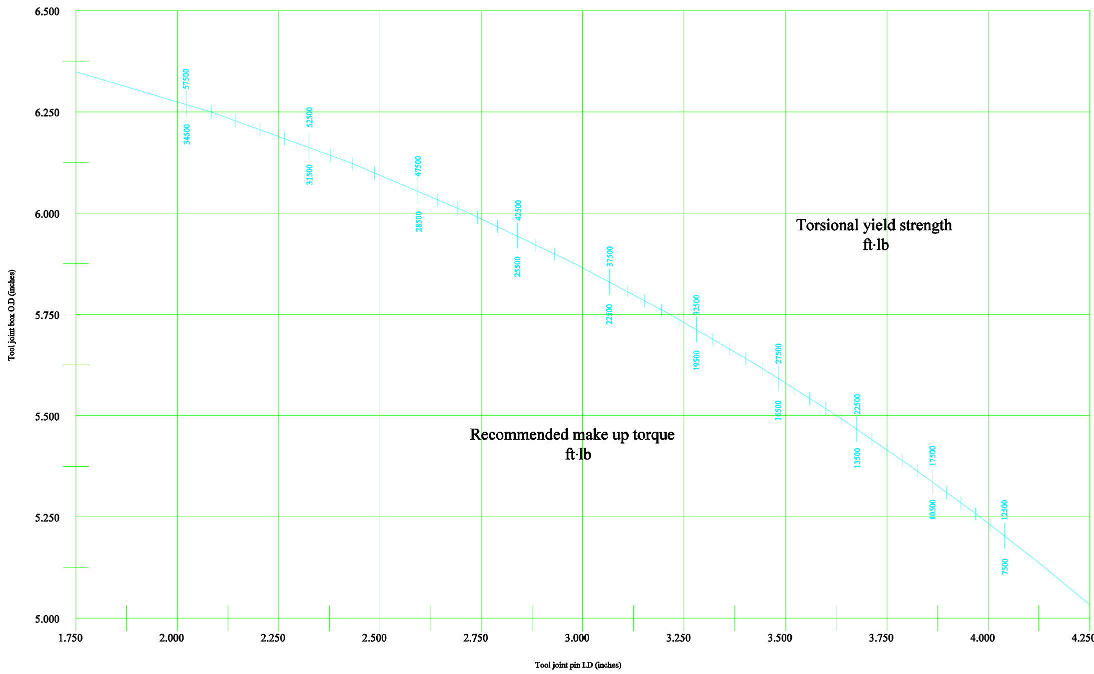


Figure 20-NC46 Torsional yield and make-up

Figure D.20 — NC46 torsional yield and make-up

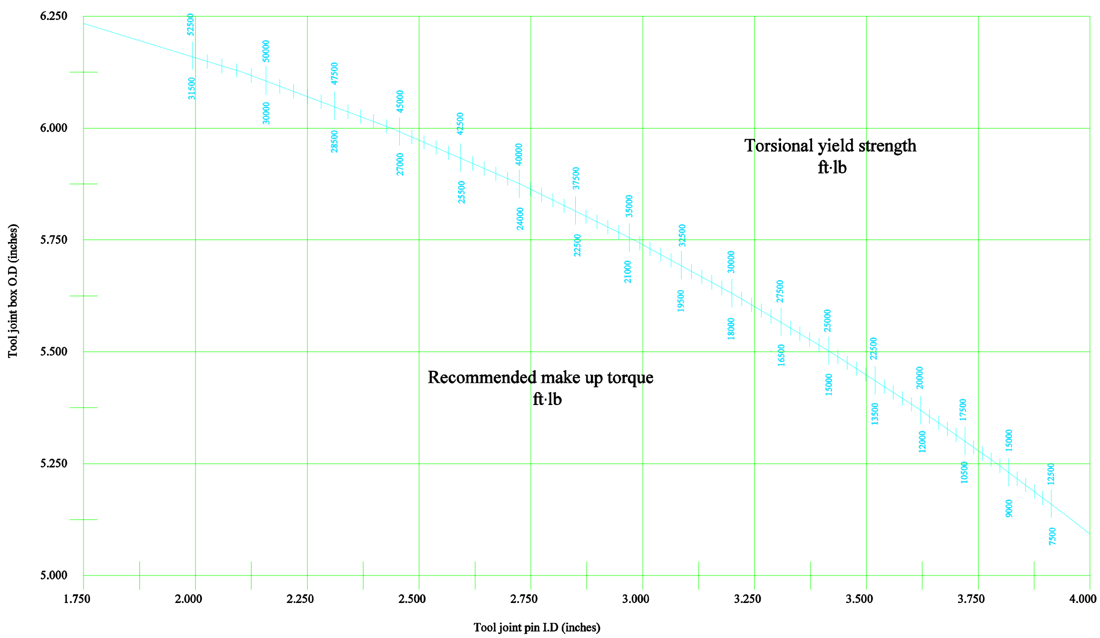


Figure 21-4 1/2 FH torsional yield and make-up

Figure D.21 — 4 ½ FH torsional yield and make-up

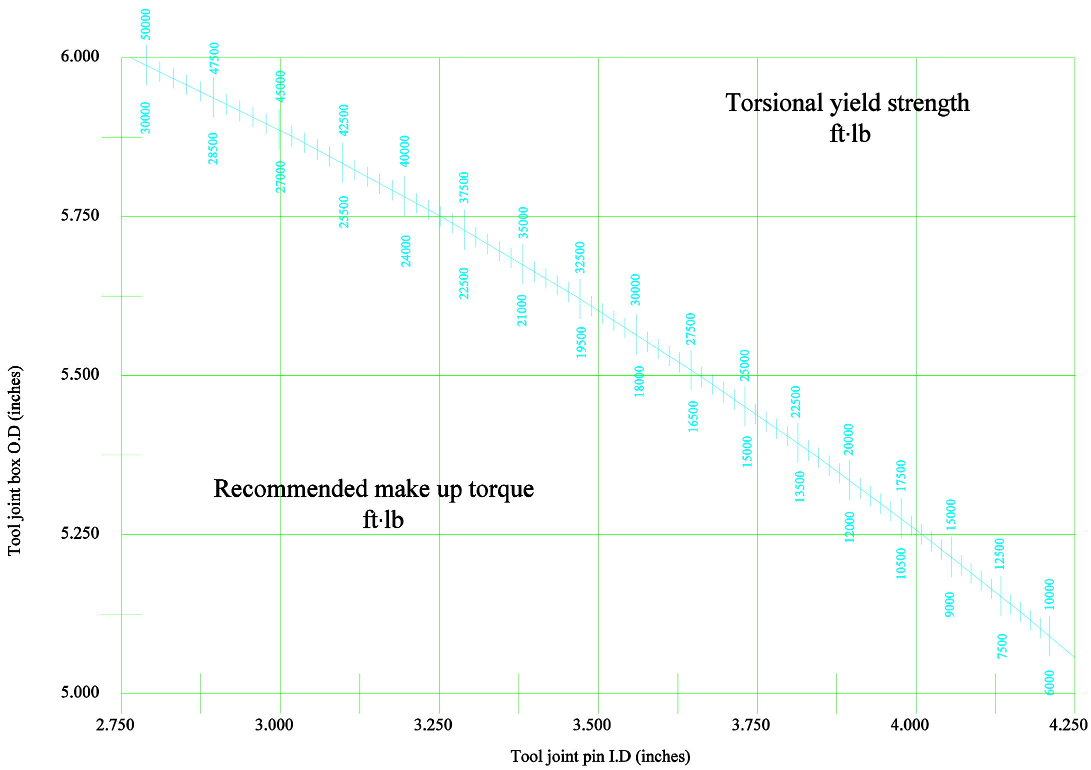


Figure 22-4 1/2 H90 torsional yield and make-up

Figure D.22 — 4 ½ H90 torsional yield and make-up

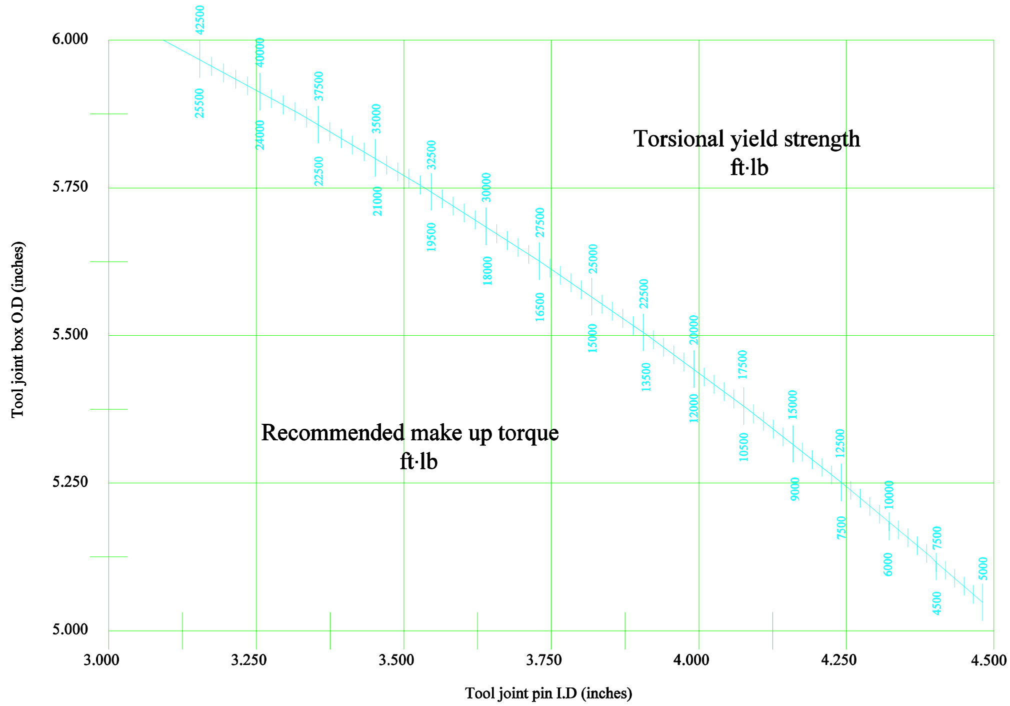


Figure 23-4 1/2 Open hole (standard weight) torsional yield and make-up

Figure D.23 — 4 ½ Open hole (standard weight) torsional yield and make-up

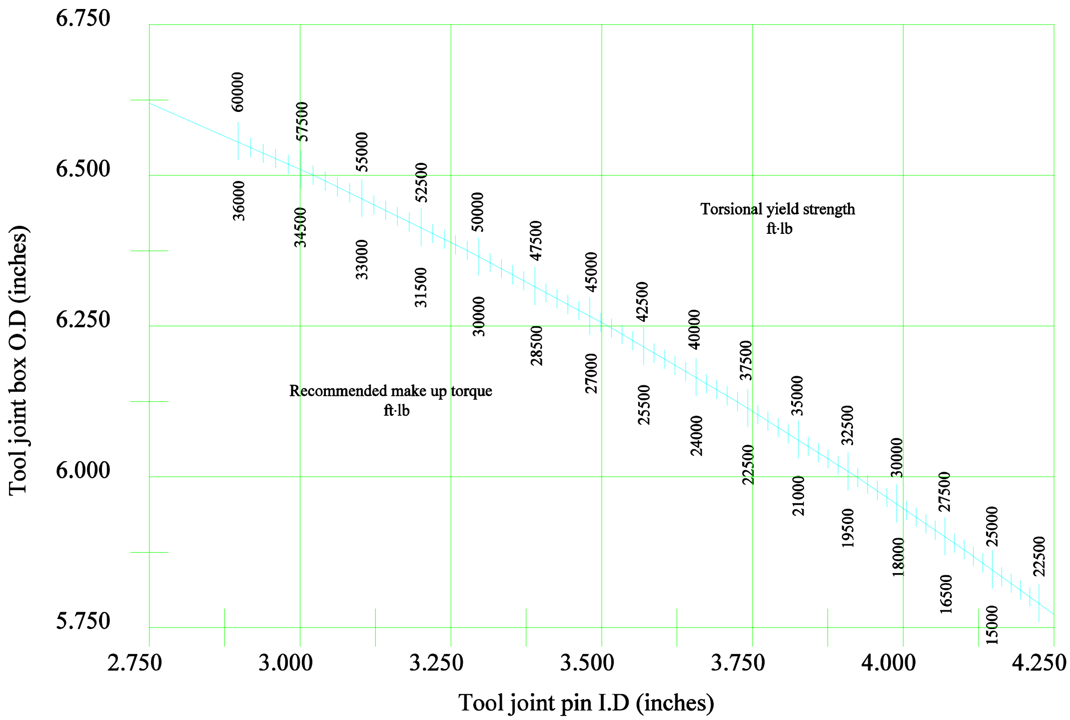


Figure 24-NC50 Torsional yield and make-up

Figure D.24 — NC50 torsional yield and make-up

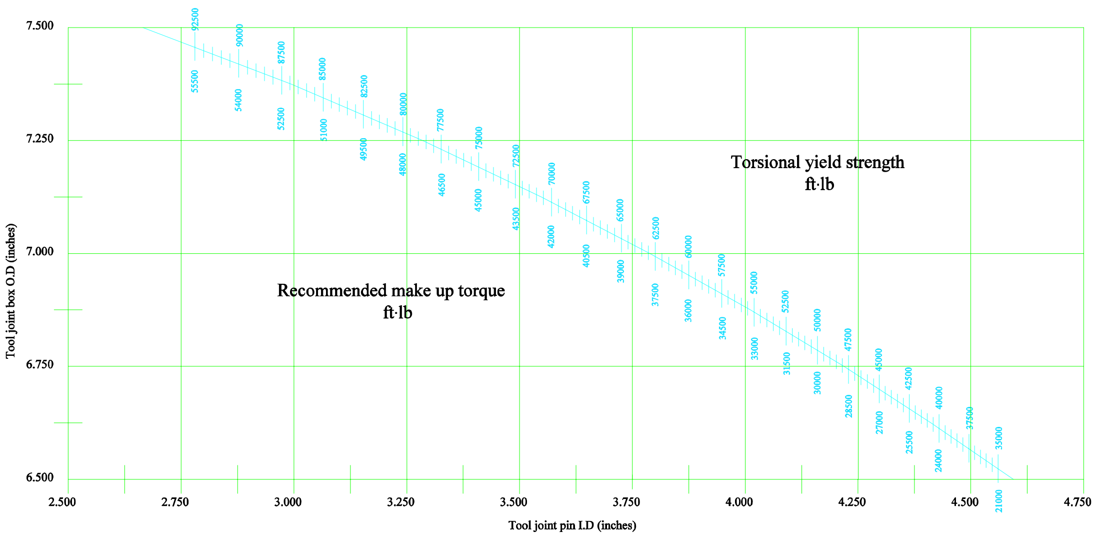
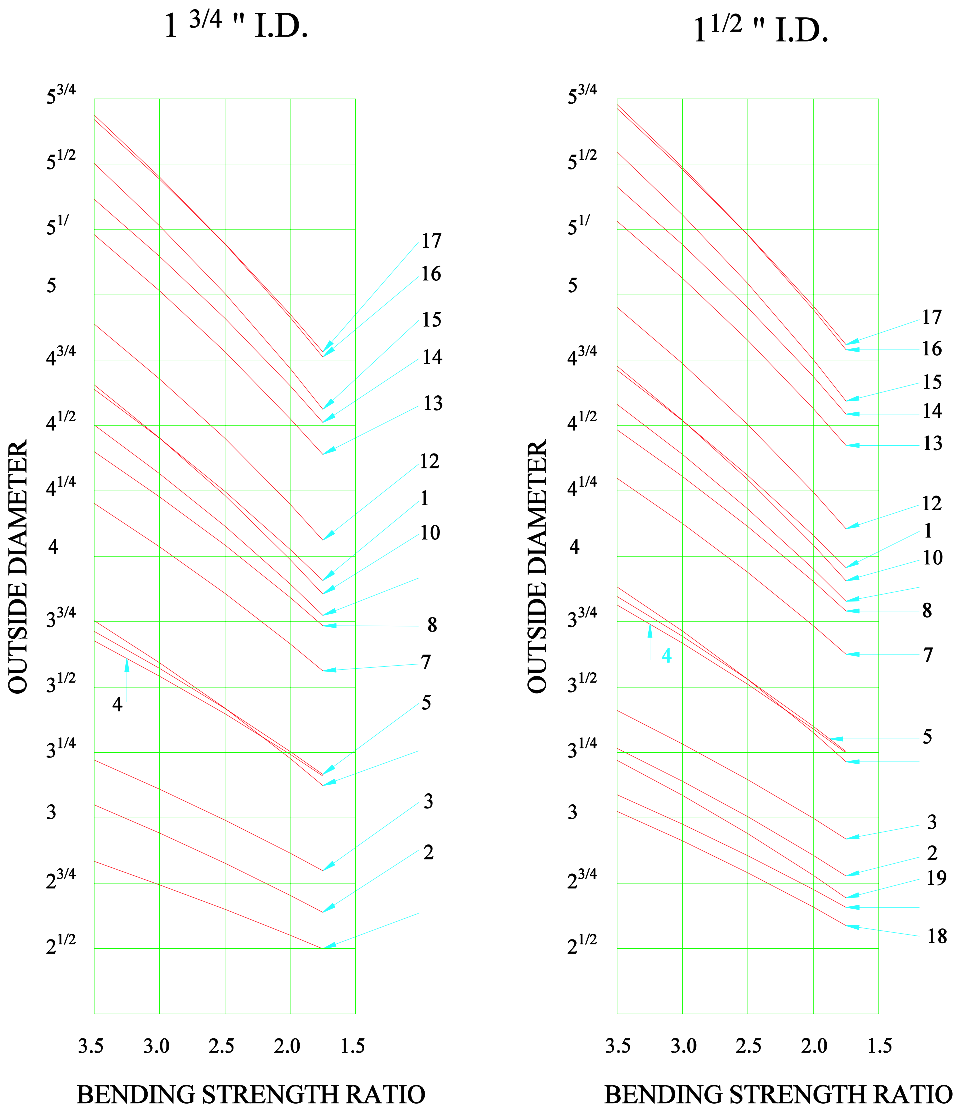


Figure 25-5 1/2 FH torsional yield and make-up

Figure D.25 — 5 ½ FH torsional yield and make-up

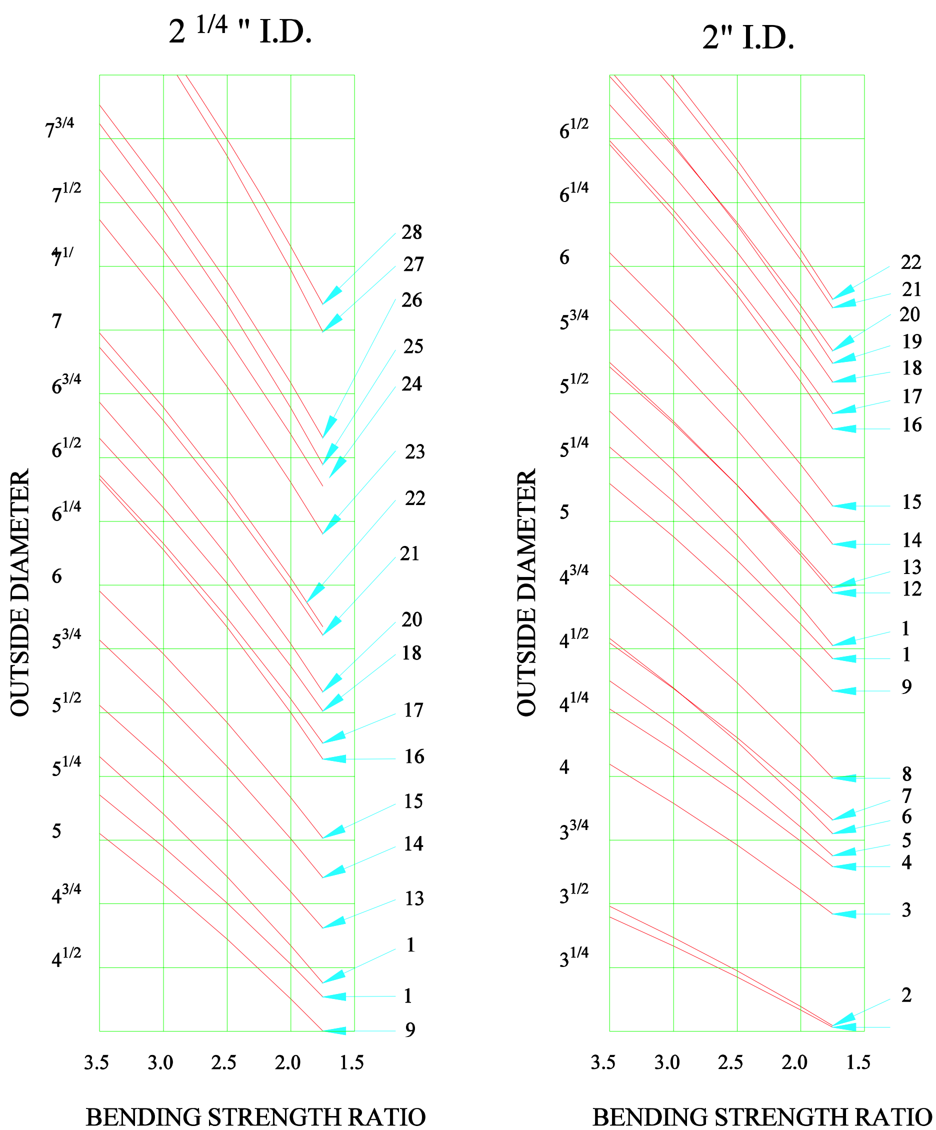


KEY:

1 2 3/8 PAC	7 3 1/2 PAC	13 NC-35
2 NC-23	8 2 7/8 OH	14 3 1/2 EH
3 2 7/8 PAC	9 2 7/8 EH	15 3 1/2 FH
4 2 3/8 OH	10 3 1/2 REG	16 NC-38
5 2 7/8 REG	11 NC-31	17 3 1/2 OH
6 NC-26	12 2 7/8 FH	18 2 3/8 SH
		19 2 3/8 REG

Figure 26-Drill collar bending strength ratios, $1^{1/2}$ -and $1\ ^{3/4}$-inch ID

Figure D.26 — Drill collar bending strength ratios, 1 ½-in and 1 ¾-in ID

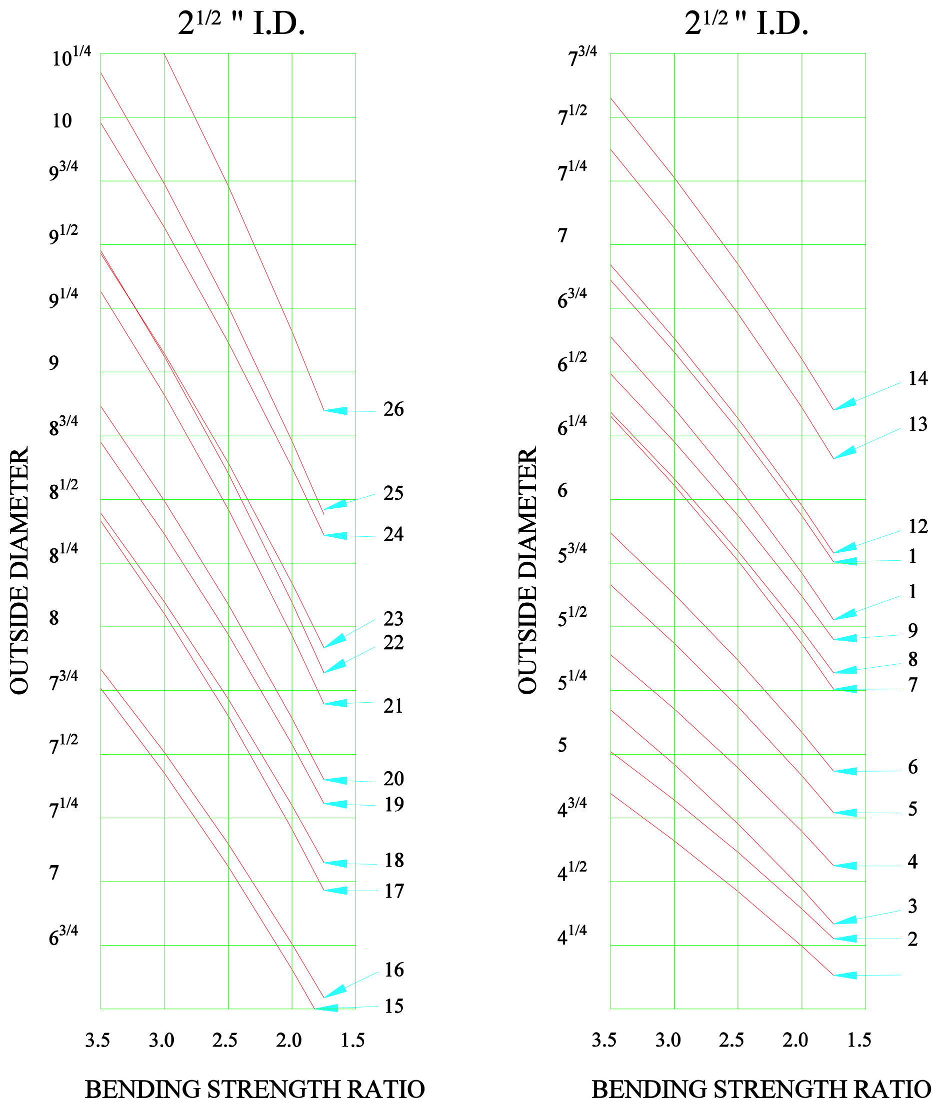


KEY:

1 2 3/8 OH	8 2 7/8 FH	15 NC-40	22 4 1/2 H90
2 NC-26	9 NC-35	16 4 1/2 REG	23 5 H90
3 3 1/2 PAC	10 3 1/2 EH	17 4 H90	24 NC-50
4 2 7/8 OH	11 3 1/2 FH	18 NC-44	25 5 1/2 REG
5 2 7/8 XH	12 3 1/2 OH	19 4 OH	26 5 1/2 H90
6 3 1/2 REG	13 NC-38	20 4 1/2 FH	27 NC-56
7 NC-31	14 3 1/2 H90	21 NC-46	28 5 1/2 FH

Figure 27-Drill collar bending strength ratios, 2-and 2 1/4-inch ID

Figure D.27 — Drill collar bending strength ratios, 2-in and 2 ¼-in ID

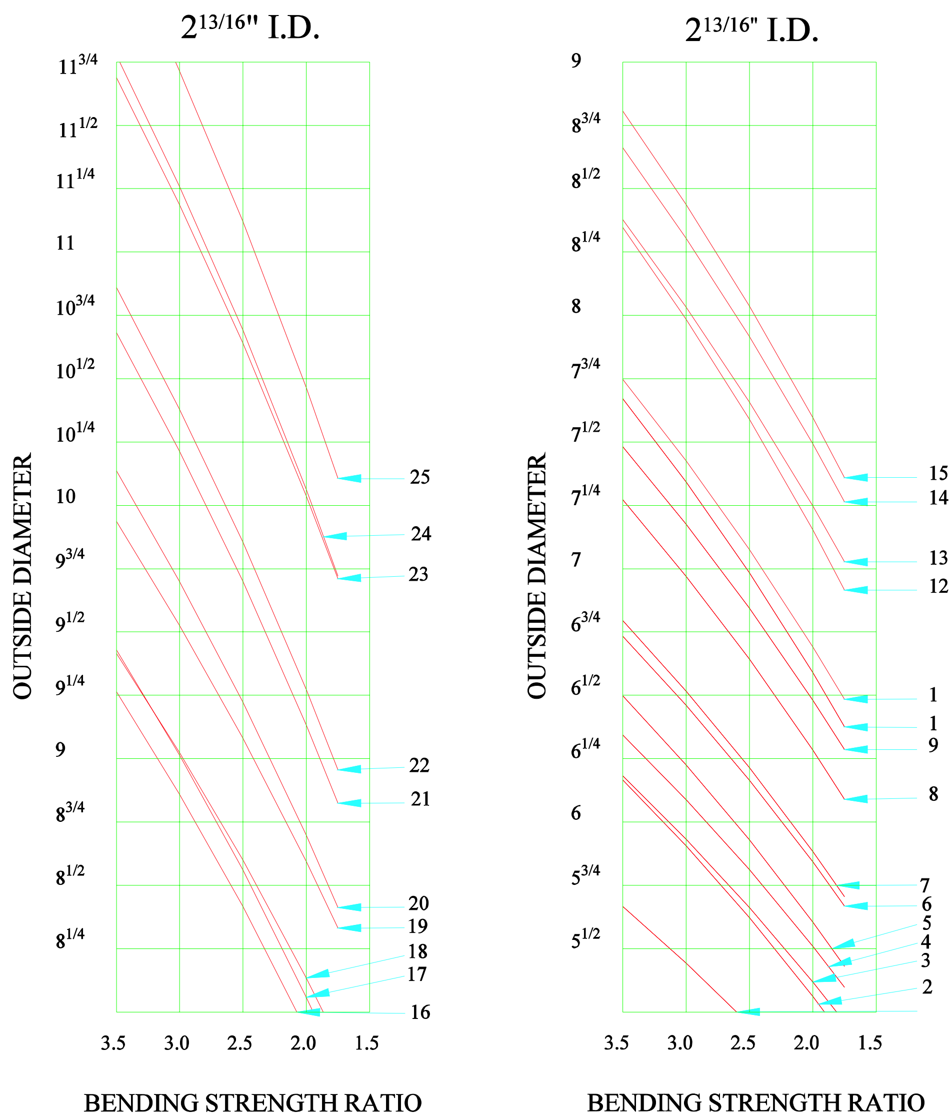


KEY:

1 NC-35	8 4 H90	15 $5^{1/2}$REG	22 7 H90
2 $3^{1/2}$EH	9 NC-44	16 $5^{1/2}$H90	23 $5^{1/2}$IF
3 $3^{1/2}$FH	10 $4^{1/2}$FH	17 NC-56	24 $6^{5/8}$FH
4 NC-38	11 NC-46	18 $5^{1/2}$FH	25 $7^{5/8}$REG
5 $3^{1/2}$H90	12 $4^{1/2}$H90	19 $6^{5/8}$REG	26 NC-70
6 NC-40	13 5 H90	20 $6^{5/8}$H90	
7 $4^{1/2}$REG	14 NC-50	21 NC-61	

Figure 28-Drill collar bending strength ratios, $2^{1/2}$-inch ID

Figure D.28 — Drill collar bending strength ratios, 2 ½-in ID

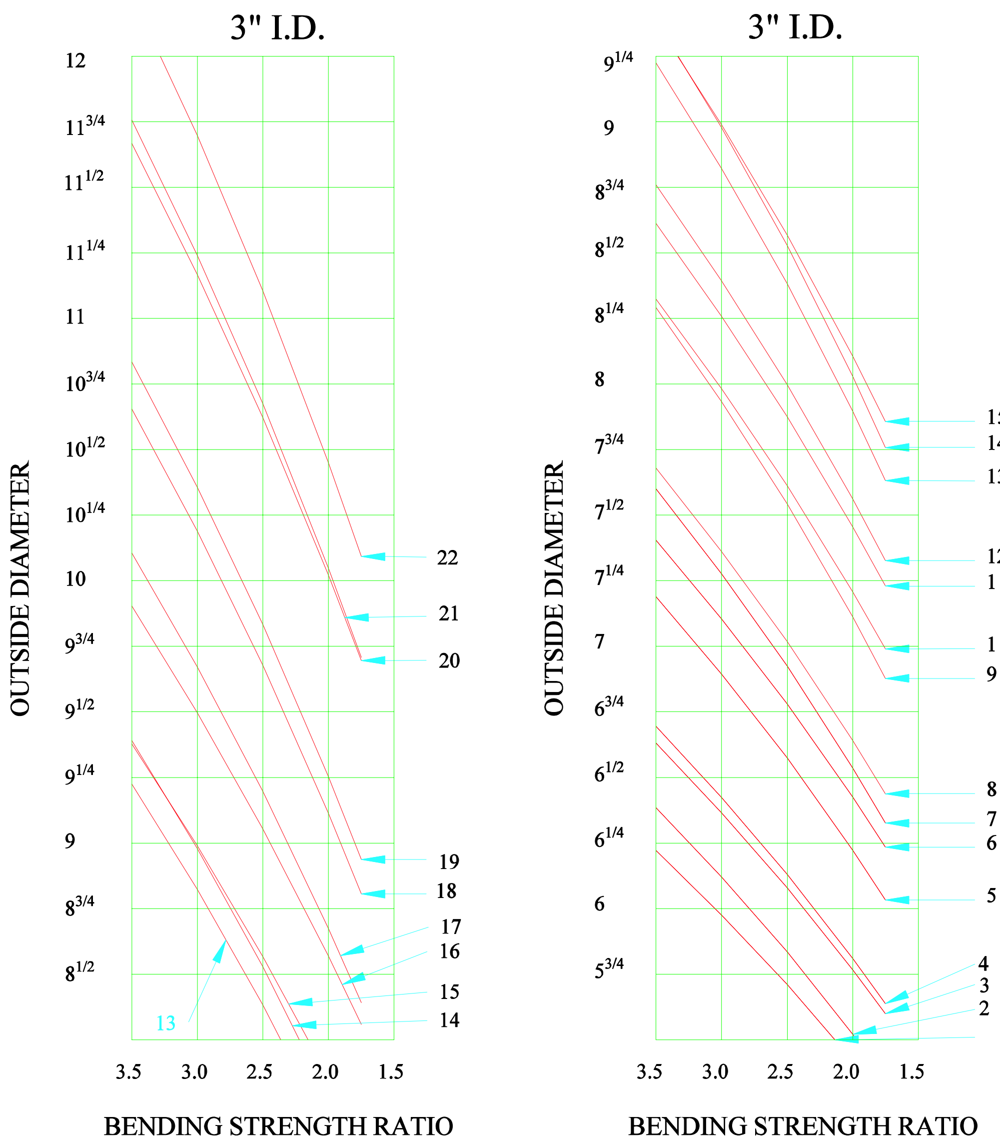


KEY:

1 NC-40	8 5 H90	15 6 5/8H90	22 7 5/8H90
2 4 1/2REG	9 NC-50	16 NC-61	23 8 5/8REG
3 4 H90	10 5 1/2REG	17 7 H90	24 NC-77
4 NC-44	11 5 1/2H90	18 5 1/2IF	25 8 5/8H90
5 4 1/2FH	12 NC-56	19 6 5/8FH	
6 NC-46	13 5 1/2FH	20 7 5/8REG	
7 4 1/2H90	14 6 5/8REG	21 NC-70	

Figure 29-Drill collar bending strength ratios, 2 13/16-inch ID

Figure D.29 — Drill collar bending strength ratios, 2 13/16-in ID

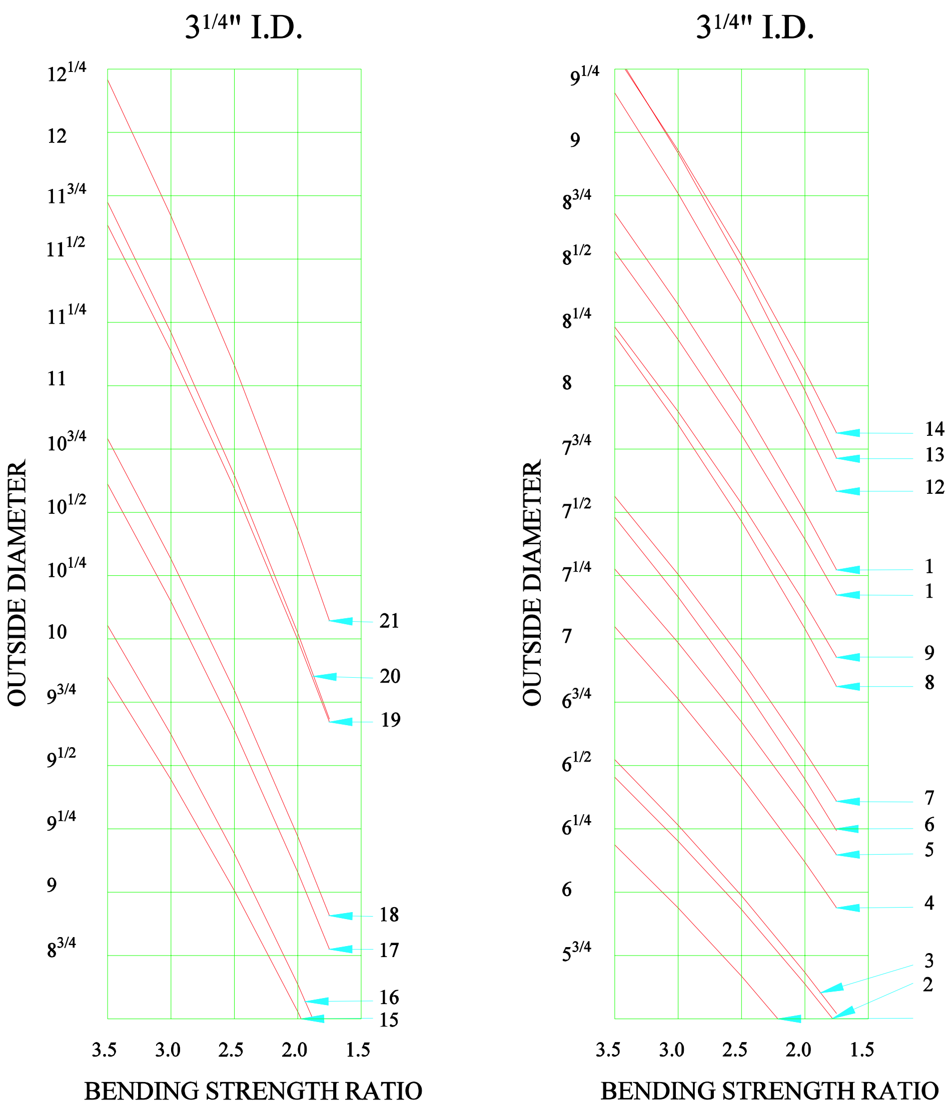


KEY:

1 NC-44
2 $4^{1/2}$FH
3 NC-46
4 $4^{1/2}$H90
5 5 H90
6 NC-50
7 $5^{1/2}$REG
8 $5^{1/2}$ H90
9 NC-56
10 $5^{1/2}$FH
11 $6^{5/8}$REG
12 $6^{5/8}$H90
13 NC-61
14 7 H90
15 $5^{1/2}$IF
16 $6^{5/8}$FH
17 $7^{5/8}$REG
18 NC-70
19 $7^{5/8}$H90
20 $8^{5/8}$REG
21 NC-77
22 $8^{5/8}$H90

Figure 30-Drill collar bending strength ratios, 3-inch ID

Figure D.30 — Drill collar bending strength ratios, 3-in ID

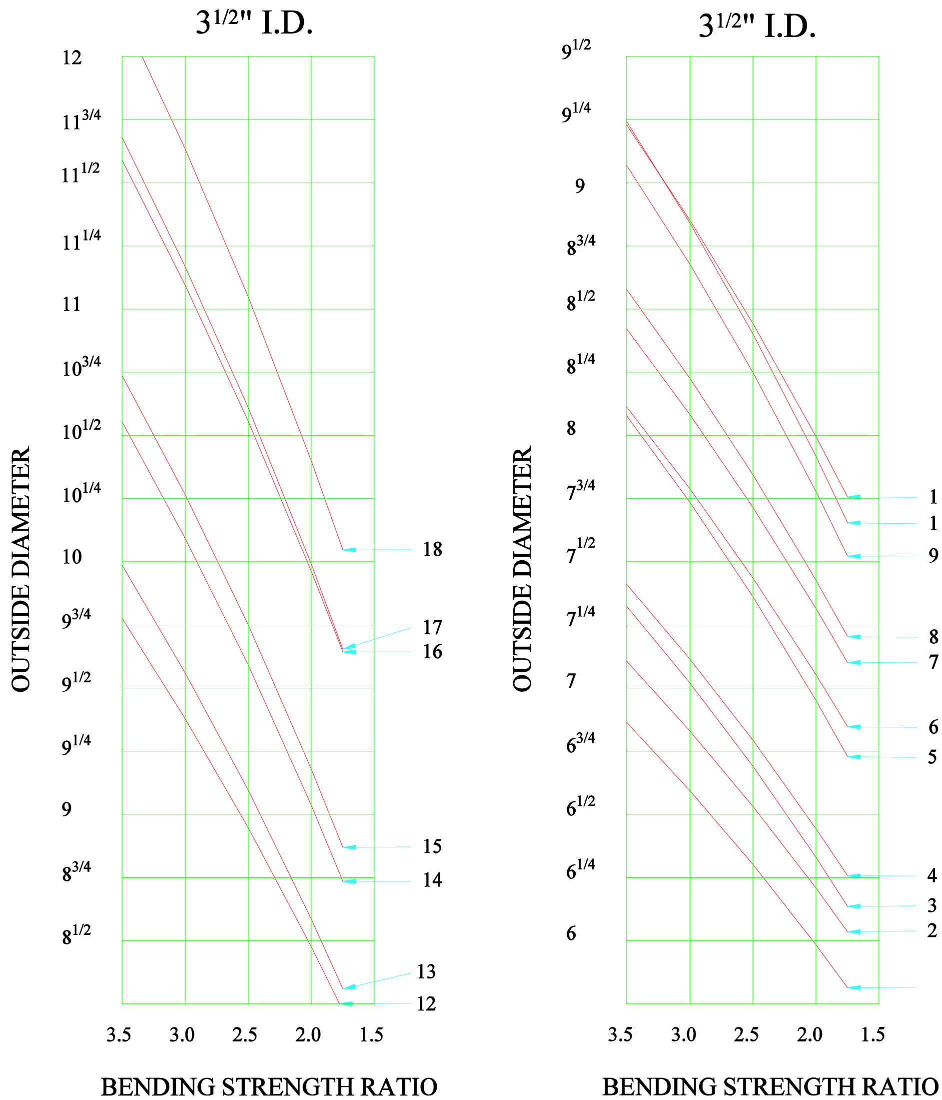


KEY:

1 $4^{1/2}$FH	8 NC-56	15 $6^{5/8}$FH
2 NC-46	9 $5^{1/2}$FH	16 $7^{5/8}$REG
3 $4^{1/2}$H90	10 $6^{5/8}$REG	17 NC-70
4 5 H90	11 $6^{5/8}$H90	18 $7^{5/8}$H90
5 NC-50	12 NC-61	19 $8^{5/8}$REG
6 $5^{1/2}$REG	13 7 H90	20 NC-77
7 $5^{1/2}$H90	14 $5^{1/2}$IF	21 $8^{5/8}$H90

Figure 31-Drill collar bending strength ratios, $3^{1/4}$-inch ID

Figure D.31 — Drill collar bending strength ratios, 3 ¼-in ID



KEY:

1 5 H90
2 NC-50
3 $5^{1/2}$REG
4 $5^{1/2}$H90
5 NC-56
6 $5^{1/2}$FH
7 $6^{5/8}$REG
8 $6^{5/8}$H90
9 NC-61
10 7 H90
11 $5^{1/2}$IF
12 $6^{5/8}$FH
13 $7^{5/8}$REG
14 NC-70
15 $7^{5/8}$H90
16 $8^{5/8}$REG
17 NC-77
18 $8^{5/8}$H90

Figure 32-Drill collar bending strength ratios, $3^{1/2}$-inch ID

Figure D.32 — Drill collar bending strength ratios, 3 ½-in ID

Figure D.33 — Delayed-failure characteristics of unnotched specimens of an SAE 4340 steel during cathodic charging with hydrogen under standardized conditions

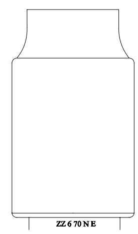


Figure D.34 — API RP 7G Figure 82

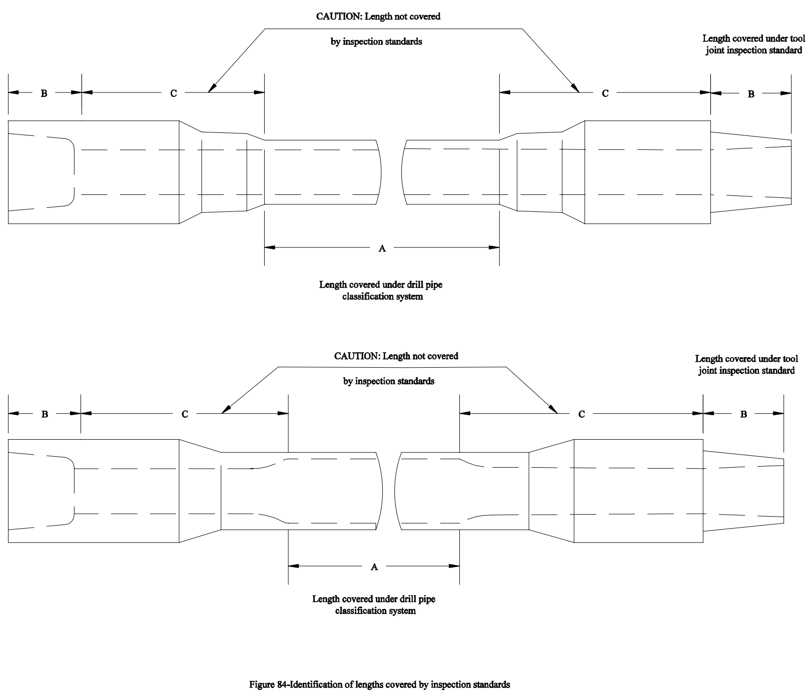


Figure D.35 — Identification of lengths covered by inspection standards

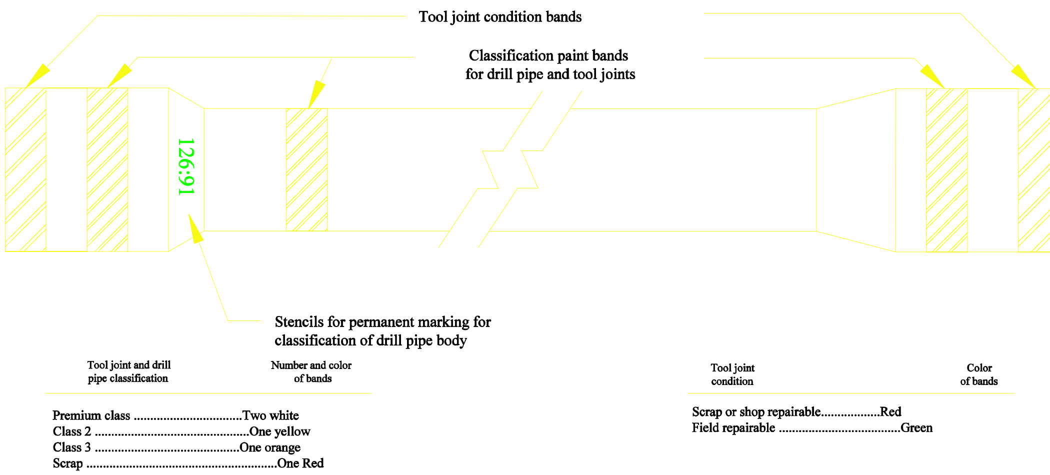


Figure 85-Drill pipe and tool joint color code identification

Figure D.36 — Drill pipe and tool joint color code identification

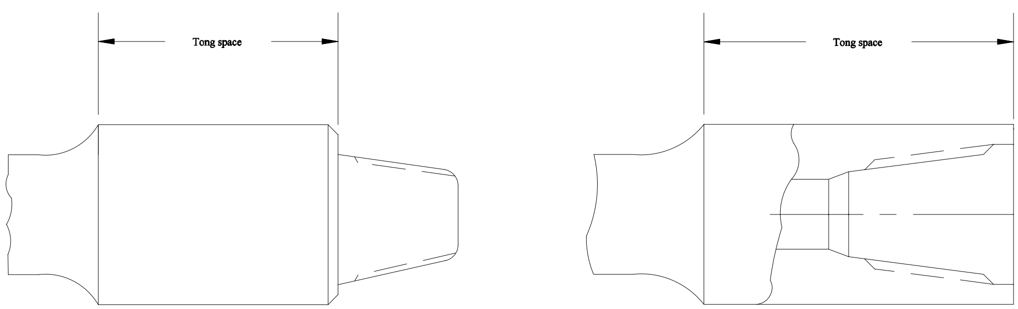


Figure 86-Tong space and bench mark position

Figure D.37 — Tong space and bench mark position

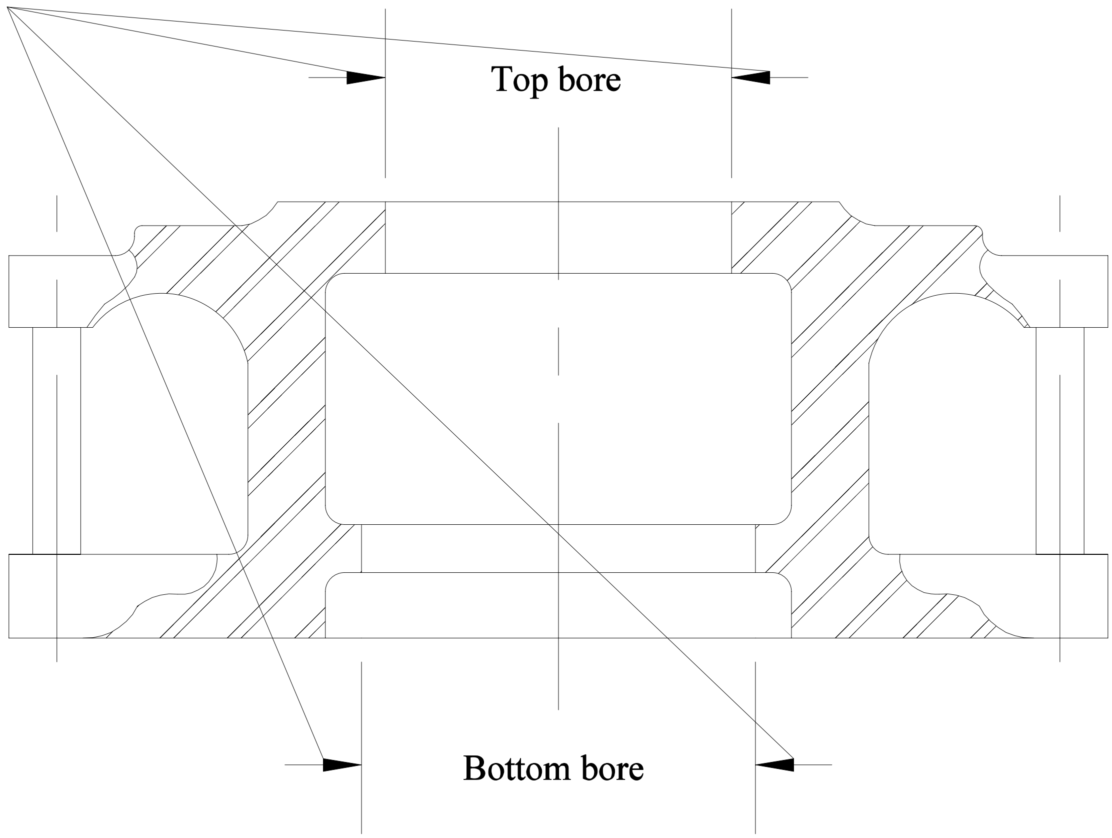


Figure 87-Drill collar elevator

Figure D.38 — Drill collar elevator

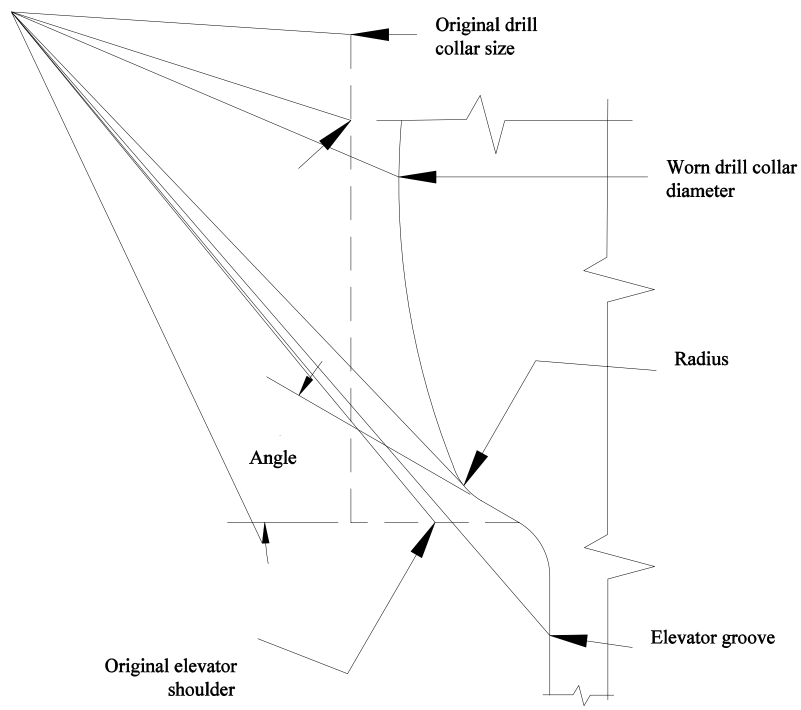


Figure 89-Drill collar wear

Figure D.39 — Drill collar wear

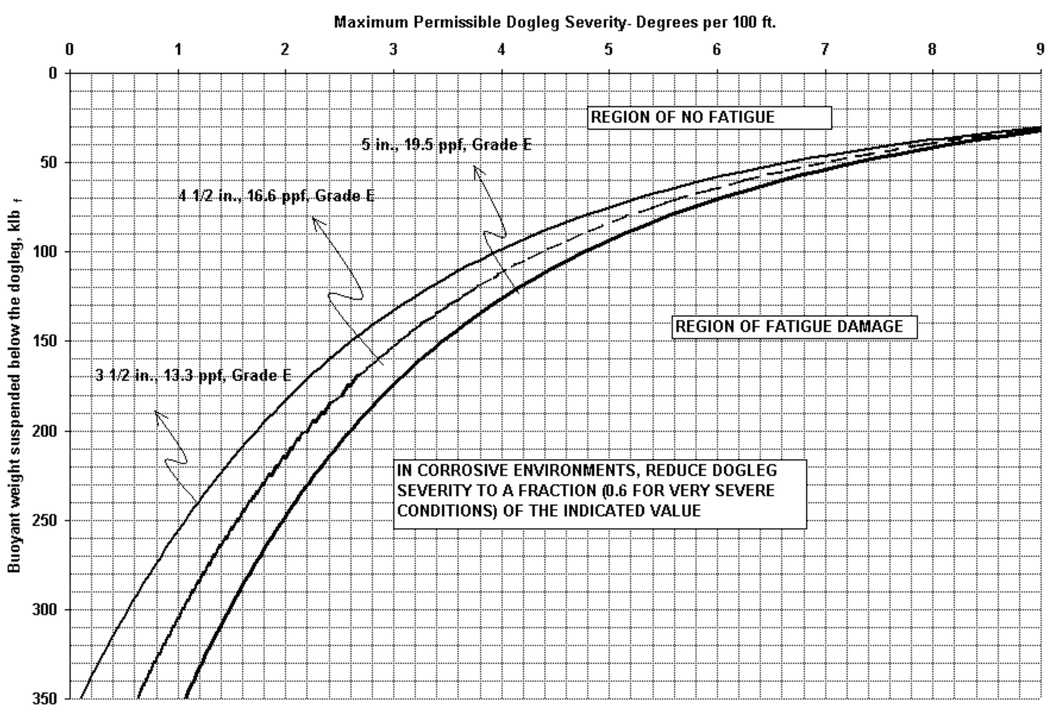


Figure D.40 — Dogleg Severity Limits for Fatigue of Grade E drill Pipe

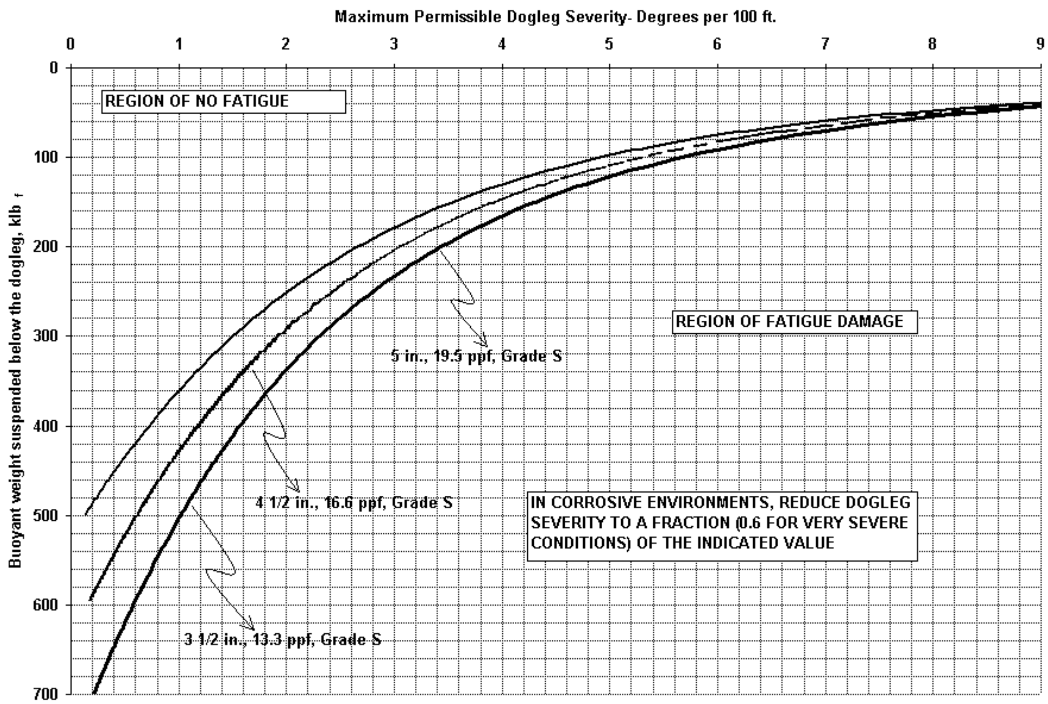


Figure D.41 — Dogleg Severity Limits for Fatigue of S-135 drill Pipe

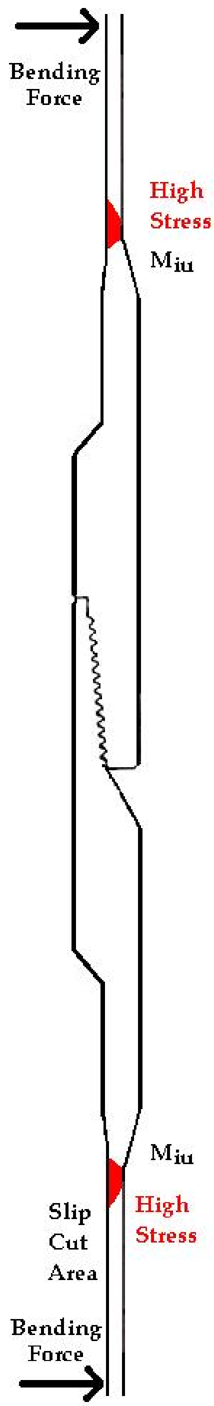


Figure D.42 — DP Cross-Section

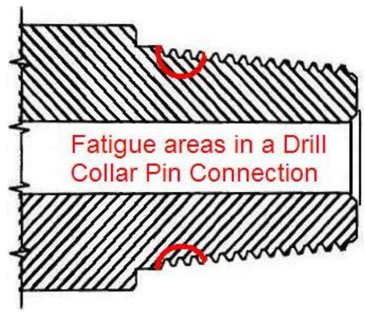


Figure D.43 — DC Pin Connection

Figure D.44 — DC Box Connection

Figure D.45 — Wash out in the slip area

Figure D.46 — CO_2 corrosion on drill pipe OD

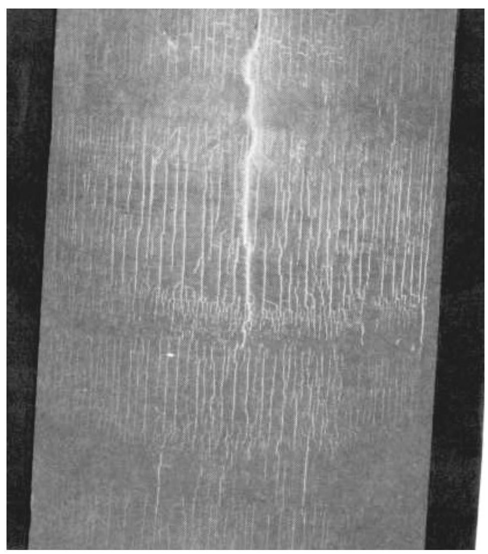
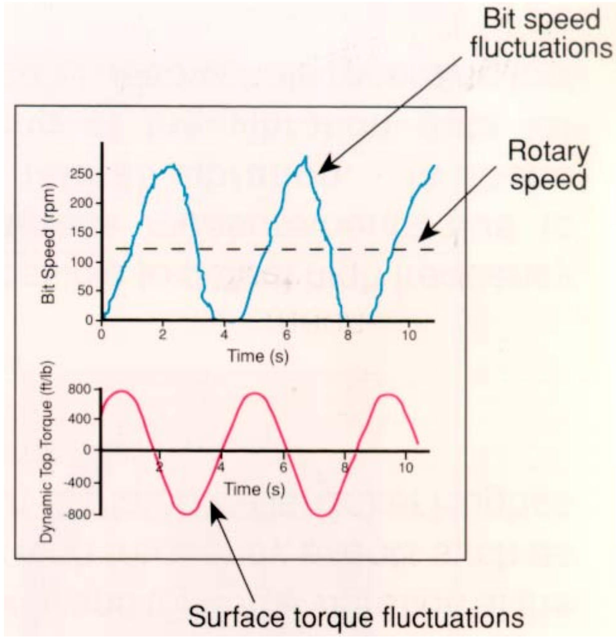


Figure D.47 — Heat Checking on Box Tool Joint Shown Under Black Light

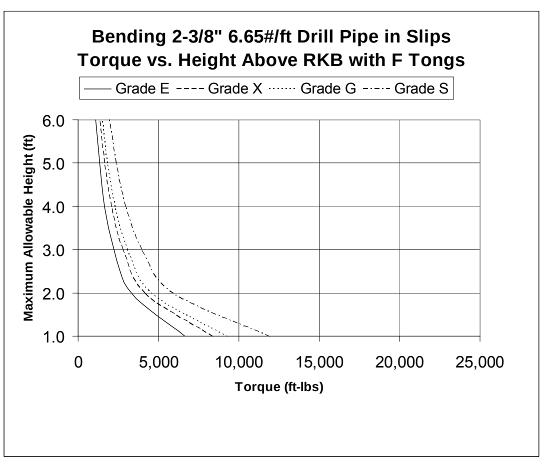


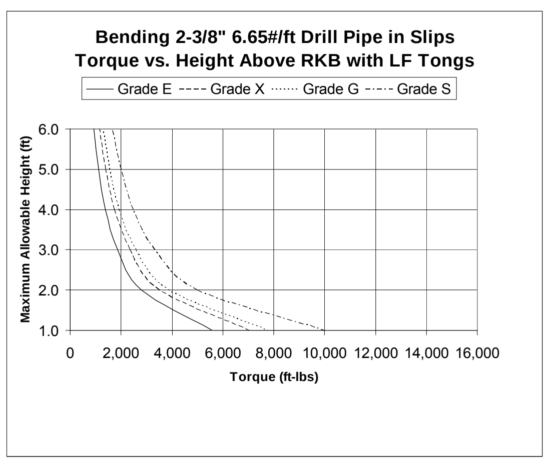
1. Bit speed fluctuations
2. Rotary speed
3. Surface torque fluctuations

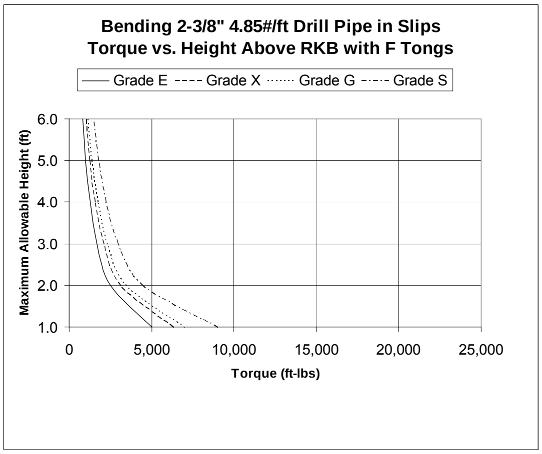
Figure D.48 — Stick-Slip Torsional Vibration

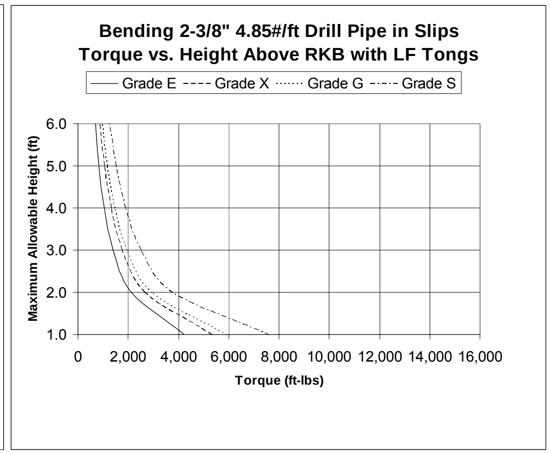
NOTE 1 These graphs are applicable for tong arms which are situated 180º from each other. Multiply the height by 1.4 to get the maximum allowable height if tong arms are 90o apart.

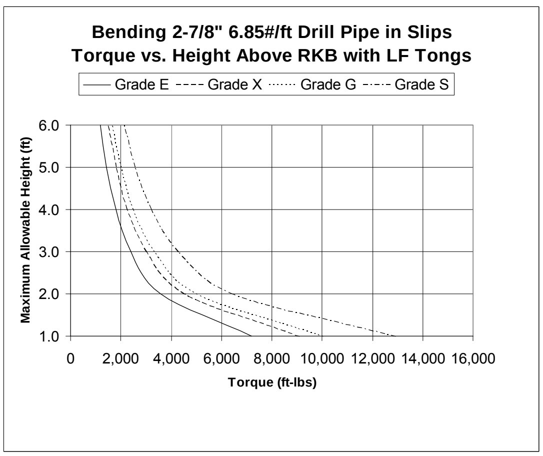
NOTE 2 These values are based on a tong arm lengths designated in Table B.44.

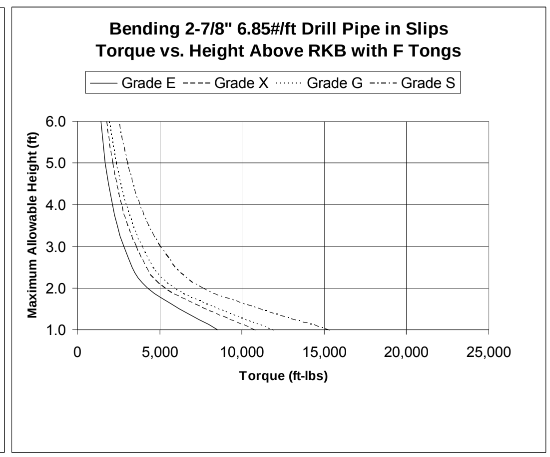


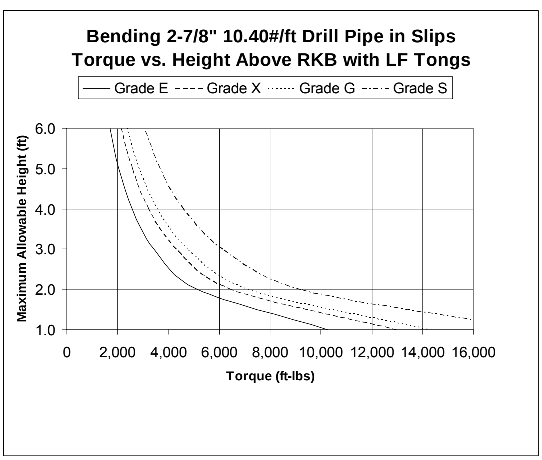


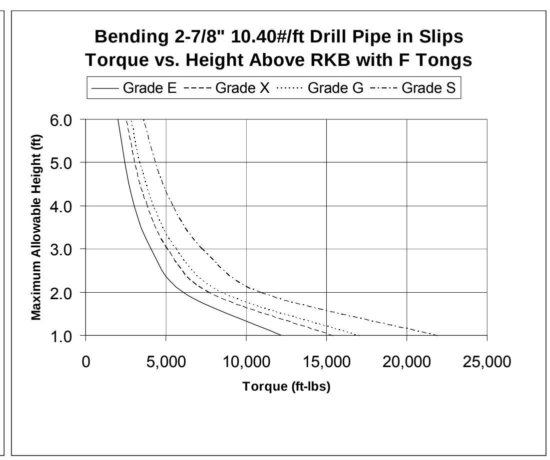

Bending 2-3/8" 4.85#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with F Tongs
Grade E Grade X Grade G Grade S
Maximum Allowable Height (ft)
6.0 5.0 4.0 3.0 2.0 1.0
0 5,000 10,000 15,000 20,000 25,000
Torque (ft-lbs)

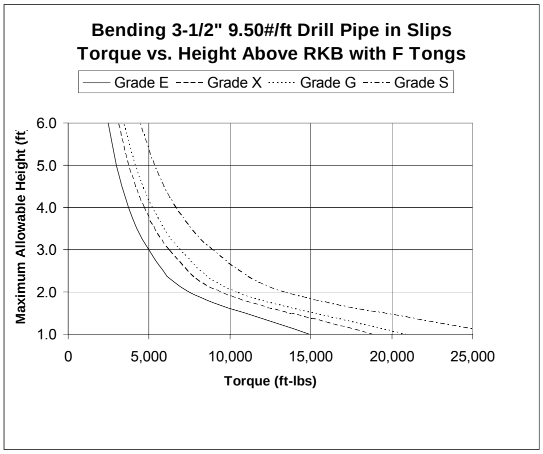

Bending 2-3/8" 4.85#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with LF Tongs
Grade E Grade X Grade G Grade S
Maximum Allowable Height (ft)
6.0 5.0 4.0 3.0 2.0 1.0
0 2,000 4,000 6,000 8,000 10,000 12,000 14,000 16,000
Torque (ft-lbs)

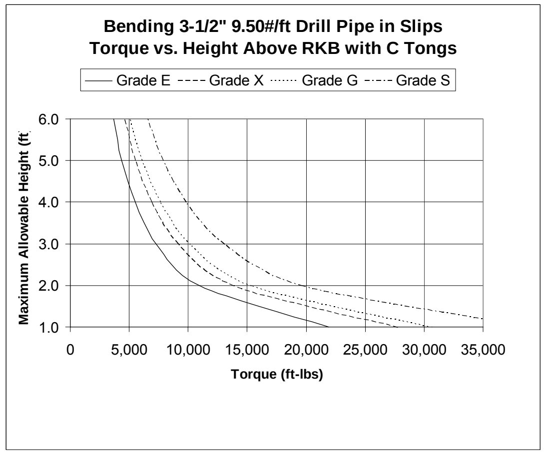

Bending 2-7/8" 6.85#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with LF Tongs
Grade E Grade X Grade G Grade S
Maximum Allowable Height (ft)
6.0 5.0 4.0 3.0 2.0 1.0
0 2,000 4,000 6,000 8,000 10,000 12,000 14,000 16,000
Torque (ft-lbs)


Bending 2-7/8" 6.85#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with F Tongs
Grade E Grade X Grade G Grade S
Maximum Allowable Height (ft)
6.0 5.0 4.0 3.0 2.0 1.0
0 5,000 10,000 15,000 20,000 25,000
Torque (ft-lbs)


Bending 2-7/8" 10.40#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with LF Tongs
Grade E Grade X Grade G Grade S
Maximum Allowable Height (ft)
6.0 5.0 4.0 3.0 2.0 1.0
0 2,000 4,000 6,000 8,000 10,000 12,000 14,000 16,000
Torque (ft-lbs)


Bending 2-7/8" 10.40#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with F Tongs
Grade E Grade X Grade G Grade S
Maximum Allowable Height (ft)
6.0 5.0 4.0 3.0 2.0 1.0
0 5,000 10,000 15,000 20,000 25,000
Torque (ft-lbs)


Bending 3-1/2" 9.50#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with F Tongs
Grade E
Grade X
Grade G
Grade S
Maximum Allowable Height (ft
6.0
5.0
4.0
3.0
2.0
1.0
0
5,000
10,000
15,000
20,000
25,000
Torque (ft-lbs)


Bending 3-1/2" 9.50#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with C Tongs
Grade E
Grade X
Grade G
Grade S
Maximum Allowable Height (ft
6.0
5.0
4.0
3.0
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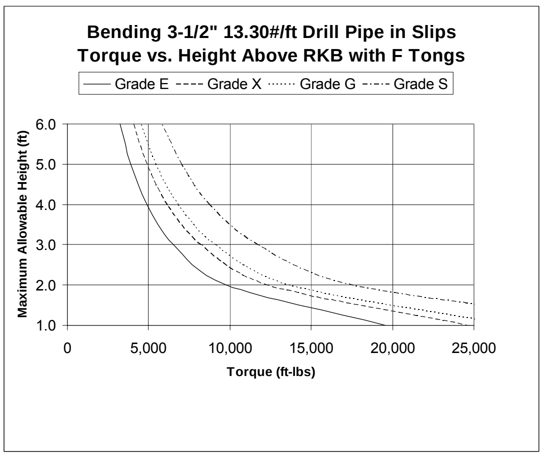
5,000
10,000
15,000
20,000
25,000
30,000
35,000
Torque (ft-lbs)


Bending 3-1/2" 13.30#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with F Tongs
Grade E
Grade X
Grade G
Grade S
Maximum Allowable Height (ft)
6.0
5.0
4.0
3.0
2.0
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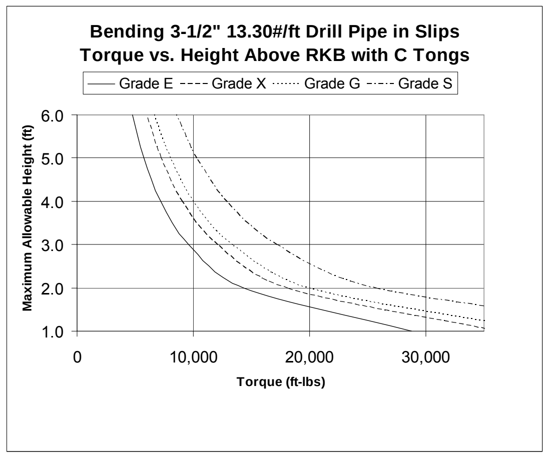
5,000
10,000
15,000
20,000
25,000
Torque (ft-lbs)


Bending 3-1/2" 13.30#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with C Tongs
Grade E
Grade X
Grade G
Grade S
Maximum Allowable Height (ft)
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5.0
4.0
3.0
2.0
1.0
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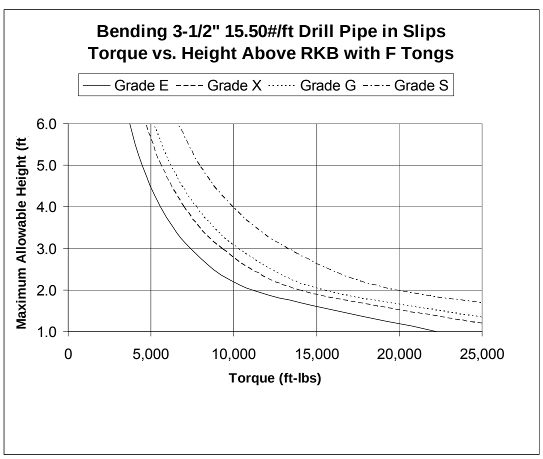
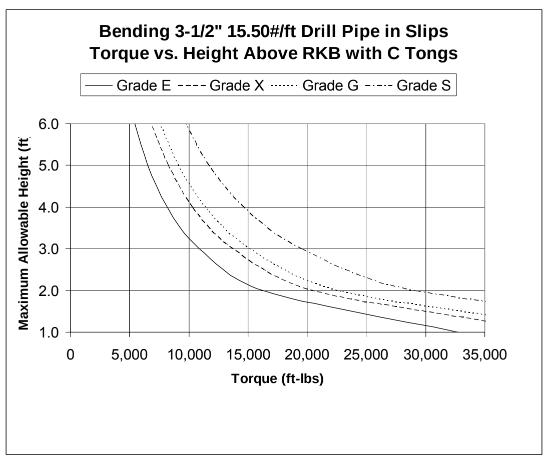
10,000
20,000
30,000
Torque (ft-lbs)


Bending 3-1/2" 15.50#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with F Tongs
Grade E
Grade X
Grade G
Grade S
Maximum Allowable Height (ft
6.0
5.0
4.0
3.0
2.0
1.0
0
5,000
10,000
15,000
20,000
25,000
Torque (ft-lbs)


Bending 3-1/2" 15.50#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with C Tongs
Grade E
Grade X
Grade G
Grade S
Maximum Allowable Height (ft
6.0
5.0
4.0
3.0
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1.0
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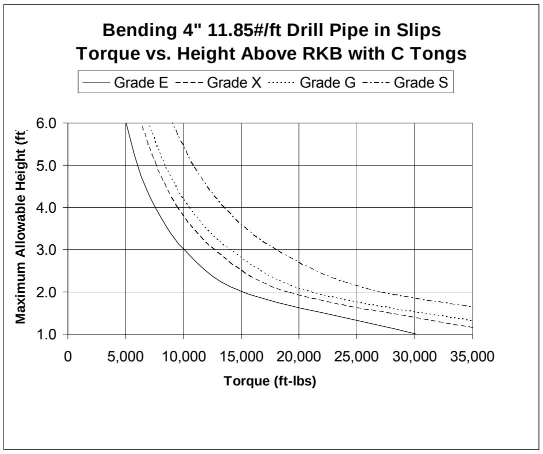
5,000
10,000
15,000
20,000
25,000
30,000
35,000
Torque (ft-lbs)


Bending 4" 11.85#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with C Tongs
Grade E
Grade X
Grade G
Grade S
Maximum Allowable Height (ft
6.0
5.0
4.0
3.0
2.0
1.0
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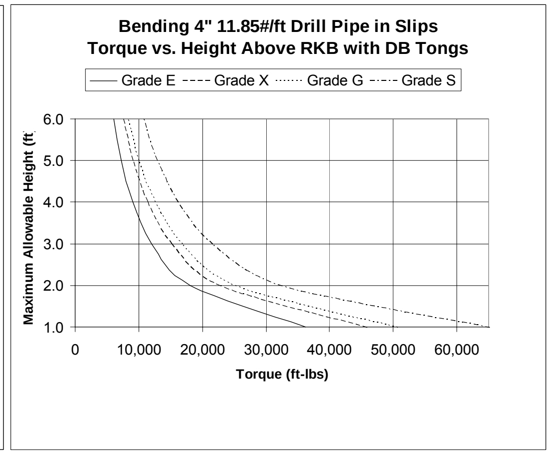
15,000
20,000
25,000
30,000
35,000
Torque (ft-lbs)


Bending 4" 11.85#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with DB Tongs
Grade E
Grade X
Grade G
Grade S
Maximum Allowable Height (ft
6.0
5.0
4.0
3.0
2.0
1.0
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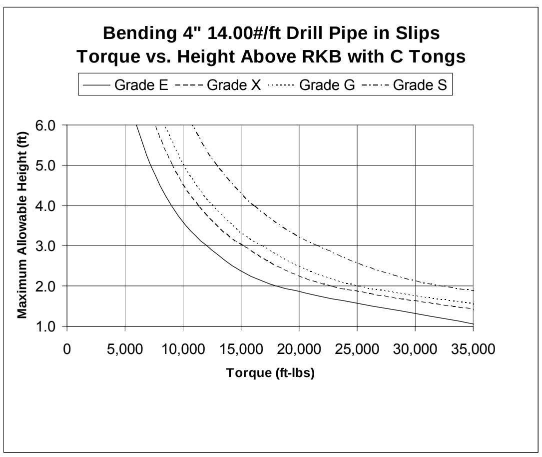
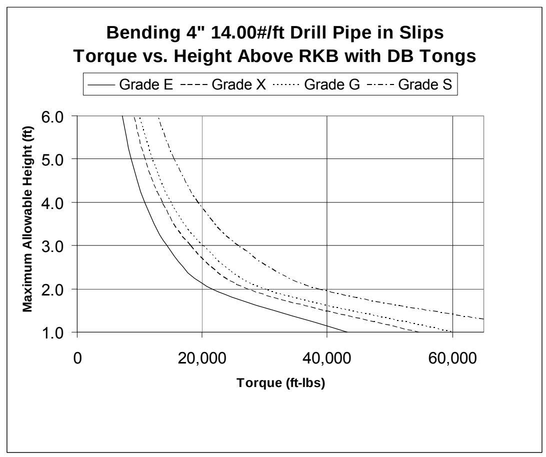
10,000
20,000
30,000
40,000
50,000
60,000
Torque (ft-lbs)


Bending 4" 14.00#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with C Tongs
Grade E
Grade X
Grade G
Grade S
Maximum Allowable Height (ft)
6.0
5.0
4.0
3.0
2.0
1.0
0
5,000
10,000
15,000
20,000
25,000
30,000
35,000
Torque (ft-lbs)


Bending 4" 14.00#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with DB Tongs
Grade E
Grade X
Grade G
Grade S
Maximum Allowable Height (ft)
6.0
5.0
4.0
3.0
2.0
1.0
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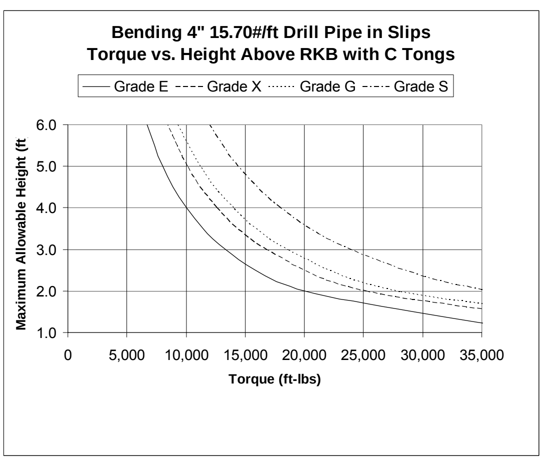
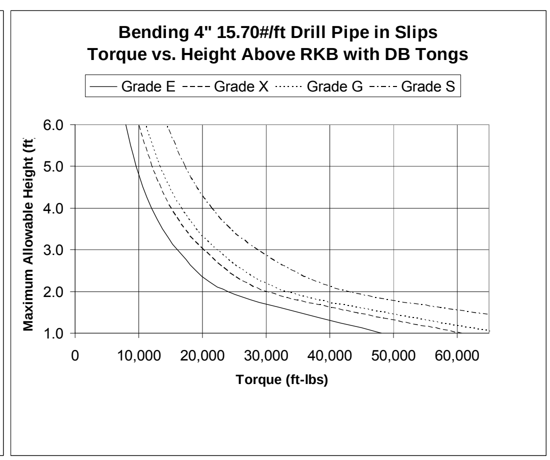
20,000
40,000
60,000
Torque (ft-lbs)


Bending 4" 15.70#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with C Tongs
Grade E
Grade X
Grade G
Grade S
Maximum Allowable Height (ft
6.0
5.0
4.0
3.0
2.0
1.0
0
5,000
10,000
15,000
20,000
25,000
30,000
35,000
Torque (ft-lbs)


Bending 4" 15.70#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with DB Tongs
Grade E
Grade X
Grade G
Grade S
Maximum Allowable Height (ft
6.0
5.0
4.0
3.0
2.0
1.0
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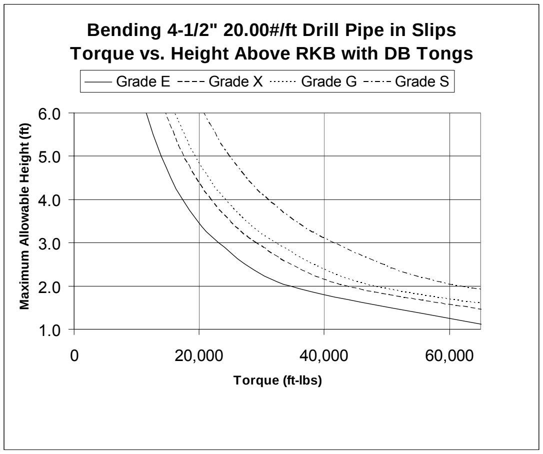
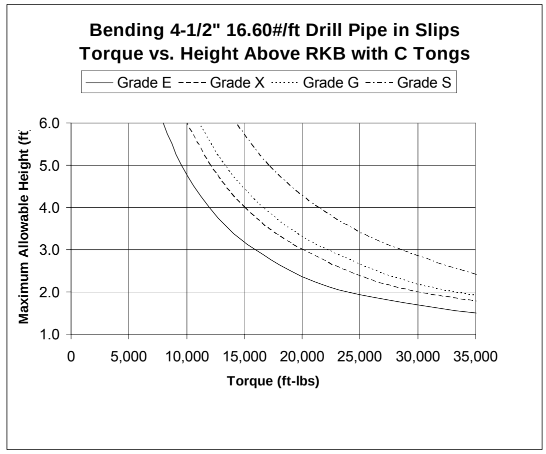
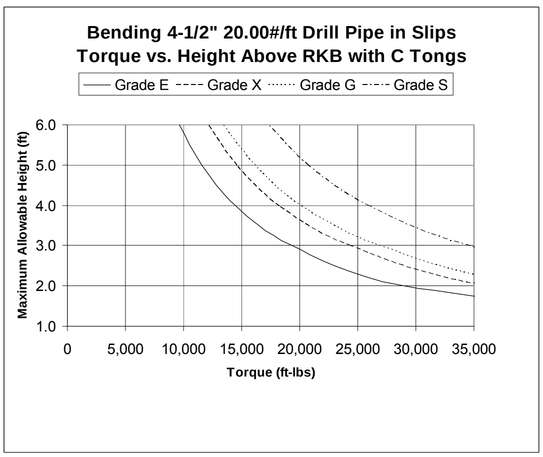
10,000
20,000
30,000
40,000
50,000
60,000
Torque (ft-lbs)


Bending 4-1/2" 20.00#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with DB Tongs
Grade E
Grade X
Grade G
Grade S
Maximum Allowable Height (ft)
6.0
5.0
4.0
3.0
2.0
1.0
0
20,000
40,000
60,000
Torque (ft-lbs)


Bending 4-1/2" 16.60#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with C Tongs
Grade E
Grade X
Grade G
Grade S
Maximum Allowable Height (ft
6.0
5.0
4.0
3.0
2.0
1.0
0
5,000
10,000
15,000
20,000
25,000
30,000
35,000
Torque (ft-lbs)


Bending 4-1/2" 20.00#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with C Tongs
Grade E
Grade X
Grade G
Grade S
Maximum Allowable Height (ft)
6.0
5.0
4.0
3.0
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1.0
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5,000
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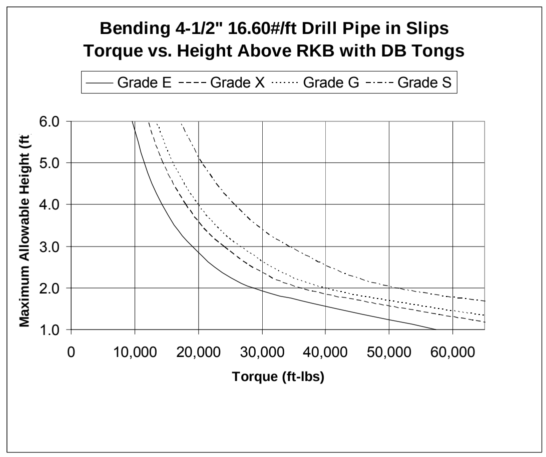
15,000
20,000
25,000
30,000
35,000
Torque (ft-lbs)


Bending 4-1/2" 16.60#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with DB Tongs
Grade E
Grade X
Grade G
Grade S
Maximum Allowable Height (ft
6.0
5.0
4.0
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2.0
1.0
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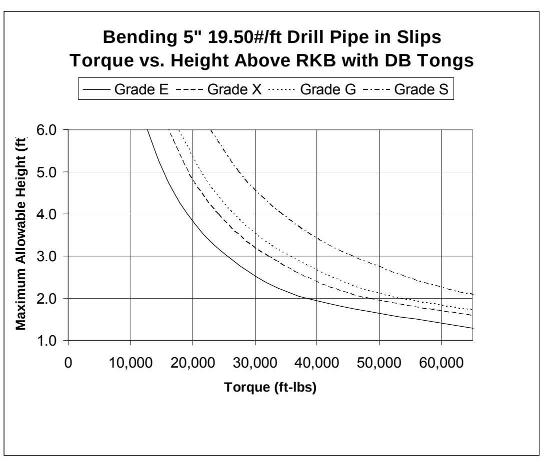
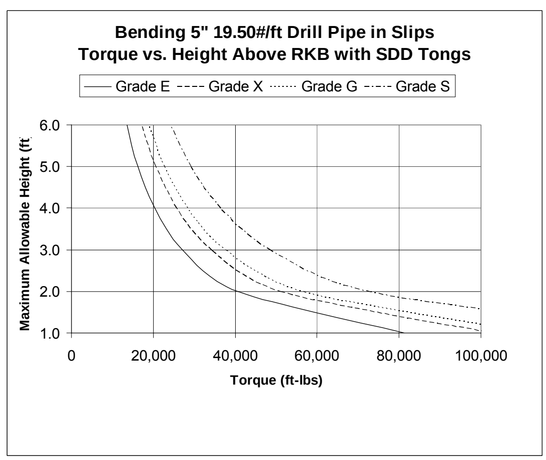
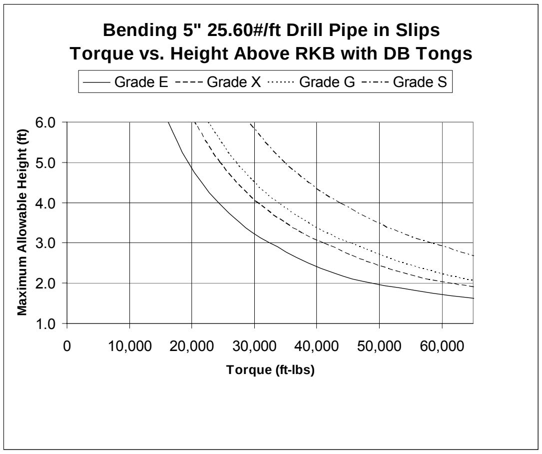
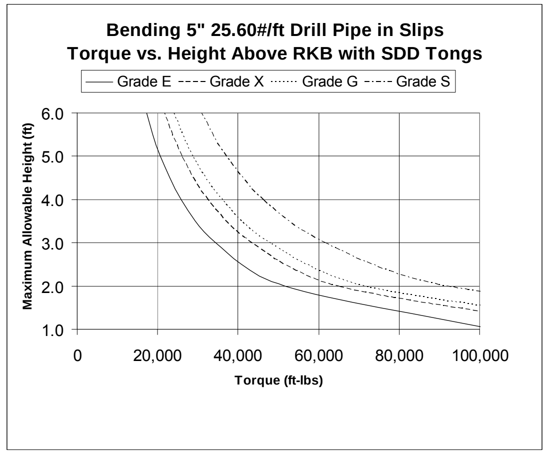
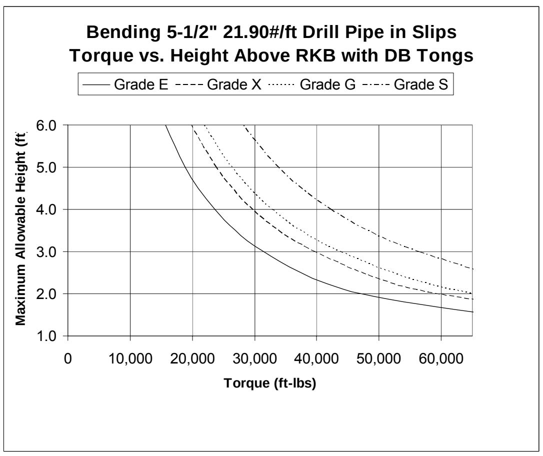
10,000
20,000
30,000
40,000
50,000
60,000
Torque (ft-lbs)

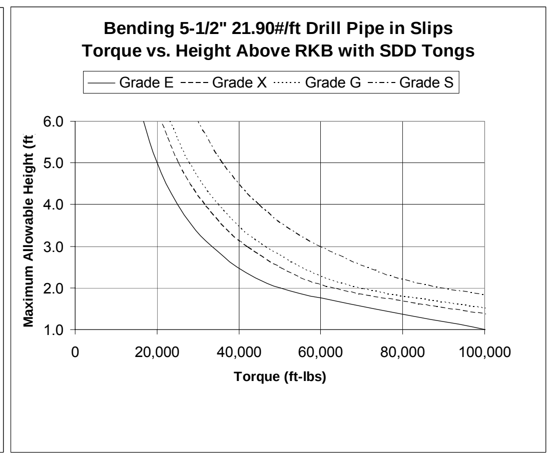

Bending 5" 19.50#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with DB Tongs
Grade E
Grade X
Grade G
Grade S
Maximum Allowable Height (ft
6.0
5.0
4.0
3.0
2.0
1.0
0
10,000
20,000
30,000
40,000
50,000
60,000
Torque (ft-lbs)

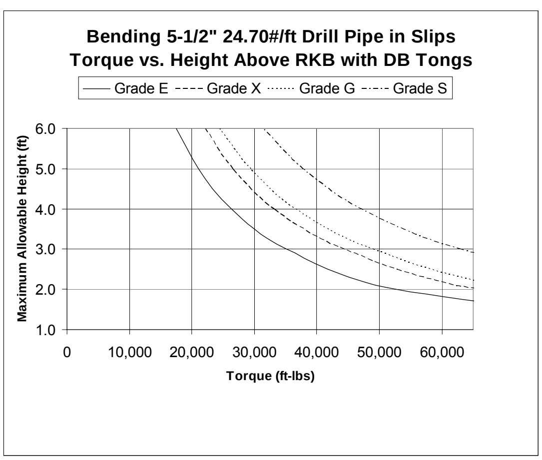

Bending 5" 19.50#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with SDD Tongs
Grade E
Grade X
Grade G
Grade S
Maximum Allowable Height (ft
6.0
5.0
4.0
3.0
2.0
1.0
0
20,000
40,000
60,000
80,000
100,000
Torque (ft-lbs)

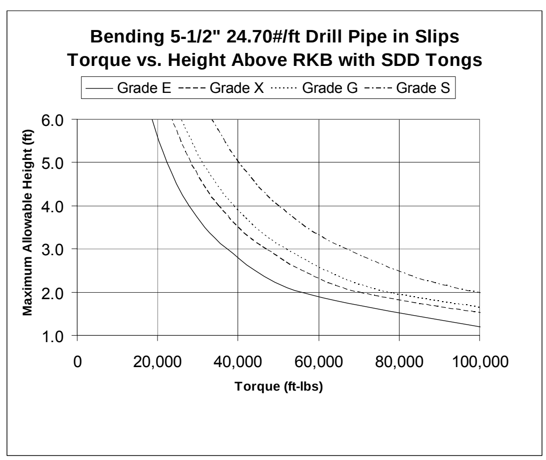

Bending 5" 25.60#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with DB Tongs
Grade E
Grade X
Grade G
Grade S
Maximum Allowable Height (ft)
Torque (ft-lbs)

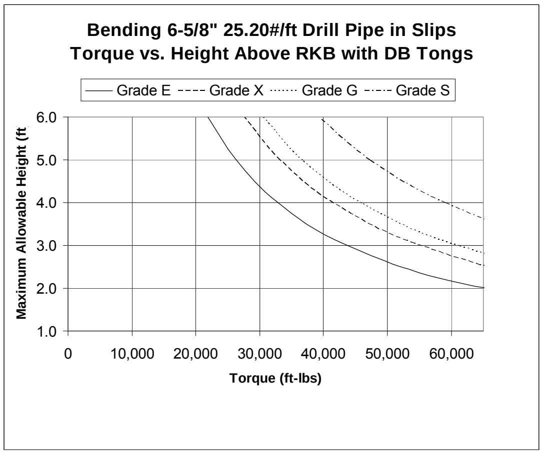

Bending 5" 25.60#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with SDD Tongs
Grade E
Grade X
Grade G
Grade S
Maximum Allowable Height (ft)
Torque (ft-lbs)

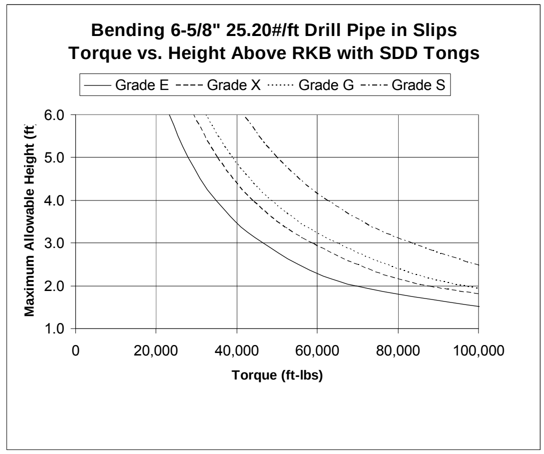

Bending 5-1/2" 21.90#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with DB Tongs
Grade E
Grade X
Grade G
Grade S
Maximum Allowable Height (ft)
Torque (ft-lbs)


Bending 5-1/2" 21.90#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with SDD Tongs
Grade E
Grade X
Grade G
Grade S
Maximum Allowable Height (ft
Torque (ft-lbs)


Bending 5-1/2" 24.70#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with DB Tongs
Grade E
Grade X
Grade G
Grade S
Maximum Allowable Height (ft)
Torque (ft-lbs)


Bending 5-1/2" 24.70#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with SDD Tongs
Grade E
Grade X
Grade G
Grade S
Maximum Allowable Height (ft)
Torque (ft-lbs)


Bending 6-5/8" 25.20#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with DB Tongs
Grade E
Grade X
Grade G
Grade S
Maximum Allowable Height (ft
6.0
5.0
4.0
3.0
2.0
1.0
0
10,000
20,000
30,000
40,000
50,000
60,000
Torque (ft-lbs)


Bending 6-5/8" 25.20#/ft Drill Pipe in Slips
Torque vs. Height Above RKB with SDD Tongs
Grade E
Grade X
Grade G
Grade S
Maximum Allowable Height (ft
6.0
5.0
4.0
3.0
2.0
1.0
0
20,000
40,000
60,000
80,000
100,000
Torque (ft-lbs)

Annex E
(informative)

Formulas

E.1 Torsional strength of eccentrically worn drill pipe

Assume 1: Eccentric hollow circular section (see Figure E.1). Reference: *Formulas for Stress & Strain*, Roark, 3rd Edition.

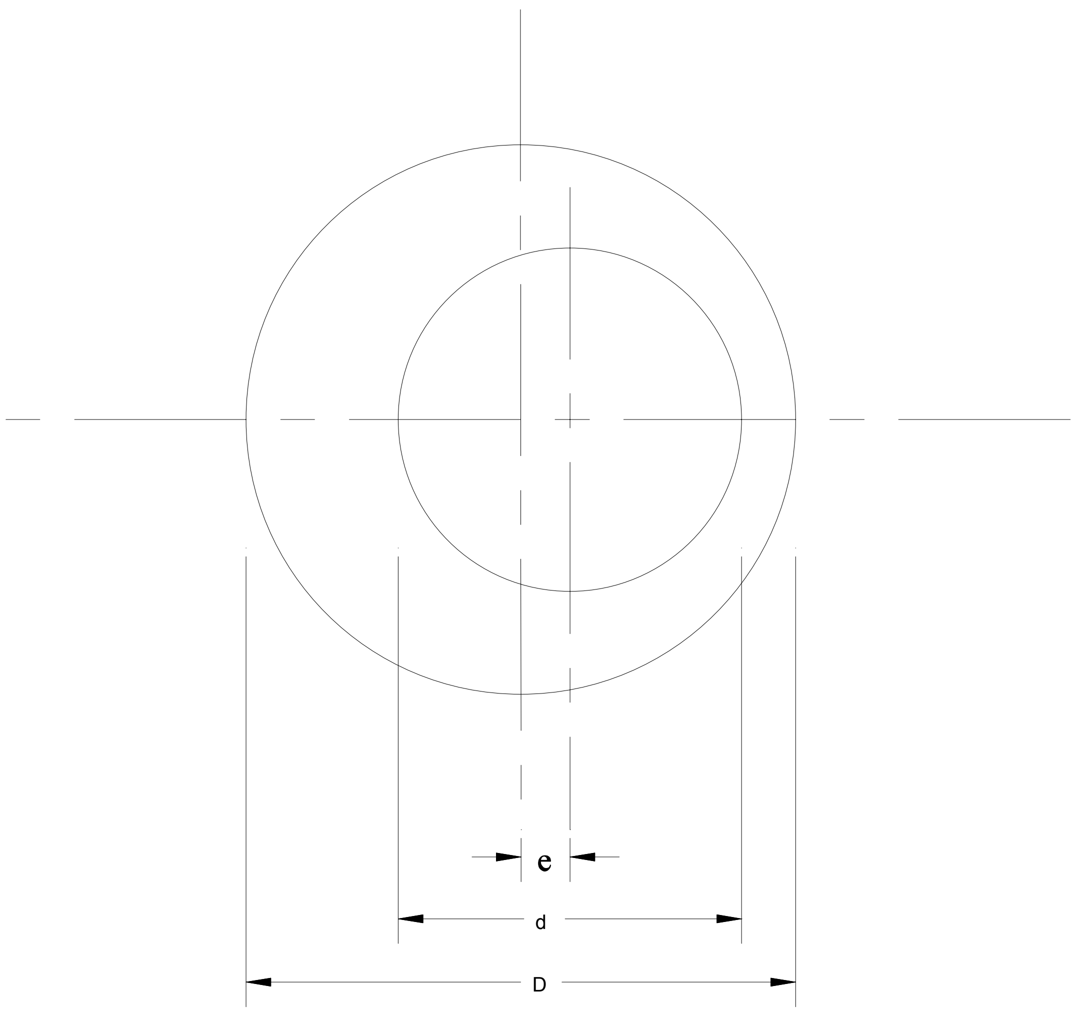


Figure E.1 — Eccentric hollow section of drill pipe

$$T = \frac{\pi S_S \left(D^4 - d^4\right)}{12 \times 16 \times D \times F'} \tag{E.1}$$

where

T is the torque, expressed in foot-pounds (ft-lbf);

S_S is the minimum shear strength, expressed in pounds per square inch (psi);

D is the outside diameter, expressed in inches (in);

d is the inside diameter, expressed in inches (in);

$$F = 1 + \frac{4N^2\phi}{\left(1-N^2\right)} + \frac{32N^2\phi^2}{\left(1-N^2\right)\left(1-N^4\right)} + \frac{48N^2\left(1+2N^2+3N^4+2N^6\right)\phi^3}{\left(1-N^2\right)\left(1-N^4\right)\left(1-N^6\right)}$$

where

$$N = \frac{d}{D}$$

$$\phi = \frac{e}{D}$$

where *e* is the distance from the inner diameter centreline to the outer diameter centreline, expressed in inches (in).

Assume 2: The internal diameter, *d*, remains constant and at the nominal ID of the pipe throughout its life.

Assume 3: The external diameter, *D*, is *d* + *t* nominal + *t* minimum; i.e., all wear occurs on one side. This diameter is not the same as diameter for uniform wear.

NOTE Torsional yield strengths for Premium Class, Table B.4,and Class 2, Table B.6, were calculated from equation E.1, using the assumption that wear is uniform on the external surface.

E.2 Safety factors

Values for various performance properties of drill pipe are given in Tables A.2 through A.7 (Tables B.2 through B.7). The values shown are minimum values and do not include factors of safety. In the design of pipe drill strings, factors of safety should be used as are considered necessary for the particular application.

E.3 Collapse pressure for drill pipe

NOTE See API Bulletin 5C3 for derivation of equations in E.3.

The minimum collapse pressures given in Tables A.3, A.5 and A.7 (Tables B.3, B.5 and B.7) are calculated values determined from equations in API Bulletin 5C3. Equations E.2 through E.5 are simplified equations that yield similar results. The *D/t* ratio determines the applicable formula, since each formula is based on a specific *D/t* ratio range.

For minimum collapse failure in the plastic range with minimum yield stress limitations: the external pressure that generates minimum yield stress on the inside wall of a tube.

$$P_C = 2Y_M\left[\frac{(D/t)-1}{(D/t)^2}\right] \tag{E.2}$$

where

P_C is the minimum collapse pressure, expressed in pounds per square inch (psi);

D is the nominal outside diameter, expressed in inches (in);

t is the nominal wall thickness, expressed in inches (in);

Y_M is the material minimum yield strength, expressed in pounds per square inch (psi).

NOTE 1 Collapse pressures for used drill pipe are determined by adjusting the nominal outside diameter, *D*, and wall thickness, *t*, as if wear is uniform on the outside of the pipe body and the inside diameter remains constant. Values of *D* and *t* for each class of drill pipe follow. These values are to be used in applicable equation E.2, E.3, E.4 or E.5, depending on the *D/t* ratio, to determine collapse pressure.

NOTE 2 Premium Class: *t*=(0,80)x(nominal wall), *D*=nominal OD-(0,40)x(nominal wall).

NOTE 3 Class 2: *t*=(0,70)x(nominal wall), *D*=nominal OD-(0,40)x(nominal wall).

Applicable *D/t* ratios for application of Equation E.2 are as follows:

Grade	D/t ratio
E75	13,60 and less
X94	12,85 and less
G105	12,57 and less
S135	11,92 and less

For minimum collapse failure in the plastic range:

$$P_C = Y_M \left[\left(\frac{A'}{D/t} \right) - B' \right] - C \quad \text{(E.3)}$$

where

P_C is the minimum collapse pressure, expressed in pounds per square inch (psi);

D is the nominal outside diameter, expressed in inches (in);

t is the nominal wall thickness, expressed in inches (in);

Y_M is the material minimum yield strength, expressed in pounds per square inch (psi);

A' is a formula factor;

B' is a formula factor;

C' is a formula factor.

NOTE 1 Collapse pressures for used drill pipe are determined by adjusting the nominal outside diameter, *D*, and wall thickness, *t*, as if wear is uniform on the outside of the pipe body and the inside diameter remains constant. Values of *D* and *t* for each class of drill pipe follow. These values are to be used in applicable equation E.2, E.3, E.4 or E.5, depending on the *D/t* ratio, to determine collapse pressure.

NOTE 2 Premium Class: t=(0,80)x(nominal wall), D=nominal OD-(0,40)x(nominal wall).

NOTE 3 Class 2: t=(0,70)x(nominal wall), D=nominal OD-(0,40)x(nominal wall).

Factors and applicable D/t ratios for application of Equation E.3 are as follows:

Grade	Formula factors			D/t ratio
	A'	B'	C′	
E75	3,054	0,0642	1806	13,60 to 22,91
X94	3,124	0,0743	2404	12,85 to 21,33
G105	3,162	0,0794	2702	12,57 to 20,70
S135	3,278	0,0946	3601	11,92 to 19,18

For minimum collapse failure in conversion or transition zone between elastic and plastic range:

$$P_C = Y_M \left[\left(\frac{A}{D/t} \right) - B \right] \tag{E.4}$$

where

P_C is the minimum collapse pressure, expressed in pounds per square inch (psi);

D is the nominal outside diameter, expressed in inches (in);

t is the nominal wall thickness, expressed in inches (in);

Y_M is the material minimum yield strength, expressed in pounds per square inch (psi);

A' is a formula factor;

B' is a formula factor;

NOTE 1 Collapse pressures for used drill pipe are determined by adjusting the nominal outside diameter, *D*, and wall thickness, *t*, as if wear is uniform on the outside of the pipe body and the inside diameter remains constant. Values of *D* and *t* for each class of drill pipe follow. These values are to be used in applicable equation E.2, E.3, E.4 or E.5, depending on the *D/t* ratio, to determine collapse pressure.

NOTE 2 Premium Class: *t*=(0,80)x(nominal wall), *D*=nominal OD-(0,40)x(nominal wall).

NOTE 3 Class 2: *t*=(0,70)x(nominal wall), *D*=nominal OD-(0,40)x(nominal wall).

Factors and applicable *D/t* ratios for application of Equation E.3 are as follows:

Grade	Formula factors		*D/t* ratio
	A′	*B′*	
E75	1,990	0,0418	22,91 to 32,05
X94	2,029	0,0482	21,33 to 28,36
G105	2,053	0,0515	20,70 to 26,89
S135	2,133	0,0615	19,18 to 23,44

For minimum collapse failure in the elastic range:

$$P_C = \frac{46.95 \times 10^6}{(D/t)\left[(D/t) - 1\right]^2} \tag{E.5}$$

where

P_C is the minimum collapse pressure, expressed in pounds per square inch (psi);

D is the nominal outside diameter, expressed in inches (in);

t is the nominal wall thickness, expressed in inches (in).

NOTE 1 Collapse pressures for used drill pipe are determined by adjusting the nominal outside diameter, *D*, and wall thickness, *t*, as if wear is uniform on the outside of the pipe body and the inside diameter remains constant. Values of *D* and *t* for each class of drill pipe follow. These values are to be used in applicable equation E.2, E.3, E.4 or E.5, depending on the *D/t* ratio, to determine collapse pressure.

NOTE 2 Premium Class: *t*=(0,80)x(nominal wall), *D*=nominal OD-(0,40)x(nominal wall).

NOTE 3 Class 2: *t*=(0,70)x(nominal wall), *D*=nominal OD-(0,40)x(nominal wall).

Applicable *D/t* ratios for application of Equation E.5 are as follows:

Grade	D/t ratio
E75	32,05 and less
X94	28,36 and less
G105	26,89 and less
S135	23,44 and less

E.4 Free length of stuck pipe

The relation between differential stretch and free length of a stuck string of steel pipe due to a differential pull is:

$$L_1 = \frac{E \times e \times W_{DP}}{40.8 P_{DIFF}} \qquad \text{(E.6)}$$

where

L_1 is the length of free drill pipe, expressed in feet (ft);

E is the modulus of elasticity, expressed in pounds per square inch (lb/in^2);

e is the differential stretch, expressed in inches (in);

W_{DP} is the weight per foot of pipe, expressed in pounds per foot (lb/ft);

P_{DIFF} is the differential pull, expressed in pounds (lb).

Where E = 30x10^6, this formula becomes:

$$L_1 = \frac{735{,}294 \times e \times W_{DP}}{P_{DIFF}} \qquad \text{(E.7)}$$

E.5 Internal pressure

E.5.1 Drill pipe

$$P_I = \frac{2Y_M t}{D} \tag{E.8}$$

where

P_I is the internal pressure, expressed in pounds per square inch (psi);

Y_M is the material minimum yield strength, expressed in pounds per square inch (psi);

t is the remaining wall thickness, expressed in inches (in);

D is the nominal outside diameter of tube, expressed in inches (in).

NOTE 1 Internal pressures for new drill pipe in Table B.3 were determined by using the nominal wall thickness for *t* in the above equation and multiplying by the factor 0,875 due to permissible wall thickness tolerance of minus 12,5 %.

NOTE 2 Internal pressures for used drill pipe were determined by adjusting the nominal wall thickness according to footnotes below Table B.5 and B.7 and using the nominal outside diameter, in the above Equation E.8.

E.5.2 Kellys

$$P_I = \frac{Y_M \left[D_{FL}{}^2 - \left(D_{FL} - 2t\right)^2\right]}{\sqrt{3\left(D_{FL}\right)^4 + \left(D_{FL} - 2t\right)^4}} \tag{E.9}$$

where

P_I is the internal pressure, expressed in pounds per square inch (psi);

Y_M is the material minimum yield strength, expressed in pounds per square inch (psi);

D_{FL} is the distance across drive section flats, expressed in inches (in).

t is the remaining wall thickness, expressed in inches (in).

NOTE The dimension *t* is the minimum wall thickness of the drive section and should be determined in each case through the use of an ultrasonic thickness gauge or similar device.

E.6 Stretch of suspended drill pipe

When pipe is freely suspended in a fluid, the stretch due to its own weight is:

$$e = \frac{L_1{}^2}{24E}\left[W_A - 2W_F\left(1 - \mu\right)\right] \tag{E.10}$$

where

e is the stretch, expressed in inches (in);

L_1 is the length of free drill pipe, expressed in feet (ft);

E is the modulus of elasticity, expressed in pounds per square inch (lb/in^2);

W_A is the weight of pip material, expressed in pounds per cubic foot (lb/ft^3);

W_F is the weight of fluid, expressed in pounds per cubic foot (lb/ft^3);

μ is Poisson's ratio.

For steel pipe where W = 489,5 lb/ft^3, E = 30 x 106 psi and μ = 0,28, this formula will be:

$$e = \frac{L_1^{\ 2}}{72 \times 10^7}\left[489.5 - 144 W_F\right] \quad \text{(E.11)}$$

or

$$e = \frac{L_1^{\ 2}}{9.625 \times 10^7}\left[65.44 - 144 W_G\right] \quad \text{(E.12)}$$

where

W_F is the weight of fluid, expressed in pounds per cubic foot (lb/ft^3);

W_G is the weight of fluid, expressed in pounds per gallon (lb/gal).

E.7 Tensile strength of drill body

$$P_{PB} = Y_M A \quad \text{(E.13)}$$

where

P_{PB} is the minimum tensile strength, expressed in pounds (lb);

Y_M is the material minimum yield strength, expressed in pounds per square inch (psi);

A is the cross-section area, expressed in square inches (in^2) (Table B.1, Column 6, for drill pipe).

E.8 Torsional yield strength of drill pipe body

E.8.1 Pure torsion

$$Q = \frac{0{,}096\,167\, J Y_M}{D} \quad \text{(E.14)}$$

where

Q is the minimum torsional yield strength, expressed in foot pounds (ft-lb);

Y_M is the material minimum yield strength, expressed in pounds per square in (psi);

D is the outside diameter, expressed in inches (in);

d is the inside diameter, expressed in inches (in);

J is the polar moment of inertia, expressed in the following equation for tubes:

$$J = \frac{\pi}{32}\left(D^4 - d^4\right) = 0{,}098\,175\left(D^4 - d^4\right)$$

E.8.2 Torsion and tension

$$Q_T = \frac{0{,}096167J}{D}\sqrt{{Y_M}^2 - \frac{{P_{TL}}^2}{A^2}} \quad \text{(E.15)}$$

where

Q_T is the minimum torsional yield strength under tension, expressed in foot pounds (ft-lb);

Y_M is the material minimum yield strength, expressed in pounds per square in (psi);

D is the outside diameter, expressed in inches (in);

d is the inside diameter, expressed in inches (in);

J is the polar moment of inertia, expressed in the following equation for tubes:

$$J = \frac{\pi}{32}\left(D^4 - d^4\right) = 0{,}098\,175\left(D^4 - d^4\right)$$

P_{TL} is the total load in tension, expressed in pounds (lb);

A is the cross section area, expressed in square inches (in^2).

E.9 Torque calculations for rotary shouldered connections[31]

E.9.1 Torque to yield a rotary shouldered connection

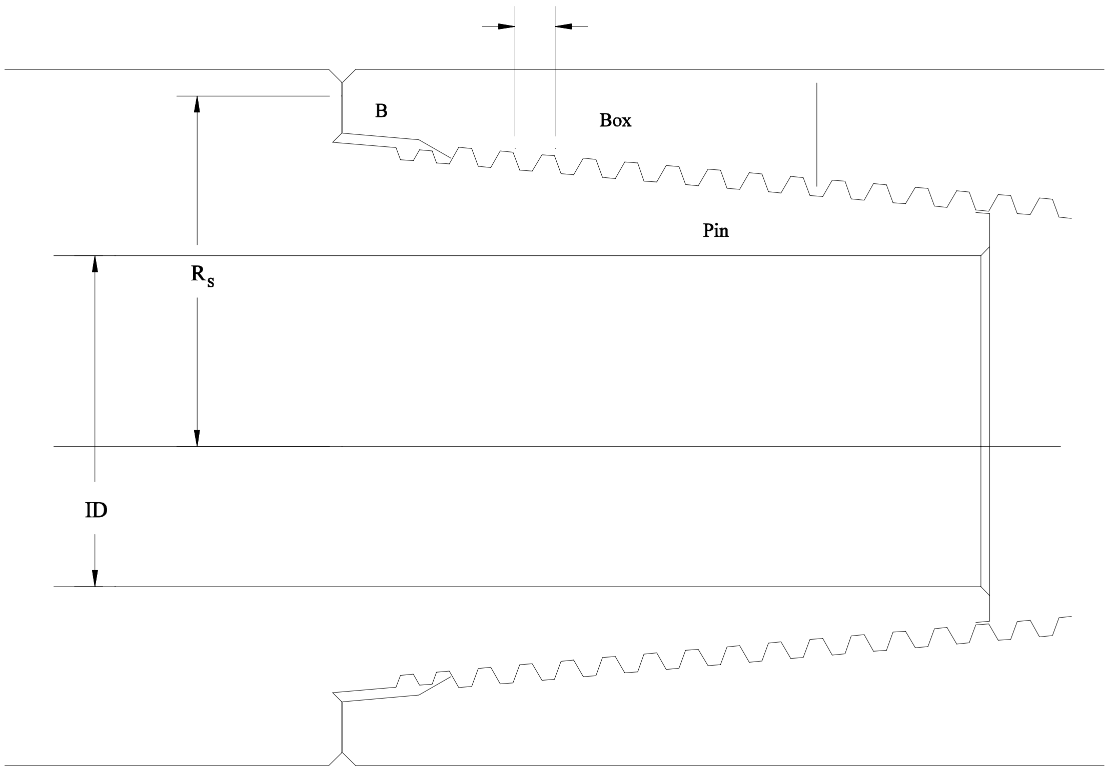


Figure E.2 — Rotary shouldered connection

[31]) See Table E-1 and Figure E-2.

Table E.1 — Rotary shouldered connection thread element information

1	2	3	4	5	6	7	8	9	10	11	12	13
Connection Type	Pitch Diameter C	Taper	Pin Length	Thread Height Not Truncated	Root Truncation	Nominal Counterbore	Thread Pitch	Thread Angle θ	Stress Relief Groove Diameter	Bore-Back Cylinder Diameter	Low Torque Counterb ore	Low Torque Bevel Diameter
NC10	1.063000	1.500000	1.500000	0.144100	0.040600	1.204000	0.166667	30	—	—	—	—
NC12	1.265000	1.500000	1.750000	0.144100	0.040600	1.406000	0.166667	30	—	—	—	—
NC13	1.391000	1.500000	1.750000	0.144100	0.040600	1.532000	0.166667	30	—	—	—	—
NC16	1.609000	1.500000	1.750000	0.144100	0.040600	1.751000	0.166667	30	—	—	—	—
NC23	2.355000	2.000000	3.000000	0.216005	0.038000	2.625000	0.250000	30	2.140625	2.218750	—	—
NC26	2.668000	2.000000	3.000000	0.216005	0.038000	2.937500	0.250000	30	2.375000	2.531250	—	—
NC31	3.183000	2.000000	3.500000	0.216005	0.038000	3.453125	0.250000	30	2.890625	2.953125	—	—
NC35	3.531000	2.000000	3.750000	0.216005	0.038000	3.812500	0.250000	30	3.231000	3.234375	—	—
NC38	3.808000	2.000000	4.000000	0.216005	0.038000	4.078125	0.250000	30	3.508000	3.468750	—	—
NC40	4.072000	2.000000	4.500000	0.216005	0.038000	4.343750	0.250000	30	3.772000	3.656250	—	—
NC44	4.417000	2.000000	4.500000	0.216005	0.038000	4.687500	0.250000	30	4.117000	4.000000	—	—
NC46	4.626000	2.000000	4.500000	0.216005	0.038000	4.906250	0.250000	30	4.326000	4.203125	—	—
NC50	5.041700	2.000000	4.500000	0.216005	0.038000	5.312500	0.250000	30	4.742000	4.625000	—	—
NC56	5.616000	3.000000	5.000000	0.215379	0.038000	5.937500	0.250000	30	5.277000	4.796875	—	—
NC61	6.178000	3.000000	5.500000	0.215379	0.038000	6.500000	0.250000	30	5.839000	5.234375	—	—
NC70	7.053000	3.000000	6.000000	0.215379	0.038000	7.375000	0.250000	30	6.714000	5.984375	—	—
NC77	7.741000	3.000000	6.500000	0.215379	0.038000	8.062500	0.250000	30	7.402000	6.546875	—	—
51/2 IF	6.189000	2.000000	5.000000	0.216005	0.038000	6.453125	0.250000	30	5.890625	5.687500	—	—
65/8 IF	7.251000	2.000000	5.000000	0.216005	0.038000	7.515625	0.250000	30	6.953125	6.750000	—	—
1 REG	1.154000	1.500000	1.500000	1.441000	0.040600	1.301000	1.66667	30	—	—	—	—
11/2 REG	1.541000	1.500000	2.000000	1.441000	0.040600	1.688000	1.66667	30	—	—	—	—

1	2	3	4	5	6	7	8	9	10	11	12	13
Connection Type	Pitch Diameter C	Taper	Pin Length	Thread Height Not Truncated	Root Truncation	Nominal Counterbore	Thread Pitch	Thread Angle θ	Stress Relief Groove Diameter	Bore-Back Cylinder Diameter	Low Torque Counterb ore	Low Torque Bevel Diameter
23/8 REG	2.365370	3.000000	3.000000	0.172303	0.020000	2.687500	0.200000	30	2.015625	2.062500	—	—
27/8 REG	2.740370	3.000000	3.500000	0.172303	0.020000	3.062500	0.200000	30	2.390625	2.312500	—	—
31/2 REG	3.239870	3.000000	3.750000	0.172303	0.020000	3.562500	0.200000	30	2.906250	2.718750	—	—
41/2 REG	4.364870	3.000000	4.250000	0.172303	0.020000	4.687500	0.200000	30	4.013000	3.718750	—	—
51/2 REG	5.234020	3.000000	4.750000	0.215379	0.025000	5.578125	0.250000	30	4.869000	4.500000	—	—
65/8 REG	5.757800	2.000000	5.000000	0.216005	0.025000	6.062500	0.250000	30	5.417000	5.281250	—	—
75/8 REG	6.714530	3.000000	5.250000	0.215379	0.025000	7.093750	0.250000	30	6.349000	5.859375	7.750000	9.250000
85/8 REG	7.666580	3.000000	5.375000	0.215379	0.025000	8.046875	0.250000	30	7.301000	6.781250	9.000000	10.500000
27/8 FH	3.365400	3.000000	3.500000	0.172303	0.020000	3.687500	0.200000	30	—	—	—	—
31/2 FH	3.734000	3.000000	3.750000	0.172303	0.020000	4.046875	0.200000	30	3.421875	3.218750	—	—
41/2 FH	4.532000	3.000000	4.000000	0.172303	0.020000	4.875000	0.200000	30	4.180000	3.953125	—	—
51/2 FH	5.591000	2.000000	5.000000	0.216005	0.025000	5.906250	0.250000	30	5.250000	5.109375	—	—
65/8 FH	6.519600	2.000000	5.000000	0.216005	0.025000	6.843750.	0.250000	30	6.171875	6.046875	—	—
23/8 SL-H90	2.578000	1.250000	2.812500	0.166215	0.034107	2.765625	0.333333	45	2.328125	2.531250	—	—
27/8 SL-H90	3.049000	1.250000	2.937500	0.166215	0.034107	3.234375	0.333333	45	2.671875	2.984375	—	—
31/2 SL-H90	3.688000	1.250000	3.187500	0.166215	0.034107	3.875000	0.333333	45	3.312500	3.593750	—	—
31/2 H-90	3.929860	2.000000	4.000000	0.141865	0.017042	4.187500	0.285710	45	3.656250	3.562500	—	—
4 H-90	4.303600	2.000000	4.250000	0.141865	0.017042	4.562500	0.285710	45	4.031250	3.875000	—	—
41/2 H-90	4.637600	2.000000	4.500000	0.141865	0.017042	4.890625	0.285710	45	4.359375	4.187500	—	—
5 H-90	4.908100	2.000000	4.750000	0.141865	0.017042	5.171875	0.285710	45	4.625000	4.406250	—	—
51/2 H-90	5.178600	2.000000	4.750000	0.141865	0.017042	5.437500	0.285710	45	4.906250	4.687500	—	—
65/8 H-90	5.803600	2.000000	5.000000	0.141865	0.017042	6.062500	0.285710	45	5.531250	5.265625	—	—

1	2	3	4	5	6	7	8	9	10	11	12	13
Connection Type	Pitch Diameter C	Taper	Pin Length	Thread Height Not Truncated	Root Truncation	Nominal Counterbore	Thread Pitch	Thread Angle θ	Stress Relief Groove Diameter	Bore-Back Cylinder Diameter	Low Torque Counterb ore	Low Torque Bevel Diameter
7 H-90	6.252300	3.000000	5.500000	0.140625	0.016733	6.562500	0.285710	45	6.031250	5.265625	7.125000	8.250000
75/8 H-90	7.141100	3.000000	6.125000	0.140625	0.016733	7.453125	0.285710	45	6.906250	6.000000	8.000000	9.250000
85/8 H-90	8.016100	3.000000	6.625000	0.140625	0.016733	8.328125	0.285710	45	7.781250	6.750000	9.375000	10.500000
23/8 PAC	2.203000	1.500000	2.375000	0.216224	0.057948	2.406250	0.250000	30	1.984375	2.171875	—	—
27/8 PAC	2.369000	1.500000	2.375000	0.216224	0.057948	2.578125	0.250000	30	2.156250	2.343750	—	—
31/2 PAC	2.884000	1.500000	3.250000	0.216224	0.057948	3.109375	0.250000	30	—	—	—	—
23/8 SH	2.230000	2.000000	2.875000	0.216005	0.038000	2.500000	0.250000	30	1.937500	2.093750	—	—
27/8 SH	2.668000	2.000000	3.000000	0.216005	0.038000	2.937500	0.250000	30	2.375000	2.531250	—	—
31/2 SH	3.183000	2.000000	3.500000	0.216005	0.038000	3.453125	0.250000	30	2.890625	2.953125	—	—
4 SH	3.604000	2.000000	3.500000	0.216005	0.038000	3.875000	0.250000	30	3.312500	3.375000	—	—
41/2 SH	3.808000	2.000000	4.000000	0.216005	0.038000	4.078125	0.250000	30	3.508000	3.468750	—	—
27/8 XH	3.119000	2.000000	4.000000	0.216005	0.038000	3.359375	0.250000	30	2.828125	2.781250	—	—
31/2 XH	3.604000	2.000000	3.500000	0.216005	0.038000	3.875000	0.250000	30	3.312500	3.375000	—	—
5 XH	5.041700	2.000000	4.500000	0.216005	0.038000	5.312500	0.250000	30	4.742000	4.625000	—	—
23/8 OH SW	2.588000	1.500000	2.375000	0.216224	0.057948	2.796875	0.250000	30	—	—	—	—
23/8 OH LW	2.588000	1.500000	2.375000	0.216224	0.057948	2.796875	0.250000	30	—	—	—	—
27/8 OH SW	2.984000	1.500000	2.875000	0.216224	0.057948	3.203125	0.250000	30	—	—	—	—
27/8 OH LW	2.984000	1.500000	2.500000	0.216224	0.057948	3.203125	0.250000	30	—	—	—	—
31/2 OH SW	3.728000	1.500000	3.250000	0.216224	0.057948	3.953125	0.250000	30	—	—	—	—
31/2 OH LW	3.728000	1.500000	3.250000	0.216224	0.057948	3.953125	0.250000	30	—	—	—	—
4 OH SW	4.416000	1.500000	4.000000	0.216224	0.057948	4.640625	0.250000	30	—	—	—	—
4 OH LW	4.416000	1.500000	3.500000	0.216224	0.057948	4.640625	0.250000	30	—	—	—	—

1	2	3	4	5	6	7	8	9	10	11	12	13
Connection Type	Pitch Diameter C	Taper	Pin Length	Thread Height Not Truncated	Root Truncation	Nominal Counterbore	Thread Pitch	Thread Angle θ	Stress Relief Groove Diameter	Bore-Back Cylinder Diameter	Low Torque Counterb ore	Low Torque Bevel Diameter
41/2 OH SW	4.752000	1.500000	3.750000	0.216224	0.057948	4.953125	0.250000	30	—	—	—	—
41/2 OH LW	4.752000	1.500000	3.750000	0.216224	0.057948	4.953125	0.250000	30	—	—	—	—
23/8 WO	2.605000	2.000000	2.375000	0.216005	0.038000	2.859375	0.250000	30	—	—	—	—
27/8 WO	3.121000	2.000000	3.000000	0.216005	0.038000	3.375000	0.250000	30	—	—	—	—
31/2 WO	3.808000	2.000000	3.500000	0.216005	0.038000	4.078125	0.250000	30	—	—	—	—
4 WO	4.626000	2.000000	4.500000	0.216005	0.038000	4.906250	0.250000	30	—	—	—	—
41/2 WO	5.041700	2.000000	4.500000	0.216005	0.038000	5.312500	0.250000	30	—	—	—	—

$$T_Y = \frac{Y_M A_{TOR}}{12}\left(\frac{p}{2\pi} + \frac{R_T f}{\cos\theta} + R_S f\right) \qquad \text{(E.16)}$$

where

T_Y is the turning moment or torque required to yield, expressed in foot pounds (ft·lb);

Y_M is the material minimum yield strength, expressed in pounds per square inch (psi);

p is the lead of thread, expressed in inches (in);

f is the coefficient of friction on mating surfaces, threads and shoulders, assumed 0,08 for thread compounds containing 40 to 60 % by weight of finely powdered metallic zinc[32)];

θ is half the included angle of thread[33], expressed in degrees;

$$R_T = \frac{C + \left[C - \left(L_{PC} - 0.625\right) \times tpr \times 1/12\right]}{4}$$

where

L_{PC} is the length of pin, expressed in inches (in)[34)] ;

tpr is the taper, expressed in inches per feet (in/ft)[35)] ;

$R_S = 1/4\left(OD + Q_C\right)$, expressed in inches (in)[36)];

A_{TOR} is the cross-section area A_B or A_P whichever is smaller, expressed in square inches (in^2)

where (without relief grooves)

$$A_P = \frac{\pi}{4}\left[\left(C - B\right)^2 - ID^2\right]$$

or (with relief grooves)

$$A_P = \frac{\pi}{4}\left[D_{RG}{}^2 - ID^2\right]$$

where

---

32) Reference the caution regarding the use of hazardous materials in Appendix G of API Specification 7.

33) Figures 21 or 22 of API Specification 7.

34) API Specification 7, Table 25, Column 9.

35) API Specification 7, Table 25, Column 4.

36) The maximum value of R_S is limited to the value obtained from the calculated OD where $A_P = A_B$.

D_{RG} is the diameter of relief groove[37], expressed in inches (in);

C is the pitch diameter of thread at gauge point[38], expressed in inches (in);

ID is the inside diameter, expressed in inches (in);

$$B = 2\left(\frac{H}{2} - S_{RS}\right) + tpr \times 1/8 \times 1/12$$

where

H is the thread height not truncated[39], expressed in inches (in);

S_{RS} is the root truncation[40], expressed in inches (in);

$$A_B = \frac{\pi}{4}\left[OD^2 - \left(Q_C - E\right)^2\right]$$

where

OD is the outside diameter, expressed in inches (in);

Q_C is the box counterbore, expressed in inches (in);

$$E = tpr \times 3/8 \times 1/12$$

E.9.2 Make-up torque for rotary shouldered connections

$$T = \frac{SA_{TOR}}{12}\left(\frac{p}{2\pi} + \frac{R_T f}{\cos\theta} + R_S f\right) \quad \text{(E.17)}$$

where

A_{TOR} is A_B or A_P whichever is smaller, expressed in square inches (in^2);

S is the recommended make-up stress level, expressed in pounds per square inch (psi).

A_P shall be based on pin connections without relief grooves.

NOTE For values of S, see 4.8.1 for tool joints and 5.2 for drill collars.

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37) API Specification 7, Table 16, Column 5.

38 API Specification 7, Table 25, Column 5.

39 API Specification 7, Table 26, Column 3.

40 API Specification 7, Table 26, Column 5.

E.10 Drill collar bending strength ratio

The bending strength ratios in Figures 26 through 32 were determined by application of Equation E.18. The effect of stress-relief features was disregarded.

$$BSR = \frac{Z_B}{Z_P} \tag{E.18}$$

$$= \frac{0.098\dfrac{\left(OD^4 - b^4\right)}{OD}}{0.098\dfrac{\left(R^4 - ID^4\right)}{R_{TR}}}$$

$$= \frac{\dfrac{\left(OD^4 - b^4\right)}{OD}}{\dfrac{\left(R^4 - ID^4\right)}{R_{TR}}}$$

where

BSR is the bending strength ratio;

Z_B is the box section modulus, expressed in cubic inches (in^3);

Z_P is the pin section modulus, expressed in cubic inches (in^3);

O*D* is the outside diameter of pin and box (Figure E.3), expressed in inches (in);

ID is the inside diameter of bore (Figure E.3), expressed in inches (in);

b thread root diameter of box threads at end of pin (Figure E.3), expressed in inches (in);

R_{TR} is the thread root diameter of pin threads ¾ in from shoulder of pin (Figure E.3), expressed in inches (in).

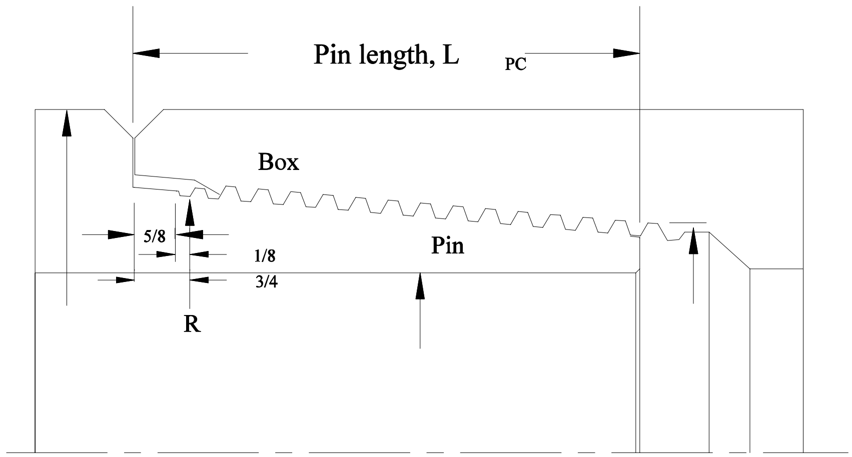

Figure E.3 — Rotary shouldered connection location of dimensions for bending strength ratio calculations

To use equation E.18, first calculate *Dedendum*, *b* and *R*:

$$Dedendum = \frac{H}{2} - f_M \tag{E.19}$$

where

H is the thread height not truncated, expressed in inches (in);

f_M is the root truncation, expressed in inches (in).

$$b = C - \frac{tpr\left(L_{PC} - 0{,}625\right)}{12} + \left(2 \times dedendum\right) \tag{E.20}$$

where

C is the pitch diameter at gauge point, expressed in inches (in);

tpr is the taper, expressed in inches per foot (in/ft).

$$R = C - \left(2 \times dedendum\right) - \left(tpr \times {}^{1}\!/_{8} \times {}^{1}\!/_{12}\right) \tag{E.21}$$

An example of the use of Equation E.18 in determining the bending strength of a typical drill collar connection is as follows:

Determine the bending strength ratio of drill collar NC46-62 (6¼ Odx2 13/16) ID connection.

OD = 6,25 (API Specification 7, Table 13, Column 2),

ID = 2 13/16 = 2,8125 (API Specification 7, Table 13, Column 3),

C = 4,626 (API Specification 7, Table 13, Column 5),

Taper = 2 (API Specification 7, Table 25, Column 4),

L_{PC} = 4,5 (API Specification 7, Table 25, Column 9),

H = 0,216005 (API Specification 7, Table 26, Column 3),

f_M = 0,038000 (API Specification 7, Table 26, Column 5).

First calculate *dedendum*, b and R:

$$Dedendum = \frac{H}{2} - f_M = \frac{0.216005}{2} - 0.038000 = 0.0700025$$

$$b = C - \frac{tpr(L_{PC} - 0.625)}{12} + (2 \times dedendum) = 4.626 - \frac{2(4.5 - 0.625)}{12} + (2 \times 0.0700025) = 4.120$$

$$R = C - (2 \times dedendum) - \left(tpr \times \tfrac{1}{8} \times \tfrac{1}{12}\right) = 4.626 - (2 \times 0.0700025) - \left(2 \times \tfrac{1}{8} \times \tfrac{1}{12}\right) = 4.465$$

Substituting these values in Equation E.27 determines the bending strength ratio as follows:

$$BSR(NC46 - 62) = \frac{\dfrac{(OD^4 - b^4)}{OD}}{\dfrac{(R^4 - ID^4)}{R}} = \frac{\dfrac{(6.25)^4 - (4.120)^4}{6.25}}{\dfrac{(4.465)^4 - (2.8125)^4}{4.465}} = 2.64 : 1$$

E.11 Torsional yield strength of Kelly drive section

The torsional yield strength of the Kelly drive section values listed in Tables B.15 and B.17 were derived from the following equation:

$$Y = \frac{0.577 Y_M \left[0.200(a^3 - b^3)\right]}{12} \tag{E.22}$$

where

Y_M is the tensile yield, expressed in pounds per square inch (psi);

a is the distance across flats, expressed in inches (in);

b is the Kelly bore, expressed in inches (in).

E.12 Bending strength, Kelly drive section

The yield in bending values of the Kelly drive section listed in Tables B.15 and B.17 were determined by one of the following equations:

a) Yield in bending through corners of the square drive section, Y_{BC}, expressed in foot-pounds (ft·lb):

$$Y_{BC} = \frac{Y_M (0.118a^4 - 0.069b^4)}{12a} \tag{E.23}$$

b) Yield in bending through the faces of the hexagonal drive section, Y_{BF}, expressed in foot-pounds (ft·lb):

$$Y_{BF} = \frac{Y_M \left(0.104a^4 - 0.085b^4\right)}{12a} \quad \text{(E.24)}$$

E.13 Approximate weight of tool joint plus drill pipe

Approximate weight of tool joint plus drill pipe assembly, expressed in pounds per foot (lb/ft):

$$= \frac{\left(\text{Approximate adjusted wt. of drill pipe} \times 29.4 + \text{Approximate wt. of tool joint}\right)}{\text{Tool joint adjusted length} + 29.4} \quad \text{(E.25)}$$

where

Approximate adjusted weight of drill pipe, expressed in pounds per foot (lb/ft):

$$= \text{Plain end wt.} + \frac{\text{Upset wt.}}{29.4} \quad \text{(E.26)}$$

Plain end weight and upset weight are found in API Specification 5D.

Approximate weight of tool joint, expressed in pounds (lb):

$$= 0.222\,L\left(D^2 - d^2\right) + 0.167\left(D^3 - D_{TE}{}^3\right) - 0.501d^2\left(D - D_{AE}\right) \quad \text{(E.27)}$$

Dimensions for L, D, d and D_{TE} are in API Specification 7, Figure 6 and Table 7.

Adjusted length of tool joint, expressed in feet (ft):

$$= \frac{L + 2.253\left(D - D_{TE}\right)}{12} \quad \text{(E.28)}$$

E.14 Critical buckling force for curved boreholes

E.14.1 The following equations define the range of hole curvatures that buckle pipe in a three dimensionally curved borehole. The pipe buckles whenever the hole curvature is between the minimum and maximum curvatures defined by the equations.

$$\text{if } F_B < \frac{4 \times E \times I}{12 \times h_C \times R_L} \text{ pipe not buckled,} \quad \text{(E.29)}$$

$$\text{if } F_B \geq \frac{4 \times E \times I}{12 \times h_C \times R_L}, \quad \text{(E.30)}$$

$$W_{EQ} = \frac{12 \times h_C \times F_B{}^2}{4 \times E \times I}, \quad \text{(E.31)}$$

$$B_{V\min} = \frac{-5730}{F_B}\left[\left(W_{EQ}{}^{2} - \left(\frac{F_B}{R_L}\right)^{2}\right)^{1/2} + W_M \times \sin\theta\right], \tag{E.32}$$

$$B_{V\max} = \frac{5730}{F_B}\left[\left(W_{EQ}{}^{2} - \left(\frac{F_B}{R_L}\right)^{2}\right)^{1/2} + W_M \times \sin\theta\right], \tag{E.33}$$

where

F_B is the critical buckling force (+ compressive), expressed in pounds (lb);

B_{VMIN} is the minimum vertical curvature rate to cause buckling (+building, -dropping), expressed in degrees per 100 feet (°/100 ft);

B_{VMAX} is the maximum vertical curvature rate that buckles pipe (+building, -dropping), expressed in degrees per 100 feet (°/100 ft);

W_{EQ} is the equivalent pipe weight required to buckle pipe at F_B axial load, expressed in pounds (lb);

E is 29,6x10^6 psi;

$$I = \frac{0.7854\left(OD^4 - ID^4\right)}{16};$$

W_M is the buoyant weight of pipe, expressed in pounds per foot (lb/ft),

$$= W_A\left(\frac{65.5 - MW}{65.5}\right)$$

where

W_A is the actual weight in air, expressed in pounds per foot (lb/ft);

MW is the mud density, expressed in pounds per gallon (lb/gal);

h_C is the radial clearance of tool joint to hole, expressed in inches (in),

$$= \left(\frac{D_H - TJOD}{2}\right)$$

where

D_H is the diameter of hole, expressed in inches (in);

$TJOD$ is the OD tool joints, expressed in inches (in);

R_L is the lateral build radius, expressed in feet (ft),

$$= \frac{5730}{B_L}$$

where

B_L is the llateral curvature rate, expressed in degrees per 100 feet (°/100 ft),

$$= \sqrt{{B_T}^2 - {B_V}^2}$$

where

B_T is the total curvature rate, expressed in degrees per 100 feet (°/100 ft);

Θ is the inclination angle, expressed in degrees (°).

E.14.2 If the hole curvature is limited to the vertical plane, the buckling equations simplify to the following:

$$W_{EQ} = \frac{12 \times h_C \times {F_B}^2}{4 \times E \times I}, \tag{E.34}$$

$$B_{V\,\min} = \frac{-5730 \times \left(W_{EQ} + W_M \times \sin\theta\right)}{F_B}, \tag{E.35}$$

$$B_{V\,\max} = \frac{5730 \times \left(W_{EQ} + W_M \times \sin\theta\right)}{F_B}, \tag{E.36}$$

where

B_{VMIN} is the minimum vertical curvature rate for buckling (+building, -dropping), expressed in degrees per 100 feet (°/100 ft);

B_{VMAX} is the maximum vertical curvature rate for buckling (+building, -dropping), expressed in degrees per 100 feet (°/100 ft);

F_B is the buckling force (+ compressive), expressed in pounds (lb);

E is 29,6x10^6 psi;

$$I = \frac{\pi}{64}\left(OD^4 - ID^4\right);$$

W_{EQ} is the buoyant weight equivalent for pipe in curved borehole, expressed in pounds per foot (lb/ft);

W_M is the buoyant weight of pipe, expressed in pounds per foot (lb/ft),

$$= W_A\left(\frac{65.5 - MW}{65.5}\right)$$

where

W_A is the actual weight of pipe in air, expressed in pounds per foot (lb/ft);

MW is the mud density, expressed in pounds per gallon (lb/gal);

h_C is the radial clearance of tool joint to hole, expressed in inches (in),

$$= \left(\frac{D_H - TJOD}{2} \right)$$

where

D_H is the diameter of hole, expressed in inches (in);

$TJOD$ is the OD tool joints, expressed in inches (in);

Θ is the inclination angle, expressed in degrees (°).

E.14.3 Figures E.4 and E.5 show the effect of hole curvature on the buckling force for 5 in and 3½ in drillpipe. Figure E.6 shows the effect of lateral curvatures on the buckling force of 5 in drillpipe. For lateral and upward curvatures, the critical buckling force increases with the total curvature rate.

Figure E.4 — Buckling force vs. hole curvature

Figure E.5 — Buckling force vs. hole curvature

Figure E.6 — Buckling force vs. hole curvature

E.15 Bending stresses on compressively loaded drillpipe in curved boreholes

E.15.1 The type of loading can be determined by comparing the actual hole curvature to calculated values of the critical curvatures that define the transition from no pipe body contact to point contact and from point contact to wrap contact. The two critical curvatures are computed from the following equations.

$$B_C = \frac{57.3 \times 100 \times 12 \times \Delta D}{J \times L\left[\tan\left(\frac{57.3 \times L}{4 \times J}\right) - \frac{L}{4 \times J}\right]} \qquad \text{(E.37)}$$

where

B_C is the critical hole curvature that defines the transition from no pipe body contact to point contact, expressed in degrees per 100 feet (°/100 ft);

$$\Delta D = (TJOD - OD)$$

where

TJOD is the tool joint outside diameter, expressed in inches (in);

OD is the pipe body outside diameter, expressed in inches (in);

$J = \left(\frac{E \times I}{F}\right)^{1/2}$, expressed in inches (in);

E is Young's modulus 30×10^6 for steel, expressed in pounds per square inch (psi);

I is the moment of inertia of pipe body, expressed in inches (in);

$$= \frac{\pi\left(OD^4 - ID^4\right)}{64}$$

where

ID is the pipe body inside diameter, expressed in inches (in);

F is the axial compressive load on pipe, expressed in pounds (lb);

L is the length of one joint of pipe, expressed in inches (in).

$$B_W = \frac{57.3 \times 100 \times 12 \times \Delta D}{J \times L\left[\frac{4J}{L} + \frac{L}{4J \times \tan^2\left(\frac{57.3L}{4J}\right)} - \frac{2}{\tan\left(\frac{57.3L}{4J}\right)}\right]}$$

where

B_W is the critical curvature that defines the transition from point contact to wrap contact, expressed in degrees per 100 feet (°/100 ft);

$$\Delta D = (TJOD - OD)$$

where

TJOD is the tool joint outside diameter, expressed in inches (in);

OD is the pipe body outside diameter, expressed in inches (in);

$$J = \left(\frac{E \times I}{F}\right)^{1/2}$$, expressed in inches (in);

E is Young's modulus 30x10^{6} for steel, expressed in pounds per square inch (psi);

I is the moment of inertia of pipe body, expressed in inches (in);

$$= \frac{\pi\left(OD^4 - ID^4\right)}{64}$$

where

ID is the pipe body inside diameter, expressed in inches (in);

F is the axial compressive load on pipe, expressed in pounds (lb);

L is the length of one joint of pipe, expressed in inches (in).

E.15.2 If the hole curvature is less than the critical curvature required to begin point contact, the maximum bending stress is given by the following:

$$S_B = \frac{B \times OD \times F \times J \times L}{57.3 \times 100 \times 12 \times 4 \times I \times \sin\left(\frac{57.3L}{2J}\right)} \tag{E.38}$$

where

SB is the maximum bending stress, expressed in pounds per square inch (psi);

B is the hole curvature

F is the axial compressive load on pipe, expressed in pounds (lb);

$$J = \left(\frac{E \times I}{F}\right)^{1/2}$$, expressed in inches (in);

E is Young's modulus 30x10^{6} for steel, expressed in pounds per square inch (psi);

I is the moment of inertia, expressed in inches (in);

$$= \frac{\pi\left(OD^4 - ID^4\right)}{64}$$

where

OD is the pipe body outside diameter, expressed in inches (in);

ID is the pipe body inside diameter, expressed in inches (in);

L is the length of one joint of pipe, expressed in inches (in).

E.15.3 If the hole curvature is between the two critical curvatures calculated, the pipe will have center body point contact and the maximum bending stress is given by the following equation:

$$S_b = \frac{E \times OD \times U^2}{4R}\left[\frac{A \times \sin\theta + B \times \cos\theta}{U \times \sin U - 4 \times \sin^2\left(\frac{U}{2}\right)}\right] \tag{E.39}$$

where

E is Young's modulus 29,6x10^6 for steel, expressed in pounds per square inch (psi);

OD is the pipe body outside diameter, expressed in inches (in);

ID is the pipe body inside diameter, expressed in inches (in);

$U = \frac{L_{WC}}{2J}$;

L_{PTC} is the length of one joint of drillpipe for point contact of pipe body, expressed in inches (in)

L_{wc} is L_E for wrap contact, expressed in inches (in);

$J = \left(\frac{E \times I}{F}\right)^{1/2}$, expressed in inches (in);

I is the moment of inertia, expressed in inches (in);

$$= \frac{\pi\left(OD^4 - ID^4\right)}{64}$$

F is the axial compressive load on pipe, expressed in pounds (lb);

$A = \left(1 + \frac{4 \times R \times \Delta D}{L^2}\right)\sin U - \frac{4}{U} \times \sin^2\left(\frac{U}{2}\right)$;

$B = 2\left[1 - \frac{\sin U}{U} - \left(1 + \frac{4 \times R \times \Delta D}{L^2}\right)\sin^2\left(\frac{U}{2}\right)\right]$;

$\theta = \arctan\left(\frac{A}{B}\right)$;

ΔD is the diameter difference tool joint minus pipe body outside diameter, expressed in inches (in);

$$= (TJOD - OD)$$

where

TJOD is the tool joint outside diameter, expressed in inches (in);

$$R = 573 \times 100 \times 12B$$

B is the hole curvature, expressed in degrees per 100 ft (°/100 ft).

E.15.4 If the hole curvature exceeds the critical curvature that separates point contact from wrap contact, we need to first compute an effective pipe length in order to calculate the maximum bending stress. The effective pipe span length is calculated from the following equation by trial and error until the calculated curvature matches the actual hole curvature.

$$B = \frac{57.3 \times 100 \times 12 \times \Delta D}{J \times L_E \left[\frac{4J}{L_E} + \frac{L_E}{4J \times \tan^2\left(\frac{57.3 L_E}{4J}\right)} - \frac{2}{\tan\left(\frac{57.3 L_E}{4J}\right)} \right]} \qquad \text{(E.40)}$$

where

L_E is the effective span length, expressed in inches (in);

B is the hole curvature, expressed in degrees per 100 ft (°/100 ft);

ΔD is the diameter difference tool joint minus pipe body outside diameter, expressed in inches (in);

$$= (TJOD - OD)$$

where

TJOD is the tool joint outside diameter, expressed in inches (in);

OD is the pipe body outside diameter, expressed in inches (in);

$J = \left(\frac{E \times I}{F}\right)^{1/2}$, expressed in inches (in);

where

E is Young's modulus $29{,}6 \times 10^6$ for steel, expressed in pounds per square inch (psi);

I is the moment of inertia, expressed in inches (in);

$$= \frac{\pi\left(OD^4 - ID^4\right)}{64}$$

F is the axial compressive load on pipe, expressed in pounds (lb);

L_W is the length of pipe body touching hole, expressed in inches (in);

$$= L - L_E$$

where

L is the length of one joint of pipe, expressed in inches (in).

E.15.5 The maximum bending stresses can then be computed using the equation for point contact and a pipe body length equal to the effective span length.

E.15.6 One of our major concerns when drilling with compressively loaded drillpipe is the magnitude of the lateral contact forces between the tool joints and the wall of the hole and the pipe body and the wall of the hole. Various authors have suggested operating limits in the range of two to three thousand pounds or more for tool joint contact faces. There are no generally accepted operating limits for compressively loaded pipe body contact forces. For loading conditions in which there is no pipe body contact, the lateral force on the tool joints is given by:

$$LF_{TJ} = \frac{F \times L \times B}{57.3 \times 100 \times 12} \tag{E.41}$$

where

LF_{TJ} is the lateral force on tool joint, expressed in pounds (lb);

L is the length of one joint of pipe, expressed in inches (in);

B is the hole curvature, expressed in degrees per 100 ft (°/100 ft).

E.15.7 For loading conditions with point or wrap contact, the following equations give the contact forces for the tool joint and the pipe body:

$$LF_{TJ} = \frac{2 \times E \times I \times U^2}{R \times L_E} \left[\frac{\left(1 - \frac{4R \times \Delta D}{L_E{}^2}\right) \sin U - \frac{4}{U} \sin^2\left(\frac{U}{2}\right)}{\sin U - \frac{4}{U} \sin^2\left(\frac{U}{2}\right)} \right] \tag{E.42}$$

where

LF_{PIPE} is the lateral force on pipe body, expressed in pounds (lb);

$$= \frac{F \times L}{R} - LF_{TJ}$$

LF_{TJ} is the lateral force on tool joint, expressed in pounds (lb);

L_W is the length of pipe for wrap contact, expressed in inches (in);

$$= L - L_E$$

L_E is the effective span length for wrap contact and equals L for point contact, expressed in inches (in);

$$R = \frac{57.3 \times 100 \times 12}{B};$$

B is the hole curvature, expressed in degrees per 100 ft (°/100 ft);

$$U = \frac{L_E}{2J};$$

$$J = \left(\frac{E \times I}{F} \right)^{1/2}$$, expressed in inches (in);

ΔD is the diameter difference tool joint minus pipe outside diameter, expressed in inches (in);

$$= TJOD - OD$$

$$I = \frac{\pi}{64}\left(OD^4 - ID^4\right).$$

Annex F
(informative)

Conversion from/to USC and Metric

Procedures used to convert from USC units to SI units

F.1 Background

The following procedures were adopted in this International Standard for converting units from the United States Customary (USC) system into the Système Internationale (SI).

F.2 General

F.2.1 Rounding

The last retained digit in a number was unchanged when the next digit was less than 5 or raised when it was greater than 5.

When the digit following the last retained digit was exactly 5 followed by all zeros, the last retained digit was unchanged if it was even, or was raised if it was odd.

F.2.2 Fractions

Fractions or numbers with fractions in USC units were converted to their full decimal equivalents in USC units without rounding. The full decimal equivalents in USC units were then converted to SI values using the following formula:

$$N_M = 25{,}4 \times N \tag{F.1}$$

where

N_M is the SI equivalent of a USC fraction or USC number with fractions (in inches), expressed in millimeters;

N is the full decimal equivalent, expressed in inches, of a USC fraction or number with fractions, which has not been rounded.

The converted SI values, in millimetres, for the equivalent of USC fractions or numbers with fractions were rounded to the appropriate number of places for the application.

F.3 Dimensions

F.3.1 Outside diameter, pipe body, pipe body upsets, tool joints, drill collars and subs

The SI values for the outside diameter of drill collars and subs that are included in ISO 10424-1 and SI values for the outside diameter of drill pipe (pipe body, upsets and tool joints) that are included in ISO 11961 were copied from the appropriate specification.

SI values for outside diameters of the pipe body, pipe body upsets, tool joints, drill collars and subs that are not included in ISO 10424-1 or ISO 11961 were converted from USC values using the following formula:

$$D_M = 25{,}4 \times D \tag{F.2}$$

where

D_M is the outside diameter, expressed in millimetres (mm);

D is the *specified* outside diameter, expressed in inches (in).

NOTE The converted SI values for the outside diameters of the pipe body, pipe body upsets and tool joints were rounded to the nearest 0,01 mm.

The converted SI values for the outside diameters of drill collars and subs were rounded to the nearest 0,1 mm.

F.3.2 Wall thickness, pipe body

The SI values that are included in ISO 11961 for the wall thickness of the pipe body were copied from ISO 11961.

SI values for the wall thickness of the pipe body that are not included in ISO 11961 were converted from USC values using the following formula:

$$t_M = 25{,}4 \times t \tag{F.3}$$

where

t_M is the wall thickness, expressed in millimetres (mm);

t is the specified wall thickness, expressed in inches (in).

The converted SI values for the wall thickness of the pipe body were rounded to the nearest 0,01 millimeter.

F.3.3 Inside diameter

F.3.3.1 Inside diameter, pipe body

The SI values that are included in ISO 11961 for inside diameters of the pipe body were copied from ISO 11961.

SI values for the inside diameters of the pipe body that are not included in ISO 11961 were calculated using the following formula:

$$d_M = D_M - (2 \times t_M) \tag{F.4}$$

where

d_M is the inside diameter, expressed in millimetres (mm);

D_M is the outside diameter, expressed in millimetres (mm);

t_M is the wall thickness, expressed in millimetres (mm).

The calculated SI values for inside diameters of the drill pipe body were rounded to the nearest 0,01 millimeters.

F.3.3.2 Inside diameter, drill pipe body upsets, tool joints, drill collars and subs

The SI values for the inside diameter of drill collars and subs that are included in ISO 10424-1 and SI values for the inside diameter of pipe upsets and tool joints that are included in ISO 11961 were copied from the appropriate specification.

SI values for inside diameters of pipe body upsets, tool joints, drill collars and subs that are not included in ISO 10424-1 or ISO 11961 were converted from USC values using the following formula:

$$d_M = 25,4 \times d \quad \text{(F.5)}$$

where

d_M is the inside diameter, expressed in millimeters;

d is the inside diameter, expressed in inches.

The converted SI values for the inside diameters of drill pipe body upsets and tool joints were rounded to the nearest 0,01 mm.

The converted SI values for the inside diameters of drill collars and subs were rounded to the nearest 0,1 mm.

F.4 Section Modulus

The USC values for the section modulus that are included in API 7G were converted to SI values using the following formula:

$$S_{X-M} = 16\,387,06 \times S_X \quad \text{(F.6)}$$

where

S_{X-M} is the section modulus, expressed in cubic millimetres (mm);

S_X is the section modulus, expressed in cubic inches (in).

The converted SI values for the section modulus were rounded to the nearest cubic millimeter.

F.5 Polar Section Modulus

The USC values for the polar section modulus that are included in API 7G were converted to SI values using the following formula:

$$Z_M = 16,387\,06 \times Z \quad \text{(F.7)}$$

where

Z_M is the polar section modulus, expressed in cubic centimetres (cm^3);

Z is the polar section modulus, expressed in cubic inches (in^3).

The converted SI values for the polar section modulus were rounded to the nearest 0,001 cm^3.

F.6 Yield and tensile strengths

The SI values that are included in ISO 10424-1 or ISO 11961 for yield strengths and tensile strengths were copied from the appropriate specification.

SI values for yield strengths and tensile strengths that are not included in ISO 10424-1 or ISO 11961 were converted from USC values using the following formula:

$$S_M = K_S \times S \qquad \text{(F.8)}$$

where

S_M is the yield strength or tensile strength, expressed in megapascals (MPa);

S is the yield strength or tensile strength, expressed in pounds per square inch (psi);

K_S is the conversion factor,

0,006 894 757 for ISO 10424-1 products

0,006 894 76 for ISO 11961 products (drill pipe and tool joints).

The converted SI values for the yield strengths and tensile strengths were rounded to the nearest megapascal.

F.7 Performance Properties, pipe body and kellys

F.7.1 Internal pressure at minimum yield strength, pipe body and kellys

The USC values for the internal pressure at minimum yield strength of the pipe body and kellys that are included in API 7G were converted to SI values using the following formula:

$$P_{IM} = 0{,}006\,894\,757 \times P_I \qquad \text{(F.9)}$$

where

P_{IM} is the internal pressure at minimum yield strength of the pipe body, expressed in megapascals (MPa);

P_I is the internal pressure at minimum yield strength of the pipe body, expressed in pounds per square inch (psi).

The converted SI values for the internal pressures at minimum yield strength of the pipe body were rounded to the nearest megapascal.

F.7.2 Collapse pressure, pipe body

The USC values for the collapse pressure of the pipe body were converted to SI values using the following formula:

$$P_{CM} = 0{,}006\,894\,757 \times P_C \qquad \text{(F.10)}$$

where

P_{CM} is the collapse pressure of the pipe body, expressed in megapascals;

P_C is the collapse pressure of the pipe body, expressed in pounds per square inch.

The converted SI values for the collapse pressure of the pipe body were rounded to the nearest megapascal.

F.7.3 Load at minimum yield strength, pipe body, tool joints and kellys

The USC values for the loads at the minimum yield strength of the pipe body, tool joints and kellys that are included in API 7G were converted to SI values using the following formula:

$$F_{YM} = 4{,}448\ 222 \times F_Y \tag{F.11}$$

where

F_{YM} is the load at minimum yield strength of the pipe body, tool joint and kelly, expressed in Newtons (N);

F_Y is the load at minimum yield strength of the pipe body, tool joint and kelly, expressed in pounds (lb).

The converted SI values for the loads at minimum yield strength for the pipe body, tool joint or kelly were rounded to the nearest megapascal.

F.8 Torque

The USC values for the torsional yield strength of the pipe body, the pipe body of eccentrically worn drill pipe, rotary shouldered connections and the drive section of kellys; and the make-up torque of rotary shouldered connections that are included in API 7G were converted to SI values using the following formula:

$$T_{Y-M} = 1{,}355\ 818 \times T_Y \tag{F.12}$$

where

$T_{Y\text{-}M}$ is the torsional yield strength of the pipe body, the pipe body of eccentrically worn drill pipe, rotary shouldered connections and the drive section of kellys; and the make-up torque of rotary shouldered connections expressed as Newton-metres (N·m);

T_Y is the torsional yield strength of the pipe body, the pipe body of eccentrically worn drill pipe, rotary shouldered connections and the drive section of kellys; and the make-up torque of rotary shouldered connections expressed as foot-pounds (ft-lb).

The converted SI values for the torsional yield strength of the pipe body, the pipe body of eccentrically worn drill pipe, rotary shouldered connections and the drive section of kellys; and the make-up torque of rotary shouldered connections were round to the nearest Newton-metres.

F.9 Temperature

Temperatures in degrees Fahrenheit (USC) were converted to temperatures in degrees Celsius (SI) using the following formula:

$$°C = (°F - 32) \times 5/9 \tag{F.13}$$

where

°C is the temperature, expressed in degrees Celsius;

°F is the temperature, expressed in degrees Fahrenheit.

The converted SI values for temperatures were rounded to the nearest degree Celsius.

F.10 Approximate weight per foot of drill pipe

The USC values for the approximate weight per foot of drill pipe were converted to SI values using the following formula:

$$W_{DP-M} = 1{,}488\,164 \times W_{DP} \tag{F.14}$$

where

W_{DP-M} is the approximate mass per meter of the drill pipe, expressed in kilogram per metre (kg/m);

W_{DP} is the approximate weight per foot of the drill pipe, expressed in pound per foot (lb/ft).

The converted SI values for the mass per meter were rounded to the nearest 0.01 kg/m.

F.11 Force to hoist 1 meter of drill pipe

The SI value for the force to hoist 1 meter of drill pipe was calculated using the following formula:

$$F_{DP-M} = 9{,}806\,650 \times W_{DP-M} \tag{F.15}$$

where:

F_{DP-M} is the force to hoist 1 m of drill pipe, expressed in Newtons (N);

W_{DP-M} is the approximate mass per meter of the drill pipe, expressed in kilogram per metre (kg/m).

The converted SI values for the force to hoist 1 meter of drill pipe were rounded to the nearest Newton.

F.12 Mass of the drill collars and bottom hole assembly

The USC value for the mass of the drill collars and bottom hole assembly was converted to an SI value using the following formula:

$$W_{D+B+M} = 0{,}453\,592\,4 \times W_{D+B} \tag{F.16}$$

where:

W_{D+B-M} is the mass of the drill collars and bottom hole assembly, expressed in kilograms (kg);

W_{D+B} is the weight of the drill collars and bottom hole assembly, expressed in pounds (lb).

The converted SI values for the mass of the drill collars and bottom hole assembly were rounded to the nearest kilogram.

F.13 Force to hoist the drill collars and bottom hole assembly

The SI value for the force to hoist the drill collars and the bottom hole assembly was calculated using the following formula:

$$F_{D+B+M} = 9{,}806\ 650 \times W_{D+B+M} \quad \text{(F.17)}$$

where:

F_{D+B-M} is the force to hoist the drill collars and bottom hole assembly, expressed in Newtons (N);

W_{D+B-M} is the weight of the drill collars and bottom hole assembly, expressed in kilograms (kg).

The converted SI values for the force to hoist the drill collars and bottom hole assembly were rounded to the nearest Newton.

F.14 Margin of overpull

The USC value for the margin of overpull was converted to an SI value using the following formula:

$$MOP_M = 4{,}448\ 222 \times MOP \quad \text{(F.18)}$$

where:

MOP_M is the margin of overpull, expressed in Newtons (N);

MOP is the margin of overpull, expressed in pounds (lb).

The converted SI values for the margin of overpull were rounded to the nearest Newton

Bibliography

[1] API Bulletin 5C3, *Formulas and calculations for casing, tubing, drill pipe and line pipe properties*

[2] ASTM D3370, *Standard practices for sampling water*

[3] Azar, J. J. and Lummus, J. L., “Mud pH vs Corrosion Performance,” *Petroleum Engineer*, Vol. 51, No. 3, pp. 72-78, 1976.

[4] Bachman, W. S., “Fatigue Testing and Development of Drill Pipe to Tool Joint Connections,” World Oil, vol, 132,, No. 1, pp. 104-116, 1951.

[5] Casner, John A., "Endurance Limit of Drill Pipw," letter 1/21/95 to John Altermann.

[6] Castleberry, G., *Machine Design*, Vol. 50, No. 4, pp. 108-110, February 23, 1978.

[7] Dale, B. A., “An experimental investigation of fatigue crack growth in drillstring tubulars,” SPE Drilling Engineering, Vol. 3, No. 4, pp. 356-362, 1988.

[8] Dawson, Rapier, and Paslay, P.R., "Drillpipe Buckling in Inclined Holes," *Journal of Petroleum Technology*, October 1984.

[9] Faupel, J. H. and Fisher, F. F., *Engineering Design, A Synthesis of Stress Analysis and Materials Engineering*, 2nd Edition, John Wiley and Sons, New York, Chapter 6, pp. 386-392 (1981).

[10] Grant, R. S. and Texter, H. G., “Causes and Prevention of Drill Pipe and Tool Joint Troubles,” *Drilling and Production Practice, American Petroleum Institute*, New York, NY., pp. 9-48, 1941.

[11] Grondin, G. Y. and Kulak, G. L., Fatigue of Drill Pipe, Report by the Department of Civil Engineering, University of Alberta, Structural Engineering report 171, 1991.

[12] Grondin, G. Y. and Kulak, G. L., Fatigue Testing of Drillpipe, *SPE Drilling and Completion,* June 1994, pp. 94-102, 1994.

[13] Hansford, John E.; Lubinski, Arthur, "Cumulative Fatigue Damage of Drill Pipe in Dog Legs," *Journal of Petroleum Technology*, March 1966.

[14] Hansford, J. E. and Lubinski, Arthur, “Effects of Drilling Vessel Pitch or Roll on Kelly and Drill Pipe Fatigue,” *Journal of Petroleum Technology,* Vol. 16., No. 1, pp. 77-86, 1964.

[15] Helbig, R. and Vogt, G. H., “Reversed Bending Fatigue Strength of Drill Strings Subject to the Attack of Drilling Fluids,” *Oil and Gas European Magazine*, No. 2, pp. 16-20, 1987.

[16] ISO 10400, *Petroleum and natural gas industries – Formulae and calculation for casing, tubing, drill pipe and line pipe properties*

[17] Johancsik, C.A.; Freisen, D.B.; and Dawson, R.; "Torque and Drag in Directional wells – Prediction and Measurement," *Journal of Petroleum Technology*, June 1984.

[18] Joosten, M. W., Shute, J., and Ferguson, R. A., “New Study Shows How to Predict Accumulated Drill Pipe Fatigue,” *World Oil*, Vol. 201, No. 5., pp. 65-70, 1985.

[19] Juvinall, R. C., *Engineeting Considerations of Stress, Strain and Strength*, McGraw Hill, New York, NY, pg 222, 1967.

[20] Kral, E., Sengupta, P. K., Newlin, L., Quan, S. S., "Fracture mechanics estimates drillpipe fatigue, part I," *Oil and Gas Journal*, Vol. 82 (33), pp. 50-55, 1984a.

[21] Kral, E., Sengupta, P. K., Newlin, L., Quan, S. S., "Fracture mechanics estimates drillpipe fatigue, part II,:" *Oil and Gas Journal*, Vol. 82 (33), pp. 115-121, 1984b.

[22] Lubinski, Arthur, "Maximum Permissible Dog-Legs in Rotary Boreholes," *Journal of Petroleum Technology*, February 1961.

[23] Miner, M. A., "Cumulative Damage in fatigue," *Journal of Applied Mechanics*., Vol. 12., 1945.

[24] Mischke, C. R., "Prediction of Stochastic Endurance Strength," *Transactions of the ASME Journal of Vibration, Acoustics, Stress and Reliability in design*, Vol. 109, No. 1, pp. 113-122, 1987.

[25] Morgan, R.P.; Roblin, M.J., "A Method for the Investigation of Fatigue Strength in Seamless Drillpipe," ASME Conference, Tulsa, Oklahoma, September 22, 1969.

[26] Nicholson, Robert W., "Minimize Drill Pipe Damage and Hole Problems. Follow Acceptable Dogleg Severity Limits," *Transactions of the 1974 International Association of Drilling Contractors (IADC) Rotary Drilling Conference*.

[27] Paslay, P. R. and Cernocky, E. P., "Bending Stress Magnification in Constant Curvature Doglegs with Impact on Drillstring and Casing," SPE Paper No. 22547 Presented at the 66th Annual Technical Conference and Exhibition, Dallas, TX., Oct. 6-9, 1991.

[28] Payne, M. L. and Sathuvalli, U. B., "An assessment of data on fatigue endurance limits of API drillpipe," A report submitted to ISO 10407 Workgroup, 2004.

[29] Payne, M.L.; Duxbury, J.K.; and Martin, J.W.; "Drillstring Design Options for Extended-Reach Drilling Operations," ASME ETCE, Houston, Texas, 30 January – 1 February, 1995.

[30] Rollins, H. M., "What we know about drill-pipe fatigue failure," *Oil and Gas Journal*, Vol. 64, No. 16, pp. 98-107, 1966.

[31] Schenk, K.D., "Calco Learns About Drilling Through Excessive Doglegs," *Oil and Gas Journal*, October 12, 1964.

[32] Shigley, J. E. and Mischke, C. R., *Mechanical Engineering Design*, Fifth Edition, McGraw Hill, Inc., New York, 1989.

[33] Stall, J.C.; Blenkarn, K.A., "Allowable Hook Load and Torque Combinations for Stuck Drill String," Mid-Continent API District Meeting, Paper No. 851-36-M, April 6, 1962.

[34] *Standard DS-1, Drill Stem Design and Inspection, Second Edition*, T.H. Hill Associates, Inc., Houston, Texas, August 1997.

[35] Wilson, G.J., "Dogleg Control in Directionally Drilled Wells," *Transactions of the American Institute of Mining, Metallurgical, and Petroleum Engineers* (AIME), Vol. 240, 1967.

[36] Wilson, G.J., "How to Select Bottom Hole Drilling Assemblies", *Petroleum Engineer International Magazine*, March and April, 1979.

[37] Young, Warren C. and Budynas, Richard. Roarks Formulas for Stress and Strain. 7th Ed. McGraw-Hill, 2002. ISBN 0-07-072542-X.