Recommended Practice for Drill Stem Design and Operating Limits

API RECOMMENDED PRACTICE 7G
SIXTEENTH EDITION, AUGUST 1998

EFFECTIVE DATE: DECEMBER 1, 1998

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Exploration and Production Department

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FOREWORD

This recommended practice is under the jurisdiction of the API Subcommittee on Standardization of Drilling and Servicing Equipment.

The purpose of this recommended practice is to standardize techniques for the procedure of drill stem design and to define the operating limits of the drill stem.

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This recommended practice shall become effective on the date printed on the cover but may be used voluntarily from the date of distribution.

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Recommended Practice for Drill Stem Design and Operating Limits

1 Scope

1.1 COVERAGE

This recommended practice involves not only the selection of drill string members, but also the consideration of hole angle control, drilling fluids, weight and rotary speed, and other operational procedures.

1.2 SECTION COVERAGE

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

2 References

(See also Appendix B.)

API

RP 5C1	*Care and Use of Casing and Tubing*
Bull 5C3	*Bulletin on Formulas and Calculations for Casing, Tubing, Drill Pipe, and Line Pipe Properties*
Spec 7	*Specification for Rotary Drill Stem Elements*
RP 7A1	*Recommended Practice for Testing of Thread Compounds for Rotary Shouldered Connections*
RP 13B-1	*Recommended Practice Standard Procedure for Field Testing Water-Based Drilling Fluids*
RP 13B-2	*Recommended Practice Standard Procedure for Field Testing Oil-Based Drilling Fluids*

ASTM[1]

D3370	*Standard Practices for Sampling Water*

NACE[2]

MR-01-75	*Sulfide Stress Cracking Resistant Metallic Material for Oil Field Equipment*

[1]American Society for Testing Materials, 100 Barr Harbor Drive, West Conshocken, Pennsylvania 19428.
[2]NACE International, P.O. Box 218340, Houston, Texas 77218-8340.

3 Definitions

3.1 bending strength ratio: The ratio of the section modulus of a rotary shouldered box at the point in the box where the pin ends when made up divided by the section modulus of the rotary shouldered pin at the last engaged thread.

3.2 bevel diameter: The outer diameter of the contact face of the rotary shouldered connection.

3.3 bit sub: A sub, usually with 2 box connections, that is used to connect the bit to the drill string.

3.4 box connection: A threaded connection on Oil Country Tubular Goods (OCTG) that has internal (female) threads.

3.5 calibration system: A documented system of gauge calibration and control.

3.6 Class 2: An API service classification for used drill pipe and tubing work strings.

3.7 cold working: Plastic deformation of metal at a temperature low enough to insure or cause permanent strain.

3.8 corrosion: The alteration and degradation of material by its environment.

3.9 critical rotary speed: A rotary speed at which harmonic vibrations occur. These vibrations may cause fatigue failures, excessive wear, or bending.

3.10 decarburization: The 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.

3.11 dedendum: The distance between the pitch line and root of thread.

3.12 dogleg: A term applied to a sharp change of direction in a wellbore or ditch. Applied also to the permanent bending of wire rope or pipe.

3.13 dogleg severity: A measure of the amount of change in the inclination and/or direction of a borehole, usually expressed in degrees per 100 feet of course length.

3.14 drift: A drift is a gauge used to check minimum ID of loops, flowlines, nipples, tubing, casing, drill pipe, and drill collars.

3.15 drill collar: Thick-walled pipe or tube designed to provide stiffness and concentration of weight at the bit.

3.16 drill pipe: A length of tube, usually steel, to which special threaded connections called tool joints are attached.

3.17 drill string element: Drill pipe with tool joints attached.

3.18 failure: Improper performance of a device or equipment that prevents completion of its design function.

3.19 fatigue: The process of progressive localized permanent structural change occurring in a material subjected to conditions that produce fluctuating stresses and strains at some point or points and that may culminate in cracks or complete fracture after a sufficient number of fluctuations.

3.20 fatigue failure: A failure which originates as a result of repeated or fluctuating stresses having maximum values less than the tensile strength of the material.

3.21 fatigue crack: A crack resulting from fatigue. See fatigue.

3.22 forging: (1) Plastically deforming metal, usually hot, into desired shapes with compressive force, with or without dies. (2) A shaped metal part formed by the forging method.

3.23 kelly: The square or hexagonal shaped steel pipe connecting the swivel to the drill string. The kelly moves through the rotary table and transmits torque to the drill string.

3.24 kelly saver sub: A short substitute that is made up onto the bottom of the kelly to protect the pin end of the kelly from wear during make-up and break-out operations.

3.25 last engaged thread: The last thread on the pin engaged with the box or the box engaged with the pin.

3.26 lower kelly valve: An essentially full-opening valve installed immediately below the kelly, with outside diameter equal to the tool joint outside diameter. Valve can be closed to remove the kelly under pressure and can be stripped in the hole for snubbing operations.

3.27 make-up shoulder: The sealing shoulder on a rotary shouldered connection.

3.28 minimum make-up torque: The minimum make-up torque is the minimum amount of torque necessary to develop an arbitrarily derived tensile stress in the pin or compressive stress in the box. This arbitrarily derived stress level is perceived as being sufficient in most drilling conditions to prevent downhole make-up and to prevent shoulder separation from bending loads.

3.29 minimum OD: For tool joints on drill pipe with rotary shouldered connections, the minimum OD is the minimum box OD that will allow the connection to remain as strong as a specified percentage of the drill pipe tube in torsion.

3.30 oil muds: The term "oil-base drilling fluid" is applied to a special type drilling fluid where oil is the continuous phase and water the dispersed phase. Such fluids contain blown asphalt and usually 1 to 5 percent water emulsified into the system with caustic soda or quick lime and an organic acid. Silicate, salt, and phosphate may also be present. Oil-base drilling fluids are differentiated from invert-emulsion drilling fluids (both water-in-oil emulsions) by the amounts of water used, method of controlling viscosity and thixotropic properties, wall-building materials, and fluid loss.

3.31 pin end: A threaded connection on Oil Country Tubular Goods (OCTG) that has external (male) threads.

3.32 plain end: Drill pipe, tubing, or casing without threads. The pipe ends may or may not be upset.

3.33 premium class: An API service classification for used drill pipe and tubing work strings.

3.34 quenched and tempered: Quench hardening—Hardening a ferrous alloy by austenitizing and then cooling rapidly enough so that some or all of the austenite transforms to martensite.
Tempering—Reheating a quenched-hardened or normalized ferrous alloy to a temperature below the transformation range and then cooling at any rate desired.

3.35 range: A length classification for API Oil Country Tubular Goods.

3.36 rotary shouldered connection: A connection used on drill string elements which has coarse, tapered threads and seating shoulders.

3.37 shear strength: The stress required to produce fracture in the plane of cross section, the conditions of loading being such that the directions of force and of resistance are parallel and opposite although their paths are offset a specified minimum amount. The maximum load divided by the original cross-sectional area of a section separated by shear.

3.38 slip area: The slip area is contained within a distance of 48 inches along the pipe body from the juncture of the tool joint OD and the elevator shoulder.

3.39 stress-relief feature: A modification performed on rotary shouldered connections which removes the unengaged threads of the pin or box. This process makes the joint more flexible and reduces the likelihood of fatigue cracking in this highly stressed area.

3.40 swivel: Device at the top of the drill stem which permits simultaneous circulation and rotation.

3.41 tensile strength: The maximum tensile stress which a material is capable of sustaining. Tensile strength is calculated from the maximum load during a tension test carried to rupture and the original cross-sectional area of the specimen.

3.42 test pressure: A pressure above working pressure used to demonstrate structural integrity of a pressure vessel.

3.43 thread form: The form of thread is the thread profile in an axial plane for a length of one pitch.

3.44 tolerance: The amount of variation permitted.

3.45 tool joint: A heavy coupling element for drill pipe having coarse, tapered threads and sealing shoulders designed to sustain the weight of the drill stem, withstand the strain of repeated make-up and break-out, resist fatigue, resist additional make-up during drilling, and provide a leak-proof seal. The male section (pin) is attached to one end of a length of drill pipe and the female section (box) is attached to the other end. Tool joints may be welded to the drill pipe, screwed onto the pipe, or a combination of screwed on and welded.

3.46 upper kelly cock: A valve immediately above the kelly that can be closed to confine pressures inside the drill string.

3.47 upset: A pipe end with increased wall thickness. The outside diameter may be increased, or the inside diameter may be reduced, or both. Upsets are usually formed by hot forging the pipe end.

3.48 working gauges: Gauges used for gauging product threads.

3.49 working pressure: The pressure to which a particular piece of equipment is subjected during normal operations.

3.50 working temperature: The temperature to which a particular piece of equipment is subjected during normal operations.

4 Properties of Drill Pipe and Tool Joints

4.1 This section contains a series of tables (Tables 1 through 11) 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.

4.2 All drill pipe and tool joint properties tables are included in Section 4.

4.3 Values listed in drill pipe tables are based on accepted standards of the industry and calculated from formulas in Appendix A.

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

4.5 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. The torque required to yield a rotary shouldered connection may be obtained from the equation in Section A.8.

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

4.7 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.

4.8 Minimum OD, box shoulder, and make-up torque values listed in Table 10 were determined using the following criteria:

4.8.1 Calculations for recommended tool joint make-up torque are based on the use of a thread compound containing 40 to 60 percent by weight of finely powdered metallic zinc applied to all threads and shoulders, and containing not more than 0.3 percent total active sulfur (reference the caution regarding the use of hazardous materials in Specification 7, Appendix G). Calculations are also based on a tensile stress of 60 percent of the minimum tensile yield for tool joints.

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

4.8.3 Premium Class Drill String is based on drill pipe having a minimum wall thickness of 80 percent.

4.8.4 Class 2 drill string allows drill pipe with a minimum wall thickness of 70 percent.

4.8.5 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 overdesigned for others.

Table 1—New Drill Pipe Dimensional Data

(1)	(2)	(3)	(4)	(5)	(6)	(7)
Size OD in. D	Nominal Weight Threads and Couplings, lb/ft	Plain End Weight[1] lb/ft	Wall Thickness in.	ID in. d	Section Area Body of Pipe[2] sq. in. A	Polar Sectional Modulus[3] cu. in. Z
$2^3/_8$	4.85	4.43	.190	1.995	1.3042	1.321
	6.65	6.26	.280	1.815	1.8429	1.733
$2^7/_8$	6.85	6.16	.217	2.441	1.8120	2.241
	10.40	9.72	.362	2.151	2.8579	3.204
$3^1/_2$	9.50	8.81	.254	2.992	2.5902	3.923
	13.30	12.31	.368	2.764	3.6209	5.144
	15.50	14.63	.449	2.602	4.3037	5.847
4	11.85	10.46	.262	3.476	3.0767	5.400
	14.00	12.93	.330	3.340	3.8048	6.458
	15.70	14.69	.380	3.240	4.3216	7.157
$4^1/_2$	13.75	12.24	.271	3.958	3.6004	7.184
	16.60	14.98	.337	3.826	4.4074	8.543
	20.00	18.69	.430	3.640	5.4981	10.232
	22.82	21.36	.500	3.500	6.2832	11.345
5	16.25	14.87	.296	4.408	4.3743	9.718
	19.50	17.93	.362	4.276	5.2746	11.415
	25.60	24.03	.500	4.000	7.0686	14.491
$5^1/_2$	19.20	16.87	.304	4.892	4.9624	12.221
	21.90	19.81	.361	4.778	5.8282	14.062
	24.70	22.54	.415	4.670	6.6296	15.688
$6^5/_8$	25.20	22.19	.330	5.965	6.5262	19.572
	27.70	24.22	.362	5.901	7.1227	21.156

[1] lb/ft = 3.3996 x A (col. 6)
[2] $A = 0.7854\,(D^2 - d^2)$
[3] $Z = 0.19635\left(\frac{D^4 - d^4}{D}\right)$

4.9 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 12.

4.10 The curves of Figures 1 through 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 Section 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 percent of the minimum yield for tool joints.

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

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

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

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

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

4.11.5 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.

4.11.6 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.

(Text continued on page 33.)

Table 2—New Drill Pipe Torsional and Tensile Data

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)
Size OD in.	Nominal Weight Threads and Couplings lb/ft	[1]Torsional Data Torsional Yield Strength, ft-lb				[2]Tensile Data Based on Minimum Values Load at the Minimum Yield Strength, lb			
		E75	X95	G105	S135	E75	X95	G105	S135
$2^3/_8$	4.85	4763.	6033.	6668.	8574.	97817.	123902.	136944.	176071.
	6.65	6250.	7917.	8751.	11251.	138214.	175072.	193500.	248786.
$2^7/_8$	6.85	8083.	10238.	11316.	14549.	135902.	172143.	190263.	244624.
	10.40	11554.	14635.	16176.	20798.	214344.	271503.	300082.	385820.
$3^1/_2$	9.50	14146.	17918.	19805.	25463.	194264.	246068.	271970.	349676.
	13.30	18551.	23498.	25972.	33392.	271569.	343988.	380197.	488825.
	15.50	21086.	26708.	29520.	37954.	322775.	408848.	451885.	580995.
4	11.85	19474.	24668.	27264.	35054.	230755.	292290.	323057.	415360.
	14.00	23288.	29498.	32603.	41918.	285359.	361454.	399502.	513646.
	15.70	25810.	32692.	36134.	46458.	324118.	410550.	453765.	583413.
$4^1/_2$	13.75	25907.	32816.	36270.	46633.	270034.	342043.	378047.	486061.
	16.60	30807.	39022.	43130.	55453.	330558.	418707.	462781.	595004.
	20.00	36901.	46741.	51661.	66421.	412358.	522320.	577301.	742244.
	22.82	40912.	51821.	57276.	73641.	471239.	596903.	659734.	848230.
5	16.25	35044.	44389.	49062.	63079.	328073.	415559.	459302.	590531.
	19.50	41167.	52144.	57633.	74100.	395595.	501087.	553833.	712070.
	25.60	52257.	66192.	73159.	94062.	530144.	671515.	742201.	954259.
$5^1/_2$	19.20	44074.	55826.	61703.	79332.	372181.	471429.	521053.	669925.
	21.90	50710.	64233.	70994.	91278.	437116.	553681.	611963.	786809.
	24.70	56574.	71660.	79204.	101833.	497222.	629814.	696111.	894999.
$6^5/_8$	25.20	70580.	89402.	98812.	127044.	489464.	619988.	685250.	881035.
	27.70	76295.	96640.	106813.	137330.	534199.	676651.	747877.	961556.

[1]Based on the shear strength equal to 57.7 percent of minimum yield strength and nominal wall thickness.
Minimum torsional yield strength calculated from Equation A.15.
[2]Minimum tensile strength calculated from Equation A.13.

Table 3—New Drill Pipe Collapse and Internal Pressure Data

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)
Size OD in.	Nominal Weight Threads and Couplings lb/ft	Collapse Pressure Based on Minimum Values, psi				Internal Pressure at Minimum Yield Strength, psi			
		E75	X95	G105	S135	E75	X95	G105	S135
$2^3/_8$	4.85	11040.	13984.	15456.	19035.	10500.	13300.	14700.	18900.
	6.65	15599.	19759.	21839.	28079.	15474.	19600.	21663.	27853.
$2^7/_8$	6.85	10467.	12940.	14020.	17034.	9907.	12548.	13869.	17832.
	10.40	16509.	20911.	23112.	29716.	16526.	20933.	23137.	29747.
$3^1/_2$	9.50	10001.	12077.	13055.	15748.	9525.	12065.	13335.	17145.
	13.30	14113.	17877.	19758.	25404.	13800.	17480.	19320.	24840.
	15.50	16774.	21247.	23484.	30194.	16838.	21328.	23573.	30308.
4	11.85	8381.	9978.	10708.	12618.	8597.	10889.	12036.	15474.
	14.00	11354.	14382.	15896.	20141.	10828.	13716.	15159.	19491.
	15.70	12896.	16335.	18055.	23213.	12469.	15794.	17456.	22444.
$4^1/_2$	13.75	7173.	8412.	8956.	10283.	7904.	10012.	11066.	14228.
	16.60	10392.	12765.	13825.	16773.	9829.	12450.	13761.	17693.
	20.00	12964.	16421.	18149.	23335.	12542.	15886.	17558.	22575.
	22.82	14815.	18765.	20741.	26667.	14583.	18472.	20417.	26250.
5	16.25	6938.	8108.	8616.	9831.	7770.	9842.	10878.	13986.
	19.50	9962.	12026.	12999.	15672.	9503.	12037.	13304.	17105.
	25.60	13500.	17100.	18900.	24300.	13125.	16625.	18375.	23625.
$5^1/_2$	19.20	6039.	6942.	7313.	8093.	7255.	9189.	10156.	13058.
	21.90	8413.	10019.	10753.	12679.	8615.	10912.	12061.	15507.
	24.70	10464.	12933.	14013.	17023.	9903.	12544.	13865.	17826.
$6^5/_8$	25.20	4788.	5321.	5500.	6036.	6538.	8281.	9153.	11768.
	27.70	5894.	6755.	7103.	7813.	7172.	9084.	10040.	12909.

Note: Calculations are based on formulas in API Bulletin 5C3.

Table 4—Used Drill Pipe Torsional and Tensile Data API Premium Class

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)
Size OD in.	New Weight Nominal W/ Threads and Couplings lb/ft	[1,2]Torsional Yield Strength Based on Uniform Wear, ft-lb				[2]Tensile Data Based on Uniform Wear Load at Minimum Yield Strength, lb			
		E75	X95	G105	S135	E75	X95	G105	S135
2 3/8	4.85	3725.	4719.	5215.	6705.	76893.	97398.	107650.	138407.
	6.65	4811.	6093.	6735.	8659.	107616.	136313.	150662.	193709.
2 7/8	6.85	6332.	8020.	8865.	11397.	106946.	135465.	149725.	192503.
	10.40	8858.	11220.	12401.	15945.	166535.	210945.	233149.	299764.
3 1/2	9.50	11094.	14052.	15531.	19968.	152979.	193774.	214171.	275363.
	13.30	14361.	18191.	20106.	25850.	212150.	268723.	297010.	381870.
	15.50	16146.	20452.	22605.	29063.	250620.	317452.	350868.	451115.
4	11.85	15310.	19392.	21433.	27557.	182016.	230554.	254823.	327630.
	14.00	18196.	23048.	25474.	32752.	224182.	283963.	313854.	403527.
	15.70	20067.	25418.	28094.	36120.	253851.	321544.	355391.	456931.
4 1/2	13.75	20403.	25844.	28564.	36725.	213258.	270127.	298561.	383864.
	16.60	24139.	30576.	33795.	43450.	260165.	329542.	364231.	468297.
	20.00	28683.	36332.	40157.	51630.	322916.	409026.	452082.	581248.
	22.82	31587.	40010.	44222.	56856.	367566.	465584.	514593.	661620.
5	16.25	27607.	34969.	38650.	49693.	259155.	328263.	362817.	466479.
	19.50	32285.	40895.	45199.	58113.	311535.	394612.	436150.	560764.
	25.60	40544.	51356.	56762.	72979.	414690.	525274.	580566.	746443.
5 1/2	19.20	34764.	44035.	48670.	62575.	294260.	372730.	411965.	529669.
	21.90	39863.	50494.	55809.	71754.	344780.	436721.	482692.	620604.
	24.70	44320.	56139.	62048.	79776.	391285.	495627.	547799.	704313.
6 5/8	25.20	55766.	71522.	79050.	101635.	387466.	490790.	542452.	697438.
	27.70	60192.	77312.	85450.	109864.	422419.	535064.	591387.	760354.

[1]Based on the shear strength equal to 57.7 percent of minimum yield strength.

[2]Torsional data based on 20 percent uniform wear on outside diameter and tensile data based on 20 percent uniform wear on outside diameter.

Table 5—Used Drill Pipe Collapse and Internal Pressure Data API Premium Class

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)
Size OD in.	Nominal Weight Threads and Couplings lb/ft	[1]Collapse Pressure Based on Minimum Values, psi				[1]Minimum Internal Yield Pressure at Minimum Yield Strength, psi			
		E75	X95	G105	S135	E75	X95	G105	S135
$2^3/_8$	4.85	8522.	10161.	10912.	12891.	9600.	12160.	13440.	17280.
	6.65	13378.	16945.	18729.	24080.	14147.	17920.	19806.	25465.
$2^7/_8$	6.85	7640.	9017.	9633.	11186.	9057.	11473.	12680.	16303.
	10.40	14223.	18016.	19912.	25602.	15110.	19139.	21153.	27197.
$3^1/_2$	9.50	7074.	8284.	8813.	10093.	8709.	11031.	12192.	15675.
	13.30	12015.	15218.	16820.	21626.	12617.	15982.	17664.	22711.
	15.50	14472.	18331.	20260.	26049.	15394.	19499.	21552.	27710.
4	11.85	5704.	6508.	6827.	7445.	7860.	9956.	11004.	14148.
	14.00	9012.	10795.	11622.	13836.	9900.	12540.	13860.	17820.
	15.70	10914.	13825.	15190.	18593.	11400.	14440.	15960.	20520.
$4^1/_2$	13.75	4686.	5190.	5352.	5908.	7227.	9154.	10117.	13008.
	16.60	7525.	8868.	9467.	10964.	8987.	11383.	12581.	16176.
	20.00	10975.	13901.	15350.	18806.	11467.	14524.	16053.	20640.
	22.82	12655.	16030.	17718.	22780.	13333.	16889.	18667.	24000.
5	16.25	4490.	4935.	5067.	5661.	7104.	8998.	9946.	12787.
	19.50	7041.	8241.	8765.	10029.	8688.	11005.	12163.	15638.
	25.60	11458.	14514.	16042.	20510.	12000.	15200.	16800.	21600.
$5^1/_2$	19.20	3736.	4130.	4336.	4714.	6633.	8401.	9286.	11939.
	21.90	5730.	6542.	6865.	7496.	7876.	9977.	11027.	14177.
	24.70	7635.	9011.	9626.	11177.	9055.	11469.	12676.	16298.
$6^5/_8$	25.20	2931.	3252.	3353.	3429.	5977.	7571.	8368.	10759.
	27.70	3615.	4029.	4222.	4562.	6557.	8306.	9180.	11803.

Note: Calculations for Premium Class drill pipe are based on formulas in API Bulletin 5C3.
[1]Data are based on minimum wall of 80 percent nominal wall. Collapse pressures are based on uniform OD wear. Internal pressures are based on uniform wear and nominal OD.

Table 6—Used Drill Pipe Torsional and Tensile Data API Class 2

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)
Size OD in.	New Weight Nominal W/ Threads and Couplings lb/ft	[1,2]Torsional Yield Strength Based on Uniform Wear, ft-lb				[2]Tensile Data Based on Uniform Wear Load at Minimum Yield Strength, lb			
		E75	X95	G105	S135	E75	X95	G105	S135
$2^3/_8$	4.85	3224.	4083.	4513.	5802.	66686.	84469.	93360.	120035.
	6.65	4130.	5232.	5782.	7434.	92871.	117636.	130019.	167167.
$2^7/_8$	6.85	5484.	6946.	7677.	9871.	92801.	117549.	129922.	167043.
	10.40	7591.	9615.	10627.	13663.	143557.	181839.	200980.	258403.
$3^1/_2$	9.50	9612.	12176.	13457.	17302.	132793.	168204.	185910.	239027.
	13.30	12365.	15663.	17312.	22258.	183398.	232304.	256757.	330116.
	15.50	13828.	17515.	19359.	24890.	215967.	273558.	302354.	388741.
4	11.85	13281.	16823.	18594.	23907.	158132.	200301.	221385.	284638.
	14.00	15738.	19935.	22034.	28329.	194363.	246193.	272108.	349852.
	15.70	17315.	21932.	24241.	31166.	219738.	278335.	307633.	395528.
$4^1/_2$	13.75	17715.	22439.	24801.	31887.	185389.	234827.	259545.	333701.
	16.60	20908.	26483.	29271.	37634.	225771.	285977.	316080.	406388.
	20.00	24747.	31346.	34645.	44544.	279502.	354035.	391302.	503103.
	22.82	27161.	34404.	38026.	48890.	317497.	402163.	444496.	571495.
5	16.25	23974.	30368.	33564.	43154.	225316.	285400.	315442.	405568.
	19.50	27976.	35436.	39166.	50356.	270432.	342548.	378605.	486778.
	25.60	34947.	44267.	48926.	62905.	358731.	454392.	502223.	645715.
$5^1/_2$	19.20	30208.	38263.	42291.	54374.	255954.	324208.	358335.	460717.
	21.90	34582.	43804.	48414.	62247.	299533.	379409.	419346.	539160.
	24.70	38383.	48619.	53737.	69090.	339533.	430076.	475347.	611160.
$6^5/_8$	25.20	48497.	61430.	67896.	87295.	337236.	427166.	472131.	607026.
	27.70	52308.	66257.	73231.	94155.	367455.	465443.	514437.	661419.

[1]Based on the shear strength equal to 57.7 percent of minimum yield strength.

[2]Torsional data based on 30 percent uniform wear on outside diameter and tensile data based on 30 percent uniform wear on outside diameter.

Table 7—Used Drill Pipe Collapse and Internal Pressure Data API Class 2

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)
Size OD in.	Nominal Weight Threads and Couplings lb/ft	[1]Collapse Pressure Based on Minimum Values, psi				[1]Minimum Internal Yield Pressure at Minimum Yield Strength, psi			
		E75	X95	G105	S135	E75	X95	G105	S135
$2^3/_8$	4.85	6852.	7996.	8491.	9664.	8400.	10640.	11760.	15120.
	6.65	12138.	15375.	16993.	21849.	12379.	15680.	17331.	22282.
$2^7/_8$	6.85	6055.	6963.	7335.	8123.	7925.	10039.	11095.	14265.
	10.40	12938.	16388.	18113.	23288.	13221.	16746.	18509.	23798.
$3^1/_2$	9.50	5544.	6301.	6596.	7137.	7620.	9652.	10668.	13716.
	13.30	10858.	13753.	15042.	18396.	11040.	13984.	15456.	19872.
	15.50	13174.	16686.	18443.	23712.	13470.	17062.	18858.	24246.
4	11.85	4311.	4702.	4876.	5436.	6878.	8712.	9629.	12380.
	14.00	7295.	8570.	9134.	10520.	8663.	10973.	12128.	15593.
	15.70	9531.	11468.	12374.	14840.	9975.	12635.	13965.	17955.
$4^1/_2$	13.75	3397.	3845.	4016.	4287.	6323.	8010.	8853.	11382.
	16.60	5951.	6828.	7185.	7923.	7863.	9960.	11009.	14154.
	20.00	9631.	11598.	12520.	15033.	10033.	12709.	14047.	18060.
	22.82	11458.	14514.	16042.	20510.	11667.	14779.	16333.	21000.
5	16.25	3275.	3696.	3850.	4065.	6216.	7874.	8702.	11189.
	19.50	5514.	6262.	6552.	7079.	7602.	9629.	10643.	13684.
	25.60	10338.	12640.	13685.	16587.	10500.	13300.	14700.	18900.
$5^1/_2$	19.20	2835.	3128.	3215.	3265.	5804.	7351.	8125.	10447.
	21.90	4334.	4733.	4899.	5465.	6892.	8730.	9649.	12405.
	24.70	6050.	6957.	7329.	8115.	7923.	10035.	11092.	14261.
$6^5/_8$	25.20	2227.	2343.	2346.	2346.	5230.	6625.	7322.	9414.
	27.70	2765.	3037.	3113.	3148.	5737.	7267.	8032.	10327.

Note: Calculations for Class 2 drill pipe are based on formulas in API Bulletin 5C3.
[1]Data are based on minimum wall of 70 percent nominal wall. Collapse pressures are based on uniform OD wear. Internal pressures are based on uniform wear and nominal OD.

Table 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)
								Mechanical Properties			
Drill Pipe Data				Tool Joint Data				Tensile Yield, lb		Torsional Yield, ft-lb	
Nominal Size in.	Nominal Weight lb/ft	Approx. Weight[1] lb/ft	Type Upset	Conn.	OD in.	ID in.	Drift Diameter[2] in.	Pipe[3]	Tool Joint[4]	Pipe[5]	Tool Joint[6]
$2\frac{3}{8}$	4.85	5.26	EU	NC26(IF)	$3\frac{3}{8}$	$1\frac{3}{4}$	1.625	97817.	313681.	4763.	6875.b
		4.95	EU	OH	$3\frac{1}{8}$	2	1.807	97817.	206416.	4763.	4521.p
		5.05	EU	SLH90	$3\frac{1}{4}$	2	1.850	97817.	202670.	4763.	5129.p
		5.15	EU	WO	$3\frac{3}{8}$	2	1.807	97817.	205369.	4763.	4311.p
	6.65	6.99	EU	NC26(IF)	$3\frac{3}{8}$	$1\frac{3}{4}$	1.625	138214.	313681.	6250.	6875.b
		6.89	EU	OH	$3\frac{1}{4}$	$1\frac{3}{4}$	1.625	138214.	294620.	6250.	6484.b
		6.71	IU	PAC	$2\frac{7}{8}$	$1\frac{3}{8}$	1.250	138214.	238504.	6250.	4688.P
		6.78	EU	SLH90	$3\frac{1}{4}$	2	1.670	138214.	202850.	6250.	5129.p
$2\frac{7}{8}$	6.85	7.50	EU	NC31(IF)	$4\frac{1}{8}$	$2\frac{1}{8}$	2.000	135902.	447130.	8083.	12053.p
		6.93	EU	OH	$3\frac{3}{4}$	$2\frac{7}{16}$	2.253	135902.	223937.	8083.	5585.P
		7.05	EU	SLH90	$3\frac{7}{8}$	$2\frac{7}{16}$	2.296	135902.	260783.	8083.	7628.p
		7.31	EU	WO	$4\frac{1}{8}$	$2\frac{7}{16}$	2.253	135902.	289264.	8083.	7197.p
	10.40	10.87	EU	NC31(IF)	$4\frac{1}{8}$	$2\frac{1}{8}$	1.963	214344.	447130.	11554.	12053.p
		10.59	EU	OH	$3\frac{7}{8}$	$2\frac{5}{32}$	1.963	214344.	345566.	11554.	8814.P
		10.27	IU	PAC	$3\frac{1}{8}$	$1\frac{1}{2}$	1.375	214344.	272938.	11554.	5730.P
		10.59	EU	SLH90	$3\frac{7}{8}$	$2\frac{5}{32}$	2.006	214344.	382765.	11554.	11288.p
		11.19	IU	XH	$4\frac{1}{4}$	$1\frac{7}{8}$	1.750	214344.	505054.	11554.	13282.p
		10.35	IU	NC26(SH)	$3\frac{3}{8}$	$1\frac{3}{4}$	1.625	214344.	313681.	11554.	6875.B
$3\frac{1}{2}$	9.50	10.58	EU	NC38(IF)	$4\frac{3}{4}$	$2\frac{11}{16}$	2.563	194264.	587308.	14146.	18107.p
		9.84	EU	OH	$4\frac{1}{2}$	3	2.804	194264.	392071.	14146.	11870.p
		9.99	EU	SLH90	$4\frac{5}{8}$	3	2.847	194264.	366705.	14146.	12650.p
		10.14	EU	WO	$4\frac{3}{4}$	3	2.804	194264.	419797.	14146.	12878.p
$3\frac{1}{2}$	13.30	14.37	EU	H90	$5\frac{1}{4}$	$2\frac{3}{4}$	2.619	271569.	664050.	18551.	23847.p
		13.93	EU	NC38(IF)	$4\frac{3}{4}$	$2\frac{11}{16}$	2.457	271569.	587308.	18551.	18107.p
		13.75	EU	OH	$4\frac{3}{4}$	$2\frac{11}{16}$	2.414	271569.	559582.	18551.	17305.p
		13.40	IU	NC31(SH)	$4\frac{1}{8}$	$2\frac{1}{8}$	2.000	271569.	447130.	18551.	11869.P
		13.91	EU	XH	$4\frac{3}{4}$	$2\frac{7}{16}$	2.313	271569.	570939.	18551.	17493.p
	15.50	16.54	EU	NC38(IF)	5	$2\frac{9}{16}$	2.414	322775.	649158.	21086.	20326.p
4	11.85	13.00	IU	H90	$5\frac{1}{2}$	$2\frac{13}{16}$	2.688	230755.	913708.	19474.	35374.p
		13.52	EU	NC46(IF)	6	$3\frac{1}{4}$	3.125	230755.	901164.	19474.	33625.p
		12.10	EU	OH	$5\frac{1}{4}$	$3\frac{15}{32}$	3.287	230755.	621357.	19474.	21976.p
		12.91	EU	WO	$5\frac{3}{4}$	$3\frac{7}{16}$	3.313	230755.	782987.	19474.	28809.p
	14.00	15.04	IU	NC40(FH)	$5\frac{1}{4}$	$2\frac{13}{16}$	2.688	285359.	711611.	23288.	23487.p
		15.43	IU	H90	$5\frac{1}{2}$	$2\frac{13}{16}$	2.688	285359.	913708.	23288.	35374.p
		15.85	EU	NC46(IF)	6	$3\frac{1}{4}$	3.125	285359.	901164.	23288.	33625.p

(Table continued on next page.)

Table 8—Mechanical Properties of New Tool Joints and New Grade E75 Drill Pipe (Continued)

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)
								Mechanical Properties			
Drill Pipe Data				Tool Joint Data				Tensile Yield, lb		Torsional Yield, ft-lb	
Nominal Size in.	Nominal Weight lb/ft	Approx. Weight[1] lb/ft	Type Upset	Conn.	OD in.	ID in.	Drift Diameter[2] in.	Pipe[3]	Tool Joint[4]	Pipe[5]	Tool Joint[6]
4	14.00	15.02	EU	OH	$5^1/_2$	$3^1/_4$	3.125	285359.	759875.	23288.	27289.p
		14.35	IU	SH	$4^5/_8$	$2^9/_{16}$	2.438	285359.	512035.	23288.	15170.P
	15.70	16.80	IU	NC40(FH)	$5^1/_4$	$2^{11}/_{16}$	2.563	324118.	776406.	25810.	25673.p
		17.09	IU	H90	$5^1/_2$	$2^{13}/_{16}$	2.688	324118.	913708.	25810.	35374.p
		17.54	EU	NC46(IF)	6	$3^1/_4$	3.095	324118.	901164.	25810.	33625.p
$4^1/_2$	13.75	15.23	IU	H90	6	$3^1/_4$	3.125	270034.	938403.	25907.	38925.p
		15.36	EU	NC50(IF)	$6^5/_8$	$3^3/_4$	3.625	270034.	939096.	25907.	37676.p
		14.04	EU	OH	$5^3/_4$	$3^{31}/_{32}$	3.770	270034.	554844.	25907.	20939.p
		14.77	EU	WO	$6^1/_8$	$3^7/_8$	3.750	270034.	849266.	25907.	33651.p
	16.60	18.14	IEU	FH	6	3	2.875	330558.	976156.	30807.	34780.p
		17.92	IEU	H90	6	$3^1/_4$	3.125	330558.	938403.	30807.	38925.p
		17.95	EU	NC50(IF)	$6^5/_8$	$3^3/_4$	3.625	330558.	939096.	30807.	37676.p
		17.07	EU	OH	$5^7/_8$	$3^3/_4$	3.625	330558.	713979.	30807.	27243.p
		16.79	IEU	NC38(SH)	5	$2^{11}/_{16}$	2.563	330558.	587308.	30807.	18346.P
		18.37	IEU	NC46(XH)	$6^1/_4$	$3^1/_4$	3.125	330558.	901164.	30807.	33993.p
	20.00	21.64	IEU	FH	6	3	2.875	412358.	976156.	36901.	34780.p
		21.64	IEU	H90	6	3	2.875	412358.	1085665.	36901.	45152.p
		21.59	EU	NC50(IF)	$6^5/_8$	$3^5/_8$	3.452	412358.	1025980.	36901.	41235.p
		22.09	IEU	NC46(XH)	$6^1/_4$	3	2.875	412358.	1048426.	36901.	39659.p
	22.82	24.11	EU	NC50(IF)	$6^5/_8$	$3^5/_8$	3.452	471239.	1025980.	40912.	41235.p
		24.56	IEU	NC46(XH)	$6^1/_4$	3	2.875	471239.	1048426.	40912.	39659.p
5	19.50	22.28	IEU	$5^1/_2$ FH	7	$3^3/_4$	3.625	395595.	1448407.	41167.	60338.b
		20.85	IEU	NC50(XH)	$6^5/_8$	$3^3/_4$	3.625	395595.	939095.	41167.	37676.p
	25.60	28.27	IEU	$5^1/_2$ FH	7	$3^1/_2$	3.375	530144.	1619231.	52257.	60338.b
		26.85	IEU	NC50(XH)	$6^5/_8$	$3^1/_2$	3.375	530144.	1109920.	52257.	44673.p
$5^1/_2$	21.90	23.78	IEU	FH	7	4	3.875	437116.	1265802.	50710.	56045.p
	24.70	26.30	IEU	FH	7	4	3.875	497222.	1265802.	56574.	56045.p
$6^5/_8$	25.20	27.28	IEU	FH	8	5	4.875	489464.	1447697.	70580.	73620.p
$6^5/_8$	27.70	29.06	IEU	FH	8	5	4.875	534198.	1447697.	76295.	73620.p

[1]Tool Joint plus drill pipe, for Range 2 steel pipe (See Appendix A for method of calculation).
[2]See Section 4.4.
[3]The tensile yield strength of Grade E drill pipe is based on 75,000 psi minimum yield strength.
[4]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 $^5/_8$ inch from the shoulder.
[5]The torsional yield strength is based on a shear strength of 57.7 percent of the minimum yield strength.
[6] p = pin limited yield; b = box limited yield; P or B indicates that tool joint could not meet 80 percent of tube torsion yield.

Table 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)
								Mechanical Properties			
Drill Pipe Data				Tool Joint Data				Tensile Yield, lb		Torsional Yield, ft-lb	
Nominal Size in.	Nominal Weight lb/ft	Approx. Weight[1] lb/ft	Type Upset	Conn.	OD in.	ID in.	Drift Diameter[2] in.	Pipe[3]	Tool Joint[4]	Pipe[3]	Tool Joint[5]
$2^3/_8$	6.65	7.11	EU-X95	NC26(IF)	$3^3/_8$	$1^3/_4$	1.625	175072.	313681.	7917.	6875.b
		6.99	EU-X95	SLH90	$3^1/_4$	$1^{13}/_{16}$	1.670	175072.	270223.	7917.	6884.p
	6.65	7.11	EU-G105	NC26(IF)	$3^3/_8$	$1^3/_4$	1.625	193500.	313681.	8751.	6875.b
		6.99	EU-G105	SLH90	$3^1/_4$	$1^{13}/_{16}$	1.670	193500.	270223.	8751.	6884.P
$2^7/_8$	10.40	11.09	EU-X95	NC31(IF)	$4^1/_8$	2	1.875	271503.	495726.	14635.	13389.p
		10.95	EU-X95	SLH90	4	2	1.875	271503.	443971.	14635.	13218.p
	10.40	11.09	EU-G105	NC31(IF)	$4^1/_8$	2	1.875	300082.	495726.	16176.	13389.p
		10.95	EU-G105	SLH90	4	2	1.875	300082.	443971.	16176.	13218.p
	10.40	11.55	EU-S135	NC31(IF)	$4^3/_8$	$1^5/_8$	1.500	385820.	623844.	20798.	17170.p
		11.26	EU-S135	SLH90	$4^1/_8$	$1^5/_8$	1.500	385820.	572089.	20798.	17213.p
$3^1/_2$	13.30	14.60	EU-X95	H90	$5^1/_4$	$2^3/_4$	2.619	343988.	664050.	23498.	23833.p
		14.62	EU-X95	NC38(IF)	5	$2^9/_{16}$	2.438	343988.	649158.	23498.	20326.p
		14.06	EU-X95	SLH90	$4^3/_4$	$2^9/_{16}$	2.438	343988.	596066.	23498.	20879.p
	13.30	14.71	EU-G105	NC38(IF)	5	$2^7/_{16}$	2.313	380197.	708063.	25972.	22213.p
		14.06	EU-G105	SLH90	$4^3/_4$	$2^9/_{16}$	2.438	380197.	596066.	25972.	20879.p
	13.30	14.92	EU-S135	NC38(IF)	5	$2^1/_8$	2.000	488825.	842440.	33392.	26515.P
		14.65	EU-S135	SLH90	5	$2^1/_8$	2.000	488825.	789348.	33392.	28078.p
		15.13	EU-S135	NC40(4FH)	$5^3/_8$	$2^7/_{16}$	2.313	488825.	897161.	33392.	29930.p
	15.50	16.82	EU-X95	NC38(IF)	5	$2^7/_{16}$	2.313	408848.	708063.	26708.	22213.p
	15.50	17.03	EU-G105	NC38(IF)	5	$2^1/_8$	2.000	451885.	842440.	29520.	26515.p
		16.97	EU-G105	NC40(4FH)	$5^1/_4$	$2^9/_{16}$	2.438	451885.	838257.	29520.	27760.p
	15.50	17.57	EU-S135	NC40(4FH)	$5^1/_2$	$2^1/_4$	2.125	580995.	979996.	37954.	32943.p
4	14.00	15.34	IU-X95	NC40(FH)	$5^1/_4$	$2^{11}/_{16}$	2.563	361454.	776406.	29498.	25673.p
		15.63	IU-X95	H90	$5^1/_2$	$2^{13}/_{16}$	2.688	361454.	913708.	29498.	35374.p
		16.19	EU-X95	NC46(IF)	6	$3^1/_4$	3.125	361454.	901164.	29498.	33625.p
	14.00	15.91	IU-G105	NC40(FH)	$5^1/_2$	$2^7/_{16}$	2.313	399502.	897161.	32603.	30114.p
		15.63	IU-G105	H90	$5^1/_2$	$2^{13}/_{16}$	2.688	399502.	913708.	32603.	35374.p
		16.19	EU-G105	NC46(IF)	6	$3^1/_4$	3.125	399502.	901164.	32603.	33625.p
	14.00	16.19	IU-S135	NC40(FH)	$5^1/_2$	2	1.875	513646.	1080135.	41918.	36363.p
		15.63	IU-S135	H90	$5^1/_2$	$2^{13}/_{16}$	2.688	513646.	913708.	41918.	35374.p
		16.42	EU-S135	NC46(IF)	6	3	2.875	513646.	1048426.	41918.	39229.p
	15.70	17.52	IU-X95	NC40(FH)	$5^1/_2$	$2^7/_{16}$	2.313	410550.	897161.	32692.	30114.p
		17.23	IU-X95	H90	$5^1/_2$	$2^{13}/_{16}$	2.688	410550.	913708.	32692.	35374.p
		17.80	EU-X95	NC46(IF)	6	$3^1/_4$	3.125	410550.	901164.	32692.	33625.p
	15.70	17.52	IU-G105	NC40(FH)	$5^1/_2$	$2^7/_{16}$	2.313	453765.	897161.	36134.	30114.p
		17.23	IU-G105	H90	$5^1/_2$	$2^{13}/_{16}$	2.688	453765.	913708.	36134.	35374.p
		17.80	EU-G105	NC46(IF)	6	$3^1/_4$	3.125	453765.	901164.	36134.	33625.p
	15.70	18.02	EU-S135	NC46(IF)	6	3	2.875	583413.	1048426.	46458.	39229.p
$4^1/_2$	16.60	18.33	IEU-X95	FH	6	3	2.875	418707.	976156.	39022.	34780.p
		18.11	IEU-X95	H90	6	$3^1/_4$	3.125	418707.	938403.	39022.	38925.p
		18.36	EU-X95	NC50(IF)	$6^5/_8$	$3^3/_4$	3.625	418707.	939095.	39022.	37676.p
		18.79	IEU-X95	NC46(XH)	$6^1/_4$	3	2.875	418707.	1048426.	39022.	39659.p
$4^1/_2$	16.60	18.33	IEU-G105	FH	6	3	2.625	462781.	976156.	43130.	34780.p
		18.33	IEU-G105	H90	6	3	3.125	462781.	1085665.	43130.	45152.p
		18.36	EU-G105	NC50(IF)	$6^5/_8$	$3^3/_4$	3.625	462781.	939095.	43130.	37676.p
		18.79	IEU-G105	NC46(XH)	$6^1/_4$	3	2.875	462781.	1048426.	43130.	39659.p
	16.60	19.19	IEU-S135	FH	$6^1/_4$	$2^1/_2$	2.375	595004.	1235337.	55453.	44769.p

(Table continued on next page.)

Table 9—Mechanical Properties of New Tool Joints and New High Strength Drill Pipe (Continued)

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)
								Mechanical Properties			
Drill Pipe Data				Tool Joint Data				Tensile Yield, lb		Torsional Yield, ft-lb	
Nominal Size in.	Nominal Weight lb/ft	Approx. Weight[1] lb/ft	Type Upset	Conn.	OD in.	ID in.	Drift Diameter[2] in.	Pipe[3]	Tool Joint[4]	Pipe[3]	Tool Joint[5]
$4\frac{1}{2}$	16.60	18.33	IEU-S135	H90	6	3	2.875	595004.	1085665.	55453.	45152.p
		18.62	EU-S135	NC50(IF)	$6\frac{5}{8}$	$3\frac{1}{2}$	3.375	595004.	1109920.	55453.	44673.p
		19.00	IEU-S135	NC46(XH)	$6\frac{1}{4}$	$2\frac{3}{4}$	2.625	595004.	1183908.	55453.	44871.p
	20.00	22.39	IEU-X95	FH	6	$2\frac{1}{2}$	2.375	522320.	1235337.	46741.	44265.p
		21.78	IEU-X95	H90	6	$3\frac{1}{4}$	3.125	522320.	938403.	46741.	38925.p
		22.08	EU-X95	NC50(IF)	$6\frac{5}{8}$	$3\frac{1}{2}$	3.375	522320.	1109920.	46741.	44673.p
		22.67	IEU-X95	NC46(XH)	$6\frac{1}{4}$	$2\frac{3}{4}$	2.625	522320.	1183908.	46741.	44871.p
	20.00	22.39	IEU-G105	FH	6	$2\frac{1}{2}$	2.375	577301.	1235337.	51661.	44265.p
		22.00	IEU-G105	H90	6	3	2.875	577301.	1085665.	51661.	45152.p
		22.08	EU-G105	NC50(IF)	$6\frac{5}{8}$	$3\frac{1}{2}$	3.375	577301.	1109920.	51661.	44673.p
		22.86	IEU-G105	NC46(XH)	$6\frac{1}{4}$	$2\frac{1}{2}$	2.375	577301.	1307608.	51661.	49630.p
	20.00	23.03	EU-S135	NC50(IF)	$6\frac{5}{8}$	3	2.875	742244.	1416225.	66421.	57800.p
		23.03	IEU-S135	NC46(XH)	$6\frac{1}{4}$	$2\frac{1}{4}$	2.125	742244.	1419527.	66421.	53936.p
	22.82	25.13	IEU-X95	FH	$6\frac{1}{4}$	$2\frac{1}{4}$	2.125	596903.	1347256.	51821.	48912.p
		24.24	EU-X95	NC50(IF)	$6\frac{5}{8}$	$3\frac{1}{2}$	3.375	596903.	1109920.	51821.	44673.p
		24.77	IEU-X95	NC46(XH)	$6\frac{1}{4}$	$2\frac{3}{4}$	2.625	596903.	1183908.	51821.	44871.p
	22.82	24.72	EU-G105	NC50(IF)	$6\frac{5}{8}$	$3\frac{1}{4}$	3.125	659735.	1268963.	57276.	51447.p
		24.96	IEU-G105	NC46(XH)	$6\frac{1}{4}$	$2\frac{1}{2}$	2.375	659735.	1307608.	57276.	49630.p
	22.82	25.41	EU-S135	NC50(IF)	$6\frac{5}{8}$	$2\frac{3}{4}$	2.625	848230.	1551706.	73641.	63406.p
5	19.50	22.62	IEU-X95	$5\frac{1}{2}$ FH	7	$3\frac{3}{4}$	3.625	501087.	1448407.	52144.	60338.b
		21.93	IEU-X95	H90	$6\frac{1}{2}$	$3\frac{1}{4}$	3.125	501087.	1176265.	52144.	51807.p
		21.45	IEU-X95	NC50(XH)	$6\frac{5}{8}$	$3\frac{1}{2}$	3.375	501087.	1109920.	52144.	44673.p
	19.50	22.62	IEU-G105	$5\frac{1}{2}$ FH	7	$3\frac{3}{4}$	3.625	553833.	1448407.	57633.	60338.b
		22.15	IEU-G105	H90	$6\frac{1}{2}$	3	2.875	553833.	1323527.	57633.	58398.p
		21.93	IEU-G105	NC50(XH)	$6\frac{5}{8}$	$3\frac{1}{4}$	3.125	553833.	1268963.	57633.	51447.p
	19.50	23.48	IEU-S135	$5\frac{1}{2}$ FH	$7\frac{1}{4}$	$3\frac{1}{2}$	3.375	712070.	1619231.	74100.	72627.p
		22.61	IEU-S135	NC50(XH)	$6\frac{5}{8}$	$2\frac{3}{4}$	2.625	712070.	1551706.	74100.	63406.p
	25.60	28.59	IEU-X95	$5\frac{1}{2}$ FH	7	$3\frac{1}{2}$	3.375	671515.	1619231.	66192.	60338.b
		27.87	IEU-X95	NC50(XH)	$6\frac{5}{8}$	3	2.875	671515.	1416225.	66192.	56984.b
	25.60	29.16	IEU-G105	$5\frac{1}{2}$ FH	$7\frac{1}{4}$	$3\frac{1}{2}$	3.375	742201.	1619231.	73159.	72627.p
		28.32	IEU-G105	NC50(XH)	$6\frac{5}{8}$	$2\frac{3}{4}$	2.625	742201.	1551706.	73159.	63406.b
	25.60	29.43	IEU-S135	$5\frac{1}{2}$ FH	$7\frac{1}{4}$	$3\frac{1}{4}$	3.125	954259.	1778274.	94062.	76156.b
$5\frac{1}{2}$	21.90	24.53	IEU-X95	FH	7	$3\frac{3}{4}$	3.625	553681.	1448407.	64233.	60338.b
		24.80	IEU-X95	H90	7	$3\frac{1}{2}$	3.125	553681.	1268877.	64233.	59091.p
	21.90	25.38	IEU-G105	FH	$7\frac{1}{4}$	$3\frac{1}{2}$	3.375	611963.	1619231.	70994.	72627.p
	21.90	26.50	IEU-S135	FH	$7\frac{1}{2}$	3	2.875	786809.	1925536.	91278.	87341.p
	24.70	27.85	IEU-X95	FH	$7\frac{1}{4}$	$3\frac{1}{2}$	3.375	629814.	1619231.	71660.	72627.p
$5\frac{1}{2}$	24.70	27.85	IEU-G105	FH	$7\frac{1}{4}$	$3\frac{1}{2}$	3.375	696111.	1619231.	79204.	72627.p
	24.70	27.77	IEU-S135	FH	$7\frac{1}{2}$	3	2.875	894999.	1925536.	101833.	87341.p
$6\frac{5}{8}$	25.20	27.15	IEU-X95	FH	8	5	4.875	619988.	1448416.	89402.	73661.p
	25.20	28.20	IEU-G105	FH	$8\frac{1}{4}$	$4\frac{3}{4}$	4.625	685250.	1678145.	98812.	86237.p
	25.20	29.63	IEU-S135	FH	$8\frac{1}{2}$	$4\frac{1}{4}$	4.125	881035.	2102260.	127044.	109226.p
	27.70	30.11	IEU-X95	FH	$8\frac{1}{4}$	$4\frac{3}{4}$	4.625	676651.	1678145.	96640.	86237.p
	27.70	30.11	IEU-G105	FH	$8\frac{1}{4}$	$4\frac{3}{4}$	4.625	747250.	1678145.	106813.	86237.p
	27.70	31.54	IEU-S135	FH	$8\frac{1}{2}$	$4\frac{1}{4}$	4.125	961556.	2102260.	137330.	109226.P

[1]Tool Joint plus drill pipe, for Range 2 steel pipe (See Appendix A for method of calculation).
[2]See Section 4.4.
[3]The torsional yield strength is based on a shear strength of 57.7 percent of the minimum yield strength.
[4]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 $\frac{5}{8}$ inch from the shoulder.
[5] p = pin limited yield; b = box limited yield; P or B indicates that tool joint could not meet 80 percent of tube torsion yield.

Table 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)
							Premium Class			Class 2		
Drill Pipe			New Tool Joint Data				Min. OD Tool Joint	Min. Box Shoulder with Eccentric Wear	Make-up Torque for Min. OD Tool Joint	Min. OD Tool Joint	Min. Box Shoulder with Eccentric Wear	Make-up Torque for Min. OD Tool Joint
Nominal Size in.	Nom Weight lb/ft	Type Upset and Grade	Conn.	New OD in.	New ID in.	Make-up Torque[6] ft-lb	in.	in.	ft-lb	in.	in.	ft-lb
$2^3/_8$	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
	4.85	EU-E75	W.O.	$3^3/_8$	2	2,586 P	$3^1/_{16}$	$^1/_{16}$	1,994	$3^1/_{32}$	$^3/_{64}$	1,746
	4.85	EU-E75	$2^3/_8$ OHLW	$3^1/_8$	2	2,713 P	3	$^1/_{16}$	1,830	$2^{31}/_{32}$	$^3/_{64}$	1,589
	4.85	EU-E75	$2^3/_8$ SL-H90	$3^1/_4$	2	3,074 P	$2^{31}/_{32}$	$^1/_{16}$	1,996	$2^{15}/_{16}$	$^3/_{64}$	1,726
$2^3/_8$	6.65	IU-E75	$2^3/_8$ PAC[2]	$2^7/_8$	$1^3/_8$	2,813 P	$2^{25}/_{32}$	$^9/_{64}$	2,455	$2^{23}/_{32}$	$^7/_{64}$	2,055
	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
	6.65	EU-E75	$2^3/_8$ SL-H90	$3^1/_4$	2	3,074 P	$3^1/_{32}$	$^3/_{32}$	2,549	$2^{31}/_{32}$	$^1/_{16}$	1,996
	6.65	EU-E75	$2^3/_8$ OHSW	$3^1/_4$	$1^3/_4$	3,891 B	$3^1/_{16}$	$^3/_{32}$	2,324	$3^1/_{32}$	$^5/_{64}$	2,075
$2^3/_8$	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[2]	$3^3/_8$	$1^3/_4$	4,125 B	$3^9/_{32}$	$^1/_8$	3,279	$3^1/_4$	$^7/_{64}$	3,005
$2^7/_8$	6.85	EU-E75	NC31	$4^1/_8$	$2^1/_8$	7,122 P	$3^{11}/_{16}$	$^5/_{64}$	3,154	$3^{21}/_{32}$	$^1/_{16}$	2,804
	6.85	EU-E75	$2^7/_8$ WO	$4^1/_8$	$2^7/_{16}$	4,318 P	$3^5/_8$	$^5/_{64}$	3,216	$3^{19}/_{32}$	$^1/_{16}$	2,876
	6.85	EU-E75	$2^7/_8$ OHLW[2]	$3^3/_4$	$2^7/_{16}$	3,351 P	$3^1/_2$	$^7/_{64}$	3,297	$3^7/_{16}$	$^5/_{64}$	2,666
	6.85	EU-E75	$2^7/_8$ SL-H90	$3^7/_8$	$2^7/_{16}$	4,575 P	$3^1/_2$	$^3/_{32}$	3,397	$3^7/_{16}$	$^1/_{16}$	2,666
$2^7/_8$	10.40	EU-E75	NC31	$4^1/_8$	$2^1/_8$	7,122 P	$3^{13}/_{16}$	$^9/_{64}$	4,597	$3^3/_4$	$^7/_{64}$	3,867
	10.40	IU-E75	$2^7/_8$ XH	$4^1/_4$	$1^7/_8$	7,969 P	$3^{23}/_{32}$	$^9/_{64}$	4,357	$3^{21}/_{32}$	$^7/_{64}$	3,664
	10.40	IU-E75	NC26[2]	$3^3/_8$	$1^3/_4$	4,125 B	$3^3/_8$	$^{11}/_{64}$	4,125	$3^{11}/_{32}$	$^5/_{32}$	3,839
	10.40	EU-E75	$2^7/_8$ OHSW[2]	$3^7/_8$	$2^5/_{32}$	5,270 P	$3^{19}/_{32}$	$^5/_{32}$	4,273	$3^9/_{16}$	$^7/_{64}$	3,941
	10.40	EU-E75	$2^7/_8$ SL-H90	$3^7/_8$	$2^5/_{32}$	6,773 P	$3^{19}/_{32}$	$^9/_{64}$	4,529	$3^{17}/_{32}$	$^7/_{64}$	3,770
	10.40	IU-E75	$2^7/_8$ PAC[2]	$3^1/_8$	$1^1/_2$	3,439 P	$3^1/_8$	$^{15}/_{64}$	3,439	$3^1/_8$	$^{15}/_{64}$	3,439
$2^7/_8$	10.40	EU-X95	NC31	$4^1/_8$	2	7,918 P	$3^{29}/_{32}$	$^3/_{16}$	5,726	$3^{27}/_{32}$	$^3/_{32}$	4,969
	10.40	EU-X95	$2^7/_8$ SL-H90[2]	$3^7/_8$	$2^5/_{32}$	6,773 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	7,918 P	$3^{15}/_{16}$	$^{13}/_{64}$	6,110	$3^7/_8$	$^{11}/_{64}$	5,345
$2^7/_8$	10.40	EU-S135	NC31	$4^3/_8$	$1^5/_8$	10,167 P	$4^1/_{16}$	$^{17}/_{64}$	7,694	4	$^{15}/_{64}$	6,893
$3^1/_2$	9.50	EU-E75	NC38	$4^3/_4$	3	7,688 P	$4^{13}/_{32}$	$^1/_8$	5,773	$4^{11}/_{32}$	$^3/_{32}$	4,797
	9.50	EU-E75	NC38	$4^3/_4$	$2^{11}/_{16}$	10,864 P	$4^{13}/_{32}$	$^1/_8$	5,773	$4^{11}/_{32}$	$^3/_{32}$	4,797
	9.50	EU-E75	$3^1/_2$ OHLW	$4^3/_4$	3	7,218 P	$4^9/_{32}$	$^1/_8$	5,340	$4^1/_4$	$^7/_{64}$	4,868
	9.50	EU-E75	$3^1/_2$ SL-H90	$4^5/_8$	3	7,584 P	$4^3/_{16}$	$^7/_{64}$	5,521	$4^5/_{32}$	$^3/_{32}$	5,003
$3^1/_2$	13.30	EU-E75	NC38	$4^3/_4$	$2^{11}/_{16}$	10,864 P	$4^1/_2$	$^{11}/_{64}$	7,274	$4^7/_{16}$	$^9/_{64}$	6,268
	13.30	IU-E75	NC31[2]	$4^1/_8$	$2^1/_8$	7,122 P	4	$^{15}/_{64}$	6,893	$3^{15}/_{16}$	$^{13}/_{64}$	6,110
	13.30	EU-E75	$3^1/_2$ OHSW	$4^3/_4$	$2^{11}/_{16}$	10,387 P	$4^{13}/_{32}$	$^3/_{16}$	7,278	$4^{11}/_{32}$	$^5/_{32}$	6,299
	13.30	EU-E75	$3^1/_2$ H90	$5^1/_4$	$2^3/_4$	14,300 P	$4^{17}/_{32}$	$^1/_8$	7,064	$4^1/_2$	$^7/_{64}$	6,487
	13.30	EU-X95	NC38	5	$2^9/_{16}$	12,196 P	$4^{19}/_{32}$	$^7/_{32}$	8,822	$4^{17}/_{32}$	$^3/_{16}$	7,785
	13.30	EU-X95	$3^1/_2$ SL-H90[2]	$4^5/_8$	$2^{11}/_{16}$	11,137 P	$4^3/_8$	$^{13}/_{64}$	8,742	$4^5/_{16}$	$^{11}/_{64}$	7,647
	13.30	EU-X95	$3^1/_2$ H90	$5^1/_4$	$2^3/_4$	14,300 P	$4^5/_8$	$^{11}/_{64}$	8,826	$4^9/_{16}$	$^9/_{64}$	7,646
$3^1/_2$	13.30	EU-G105	NC38	5	$2^7/_{16}$	13,328 P	$4^{21}/_{32}$	$^1/_4$	9,879	$4^{19}/_{32}$	$^7/_{32}$	8,822
$3^1/_2$	13.30	EU-S135	NC40	$5^3/_8$	$2^7/_{16}$	17,958 P	5	$^9/_{32}$	12,569	$4^{29}/_{32}$	$^{15}/_{64}$	10,768
	13.30	EU-S135	NC38	5	$2^1/_8$	15,909 P	$4^{13}/_{16}$	$^{21}/_{64}$	12,614	$4^{23}/_{32}$	$^9/_{32}$	10,957
$3^1/_2$	15.50	EU-E75	NC38	5	$2^9/_{16}$	12,196 P	$4^{17}/_{32}$	$^3/_{16}$	7,785	$4^{15}/_{32}$	$^5/_{32}$	6,769
$3^1/_2$	15.50	EU-X95	NC38	5	$2^7/_{16}$	13,328 P	$4^{21}/_{32}$	$^1/_4$	9,879	$4^{19}/_{32}$	$^7/_{32}$	8,822
$3^1/_2$	15.50	EU-G105	NC38	5	$2^1/_8$	15,909 P	$4^{23}/_{32}$	$^9/_{32}$	10,957	$4^5/_8$	$^{15}/_{64}$	9,348
	15.50	EU-G105	NC40	$5^1/_4$	$2^9/_{16}$	16,656 P	$4^{15}/_{16}$	$^1/_4$	11,363	$4^{27}/_{32}$	$^{13}/_{64}$	9,595
$3^1/_2$	15.50	EU-S135	NC40	$5^1/_2$	$2^1/_4$	19,766 P	$5^3/_{32}$	$^{21}/_{64}$	14,419	$4^{31}/_{32}$	$^{17}/_{64}$	11,963
4	11.85	EU-E75	NC46	6	$3^1/_4$	20,175 P	$5^7/_{32}$	$^7/_{64}$	7,843	$5^5/_{32}$	$^5/_{64}$	6,476
	11.85	EU-E75	4 WO	$5^3/_4$	$3^7/_{16}$	17,285 P	$5^7/_{32}$	$^7/_{64}$	7,843	$5^5/_{32}$	$^5/_{64}$	6,476

(Table continued on next page.)

Table 10—Recommended Minimum OD* and Make-up Torque of Weld-on Type Tool Joints Based on Torsional Strength of Box and Drill Pipe (Continued)

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)
							Premium Class			Class 2		
Drill Pipe			New Tool Joint Data				Min. OD Tool Joint in.	Min. Box Shoulder with Eccentric Wear in.	Make-up Torque for Min. OD Tool Joint ft-lb	Min. OD Tool Joint in.	Min. Box Shoulder with Eccentric Wear in.	Make-up Torque for Min. OD Tool Joint ft-lb
Nominal Size in.	Nom Weight lb/ft	Type Upset and Grade	Conn.	New OD in.	New ID in.	Make-up Torque[6] ft-lb						
	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
	11.85	IU-E75	4 H90	5 1/2	2 13/16	21,224 P	4 7/8	7/64	7,630	4 27/32	3/32	6,962
4	14.00	IU-E75	NC40	5 1/4	2 13/16	14,092 P	4 13/16	3/16	9,017	4 3/4	5/32	7,877
	14.00	EU-E75	NC46	6	3 1/4	20,175 P	5 9/32	9/64	9,233	5 7/32	7/64	7,843
	14.00	IU-E75	4 SH[2]	4 5/8	2 9/16	9,102 P	4 7/16	15/64	8,782	4 3/8	13/64	7,817
	14.00	EU-E75	4 OHSW	5 1/2	3 1/4	16,320 P	5 1/16	11/64	9,131	5	9/64	7,839
	14.00	IU-E75	4 H90	5 1/2	2 13/16	21,224 P	4 15/16	9/64	8,986	4 7/8	7/64	7,630
4	14.00	IU-X95	NC40	5 1/4	2 11/16	15,404 P	4 15/16	1/4	11,363	4 27/32	13/64	9,595
	14.00	EU-X95	NC46	6	3 1/4	20,175 P	5 3/8	3/16	11,363	5 5/16	5/32	9,937
	14.00	IU-X95	4 H90	5 1/2	2 13/16	21,224 P	5 1/32	3/16	11,065	4 31/32	5/32	9,673
4	14.00	IU-G105	NC40	5 1/2	2 7/16	18,068 P	5	9/32	12,569	4 29/32	15/64	10,768
	14.00	EU-G105	NC46	6	3 1/4	20,175 P	5 7/16	7/32	12,813	5 11/32	11/64	10,647
	14.00	IU-G105	4 H90	5 1/2	2 13/16	21,224 P	5 3/32	7/32	12,481	5 1/32	3/16	11,065
4	14.00	EU-S135	NC46	6	3	23,538 P	5 9/16	9/32	15,787	5 1/2	1/4	14,288
4	15.70	IU-E75	NC40	5 1/4	2 11/16	15,404 P	4 7/8	7/32	10,179	4 25/32	11/64	8,444
	15.70	EU-E75	NC46	6	3 1/4	20,175 P	5 5/16	5/32	9,937	5 1/4	1/8	8,535
	15.70	IU-E75	4 H90	5 1/2	2 13/16	21,224 P	4 31/32	5/32	9,673	4 29/32	1/8	8,305
4	15.70	IU-X95	NC40	5 1/2	2 7/16	18,068 P	5	9/32	12,569	4 29/32	15/64	10,768
	15.70	EU-X95	NC46	6	3	23,538 P	5 7/16	7/32	12,813	5 11/32	11/64	10,647
	15.70	IU-X95	4 H90	5 1/2	2 13/16	21,224 P	5 3/32	7/32	12,481	5 1/32	3/16	11,065
4	15.70	EU-G105	NC46	6	3	23,538 P	5 13/32	15/64	13,547	5 13/32	13/64	12,085
	15.70	IU-G105	4 H90	5 1/2	2 13/16	21,224 P	5 5/32	1/4	13,922	5 1/16	13/64	11,770
4	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
	15.70	EU-S135	NC46	6	2 7/8	25,118 P	5 21/32	21/64	18,083	5 17/32	17/64	15,035
4 1/2	16.60	IEU-E75	4 1/2 FH	6	3	20,868 P	5 3/8	13/64	12,125	5 9/32	5/32	10,072
	16.60	IEU-E75	NC46	6 1/4	3 1/4	20,396 P	5 13/32	13/64	12,085	5 11/32	11/64	10,647
	16.60	IEU-E75	4 1/2 OHSW	5 7/8	3 3/4	16,346 P	5 7/16	13/64	11,862	5 3/8	11/64	10,375
	16.60	EU-E75	NC50	6 5/8	3 3/4	22,836 P	5 23/32	5/32	11,590	5 11/16	9/64	10,773
	16.60	IEU-E75	4 1/2 H-90	6	3 1/4	23,355 P	5 11/32	3/16	12,215	5 9/32	5/32	10,642
4 1/2	16.60	IEU-X95	4 1/2 FH	6	2 3/4	23,843 P	5 1/2	17/64	14,945	5 13/32	7/32	12,821
	16.60	IEU-X95	NC46	6 1/4	3 1/4	20,396 P	5 17/32	17/64	15,035	5 7/16	7/32	12,813
	16.60	EU-X95	NC50	6 5/8	3 3/4	22,836 P	5 27/32	7/32	14,926	5 25/32	3/16	13,245
	16.60	IEU-X95	4 1/2 H-90	6	3	27,091 P	5 15/32	1/4	15,441	5 3/8	13/64	13,102
4 1/2	16.60	IEU-G105	4 1/2 FH	6	2 3/4	23,843 P	5 9/16	19/64	16,391	5 15/32	1/4	14,231
	16.60	IEU-G105	NC46	6 1/4	3	23,795 P	5 19/32	19/64	16,546	5 1/2	1/4	14,288
	16.60	EU-G105	NC50	6 5/8	3 3/4	22,836 P	5 29/32	1/4	16,633	5 13/16	13/64	14,082
	16.60	IEU-G105	4 1/2 H-90	6	3	27,091 P	5 1/2	17/64	16,264	5 7/16	15/64	14,625
4 1/2	16.60	IEU-S135	NC46	6 1/4	2 3/4	26,923 P	5 25/32	25/64	21,230	5 21/32	21/64	18,083
	16.60	EU-S135	NC50	6 5/8	3 1/2	27,076 P	6 1/16	21/64	21,017	5 31/32	9/32	18,367
4 1/2	20.00	IEU-E75	4 1/2 FH	6	3	20,868 P	5 15/32	1/4	14,231	5 3/8	13/64	12,125
	20.00	IEU-E75	NC46	6 1/4	3	23,795 P	5 1/2	1/4	14,288	5 13/32	13/64	12,085
	20.00	EU-E75	NC50	6 5/8	3 5/8	24,993 P	5 13/16	13/64	14,082	5 3/4	12/64	12,415
	20.00	IEU-E75	4 1/2 H-90	6	3	27,091 P	5 13/32	7/32	13,815	5 11/32	3/16	12,215
4 1/2	20.00	IEU-X95	4 1/2 FH	6	2 1/2	26,559 P	5 5/8	21/64	17,861	5 17/32	9/32	15,665
	20.00	IEU-X95	NC46	6 1/4	2 3/4	26,923 P	5 21/32	21/64	18,083	5 9/16	9/32	15,787
	20.00	EU-X95	NC50	6 5/8	3 1/2	27,076 P	5 15/16	17/64	17,497	5 7/8	15/64	15,776
	20.00	IEU-X95	4 1/2 H-90	6	3	27,091 P	5 9/16	19/64	17,929	5 15/32	1/4	15,441

(Table continued on next page.)

Table 10—Recommended Minimum OD* and Make-up Torque of Weld-on Type Tool Joints Based on Torsional Strength of Box and Drill Pipe (Continued)

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)
							Premium Class			Class 2		
Drill Pipe			New Tool Joint Data				Min. OD	Min. Box Shoulder	Make-up Torque for	Min. OD	Min. Box Shoulder	Make-up Torque for
Nominal Size in.	Nom Weight lb/ft	Type Upset and Grade	Conn.	New OD in.	New ID in.	Make-up Torque[6] ft-lb	Tool Joint in.	with Eccentric Wear in.	Min. OD Tool Joint ft-lb	Tool Joint in.	with Eccentric Wear in.	Min. OD Tool Joint ft-lb
$4\frac{1}{2}$	20.00	IEU-G105	NC46	$6\frac{1}{4}$	$2\frac{1}{2}$	29,778 P	$5\frac{23}{32}$	$\frac{23}{64}$	19,644	$5\frac{5}{8}$	$\frac{5}{16}$	17,311
	20.00	EU-G105	NC50	$6\frac{5}{8}$	$3\frac{1}{2}$	27,076 P	$6\frac{1}{32}$	$\frac{5}{16}$	20,127	$5\frac{29}{32}$	$\frac{1}{4}$	16,633
$4\frac{1}{2}$	20.00	EU-S135	NC50	$6\frac{5}{8}$	$2\frac{7}{8}$	36,398 P	$6\frac{7}{32}$	$\frac{13}{32}$	25,569	$6\frac{3}{32}$	$\frac{11}{32}$	21,914
5	19.50	IEU-E75	NC50	$6\frac{5}{8}$	$3\frac{3}{4}$	22,836 P	$5\frac{7}{8}$	$\frac{15}{64}$	15,776	$5\frac{13}{16}$	$\frac{13}{64}$	14,082
5	19.50	IEU-X95	NC50	$6\frac{5}{8}$	$3\frac{1}{2}$	27,076 P	$6\frac{1}{32}$	$\frac{5}{16}$	20,127	$5\frac{15}{16}$	$\frac{17}{64}$	17,497
	19.50	IEU-X95	5 H-90	$6\frac{1}{2}$	$3\frac{1}{4}$	31,084 P	$5\frac{27}{32}$	$\frac{19}{64}$	19,862	$5\frac{3}{4}$	$\frac{1}{4}$	17,116
5	19.50	IEU-G105	NC50	$6\frac{5}{8}$	$3\frac{1}{4}$	31,025 P	$6\frac{3}{32}$	$\frac{11}{32}$	21,914	6	$\frac{19}{64}$	19,244
	19.50	IEU-G105	5 H-90	$6\frac{1}{2}$	3	35,039 P	$5\frac{29}{32}$	$\frac{21}{64}$	21,727	$5\frac{13}{16}$	$\frac{9}{32}$	18,940
5	19.50	IEU-S135	NC50	$6\frac{5}{8}$	$2\frac{3}{4}$	38,044 P	$6\frac{5}{16}$	$\frac{29}{64}$	28,381	$6\frac{3}{16}$	$\frac{25}{64}$	24,645
	19.50	IEU-S135	$5\frac{1}{2}$ FH	$7\frac{1}{4}$	$3\frac{1}{2}$	43,490 P	$6\frac{3}{4}$	$\frac{3}{8}$	28,737	$6\frac{5}{8}$	$\frac{5}{16}$	24,412
5	25.60	IEU-E75	NC50	$6\frac{5}{8}$	$3\frac{1}{2}$	27,076 P	$6\frac{1}{32}$	$\frac{5}{16}$	20,127	$5\frac{15}{16}$	$\frac{17}{64}$	17,497
	25.60	IEU-E75	$5\frac{1}{2}$ FH	7	$3\frac{1}{2}$	37,742 B	$6\frac{1}{2}$	$\frac{1}{4}$	20,205	$6\frac{13}{32}$	$\frac{13}{64}$	17,127
5	25.60	IEU-X95	NC50	$6\frac{5}{8}$	3	34,680 P	$6\frac{7}{32}$	$\frac{13}{32}$	25,569	$6\frac{3}{32}$	$\frac{11}{32}$	21,914
	25.60	IEU-X95	$5\frac{1}{2}$ FH	7	$3\frac{1}{2}$	37,742 B	$6\frac{21}{32}$	$\frac{21}{64}$	25,483	$6\frac{9}{16}$	$\frac{9}{32}$	22,294
5	25.60	IEU-G105	NC50	$6\frac{5}{8}$	$2\frac{3}{4}$	38,044 P	$6\frac{9}{32}$	$\frac{7}{16}$	27,437	$6\frac{5}{32}$	$\frac{3}{8}$	23,728
	25.60	IEU-G105	$5\frac{1}{2}$ FH	$7\frac{1}{4}$	$3\frac{1}{2}$	43,490 P	$6\frac{23}{32}$	$\frac{23}{64}$	27,645	$6\frac{5}{8}$	$\frac{5}{16}$	24,412
5	25.60	IEU-S135	$5\frac{1}{2}$ FH	$7\frac{1}{4}$	$3\frac{1}{4}$	47,230 B	$6\frac{15}{16}$	$\frac{15}{32}$	35,446	$6\frac{13}{16}$	$\frac{13}{32}$	30,943
$5\frac{1}{2}$	21.90	IEU-E75	$5\frac{1}{2}$ FH	7	4	33,560 P	$6\frac{15}{32}$	$\frac{15}{64}$	19,172	$6\frac{13}{32}$	$\frac{13}{64}$	17,127
$5\frac{1}{2}$	21.90	IEU-X95	$5\frac{1}{2}$ FH	7	$3\frac{3}{4}$	37,742 B	$6\frac{5}{8}$	$\frac{5}{16}$	24,412	$6\frac{17}{32}$	$\frac{17}{64}$	21,246
	21.90	IEU-X95	$5\frac{1}{2}$ H-90	7	$3\frac{1}{2}$	35,454 P	$6\frac{3}{16}$	$\frac{21}{64}$	24,414	$6\frac{3}{32}$	$\frac{9}{32}$	21,349
$5\frac{1}{2}$	21.90	IEU-G105	$5\frac{1}{2}$ FH	$7\frac{1}{4}$	$3\frac{1}{2}$	43,490 P	$6\frac{23}{32}$	$\frac{23}{64}$	27,645	$6\frac{19}{32}$	$\frac{19}{64}$	23,350
$5\frac{1}{2}$	21.90	IEU-S135	$5\frac{1}{2}$ FH	$7\frac{1}{2}$	3	53,302 P	$6\frac{15}{16}$	$\frac{15}{32}$	35,446	$6\frac{13}{16}$	$\frac{13}{32}$	30,943
$5\frac{1}{2}$	24.70	IEU-E75	$5\frac{1}{2}$ FH	7	4	33,560 P	$6\frac{9}{16}$	$\frac{9}{32}$	22,294	$6\frac{15}{32}$	$\frac{15}{64}$	19,172
$5\frac{1}{2}$	24.70	IEU-X95	$5\frac{1}{2}$ FH	$7\frac{1}{4}$	$3\frac{1}{2}$	43,490 P	$6\frac{23}{32}$	$\frac{23}{64}$	27,645	$6\frac{19}{32}$	$\frac{19}{64}$	23,350
$5\frac{1}{2}$	24.70	IEU-G105	$5\frac{1}{2}$ FH	$7\frac{1}{4}$	$3\frac{1}{2}$	43,490 P	$6\frac{25}{32}$	$\frac{25}{64}$	29,836	$6\frac{11}{16}$	$\frac{11}{32}$	26,560
$5\frac{1}{2}$	24.70	IEU-S135	$5\frac{1}{2}$ FH	$7\frac{1}{2}$	3	52,302 P	$7\frac{1}{32}$	$\frac{33}{64}$	38,901	$6\frac{7}{8}$	$\frac{7}{16}$	33,180
$6\frac{5}{8}$	25.20	IEU-E75	$6\frac{5}{8}$ FH	8	5	44,196 P	$7\frac{7}{16}$	$\frac{1}{4}$	26,810	$7\frac{3}{8}$	$\frac{7}{32}$	24,100
		IEU-X95	$6\frac{5}{8}$ FH	8	5	44,196 P	$7\frac{5}{8}$	$\frac{11}{32}$	35,139	$7\frac{1}{2}$	$\frac{9}{32}$	29,552
		IEU-G105	$6\frac{5}{8}$ FH	$8\frac{1}{4}$	$4\frac{3}{4}$	51,742 P	$7\frac{11}{16}$	$\frac{3}{8}$	37,983	$7\frac{19}{32}$	$\frac{21}{64}$	33,730
		IEU-S135	$6\frac{5}{8}$ FH	$8\frac{1}{2}$	$4\frac{1}{4}$	65,535 P	$7\frac{29}{32}$	$\frac{31}{64}$	48,204	$7\frac{25}{32}$	$\frac{27}{64}$	42,312
$6\frac{5}{8}$	27.70	IEU-E75	$6\frac{5}{8}$ FH	8	5	44,196 P	$7\frac{1}{2}$	$\frac{9}{32}$	29,552	$7\frac{13}{32}$	$\frac{15}{64}$	25,451
		IEU-X95	$6\frac{5}{8}$ FH	$8\frac{1}{4}$	$4\frac{3}{4}$	51,742 P	$7\frac{11}{16}$	$\frac{3}{8}$	37,983	$7\frac{9}{16}$	$\frac{5}{16}$	32,329
		IEU-G105	$6\frac{5}{8}$ FH	$8\frac{1}{4}$	$4\frac{3}{4}$	51,742 P	$7\frac{3}{4}$	$\frac{13}{32}$	40,860	$7\frac{21}{32}$	$\frac{23}{64}$	36,556
		IEU-S135	$6\frac{5}{8}$ FH	$8\frac{1}{2}$	$4\frac{1}{4}$	65,535 P	8	$\frac{17}{32}$	52,714	$7\frac{27}{64}$	$\frac{29}{64}$	45,241

[1]The use of outside diameters (OD) smaller than those listed in the table may be acceptable due to special service requirements.

[2]Tool joint with dimensions shown has lower torsional yield ratio than the 0.80 which is generally used.

[3]Recommended make-up torque is based on 72,000 psi stress.

[4]In calculation of torsional strengths of tool joints, both new and worn, the bevels of the tool joint shoulders are disregarded. This thickness measurement should be made in the plane of the face from the I.D. of the counter bore to the outside diameter of the box, disregarding the bevels.

[5]Any tool joint with an outside diameter less than API bevel diameter should be provided with a minimum $\frac{1}{32}$ inch depth x 45 degree bevel on the outside and inside diameter of the box shoulder and outside diameter of the pin shoulder.

[6] p = Pin limit; B = Box limit.

*Tool joint diameters specified are required to retain torsional strength in the tool joint comparable to the torsional strength of the attached drill pipe. These should be adequate for all service. Tool joints with torsional strengths considerably below that of the drill pipe may be adequate for much drilling service.

Table 11—Buoyancy Factors

(1)	(2)	(3)
Mud Density lb/gal	Mud Density lb/cu ft	Buoyancy Factor, Kb
8.4	62.84	.872
8.6	64.33	.869
8.8	65.83	.866
9.0	67.32	.862
9.2	68.82	.859
9.4	70.32	.856
9.6	71.81	.853
9.8	73.31	.850
10.0	74.80	.847
10.2	76.30	.844
10.4	77.80	.841
10.6	79.29	.838
10.8	80.79	.835
11.0	82.29	.832
11.2	83.78	.829
11.4	85.28	.826
11.6	86.77	.823
11.8	88.27	.820
12.0	89.77	.817
12.2	91.26	.814
12.4	92.76	.811
12.6	94.25	.807
12.8	95.75	.804
13.0	97.25	.801
13.2	98.74	.798
13.4	100.24	.795
13.6	101.74	.792
13.8	103.23	.789
14.0	104.73	.786
14.2	106.22	.783
14.4	107.72	.780
14.6	109.22	.777
14.8	110.71	.774
15.0	112.21	.771
15.2	113.70	.768
15.4	115.20	.765
15.6	116.70	.762
15.8	118.19	.759
16.0	119.69	.756
16.2	121.18	.752
16.4	122.68	.749
16.6	124.18	.746
16.8	125.67	.743
17.0	127.17	.740
17.2	128.66	.737
17.4	130.16	.734
17.6	131.66	.731
17.8	133.15	.728
18.0	134.65	.725
18.5	138.39	.717
19.0	142.13	.710
19.5	145.87	.702
20.0	149.61	.694

Table 12—Rotary Shouldered Connection Interchange List

Common Name: Style	Common Name: Size	Pin Base Diameter (Tapered)	Threads Per in.	Taper in/ft	Thread Form[1]	Same as or Interchanges With
Internal Flush (I.F.)	$2^3/_8$"	2.876	4	2	V-0.065 (V-0.038 rad)	$2^7/_8$" Slim Hole N.C. 26[2]
	$2^7/_8$"	3.391	4	2	V-0.065 (V-0.038 rad)	$3^1/_2$" Slim Hole N.C. 31[2]
	$3^1/_2$"	4.016	4	2	V-0.065 (V-0.038 rad)	$4^1/_2$" Slim Hole N.C. 38[2]
	4"	4.834	4	2	V-0.065 (V-0.038 rad)	$4^1/_2$" Extra Hole N.C. 46[2]
	$4^1/_2$"	5.250	4	2	V-0.065 (V-0.038 rad)	5" Extra Hole N.C. 50[2] $5^1/_2$" Double Streamline
Full Hole (F.H.)	4	4.280	4	2	V-0.065 (V-0.038 rad)	$4^1/_2$" Double Streamline N.C. 40[2]
Extra Hole (X.H.) (E.H.)	$2^7/_8$"	3.327	4	2	V-0.065 (V-0.038 rad)	$3^1/_2$" Double Streamline
	$3^1/_2$"	3.812	4	2	V-0.065 (V-0.038 rad)	4" Slim Hole $4^1/_2$" External Flush
	$4^1/_2$"	4.834	4	2	V-0.065 (V-0.038 rad)	4" Internal Flush N.C. 46[2]
	5"	5.250	4	2	V-0.065 (V-0.038 rad)	$4^1/_2$" Internal Flush N.C. 50[2] $5^1/_2$" Double Streamline
Slim Hole (S.H.)	$2^7/_8$"	2.876	4	2	V-0.065 (V-0.038 rad)	$2^3/_8$" Internal Flush N.C. 26[2]
	$3^1/_2$"	3.391	4	2	V-0.065 (V-0.038 rad)	$2^7/_8$" Internal Flush N.C. 31[2]
	4"	3.812	4	2	V-0.065 (V-0.038 rad)	$3^1/_2$" Extra Hole $4^1/_2$" External Flush
	$4^1/_2$"	4.016	4	2	V-0.065 (V-0.038 rad)	$3^1/_2$" Internal Flush N.C. 38[2]
Double Streamline (DSL)	$3^1/_2$"	3.327	4	2	V-0.065 (V-0.038 rad)	$2^7/_8$" Extra Hole
	$4^1/_2$"	4.280	4	2	V-0.065 (V-0.038 rad)	4" Full Hole N.C. 40[2]
	$5^1/_2$"	5.250	4	2	V-0.065 (V-0.038 rad)	$4^1/_2$" Internal Flush 5" Extra Hole N.C. 50[2]
Numbered Connection (N.C.)	26	2.876	4	2	V-0.038 rad	$2^3/_8$" Internal Flush $2^7/_8$" Slim Hole
	31	3.391	4	2	V-0.038 rad	$2^7/_8$" Internal Flush $3^1/_2$" Slim Hole
	38	4.016	4	2	V-0.038 rad	$3^1/_2$" Internal Flush $4^1/_2$" Slim Hole
	40	4.280	4	2	V-0.038 rad	4" Full Hole $4^1/_2$" Double Streamline
	46	4.834	4	2	V-0.038 rad	4" Internal Flush $4^1/_2$" Extra Hole
	50	5.250	4	2	V-0.038 rad	$4^1/_2$" Internal Flush 5" Extra Hole
External Flush (E.F.)	$4^1/_2$"	3.812	4	2	V-0.065 (V-0.038 rad)	4" Slim Hole $3^1/_2$" Extra Hole

[1]Connections with two thread forms shown may be machined with either thread form without affecting gauging or interchangeability.
[2]Numbered connections (N.C.) may be machined only with the V-0.038 radius thread form.

Figures 1-25—Torsional Strength and Recommended Make-up Torque Curves
(All curves based on 120,000 psi minimum yield strength and 60 percent
of minimum yield strength for recommended make-up torque.)

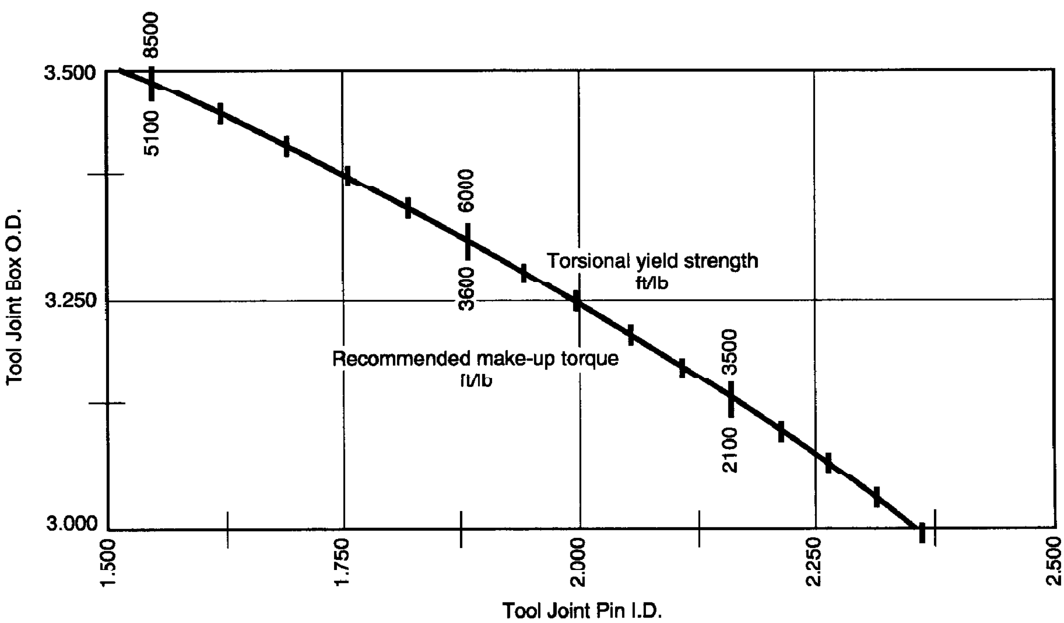


Figure 1—NC26 Torsional Yield and Make-up

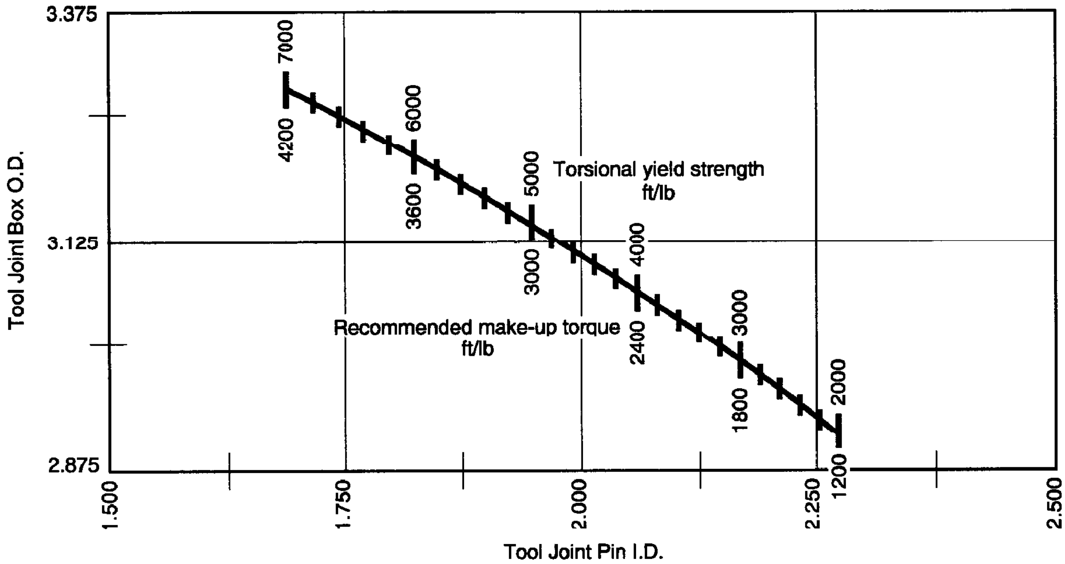


Figure 2—$2^3/_8$ Open Hole Torsional Yield and Make-up

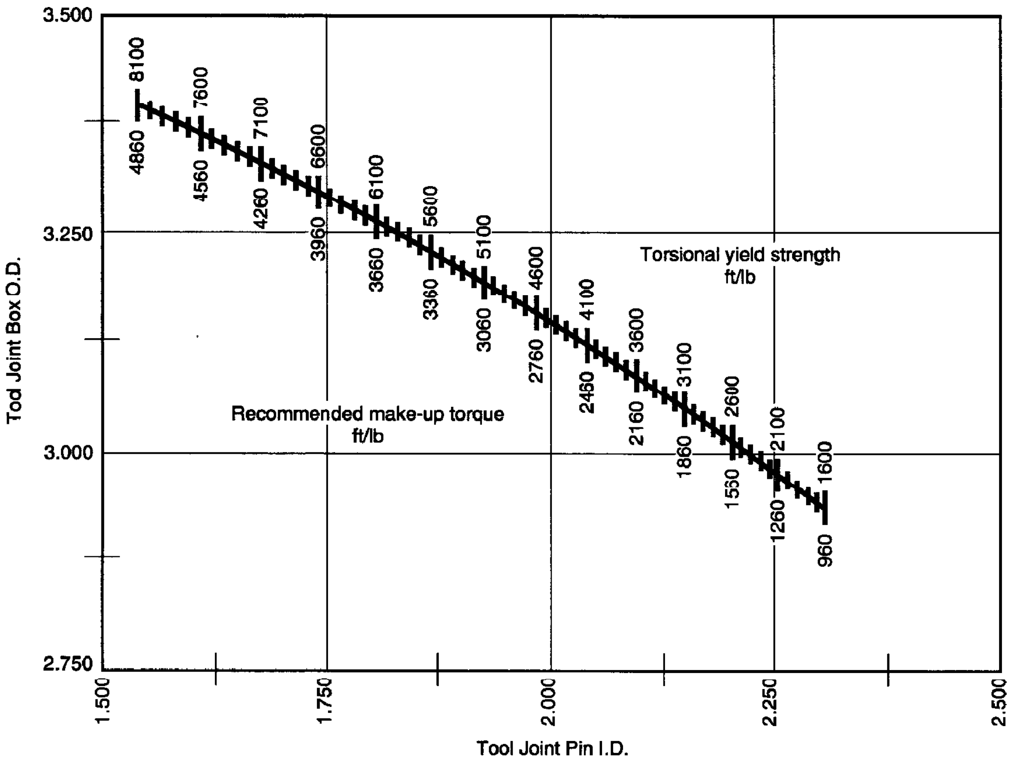


Figure 3—2$^3/_8$ Wide Open Torsional Yield and Make-up

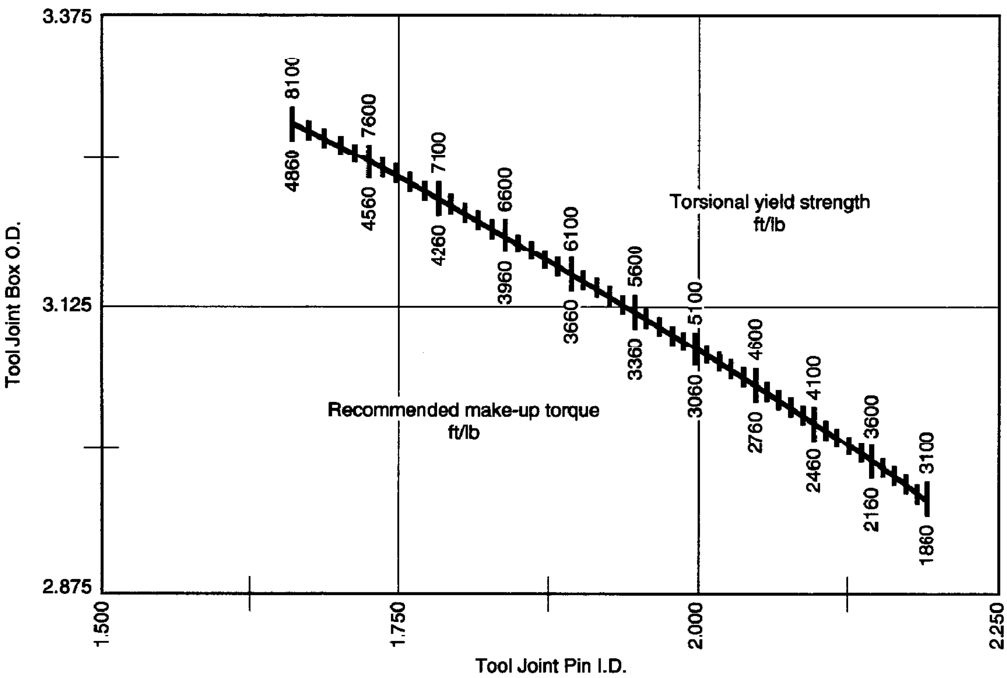


Figure 4—2$^3/_8$ SLH90 Torsional Yield and Make-up

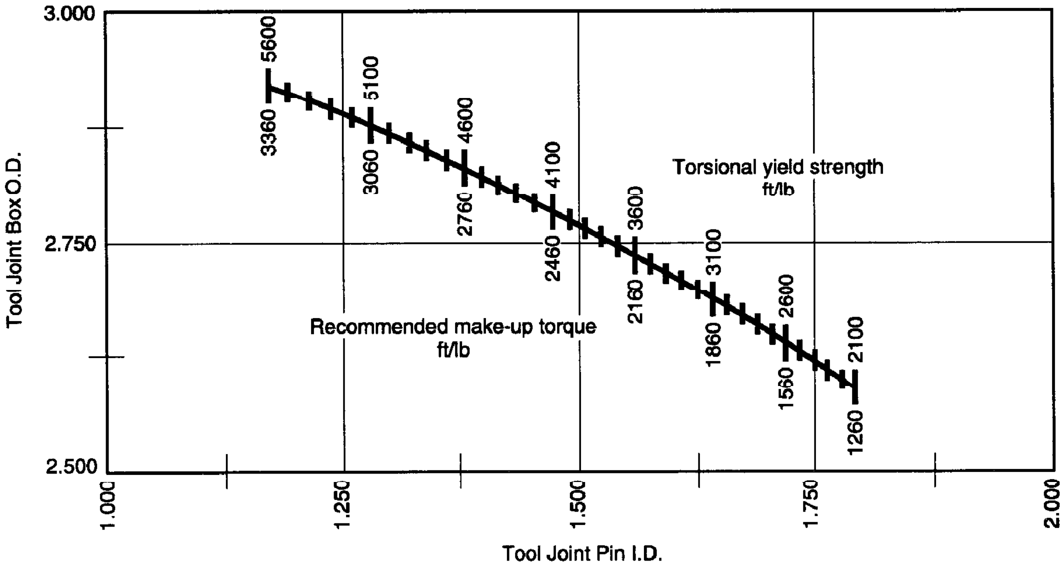


Figure 5—$2^3/_8$ PAC Torsional Yield and Make-up

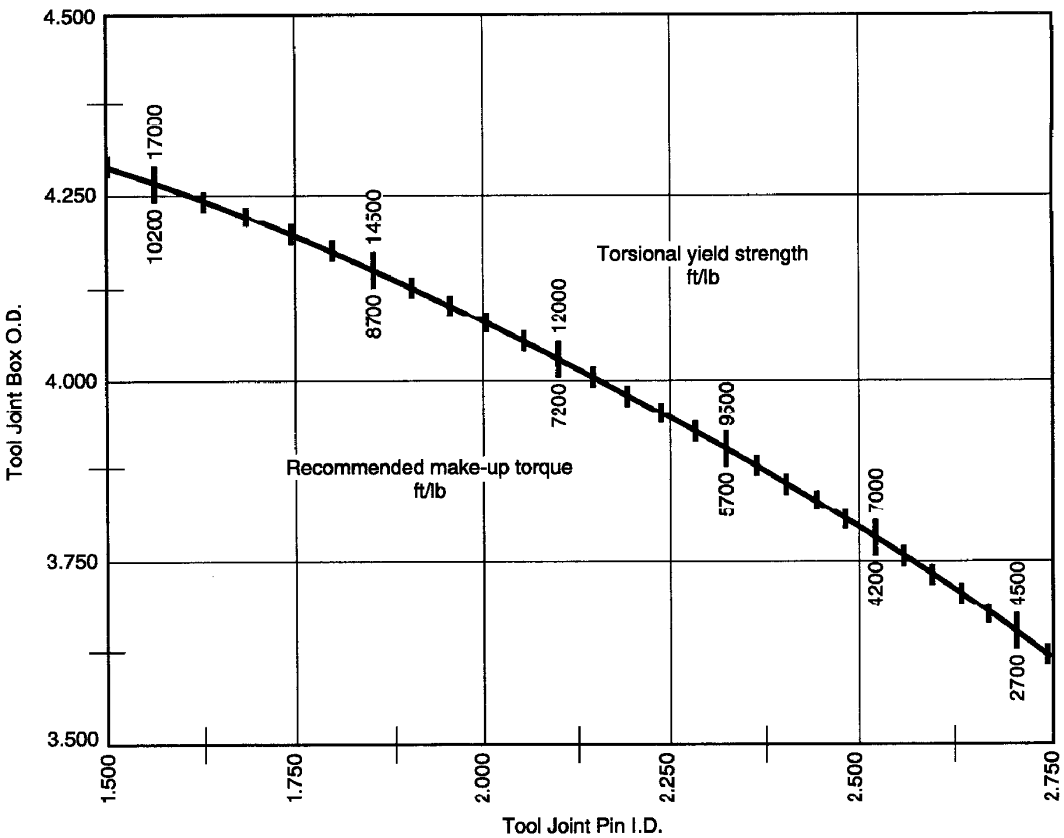


Figure 6—NC31 Torsional Yield and Make-up

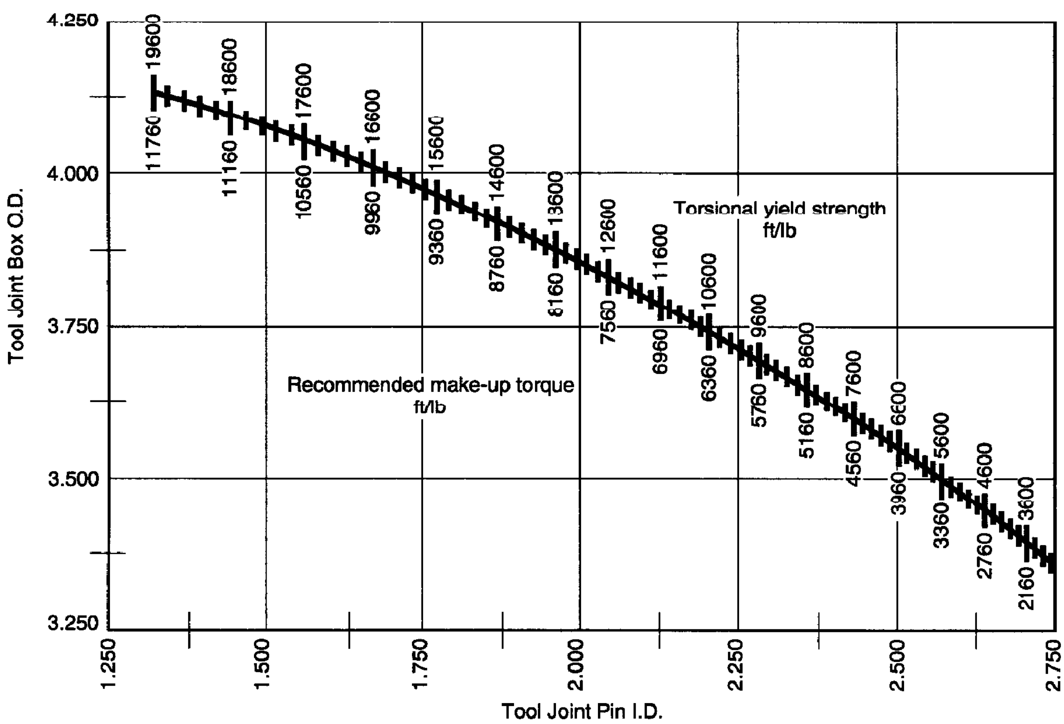


Figure 7—2⁷/₈ SLH90 Torsional Yield and Make-up

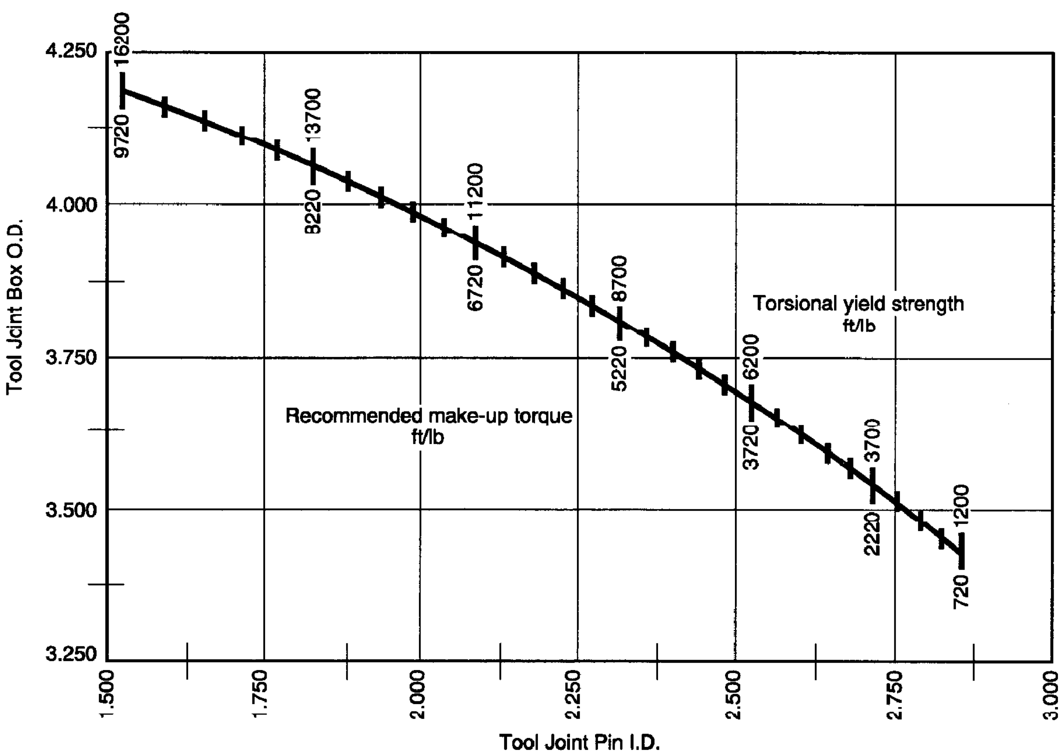


Figure 8—2⁷/₈ Wide Open Torsional Yield and Make-up

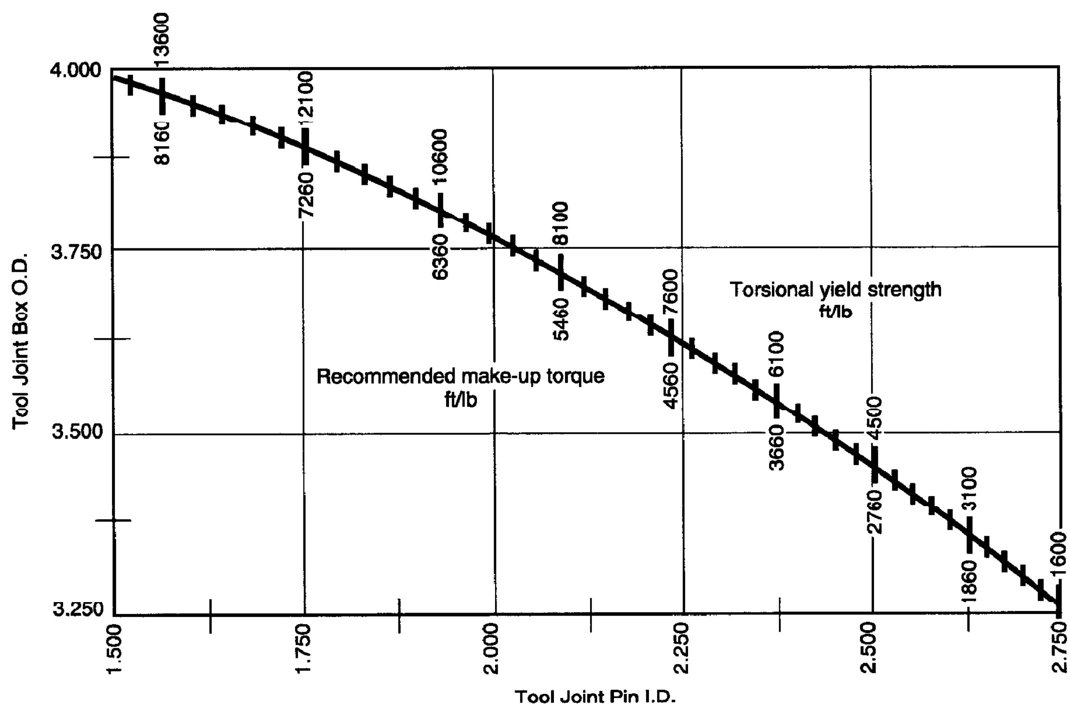


Figure 9—2⁷/₈ Open Hole Torsional Yield and Make-up

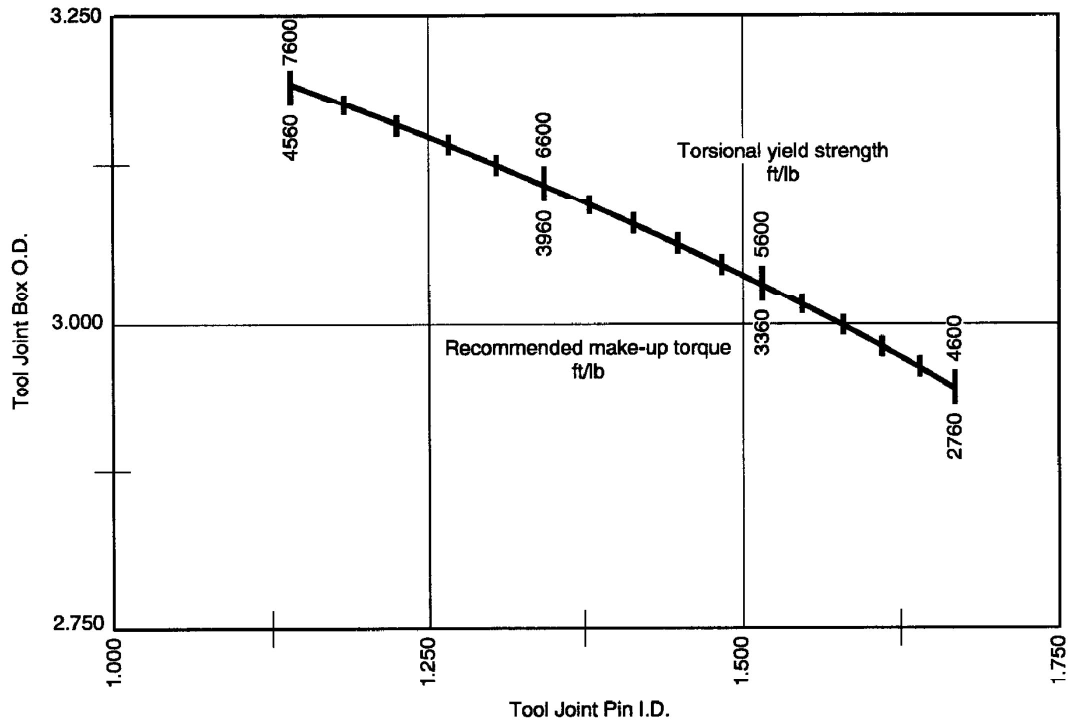


Figure 10—2⁷/₈ PAC Torsional Yield and Make-up

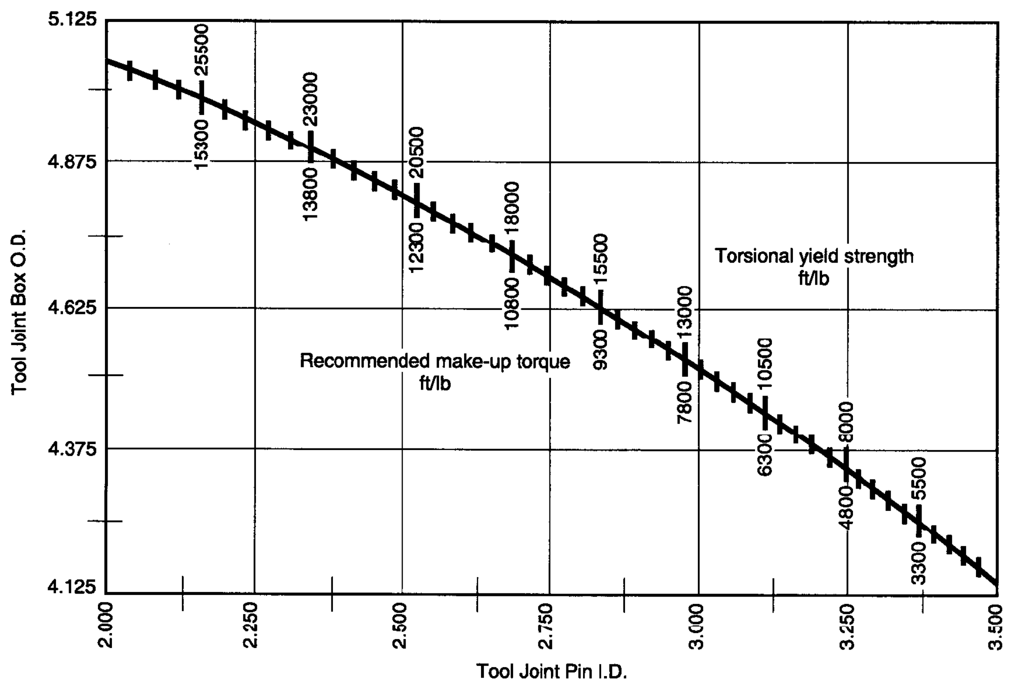


Figure 11—NC38 Torsional Yield and Make-up

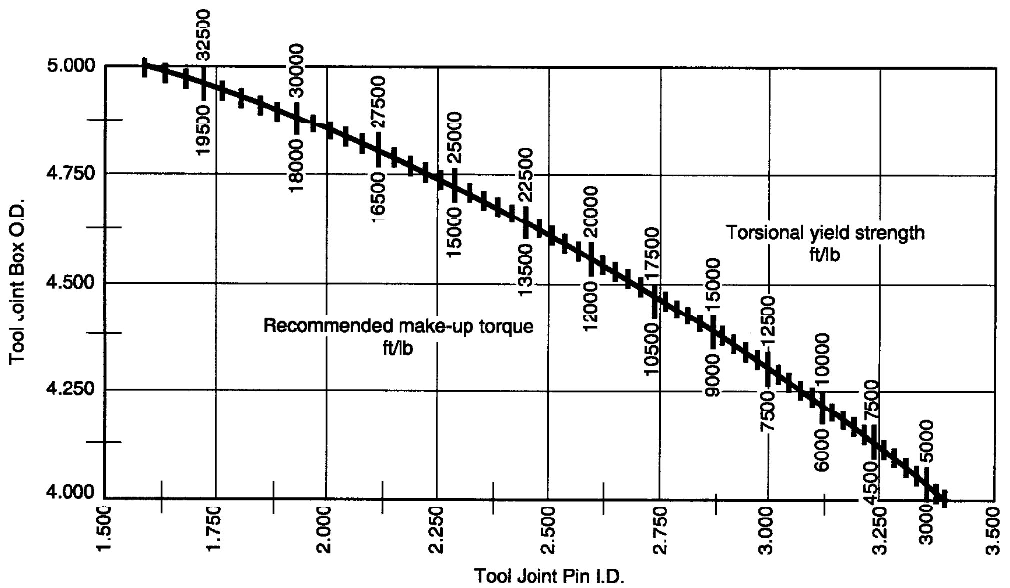


Figure 12—3$^1/_2$ SLH90 Torsional Yield and Make-up

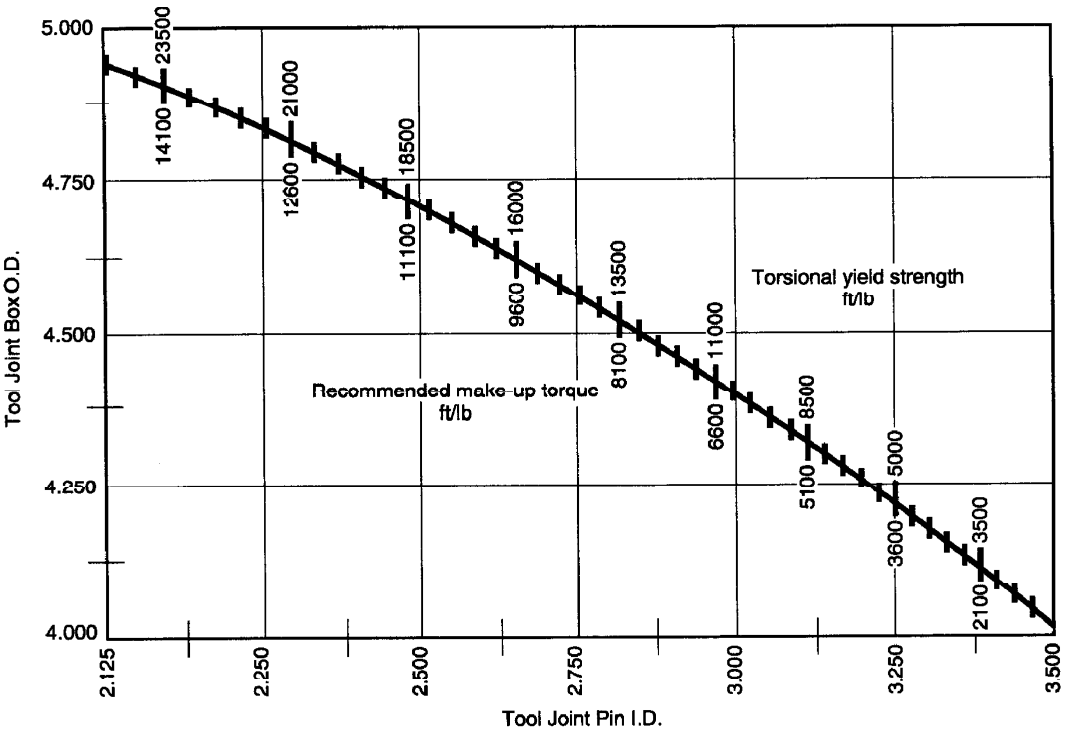


Figure 13—3½ FH Torsional Yield and Make-up

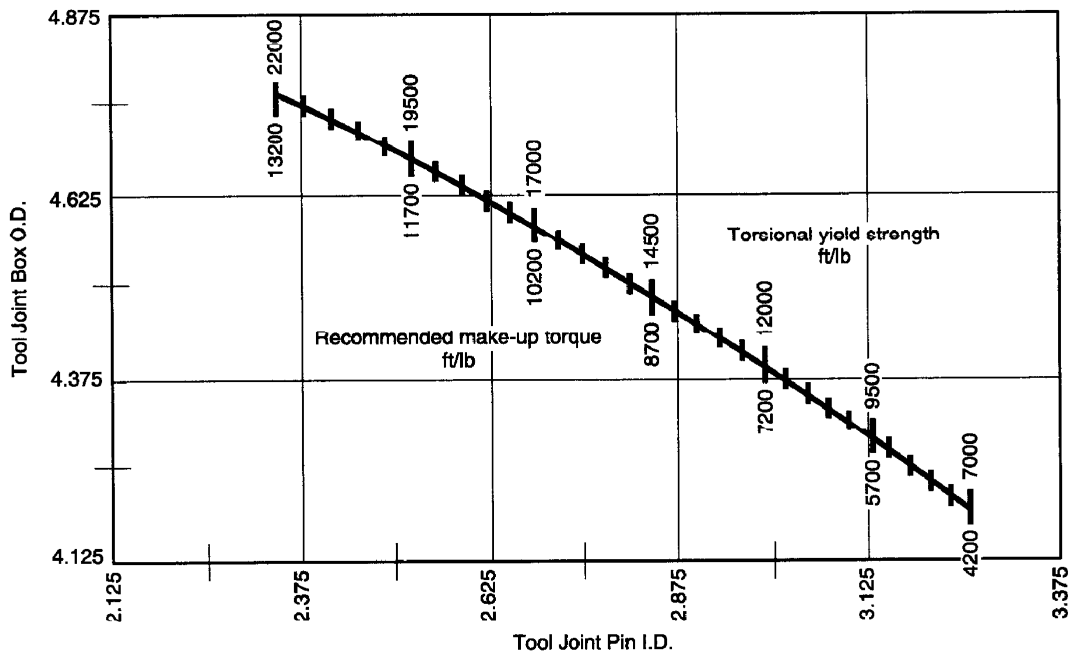


Figure 14—3½ Open Hole Torsional Yield and Make-up

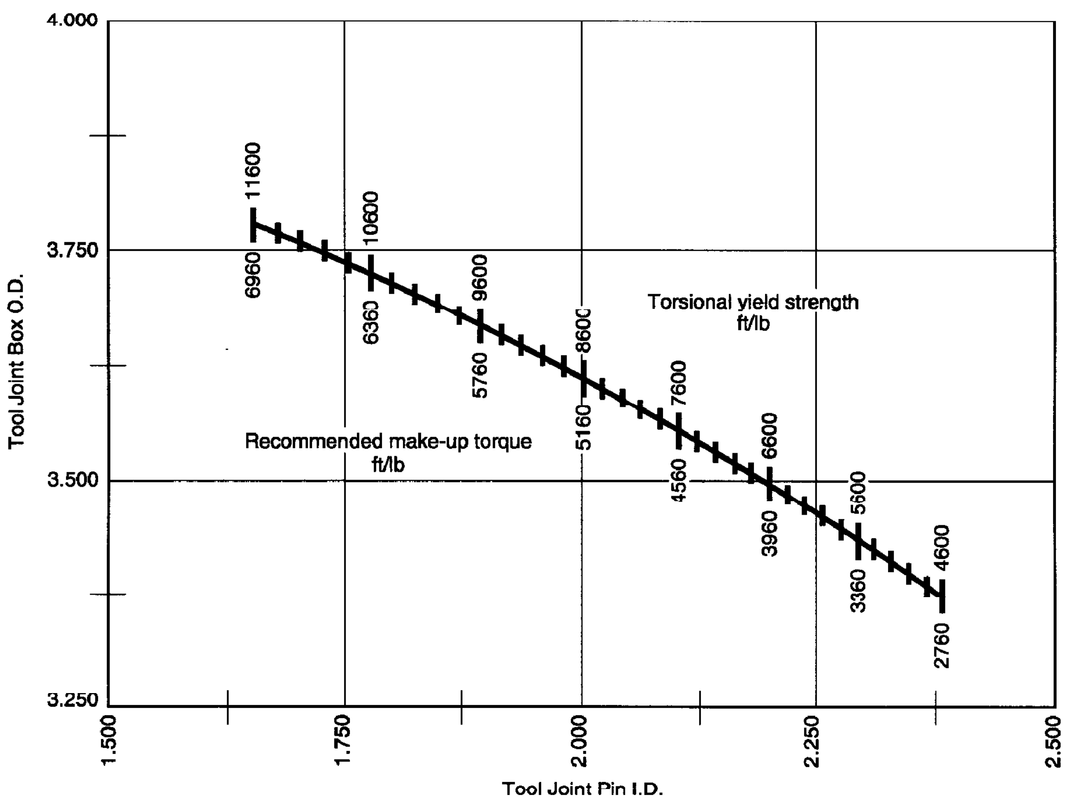


Figure 15—3½ PAC Torsional Yield and Make-up

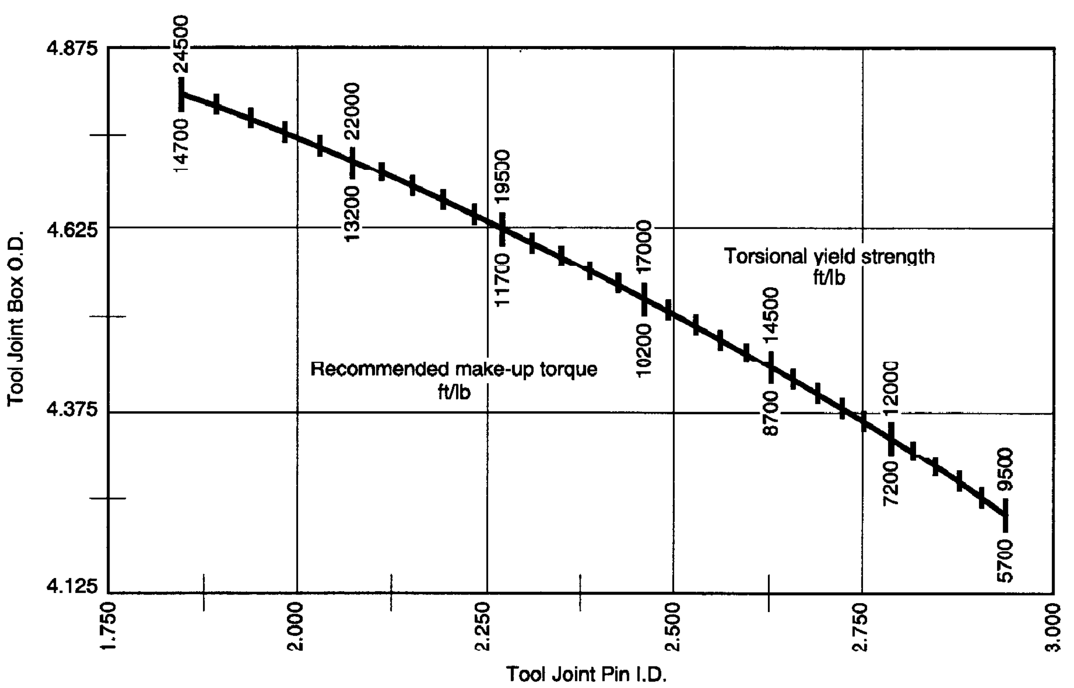


Figure 16—3½ XH Torsional Yield and Make-up

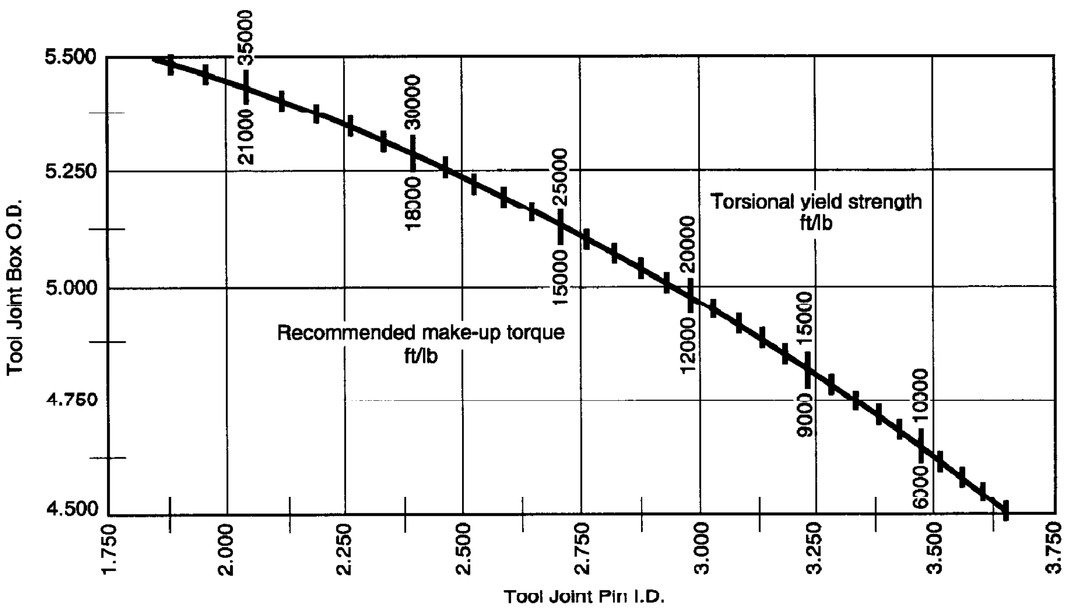


Figure 17—NC40 Torsional Yield and Make-up

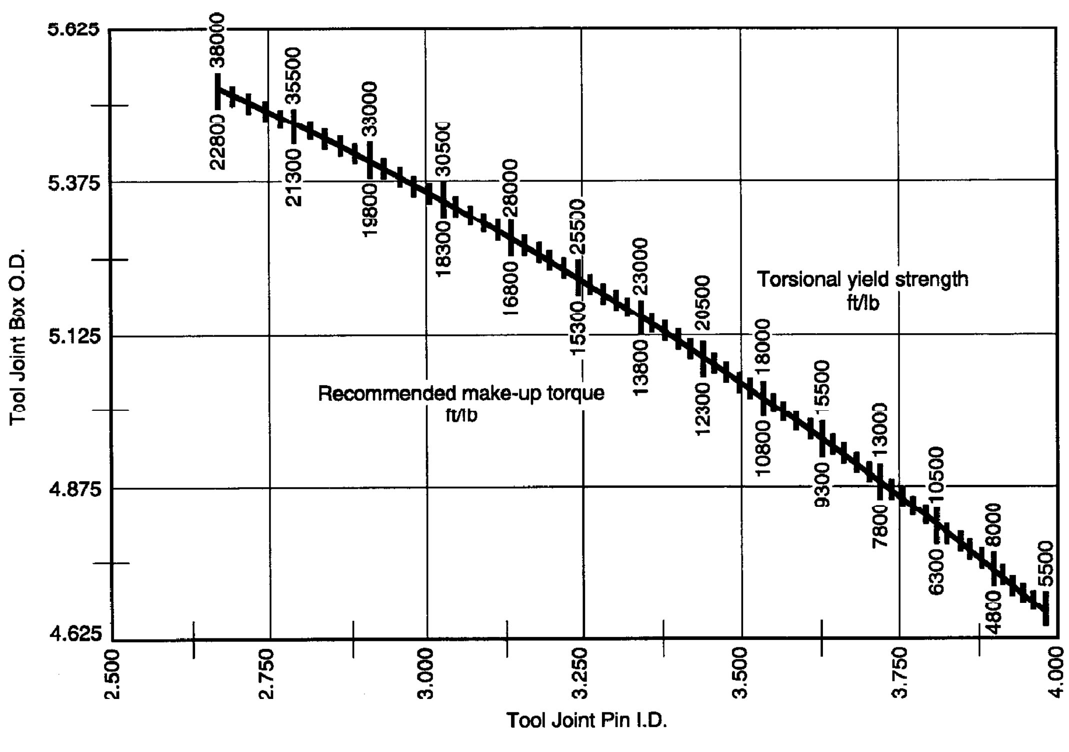


Figure 18—4-Inch H90 Torsional Yield and Make-up

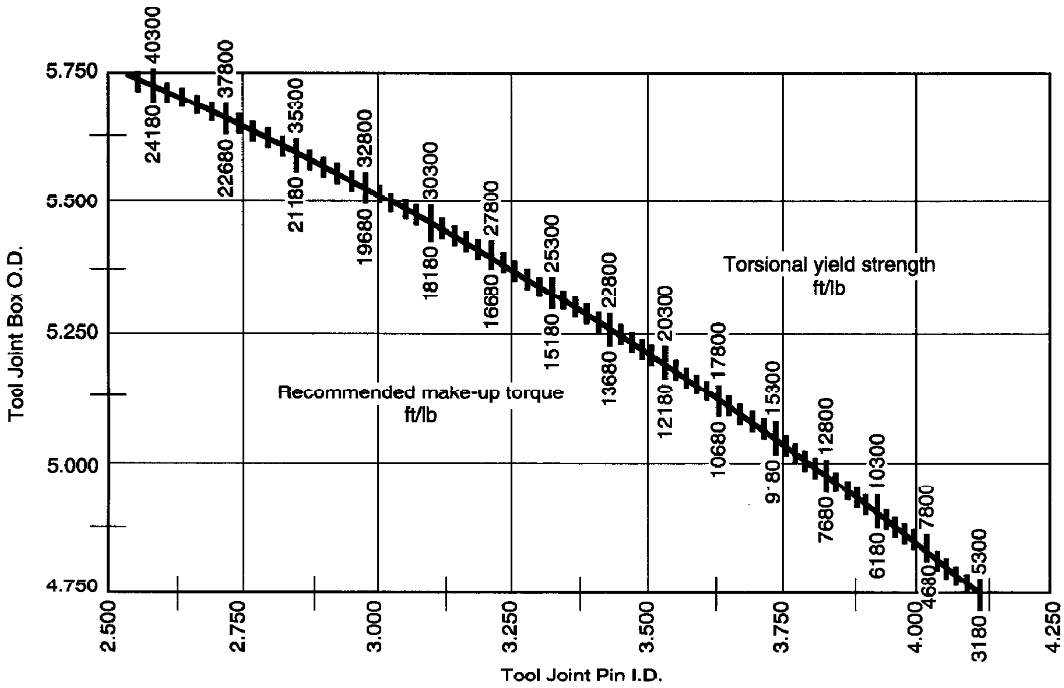


Figure 19—4-Inch Open Hole Torsional Yield and Make-up

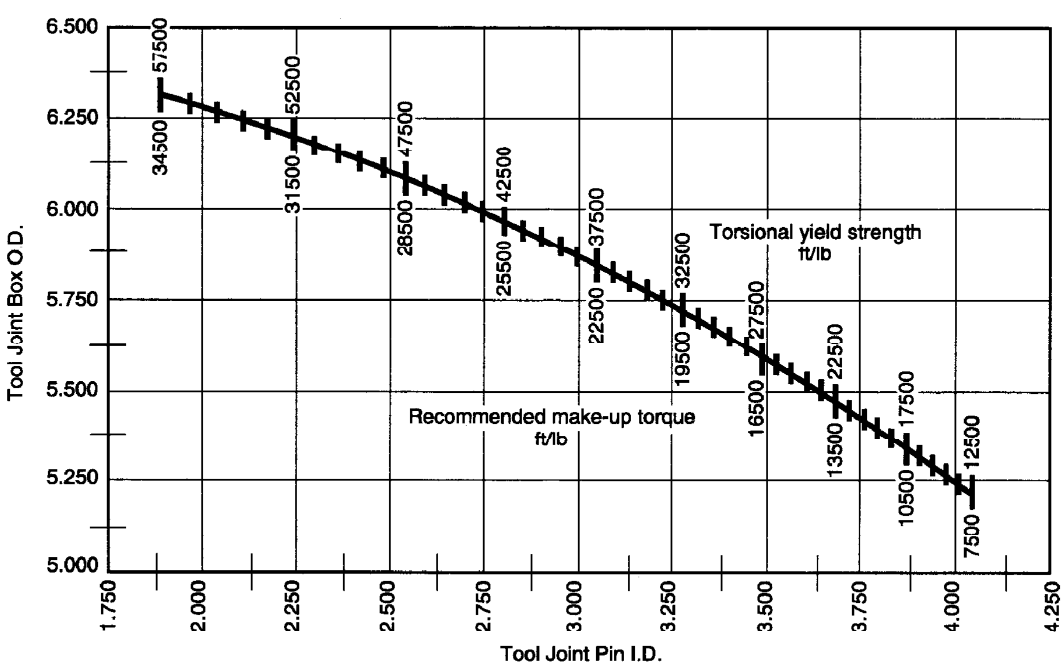


Figure 20—NC46 Torsional Yield and Make-up

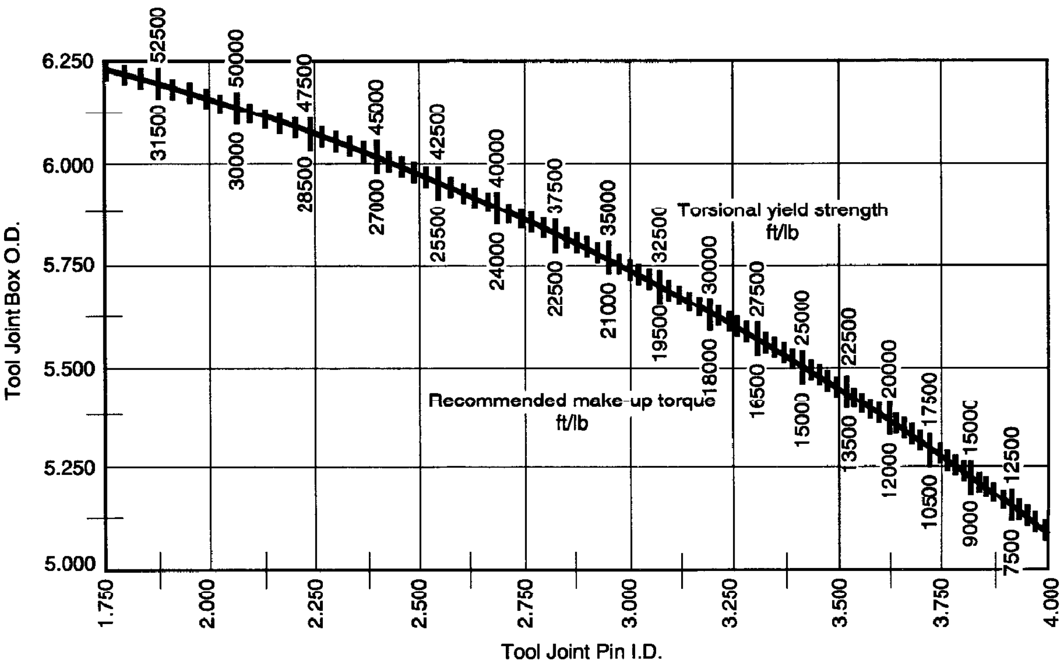


Figure 21—4½ FH Torsional Yield and Make-up

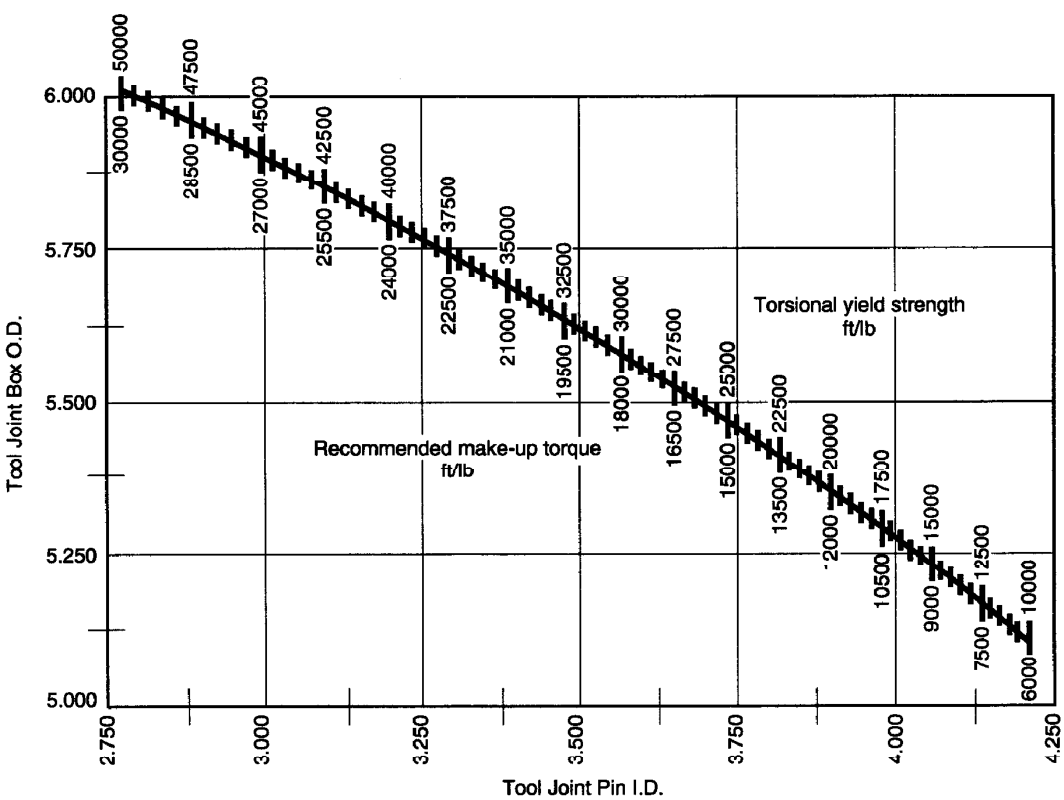


Figure 22—4½ H90 Torsional Yield and Make-up

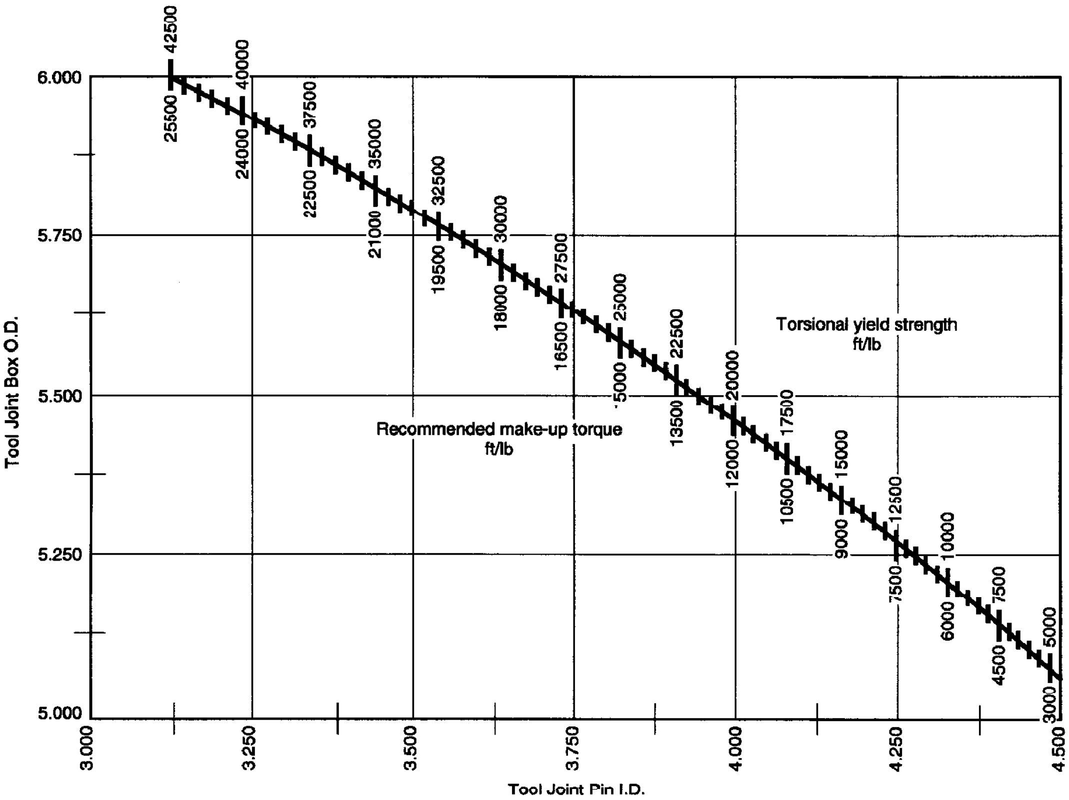


Figure 23—4½ Open Hole (Standard Weight) Torsional Yield and Make-up

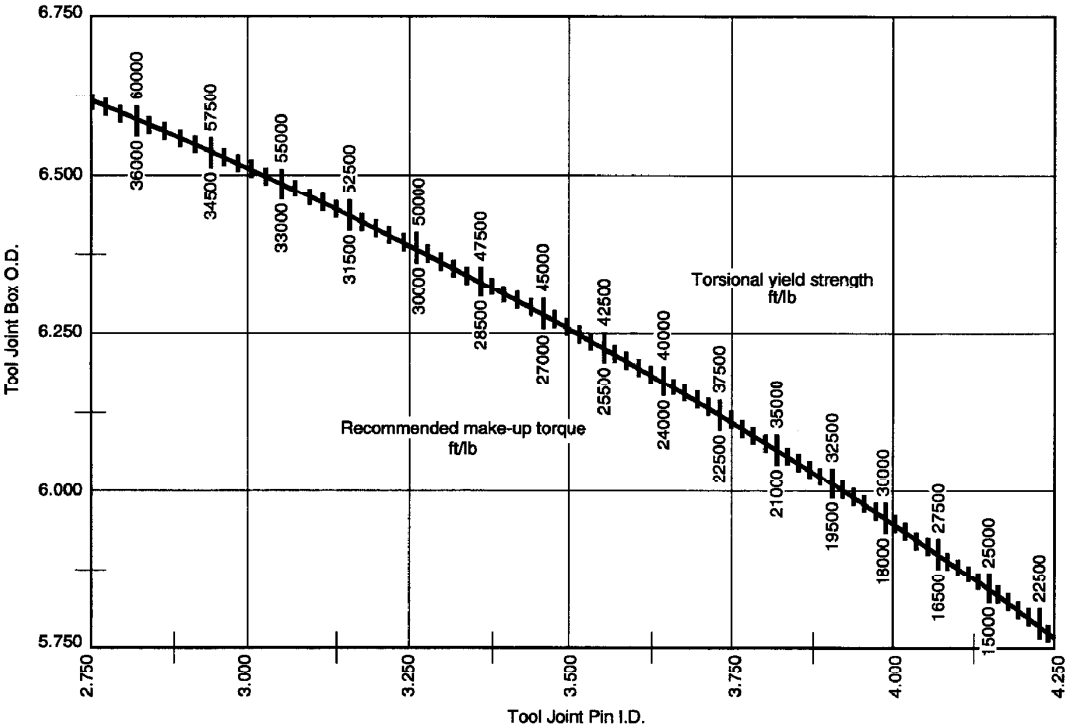


Figure 24—NC50 Torsional Yield and Make-up

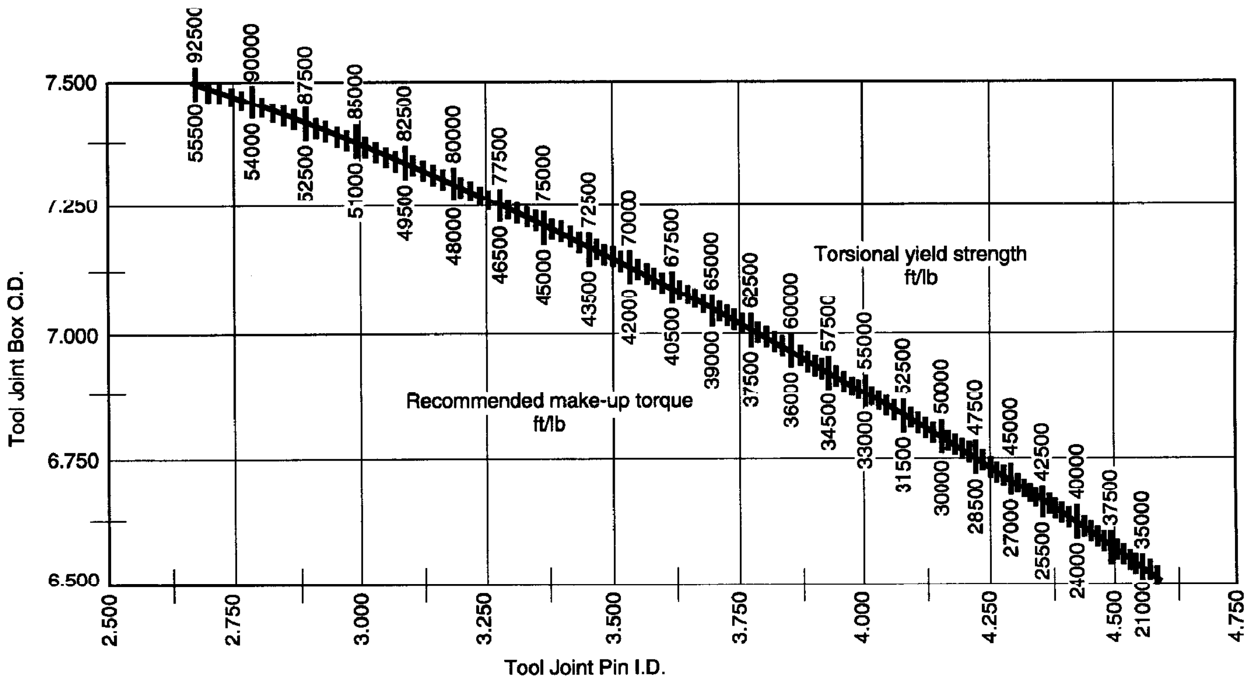


Figure 25—5$^1/_2$ FH Torsional Yield and Make-up

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

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

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

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

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

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

5 Properties Of Drill Collars

5.1 Table 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.

5.2 Recommended make-up torque values for rotary shouldered drill collar connections are listed in Table 14. These values are listed for various connection styles and for commonly used drill collar OD and ID sizes. The table also includes a designation of the weak member (pin or box) for each connection size and style.

5.3 Many drill collar connection failures are a result of bending stresses rather than torsional stresses. Figures 26 through 32 may be used for determining the most suitable connection to be used on new drill collars or for selecting the new connection to be used on collars which have been worn down on the outside diameter.

5.4 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 1.90:1 depending upon the drilling conditions.

5.5 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.

5.6 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.

5.7 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.

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

5.9 The curves in Figures 26 through 32 were determined from bending strength ratios calculated by using the Section Modulus (Z) as the measure of the capacity of a section to resist any bending moment to which it may be subjected. The effect of stress-relief features is disregarded. The equation, its derivation, and an example of its use are included in A.10.

6 Properties of Kellys

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

6.2 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.

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

6.3.1 Full length quenched and tempered before machining; or

6.3.2 The drive section normalized and tempered and the ends quenched and tempered.

It should be noted that 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.

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

6.4.1 It may be noted from Table 15 that the drive section of the hexagonal kelly is stronger than the drive section of the square kelly when the appropriate kelly is selected for a given casing size.

Example: A $4^1/_4$-inch square kelly or a $5^1/_4$-inch hexagonal kelly would be selected for use in $8^5/_8$-inch casing.

It should be noted, however, that 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.

(Text continued on page 46.)

Table 13—Drill Collar Weight (Steel) (pounds per foot)

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)
Drill Collar OD, inches	Drill Collar ID, 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	22	22	20										
3 1/4	26	24	22										
3 1/2	30	29	27										
3 3/4	35	33	32										
4	40	39	37	35	32	29							
4 1/8	43	41	39	37	35	32							
4 1/4	46	44	42	40	38	35							
4 1/2	51	50	48	46	43	41							
4 3/4			54	52	50	47	44						
5			61	59	56	53	50						
5 1/4			68	65	63	60	57						
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	91	89	85	80		
6 3/4			116	114	111	108	105	100	98	93	89		
7			125	123	120	117	114	110	107	103	98	93	84
7 1/4			134	132	130	127	124	119	116	112	108	103	93
7 1/2			144	142	139	137	133	129	126	122	117	113	102
7 3/4			154	152	150	147	144	139	136	132	128	123	112
8			165	163	160	157	154	150	147	143	138	133	122
8 1/4			176	174	171	168	165	160	158	154	149	144	133
8 1/2			187	185	182	179	176	172	169	165	160	155	150
9			210	208	206	203	200	195	192	188	184	179	174
9 1/2			234	232	230	227	224	220	216	212	209	206	198
9 3/4			248	245	243	240	237	232	229	225	221	216	211
10			261	259	257	254	251	246	243	239	235	230	225
11			317	315	313	310	307	302	299	295	291	286	281
12			379	377	374	371	368	364	361	357	352	347	342

Notes:

1. See API Specification 7, Table 13 for API standard drill collar dimensions.
2. For special configurations of drill collars, consult manufacturer for reduction in weight.

Table 14—Recommended Make-up Torque[1] for Rotary Shouldered Drill Collar Connections (See footnotes for use of this table.)

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)
Connection			Minimum Make-up Torque ft-lb[2]									
			Bore of Drill Collar, inches									
Size, in.	Type	OD, in.	1	1 1/4	1 1/2	1 3/4	2	2 1/4	2 1/2	2 13/16	3	3 3/4
API	NC 23	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	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[3]	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 1/2		*4,606	*4,606	3,697						
API	NC 26	3 3/4		5,501	4,668	3,697						
2 7/8	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	Slim Hole											
2 7/8	Extra Hole	3 3/4		*4,089	*4,089	*4,089						
3 1/2	Double Streamline	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					
API	NC 31	4 1/8		*7,390	*7,390	*7,390	6,853					
3 1/2	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				
API	NC 35	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		
API	NC 38	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[4]	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	Full Hole	5				*10,910	*10,910	*10,910	*10,910	*10,910		
API	NC 40	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	Double Streamline	5 3/4				20,539	18,886	17,028	14,969	12,125		
		6				20,539	18,886	17,028	14,969	12,125		

(Table continued on next page.)

Table 14—Recommended Make-up Torque[1] for Rotary Shouldered Drill Collar Connections (Continued)
(See footnotes for use of this table.)

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)
Connection			Minimum Make-up Torque ft-lb[2]									
			Bore of Drill Collar, inches									
Size, in.	Type	OD, in.	1	1 1/4	1 1/2	1 3/4	2	2 1/4	2 1/2	2 13/16	3	3 3/4
	H-90[4]	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		
API	NC 44	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	22,028	19,921	17,900	
		6 1/2					27,294	25,272	22,028	19,921	17,900	
4 1/2	Extra Hole	5 3/4						*17,738	*17,738	*17,738	*17,738	
API	NC 46	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	Double Streamline	6 3/4						28,021	25,676	22,426	20,311	
4 1/2	Mod. Open											
4 1/2	H-90[4]	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[4]	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					
API	NC 50	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	Double Streamline	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[4]	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					

(Table continued on next page.)

Table 14—Recommended Make-up Torque[1] for Rotary Shouldered Drill Collar Connections (Continued) (See footnotes for use of this table.)

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)
	Connection		Minimum Make-up Torque ft-lb[2]									
			Bore of Drill Collar, inches									
Size, in.	Type	OD, in.	1	1 1/4	1 1/2	1 3/4	2	2 1/4	2 1/2	2 13/16	3	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					
API	NC 56	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	46,936					
		8		57,393	53,346	50,704	46,936					
		8 1/4		57,393	53,346	50,704	46,936					
6 5/8	H 90[4]	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					
API	NC 61	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			
API	NC70	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			
API	NC77	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[4]	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			

(Table continued on next page.)

Table 14—Recommended Make-up Torque[1] for Rotary Shouldered Drill Collar Connections (Continued) (See footnotes for use of this table.)

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)
Connection			Minimum Make-up Torque ft-lb[2]									
			Bore of Drill Collar, inches									
Size, in.	Type	OD, in.	1	1 1/4	1 1/2	1 3/4	2	2 1/4	2 1/2	2 13/16	3	3 3/4
		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[4]	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[4]	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[4]	8 3/4			*68,061	*68,061	67,257	62,845	58,131			
	(with low torque face)	9			74,235	71,361	67,257	62,845	58,131			
7 5/8	API Regular	9 1/4				*73,099	*73,099	*73,099	*73,099			
	(with low torque face)	9 1/2				*86,463	*86,463	82,457	77,289			
		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[4]	9 3/4			*91,667	*91,667	*91,667	*91,667	*91,667			
	(with low torque face)	10			*106,260	*106,260	*106,260	104,171	98,804			
		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 torque face)	11				*130,672	*130,672	*130,672	*130,672			
		11 1/4				147,616	142,430	136,846	130,871			
8 5/8	H-90[4]	10 3/4				*92,960	*92,960	*92,960	*92,960			
	(with low torque face)	11				*110,781	*110,781	*110,781	*110,781			
		11 1/4				*129,203	*129,203	*129,203	*129,203			

*Notes:

1. 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.
2. 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, API NC 26, and 2 7/8 Slim Hole connections used with 3 1/2 x 1 1/4 drill collars all have the same minimum make-up torque of 4600 ft. lb., and the BOX is the weaker member.
3. Stress-relief features are disregarded for make-up torque.

[1]Basis of calculations for recommended make-up torque assumed the use of a thread compound containing 40-60 percent by weight of finely powdered metallic zinc or 60 percent by weight of finely powdered metallic lead, with not more than 0.3 percent total active sulfur (reference the caution regarding the use of hazardous materials in Appendix G of Specification 7) applied thoroughly to all threads and shoulders and using the modified Screw Jack formula in A.8, and a unit stress of 62,500 psi in the box or pin, whichever is weaker.

[2]Normal torque range is tabulated value plus 10 percent. Higher torque values may be used under extreme conditions.

[3]Make-up torque for 2 7/8 PAC connection is based on 87,500 psi stress and other factors listed in footnote 1.

[4]Make-up torque for H-90 connection is based on 56,200 psi stress and other factors listed in footnote 1.

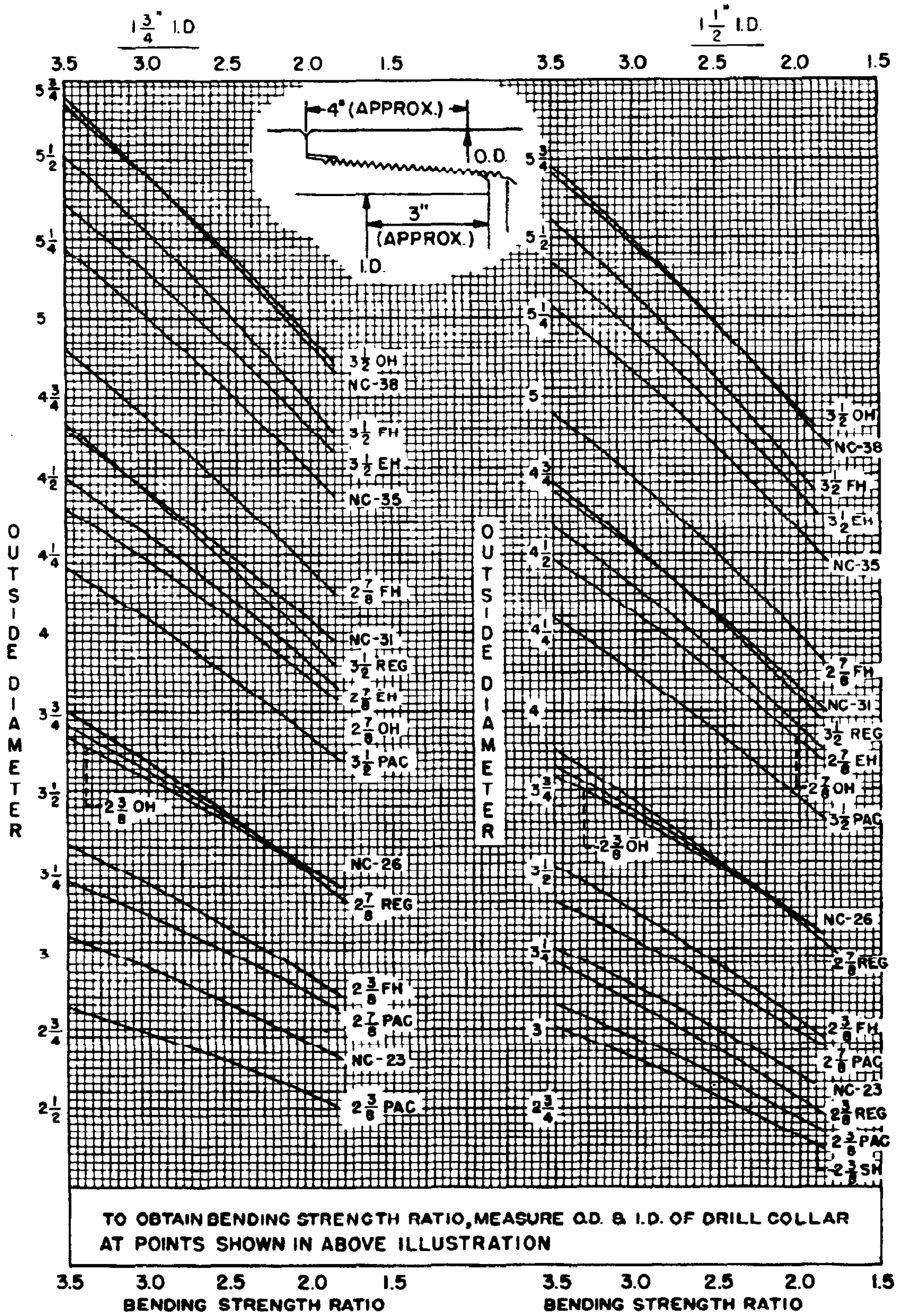


Figure 26—Drill Collar Bending Strength Ratios, 1 1/2- and 1 3/4-Inch ID

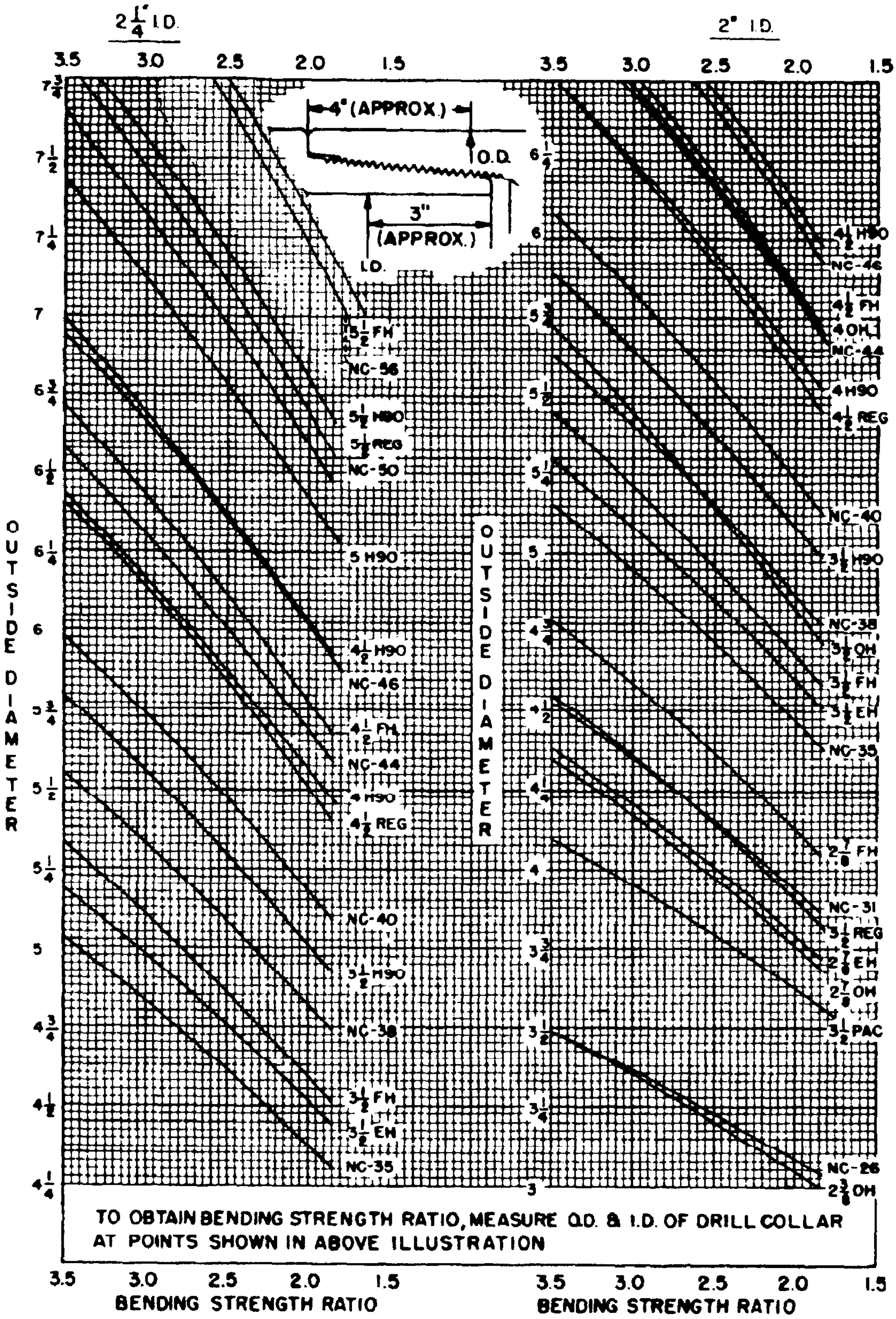


Figure 27—Drill Collar Bending Strength Ratios, 2- and $2^1/_4$-Inch ID

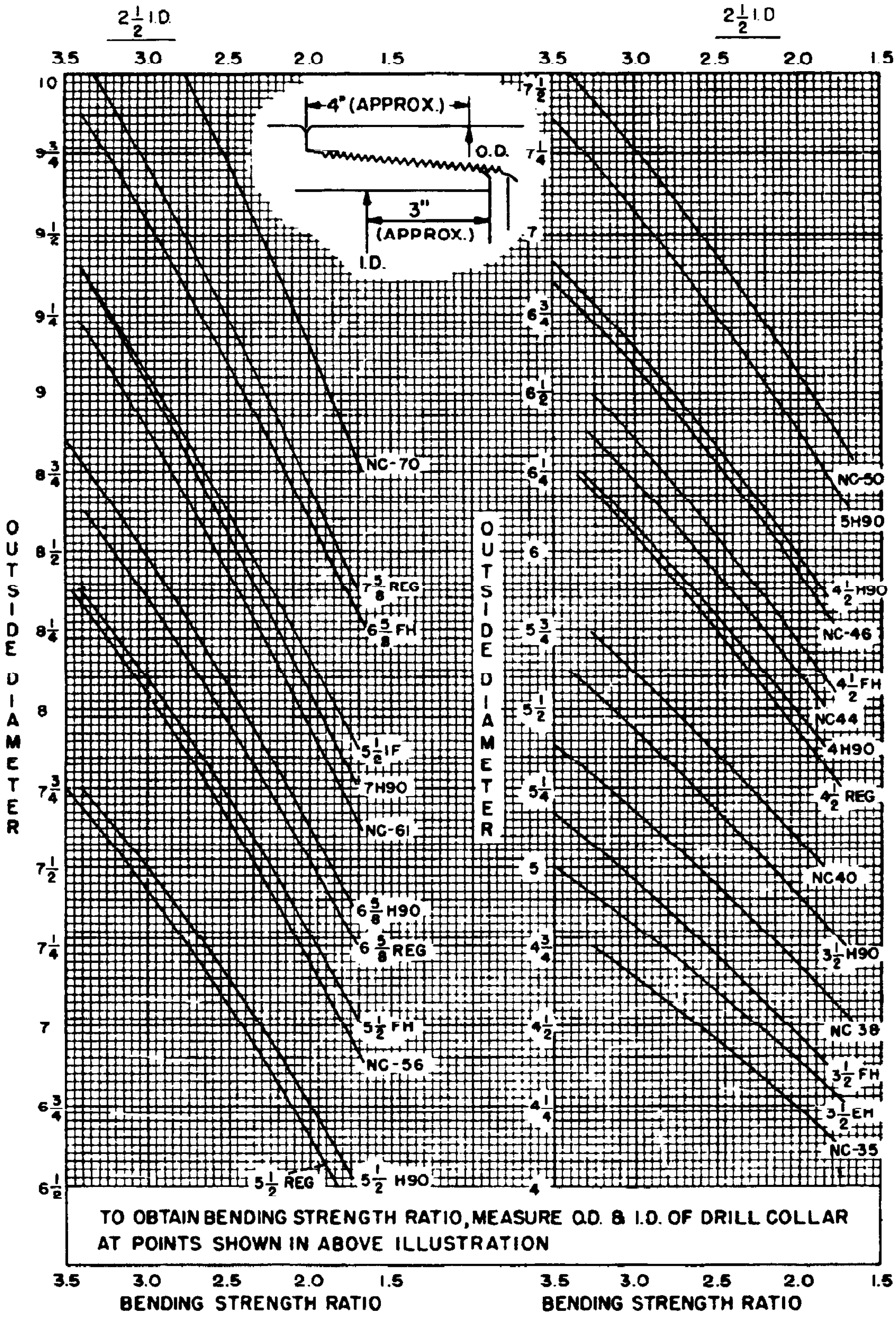


Figure 28—Drill Collar Bending Strength Ratios, 2$^1/_2$-Inch ID

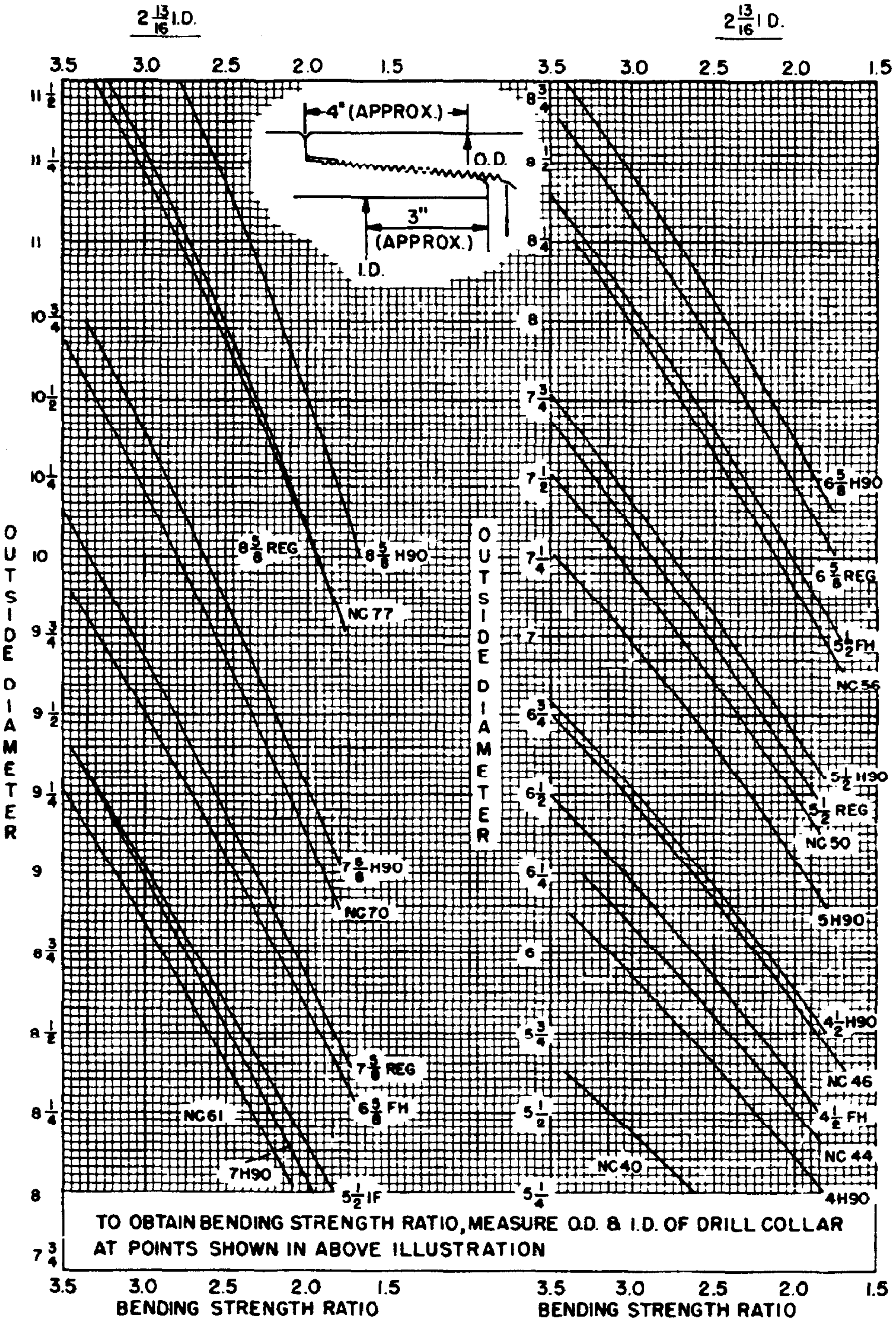


Figure 29—Drill Collar Bending Strength Ratios, $2^{13}/_{16}$-Inch ID

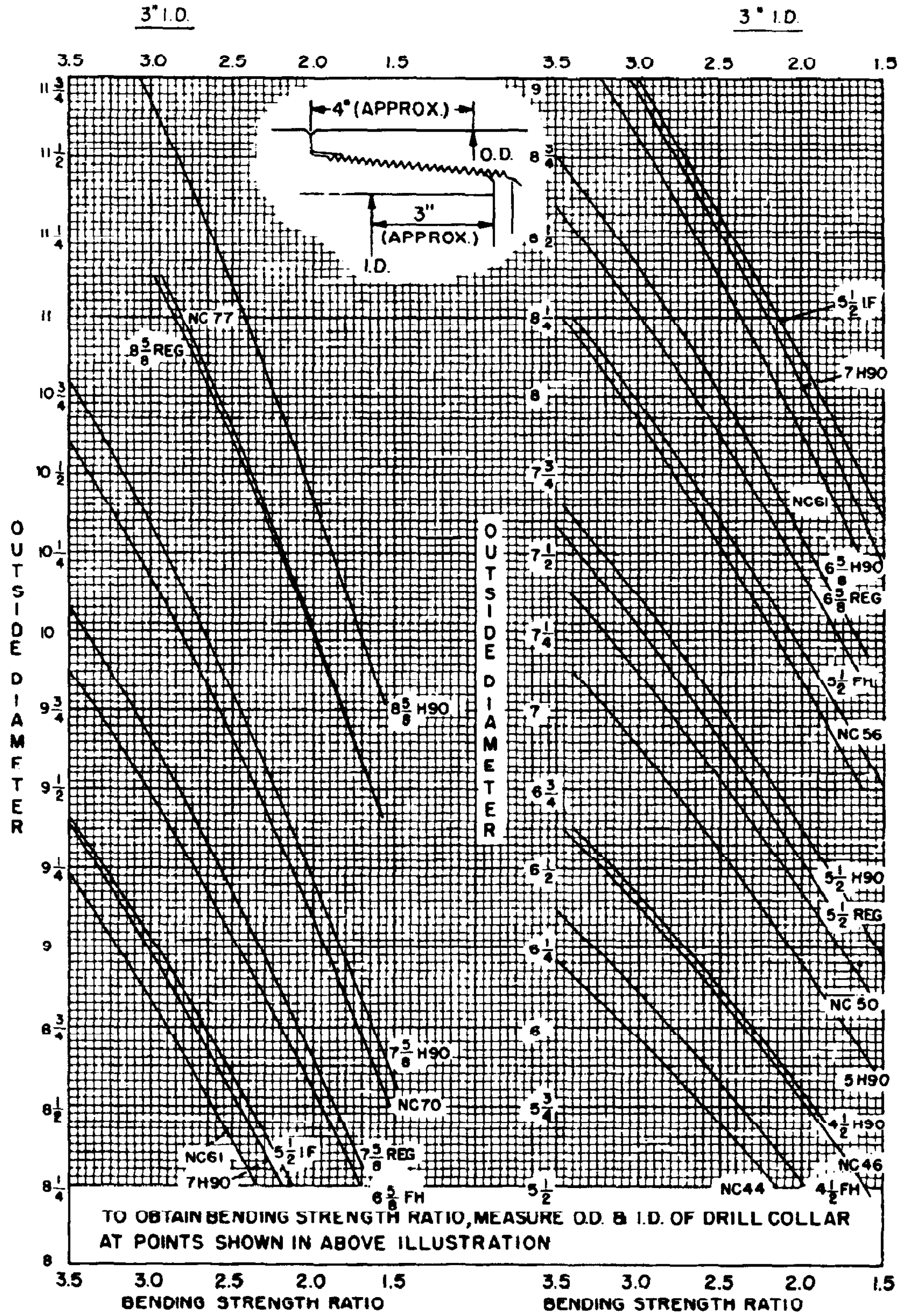


Figure 30—Drill Collar Bending Strength Ratios, 3-Inch ID

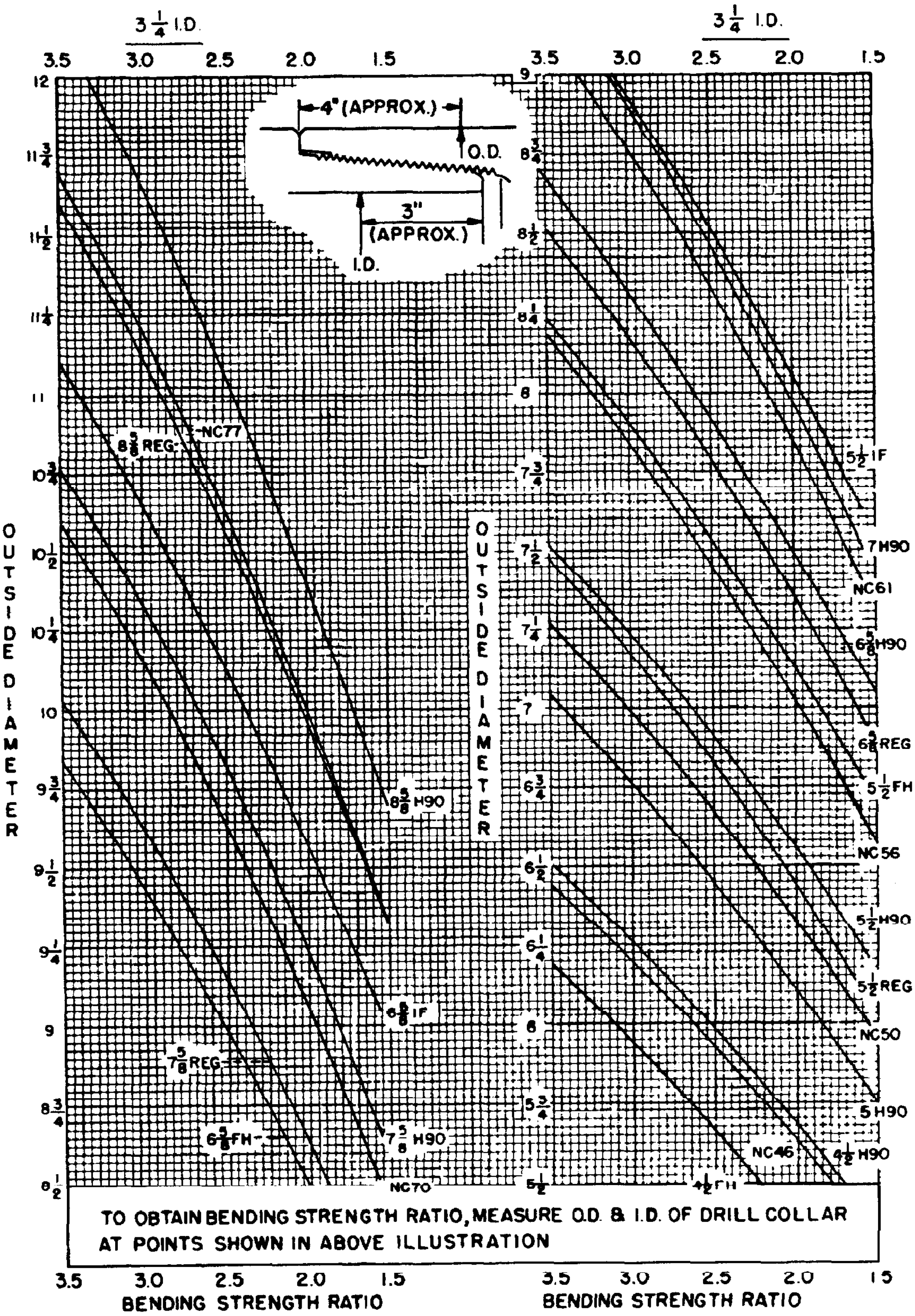


Figure 31—Drill Collar Bending Strength Ratios, $3^1/_4$-Inch ID

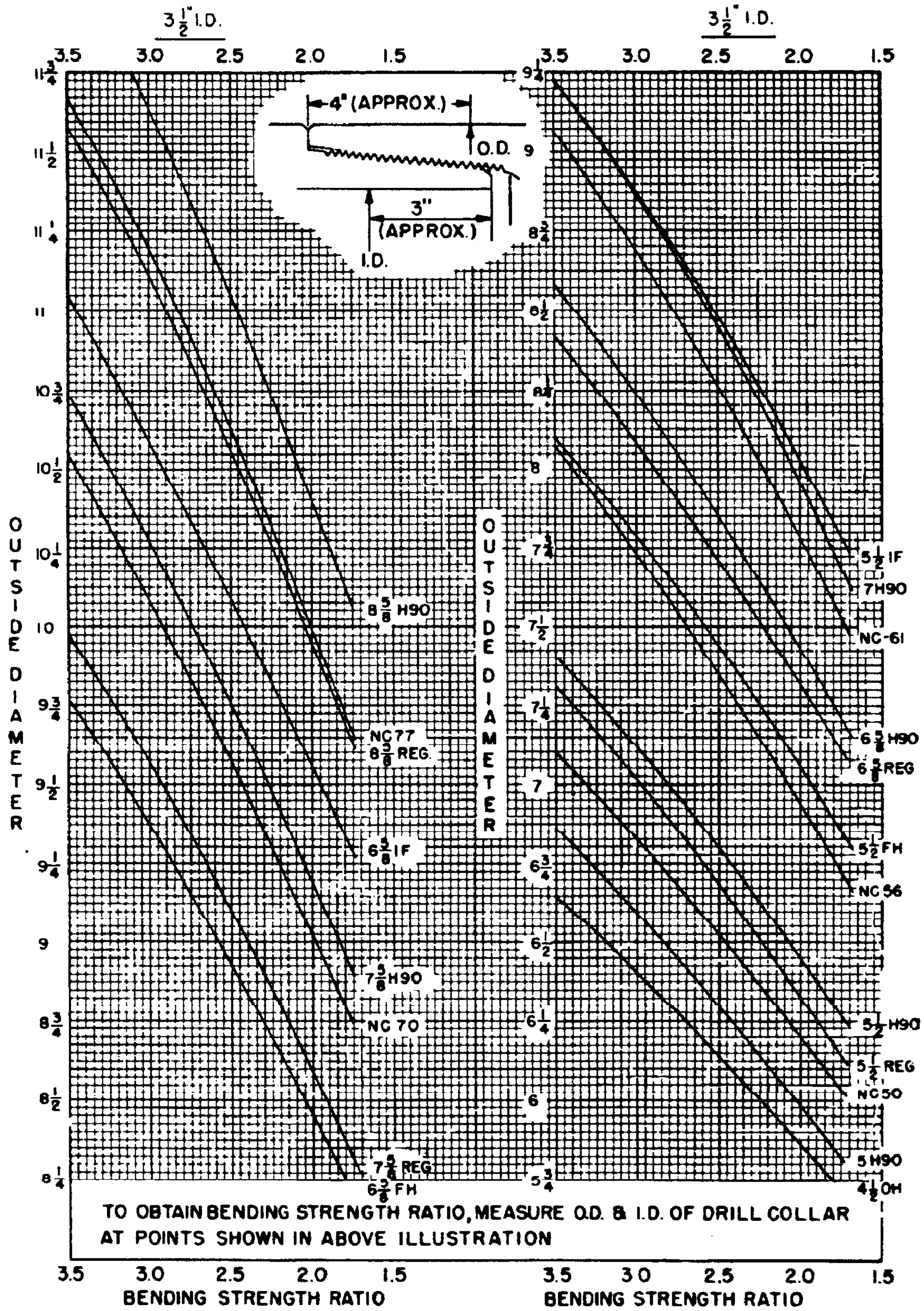


Figure 32—Drill Collar Bending Strength Ratios, $3^1/_2$-Inch ID

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

6.4.3 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.

6.4.4 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.

6.4.5 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.

6.4.6 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 33 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 16. The information in Table 16 and Figures 33 and 34 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.

Example: At the time of evaluation, the wear pattern width for a $5^1/_4$-inch hexagonal kelly is 1.00 inch.

This could mean one of the following two conditions exist:

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

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

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

6.5.1 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 17 for drive section dimensions and strengths.)

6.5.2 Reversing Ends

Usually both ends of the kelly must 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 must be used to prevent cracking and to recondition the sections where welding has been performed.

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

7 Design Calculations

7.1 DESIGN PARAMETERS

It is intended to outline a step-by-step procedure to ensure complete consideration of factors, and to simplify calculations. Derivation of formulas may be reviewed in Appendix A. The following design criteria must be established:

a. Anticipated total depth with this string.
b. Hole size.
c. Expected mud weight.
d. Desired Factor of Safety in tension and/or Margin of Over Pull.
e. Desired Factor of Safety in collapse.
f. Length of drill collars, OD, ID, and weight per foot.
g. Desired drill pipe sizes, and inspection class.

7.2 SPECIAL DESIGN PARAMETERS

If the actual wall thickness has been determined by inspection to exceed that in API tables, higher tensile, collapse, and internal pressure values may be used for drill stem design.

7.3 SUPPLEMENTAL DRILL STEM MEMBERS

Machining of the connections to API specifications and the proper heat treatment of the material shall be done on all supplemental drill stem members, such as subs, stabilizers, tools, etc.

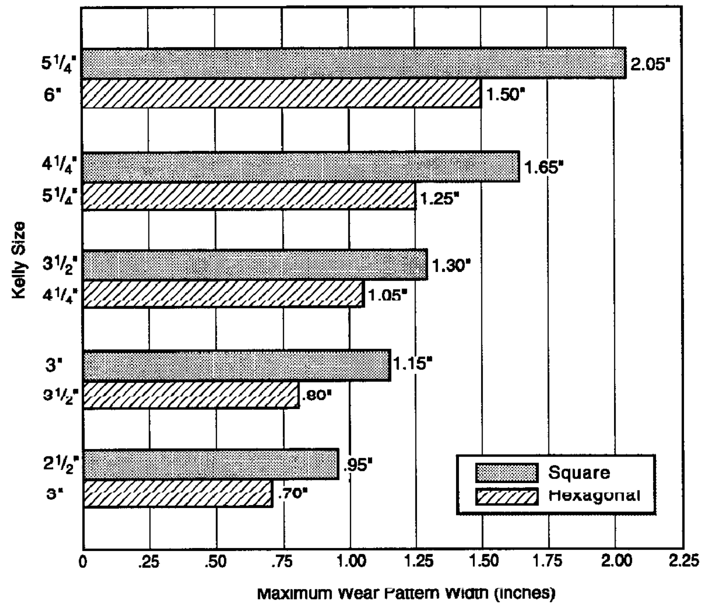
7.4 TENSION LOADING

The design of the drill string for static tension loads requires sufficient strength in the topmost joint of each size, weight, grade, and classification of drill pipe to support the submerged weight of all the drill pipe plus the submerged

Table 15—Strength of Kellys[1]

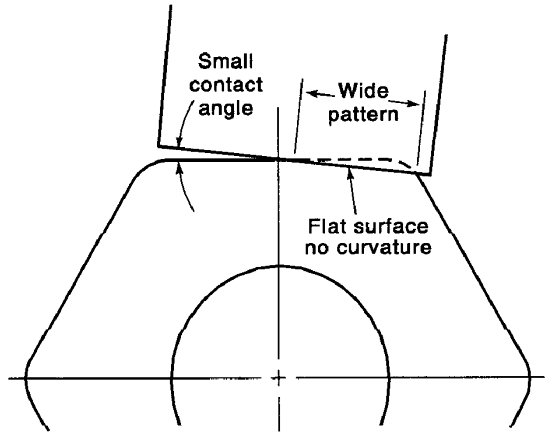
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)
		Lower Pin Connection		Minimum[2] Recommended Casing OD in.	Tensile Yield		Torsional Yield		Yield in Bending	Internal Pressure at Minimum Yield
Kelly Size and Type in.	Kelly Bore in.	Size and Style	OD in.		Lower Pin Connection[3] lb	Drive Section lb	Lower Pin Connection ft-lb	Drive Section ft-lb	Through Drive Section ft-lb	Drive Section psi
$2^1/_2$ Square	$1^1/_4$	NC26 ($2^3/_8$ 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	$2^1/_4$	NC38 ($3^1/_2$ IF)	$4^3/_4$	$6^5/_8$	724,000	725,200	22,700	28,300	34,200	22,200
$4^1/_4$ Square	$2^{13}/_{16}$	NC46 (4IF)	$6^1/_4$	$8^5/_8$	1,054,000	1,047,000	39,350	49,100	60,300	19,500
$4^1/_4$ Square	$2^{13}/_{16}$	NC50 ($4^1/_2$ 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	$3^1/_4$	51/2 FH	7	$9^5/_8$	1,609,000	1,703,400	72,950	99,400	117,000	20,600
3 Hex	$1^1/_2$	NC26 ($2^3/_8$ IF)	$3^3/_8$	$4^1/_2$	356,000	540,500	8,300	20,400	20,000	26,700
$3^1/_2$ Hex	$1^7/_8$	NC31 ($2^7/_8$ IF)	$4^1/_8$	$5^1/_2$	495,000	710,000	13,400	31,400	31,200	25,500
$4^1/_4$ Hex	$2^1/_4$	NC38 ($3^1/_2$ IF)	$4^3/_4$	$6^5/_8$	724,000	1,046,600	22,700	56,600	56,000	25,000
$5^1/_4$ Hex	3	NC46 (4 IF)	$6^1/_4$	$8^5/_8$	960,000	1,507,600	35,450	101,900	103,000	20,600
$5^1/_4$ Hex	$3^1/_4$	NC50 ($4^1/_2$ IF)	$6^3/_8$	$8^5/_8$	1,162,000	1,397,100	46,750	95,500	99,300	20,600
6 Hex	$3^1/_2$	$5^1/_2$ FH	7	$9^5/_8$	1,463,000	1,935,500	66,350	149,800	152,500	18,200

[1]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 percent of the minimum tensile yield strength.
[2]Clearance between protector rubber on kelly saver sub and casing inside diameter should also be checked.
[3]Tensile area calculated at root of thread $^3/_4$ inch from pin shoulder.



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 33—New Kelly-New Drive Assembly



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

Figure 34—New Kelly-New Drive Assembly

Table 16—Contact Angle Between Kelly and Bushing for Development of Maximum Width Wear Pattern

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)
	Hexagon				Square			
Kelly Size in.	For Minimum Clearance in.	Contact Angle Deg. Min.	For Maximum Clearance in.	Contact Angle Deg. Min.	For Minimum Clearance in.	Contact Angle Deg. Min.	For Maximum Clearance in.	Contact Angle Deg. Min.
2½	—	—	—	—	.015	6°10'	.107	16°29'
3	.015	5°41'	.060	11°22'	.015	5°39'	.107	15°5'
3½	.015	5°16'	.060	10°32'	.015	5°14'	.107	14°2'
4¼	.015	4°48'	.060	9°34'	.015	4°45'	.123	13°36'
5¼	.015	4°19'	.060	8°37'	.015	4°17'	.123	12°16'
6	.015	4°2'	.060	8°4'	—	—	—	—

Table 17 Strength of Remachined Kellys[1]

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)
Original Kelly Size and Type in.	Remachined Kelly Size and Type in.	Lower Pin Connection Kelly Bore in.	Size and Style	OD in.	Tensile Yield Lower Pin Connection[2] lb.	Drive Section lb.	Torsional Yield Lower Pin Connection ft/lb	Drive Section ft/lb	Yield in Bending Through Drive Section ft/lb
$4^1/_4$ Square	4 Square	$2^7/_8$	NC50 ($4^1/_2$ IF)	$6^3/_8$	1,344,200	834,400	55,500	36,200	47,800
$4^1/_4$ Square	4 Square	$2^7/_8$	NC46 (4 IF)	$6^1/_4$	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
$5^1/_4$ Hex	$4^{27}/_{32}$ Hex	$3^1/_4$	NC46 (4 IF)	$6^1/_4$	809,800	1,077,100	30,600	68,600	74,000
$5^1/_4$ Hex	5 Hex	$3^1/_4$	NC46 (4 IF)	$6^1/_4$	809,800	1,196,800	30,600	78,500	83,300
$5^1/_4$ Hex	5 Hex	$3^1/_2$	NC50 ($4^1/_2$ IF)	$6^3/_8$	999,900	1,077,600	40,800	71,100	78,400
6 Hex	$5^3/_4$ Hex	4	$5^1/_2$ FH	7	1,189,500	1,443,400	51,300	109,100	119,900
6 Hex	$5^3/_4$ Hex	$4^1/_8$	$5^1/_2$ IF	$7^3/_8$	1,669,200	1,371,500	80,400	103,800	116,200

[1]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 percent of the minimum tensile yield strength.
[2]Tensile area calculated at root of thread $^3/_4$ inch from pin shoulder.
Note: Kelly bushings are normally available for kellys in above table.

weight of the collars, stabilizer, and bit. This load may be calculated as shown in Equation 1. The bit and stabilizer weights are either neglected or included with the drill collar weight.

$$P = [(L_{dp} \times W_{dp}) + (L_c \times W_c)] K_b \tag{1}$$

where

P = submerged load hanging below this section of drill pipe, lb.,
L_{dp} = length of drill pipe, ft.,
L_c = length of drill collars, ft.,
W_{dp} = weight per foot of drill pipe assembly in air,
W_c = weight per foot of drill collars in air,
K_b = buoyancy factor—see Table 11.

Any body floating or immersed in a liquid is acted on by a buoyant force equal to the weight of the liquid displaced. This force tends to reduce the effective weight of the drill string and can become of appreciable magnitude in the case of the heavier muds. For example, from Table 11, a one-pound weight submerged in a 14 lb./gal. mud would have an apparent weight of 0.786 lb.

Tension load data is given in Tables 2, 4, 6, and 8 for the various sizes, grades and inspection classes of drill pipe.

It is important to note that the tension strength values shown in the tables are theoretical values based on minimum areas, wall thickness and yield strengths. The yield strength as defined in API specifications is not the specific point at which permanent deformation of the material begins, but the stress at which a certain total deformation has occurred. This deformation includes all of the elastic deformation as well as some plastic (permanent) deformation. If the pipe is loaded to the extent shown in the tables it is likely that some permanent stretch will occur and difficulty may be experienced in keeping the pipe straight. To prevent this condition a design factor of approximately 90 percent of the tabulated tension value from the table is sometimes used; however, a better practice is to request a specific factor for the particular grade of pipe involved from the drill pipe supplier.

$$P_a = P_t \times 0.9 \tag{2}$$

where

P_a = maximum allowable design load in tension, lb.,
P_t = theoretical tension load from table, lb.,
0.9 = a constant relating proportional limit to yield strength.

The difference between the calculated load P and the maximum allowable tension load represents the Margin of Over Pull (MOP).

$$MOP = P_a - P \tag{3}$$

The same values expressed as a ratio may be called the Safety Factor (SF).

$$SF = \frac{P_a}{P} \tag{4}$$

The selection of the proper safety factor and/or margin of over pull is of critical importance and should be approached with caution. Failure to provide an adequate safety factor can result in loss or damage to the drill pipe while an overly conservative choice will result in an unnecessarily heavy and more expensive drill string. The designer should consider the overall drilling conditions in the area, particularly hole drag and the likelihood of becoming stuck. The designer must also consider the degree of risk which is acceptable for the particular well for which the drill string is being designed. Frequently, the safety factor also includes an allowance for slip crushing and for the dynamic loading, which results from accelerations and decelerations during hoisting.

Slip crushing is not a problem if slips and master bushings are maintained. Inspection class also grades the pipe with regard to slip crushing.

Normally the designer will desire to determine the maximum length of a specific size, grade and inspection class of drill pipe which can be used to drill a certain well. By combining Equation 1 and either Equation 2 or 3, the following equations result:

$$\frac{P_t \times 0.9}{SF \times W_{dp} \times K_b} - \frac{W_c L_c}{W_{dp}} = L_{dp} \tag{5}$$

and/or

$$\frac{P_t \times 0.9 - MOP}{W_{dp} \times K_b} - \frac{W_c L_c}{W_{dp}} = L_{dp} \tag{6}$$

If the string is to be a tapered string, i.e., to consist of more than one size, grade or inspection class of drill pipe, the pipe having the lowest load capacity should be placed just above the drill collars and the maximum length is calculated as shown previously. The next stronger pipe is placed next in the string and the $W\,L$ term in Equation 5 or 6 is replaced by a term representing the weight in air of the drill collars plus the drill pipe assembly in the lower string. The maximum length of the next stronger pipe may then be calculated. An example calculation using the above formulas is included in 7.8.

7.5 COLLAPSE DUE TO EXTERNAL FLUID 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 3, 5, 7, and 9. The tabulated values should be divided by a suitable factor of safety to establish the allowable collapse pressure.

$$\frac{P_p}{SF} = P_{ac} \tag{7}$$

where

P_p = theoretical collapse pressure from tables, psi,
SF = safety factor,
P_{ac} = allowable collapse pressure, 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 following equation:

$$P_c = \frac{L\,W_g}{19.251} \tag{8}$$

or

$$P_c = \frac{L\,W_f}{144} \tag{9}$$

where

P_c = net collapse pressure, psi,
L = the depth at which P_c acts, ft.,
W_g = weight of drilling fluid, lb/gal,
W_f = weight of drilling fluid, 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:

$$P_c = \frac{L\,W_g - (L - Y)W_g'}{19.251} \tag{10}$$

or

$$P_c = \frac{L\,W_f - (L - Y)W_f'}{144} \tag{11}$$

where

Y = depth to fluid inside drill pipe, ft.,
W_g' = weight of drilling fluid inside pipe, lb/gal,
W_f' = weight of drilling fluid inside pipe, lb/cu ft.

7.6 INTERNAL PRESSURE

Occasionally the drill pipe may also be subjected to a net internal pressure. Tables 3, 5, 7, and 9 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.

7.7 TORSIONAL STRENGTH

The torsional strength of drill pipe becomes critical when drilling deviated holes, deep holes, reaming, or when the pipe is stuck. This is discussed under Sections 8 and 12. Calculated values of torsional strength for various sizes, grades, and inspection classes of drill pipe are provided in Tables 2, 4, 6, and 8. The basis for these calculations is shown in Appendix A. The actual torque applied to the pipe during drilling is difficult to measure, but may be approximated by the following equation:

$$T = \frac{HP \times 5{,}250}{RPM} \tag{12}$$

where

T = torque delivered to drill pipe, ft-lbs,

HP = horse power used to produce rotation of pipe,

RPM = revolutions per minute.

Note: The torque applied to the drill string should not exceed the actual tool joint make-up torque. The recommended tool joint make-up torque is shown in Table 10.

7.8 EXAMPLE CALCULATION OF A TYPICAL DRILL STRING DESIGN—BASED ON MARGIN OF OVERPULL

Design parameters are as follows:

a. Depth—12,700 feet.

b. Hole size—$7^7/_8$ inches.

c. Mud weight—10 lb/gal.

d. Margin of overpull (MOP)—50,000 lb. (assumed for this calculation).

e. Desired safety factor in collapse $1^1/_8$ (assumed for this calculation).

f. Drill collar data:

1. Length 630 feet.
2. OD—$6^1/_4$ inches.
3. ID—$2^1/_4$ inches.
4. Weight per foot—90 lb.

If the length of drill collars is not known, the following formula may be used:

$$L_c = \frac{Bit_{wm}}{\cos\alpha \times NP \times k_b \times W_c}$$

where

L_c = length of drill collars, feet,

Bit_{wm} = maximum weight on bit, lb,

α = hole angle from vertical, 3 degrees,

NP = neutral point design factor determines neutral point position e.g., .85 means the neutral point will be 85 percent of the drill collar string length measured from the bottom (.85 assumed for this calculation),

K_b = buoyancy factor, see Table 11,

W_c = weight per foot of drill collars in air, lb,

$$L_c = \frac{40{,}000}{.998 \times .85 \times .847 \times 90},$$

= 618 feet, closest length based on 30 foot collars,

= 630 feet or 21 drill collars.

g. Drill string size: weight and grade—$4^1/_2$ in. x 16.60 lb/ft x Grade E75, with $4^1/_2$ in., NC46 tool joints, $6^1/_4$ in. OD x $3^1/_4$ in. ID, Inspection Class 2.

From Equation 5:

$$L_{dp1} = \frac{(P_{t1} \times .9) - MOP}{W_{dp1} \times K_b} - \frac{W_c \times L_c}{W_{dp1}}$$

$$= \frac{(225{,}771 \times .9) - 50{,}000}{18.37 \times .847} - \frac{90 \times 630}{18.37}$$

$$= 9846 - 3087 = 6759 \text{ feet}$$

It is apparent that drill pipe of a higher strength will be required to reach 12,700 feet. Add $4^1/_2$ in. x 16.60 lb/ft Grade X-95, with $4^1/_2$ in. X.H. tool joints, $6^1/_4$ in. OD x 3 in. ID (18.88 lb/ft) Inspection Class Premium.

Air weight of Number 1 drill pipe and drill collars:

Total weight $= (L_{dp1} \times W_{dp1}) + (L_c \times W_c)$

$= (6759 \times 18.37) + (630 \times 90)$

$= 124{,}163 + 56{,}700 = 180{,}863$ lb.

From Equation 5:

$$L_{dp2} = \frac{(P_{t2} \times .9) - MOP}{W_{dp2} \times K_b} - \frac{(L_{dp1} \times W_{dp1}) + (L_c \times W_c)}{W_{dp2}}$$

$$= \frac{(329{,}542 \times .9) - 50{,}000}{18.88 \times .847} - \frac{180{,}863}{18.88}$$

$$= 15{,}420 - 9580 = 5840 \text{ feet}$$

This is more drill pipe than required to reach 12,700 feet, so final drill string will consist of the following:

Item	Length (feet)	Weight in Air (pounds)	Weight in 10 lb/gal Mud (pounds)
Drill Collars $6^1/_4$" O.D. x $2^1/_4$" I.D.	630	56,700	48,025
No. 1 Drill Pipe $4^1/_2$" x 16.60 lb, Grade E75, Class 2	6759	124,163	105,166
No. 2 Drill Pipe $4^1/_2$" x 16.60 lb, Grade X-95, Premium Class	5311	100,272	84,930
	12,700	281,135	238,121

Torsional Yield of $4^1/_2$" x 16.60 lb x Grade E75 x Inspection Class 2 = 20,902 ft-lb.

Collapse Pressure of $4^1/_2$" x 16.60 lb x Grade E75 x Inspection Class 2 = 5951 psi.

Collapse pressure of $4^1/_2$" x 16.60 lb x Grade X-95 x Premium Inspection Class = 8868 psi.

From Equation 8:

Pressure at bottom of drill pipe: $P_c = \dfrac{LW_g}{19.251}$,

L = 12,070 feet, W_g = 10 lb/gal,

$$P_c = \frac{12{,}070 \times 10}{19.251} = 6270 \text{ psi}$$

Therefore, this drill pipe has a lower collapse pressure than may be encountered in drilling to 12,700 feet. Precautions should be taken to prevent damage to the drill pipe when running the string dry below 10,183 feet. This is determined by solving Equation 8 for maximum length of drill pipe, and dividing by the safety factor in collapse of $1^1/_8$:

$$L_{max} = \frac{P_c \times 19.251}{W_g} \div 1.125$$

$$= \frac{5951 \times 19.251}{10} \div 1.125$$

$$= 11{,}456 \div 1.125 = 10{,}183 \text{ feet}$$

7.9 DRILL PIPE BENDING RESULTING FROM TONGING OPERATIONS

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.

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.

Many factors govern this height limitation. Several of these which should be taken into most serious consideration are:

a. The angle of separation between the make-up and break-out tongs, illustrated by Case I and Case II, Figure 35. Case I indicates tongs at 90 degrees and Case II indicates tongs at 180 degrees.
b. The minimum yield strength of the pipe.
c. The length of the tong handle.
d. The maximum recommended make-up torque.

$$H_{max} = \frac{.053\ Y_m L_T\ (I/C)}{T} \text{ (Case I)} \quad (13)$$

$$H_{max} = \frac{.038\ Y_m L_T\ (I/C)}{T} \text{ (Case II)} \quad (14)$$

where

H_{max} = height of tool joint shoulder above slips, ft.,
Y_m = minimum tensile yield stress of pipe, psi,
L_T = tong arm length, ft.,
P = line pull (load), lbs,
T = make-up torque applied to tool joint ($P_x L_T$), ft-lb,
I/C = section modulus of pipe—in³. (see Table 18).

Constants 0.053 and 0.038 include a factor of 0.9 to reduce Y_m to proportional limit (see 7.3).

Sample calculation:

Assume: $4^1/_2$-in., 16.60 lb/ft, Grade E drill pipe, with $4^1/_2$-in. X.H. $6^1/_4$-in. OD, $3^1/_4$-in. ID tool joints.

Tong arm $3^1/_2$-ft.

Tongs at 90 (Case I).

Using Equation 13:

$$H_{max} = \frac{.053\ (Y_m)\ (I/C)\ (L_T)}{T},$$

Y_m = 75,000 psi (for Grade E),

I/C = 4.27 in³ (Table 18),

L_T = 3.5 ft.,

T = 16,997 ft-lb (from Table 10),

$$H_{max} = \frac{.053(75{,}000)(4.27)(3.5)}{16{,}997} = 3.4 \text{ ft.}$$

Table 18—Section Modulus Values

(1)	(2)	(3)
Pipe OD in.	Pipe Weight Nominal lbs/ft	I/C 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
	21.90	7.03
	24.70	7.84
$6^5/_8$	25.20	9.79

8 Limitations Related to Hole Deviation

8.1 FATIGUE DAMAGE

Most drill pipe failures are a result of fatigue (see 11.2). Drill pipe will suffer fatigue when it is rotated in a section of hole in which there is a change of hole angle and/or direction, commonly called a dogleg. The amount of fatigue damage which results depends on the following:

8.1.1 Tensile Load in the Pipe at the Dogleg

Following is an example calculation:

a. 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.

b. Solution:

Tensile load in the pipe at the dogleg:

$$[(11,000 - 3,000)\ 17.8 + 100 \times 147]\ 0.771 = 121,124 \text{ lb}$$

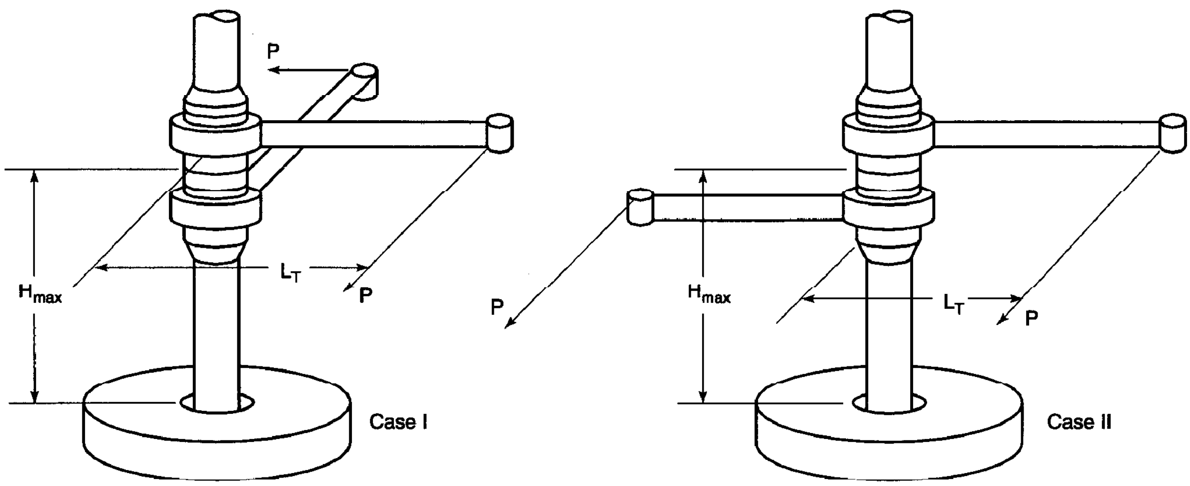


Figure 35—Maximum Height of Tool Joint Above Slips to Prevent Bending During Tonging

8.1.2 The Severity of the Dogleg

The number of cycles experienced in the dogleg, as well as the mechanical dimensions and properties of the pipe itself.

Because tension in the pipe is critical, a shallow dogleg in a deep hole often becomes a source of difficulty. Rotating off bottom is not a good practice since additional tensile load results from the suspended drill collars. Lubinski[1] and Nicholson[2] have published methods of calculating forces on tool joints and conditions necessary for fatigue damage to occur. Referring to Figures 36 and 37, note that it is necessary to remain to the left of fatigue curves to reduce fatigue damage. Programs to plan and drill wells to minimize fatigue have been reported by Schenck[3] and Wilson[4]. Such programs are necessary to reduce fatigue damage.

The curves on Figures 36, 37, and 38 (also Figures 41, 42, and 43) are for Range 2 drill pipe, i.e. for joint lengths of 30 feet. This length has an effect on the curves. Information is available on fatigue of Range 3 (45 feet) drill pipe.[14] The curves on Figures 36, 37, and 38 are independent of tool joint OD; however, the portion of the curve for which there is pipe-to-hole contact between tool joints (dashed lines on Figures 36 and 38) becomes longer when tool joint OD becomes smaller, and conversely.

The advent of electronic pocket calculators makes it easy to use the following equations instead of the curves in Figures 36 and 37.[1,4]

$$c = \frac{432{,}000}{\pi} \frac{\delta_b}{ED} \frac{\tanh KL}{KL} \tag{15}$$

$$K = \sqrt{\frac{T}{EI}} \tag{16}$$

where

c = maximum permissible dogleg severity (hole curvature), degrees per 100 feet,
E = Young's modulus, psi,
= 30 x 106 psi, for steel,
= 10.5 x 106 psi, for aluminum,
D = drill pipe OD, inches,
L = half the distance between tool joints, inches,
= 180 inches, for Range 2.

Note: Equation 15 does not hold true for Range 3.[14]

T = buoyant weight (including tool joints) suspended below the dogleg, pounds.
δ_b = maximum permissible bending stress, psi,
I = drill pipe moment of inertia with respect to its diameter, in.^{4}, calculated by Equation 17.

$$I = \frac{\pi}{64}\left(D^4 - d^4\right), \tag{17}$$

where

D = drill pipe OD, inches,
d = drill pipe ID, inches.

The maximum permissible bending stress, δ_b, is calculated from the buoyant tensile stress, δ_t (psi), in the dogleg with Equations 19 and 20 below. δ_t is calculated with Equation 18:

$$\delta_t = \frac{T}{A} \tag{18}$$

where

A = cross sectional area of drill pipe body, square inches.

For Grade E:[14]

$$\delta_b = 19{,}500 - \frac{10}{67}\delta_t - \frac{0.6}{(670)^2}(\delta_t - 33{,}500)^2 \tag{19}$$

Equation 19 holds true for values of δ_t up to 67,000 psi.

For Grade S-135:[2]

$$\sigma_b = 2000\left(1 - \frac{\sigma_t}{145{,}000}\right) \tag{20}$$

Equation 20 holds true for values of up to 133,400 psi.

The following equation may be used instead of Figure 38:

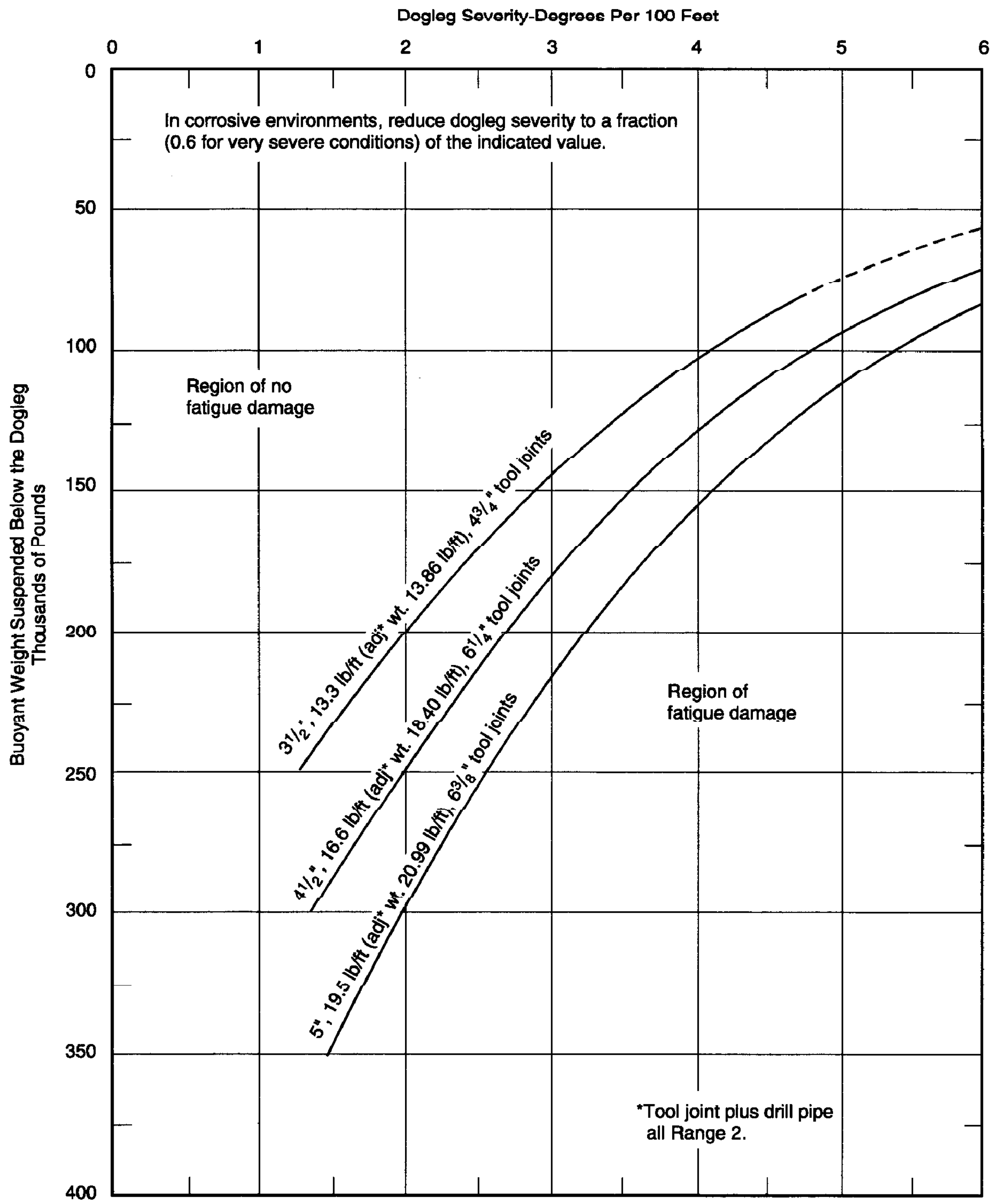
$$c = \frac{108{,}000}{\pi L} \frac{F}{T} \tag{21}$$

in which F is the lateral force on tool joint (1000, 2000, or 3000 pounds in Figure 38), and the meaning of the other symbols is the same as previously.

8.2 REMEDIAL ACTION TO REDUCE FATIGUE

If doglegs of sufficient magnitude are present or suspected, it is good practice to string ream the dogleg area. This reduces the severity of the hole angle change. With reference to Figure 40, the fatigue life of drill pipe will be decreased considerably when it is used in a corrosive drilling fluid. 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. Refer to 10.1.4 for description of a corrosion monitoring system.

Several methods are available for monitoring and controlling the corrosivity of drilling fluids. The most commonly used monitoring technique is the use of a corrosion ring inserted in the drill stem. For a description of this technique see API Recommended Practice 13B-1, *Recommended Practice Standard Procedure for Field Testing Water-Based Drilling Fluids*.



Note:
Dashed curve corresponds to condition when drill pipe contacts the hole between tool joints, and then the permissible dogleg severity is greater than indicated.

Figure 36—Dogleg Severity Limits for Fatigue of Grade E75 Drill Pipe

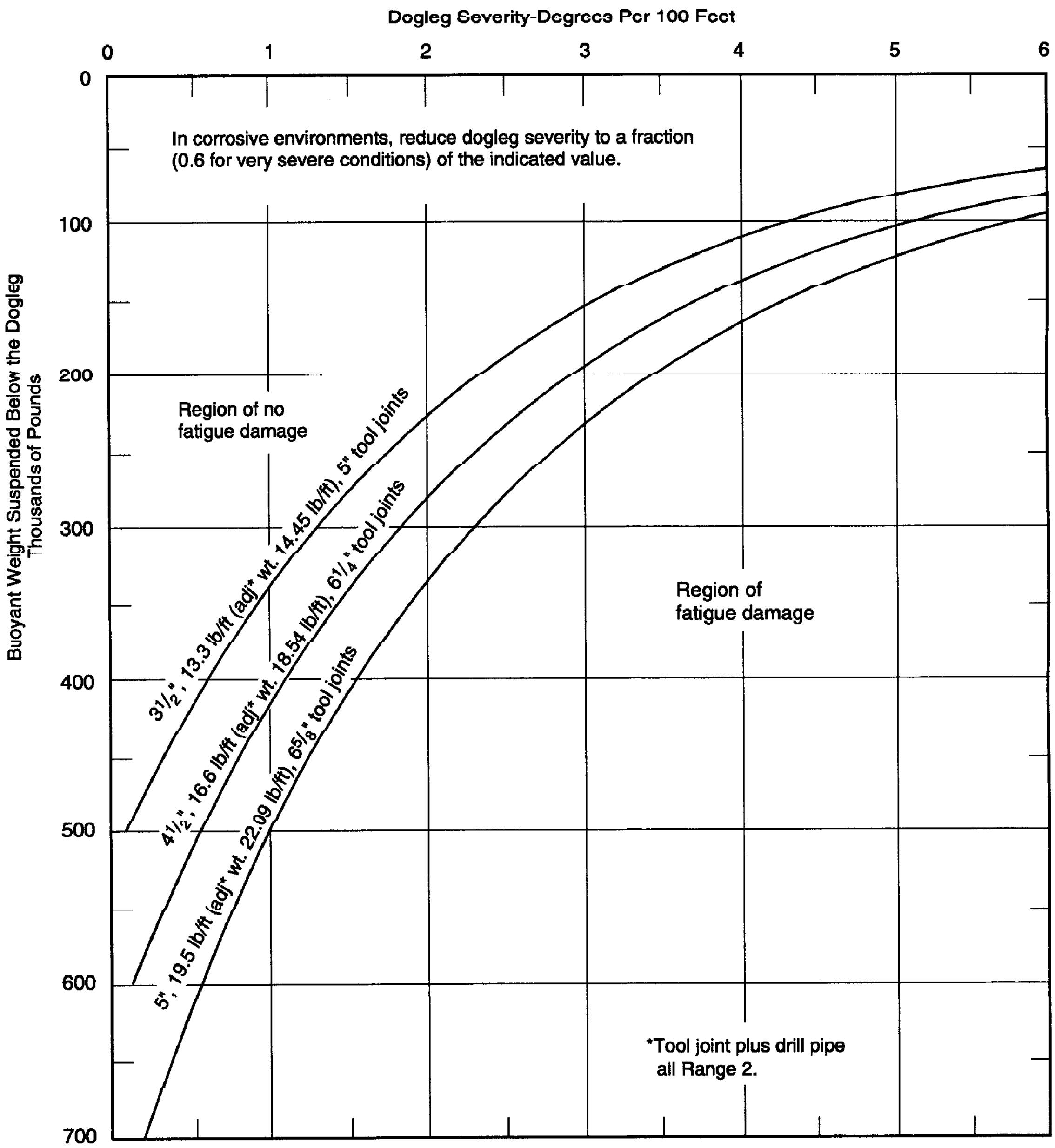
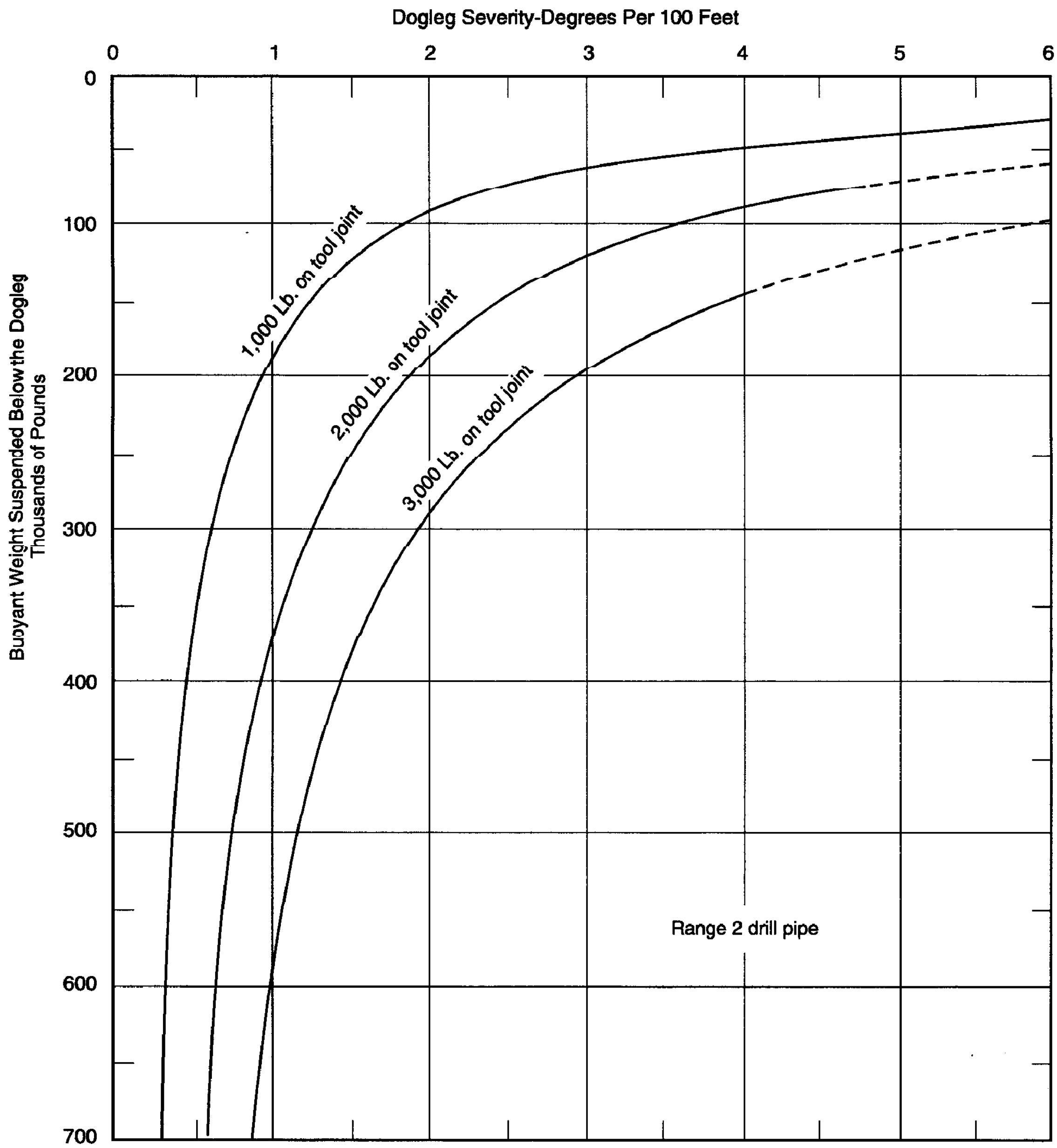


Figure 37—Dogleg Severity Limits for Fatigue of S-135 Drill Pipe



Note:
Dashed curves correspond to condition when drill pipe may contact the hole between tool joints, and then the permissible dogleg severity may be greater than indicated.

Figure 38—Lateral Force on Tool Joint

8.3 ESTIMATION OF CUMULATIVE FATIGUE DAMAGE

Hansford and Lubinski[5] have developed a method for estimating the cumulative fatigue damage to joints of pipe which have been rotated through severe doglegs (see Figures 39 and 40). While insufficient field checks of the results of this method have been made to verify its reliability, it is available as a simple analytic device to use as a guide in the identification of suspect joints. A correction formula to use for other penetration rates and rotary speeds is as follows:

$$\% \text{ Life Expended} = \% \text{ Life Expended from Figure 39 or 40} \times \frac{\text{Actual RPM}}{100 \text{ RPM}} \times \frac{10 \text{ ft/hr}}{\text{Actual ft/hr}}$$

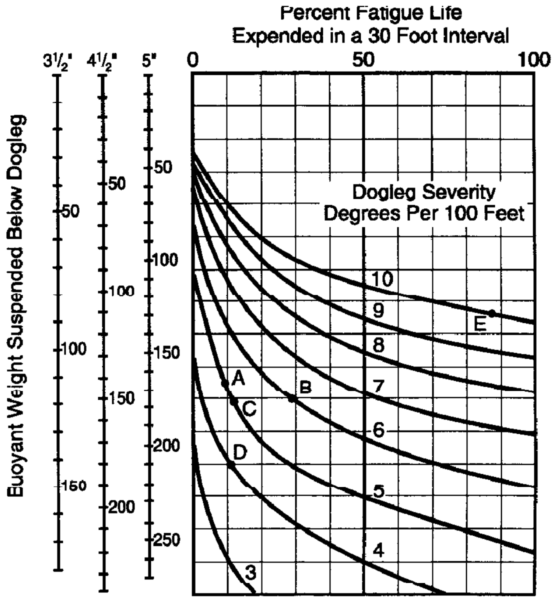
8.4 IDENTIFICATION OF FATIGUED JOINTS

As mentioned, insufficient data is available to verify the results of the method explained in 8.3. However, it is the only method presently available for estimating cumulative fatigue damage and should be used if it is possible to identify and classify fatigued joints. The difficulty lies in identifying and recording each separate joint fatigue history. Joints which have been calculated to have more than 100 percent 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.

8.5 WEAR OF TOOL JOINTS AND DRILL PIPE

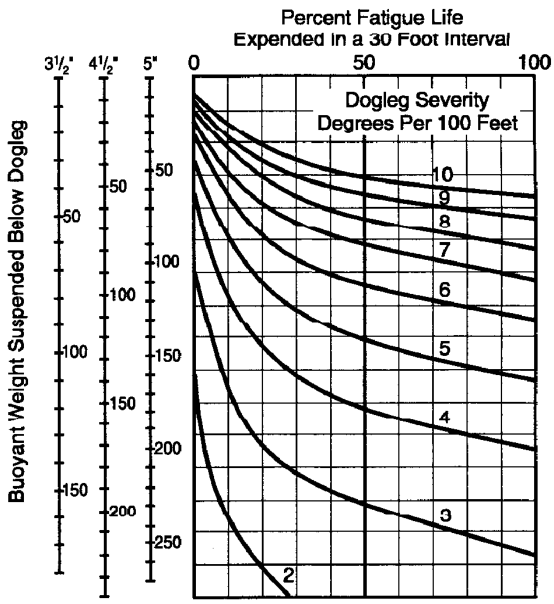
When drill pipe in a dogleg is in tension it is pulled to the inside of the bend with substantial force. The lateral force will increase the wear of the pipe and tool joints. When abrasion is a problem it is desirable to limit the amount of lateral force to less than about 2000 lb on the tool joints by controlling the rate of change of hole angle. Values either smaller or greater than 2000 lb might be in order, depending on formation at the dogleg. Figure 38 shows curves for 1000, 2000, or 3000 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, Range 2. Figure 44 is for 5-inch, 19.5 lb per foot, Range 3 drill pipe.

8.5.1 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



For:
Drill pipe, 3½", 4½", and 5" Grade E steel; rotary speed, 100 rpm; drilling rate, 10 feet/hour.

Figure 39—Fatigue Damage in Gradual Doglegs (Noncorrosive Environment)



For:
Drill pipe, 3½", 4½", and 5" Grade E steel; rotary speed, 100 rpm; drilling rate, 10 feet/hour.

Figure 40—Fatigue Damage in Gradual Doglegs (In Extremely Corrosive Environment)

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.

8.5.2 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.

8.5.3 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.

8.5.4 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.

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 170.000 lb. and if dogleg severity (hole curvature) is 10.1 degrees per 100 feet, then the force on tool joint is 6000 lb, and the force on drill pipe body is 3000 lb.

8.6 HEAT CHECKING OF TOOL JOINTS

Tool joints which are rotated under high lateral force against the wall of the hole may be damaged as a result of friction heat checking. The heat generated at the surface of the tool joint by friction with the wall of the hole when under high radial thrust loads may raise the temperature of the tool joint steel above its critical temperature. Metallurgical examination of such joints has indicated affected zones with varying hardness as much as $^3/_{16}$-in. below OD surface. If the radial thrust load is sufficiently high, surface heat checking can occur in the presence of drilling mud alternately being heated and quenched as it rotates. This action produces numerous irregular heat check cracks often accompanied by longer axial cracks sometimes extending through the full section of the joint and washouts may occur in these splits or windows. (See Lubinski, "Maximum Permissible Dog Legs in Rotary Boreholes," *Journal of Petroleum Technology*, 1961.)[1] Maintaining hole angle control so that 2000 lb lateral force is not exceeded will minimize or eliminate heat checking of tool joints.

9 Limitations Related To Floating Vessels

9.1 All possible steps should be taken to avoid subjecting drill pipe to fatigue; i.e., to cyclic stresses due to rotation of the drill string under bending and tension. Two major factors which are specific to drilling from a floater that contribute to fatigue of drill pipe are as follows:

9.1.1 The rotary table is not centered at all times exactly above the subsea borehole.

9.1.2 The derrick is not always vertical but follows the roll and pitch motions of the floater.

9.2 This text pertains to prevention of fatigue due to factor b, above. When the derrick is inclined during a part of the roll or pitch motion, the upper extremity of the drill string is not vertical while the drill pipe at some distance below the rotary table remains vertical. Thus the drill string is bent. As drill pipe is much less rigid than the kelly, most of the bending occurs in the first length of drill pipe below the kelly. This subject is studied in a paper titled, *The Effect of Drilling Vessel Pitch or Roll on Kelly and Drill Pipe Fatigue*, by John E. Hansford and Arthur Lubinski.[6]

9.3 Based on the Hansford and Lubinski paper[6], the following practices are recommended to minimize bending and, therefore, fatigue of the first joint of drill pipe, due to roll and/or pitch of a floater.

9.3.1 Multiplane bushings should not be used. Either a gamboled kelly bushing, or a one-plane roller bushing is preferable.

9.3.2 An extended length kelly should be used to relieve the severe bending of the limber drill pipe through less severe bending of the rigid kelly extension. This extension may be accomplished by any of the following means:

a. For Range 2 drill pipe, use a 54-foot kelly which is ordinarily used with Range 3 pipe, rather than the usual 40-foot kelly.
b. Use a specially made kelly at least 8 feet longer than the standard length.
c. Use at least 8 feet of kelly saver subs between the kelly and drill pipe.

9.3.3 If b, above, is not implemented, avoid rotating off bottom with the kelly more than half way up for long periods of time if the maximum angular vessel motion is more than 5 degrees single amplitude. In this text, long periods of time are:

a. More than 30 minutes for large hookloads.
b. More than 2 hours for light hookloads.

9.3.4 If conditions prevent implementing b or c, above, the first joint of drill pipe below the kelly should be removed from the string at the first opportunity and discarded.

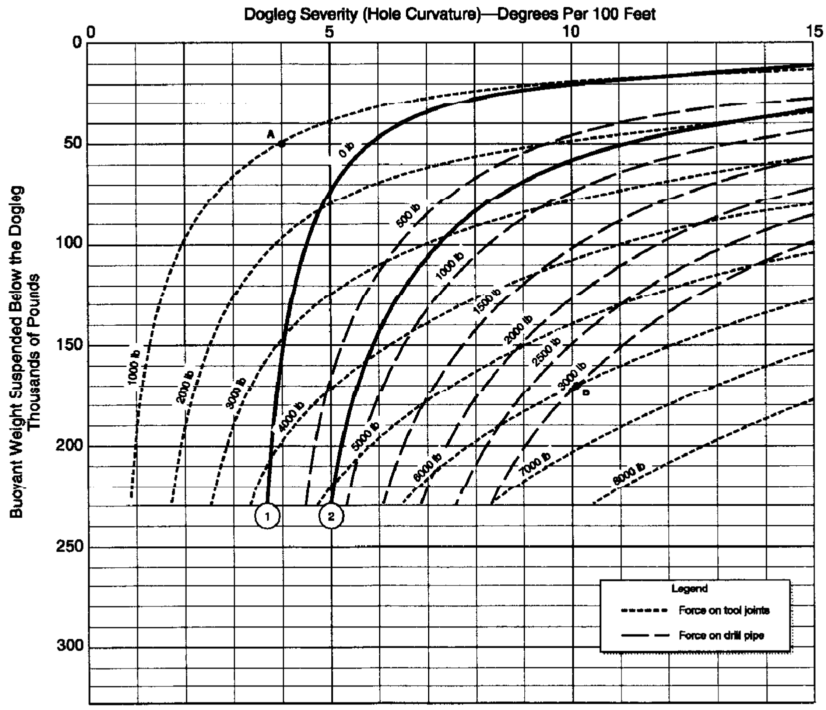


Figure 41—Lateral Forces on Tool Joints and Range 2 Drill Pipe $3^1/_2$-Inch, 13.3 Pounds per Foot, Range 2 Drill Pipe, $4^3/_4$-Inch Tool Joints

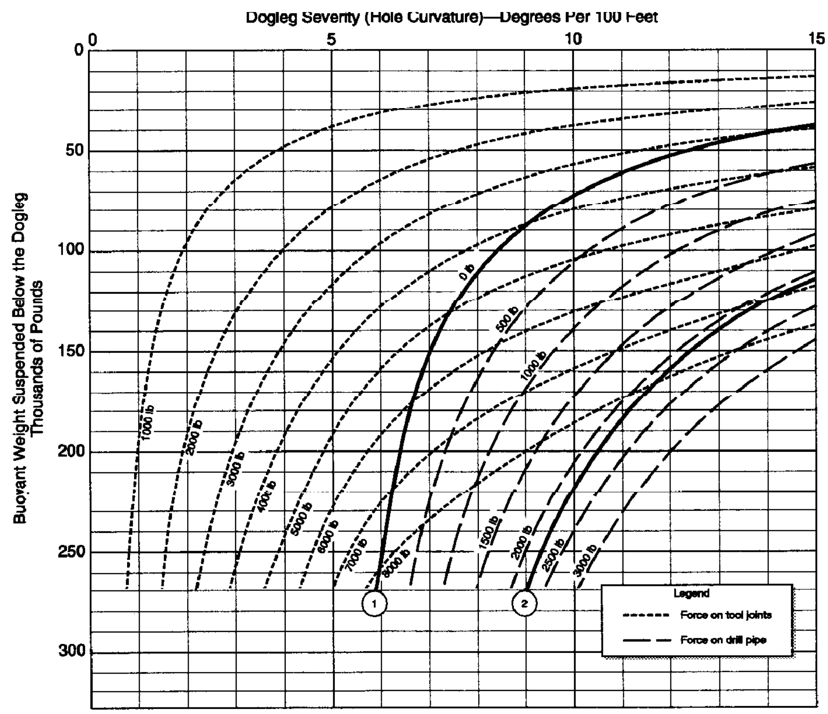


Figure 42—Lateral Forces on Tool Joints and Range 2 Drill Pipe $4^1/_2$-Inch, 16.6 Pounds per Foot, Range 2 Drill Pipe, $6^1/_4$-Inch Tool Joints

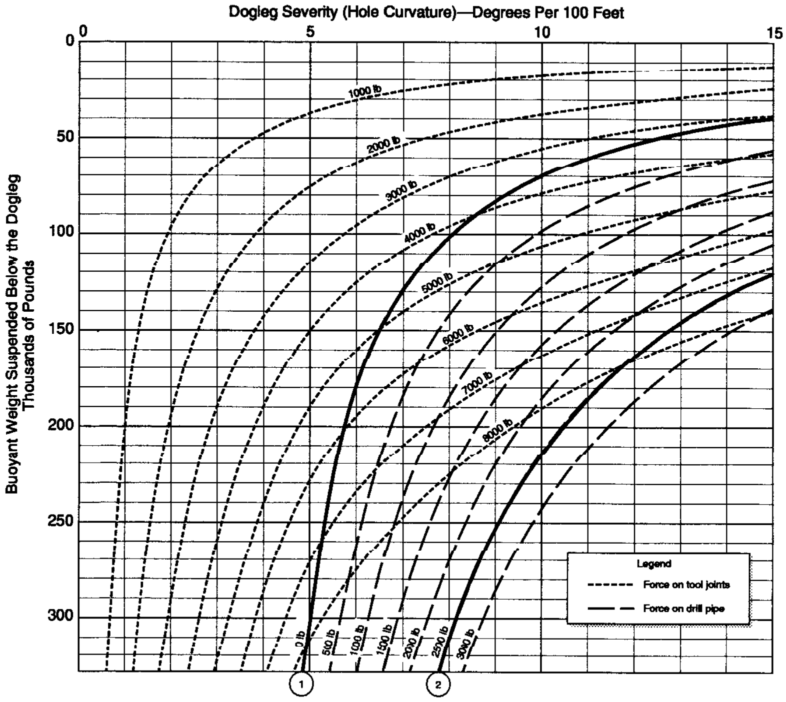


Figure 43—Lateral Forces on Tool Joints and Range 2 Drill Pipe 5-Inch, 19.5 Pounds per Foot, Range 2 Drill Pipe, $6^3/_8$-Inch Tool Joints

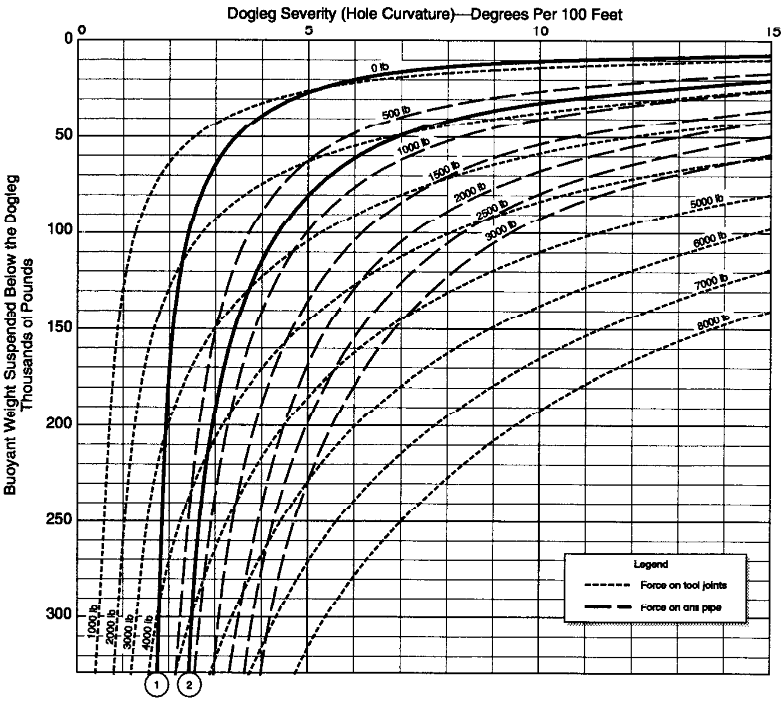


Figure 44—Lateral Forces on Tool Joints and Range 3 Drill Pipe 5-Inch, 19.5 Pounds per Foot, Range 3 Drill Pipe, $6^3/_8$-Inch Tool Joints

10 Drill Stem Corrosion and Sulfide Stress Cracking

10.1 CORROSION

10.1.1 Corrosive Agents

Corrosion may be defined as the alteration and degradation of material by its environment. The principal corrosive agents affecting drill stem materials in water-base drilling fluids are dissolved gases (oxygen, carbon dioxide, and hydrogen sulfide), dissolved salts, and acids.

10.1.1.1 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, twistoffs, and fatigue failures. Since oxygen is soluble in water, and most drilling fluid systems are open to the air, the drill stem is continually exposed to potentially severe corrosive conditions.

10.1.1.2 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 higher pH values, 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 organic drilling fluid additives, or bacterial action on organic material in the makeup water or drilling fluid additives.

10.1.1.3 Hydrogen Sulfide

Hydrogen sulfide dissolves in water to form an acid somewhat weaker and less corrosive than carbonic acid, although it may cause pitting, particularly in the presence of oxygen and/or carbon dioxide. A more significant action of hydrogen sulfide is its effect on a form of hydrogen embrittlement known as sulfide stress cracking (see 10.2 for details).

Hydrogen sulfide in drilling fluids may come from the makeup water, gas-bearing formation fluid inflow, bacterial action on dissolved sulfates, or thermal degradation of sulfur-containing drilling fluid additives.

10.1.1.4 Dissolved Salts

Dissolved salts (chlorides, carbonates, and sulfates) increase the electrical conductivity of drilling fluids. Since most corrosion processes involve electrochemical reactions, the increased conductivity may result in higher corrosion rates. Concentrated salt solutions are usually less corrosive than dilute solutions; however, because of decreased oxygen solubility. Dissolved salts also may serve as a source of carbon dioxide or hydrogen sulfide in drilling fluids.

Dissolved salts in drilling fluids may come from the makeup water, formation fluid inflow, drilled formations, or drilling fluid additives.

10.1.1.5 Acids

Acids corrode metals by lowering the pH (causing hydrogen evolution) and by dissolving protective films. Dissolved oxygen appreciably accelerates the corrosion rates of acids, and dissolved hydrogen sulfide greatly accelerates hydrogen embrittlement.

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.

10.1.2 Factors Affecting Corrosion Rates

Among the many factors affecting corrosion rates of drill stem materials the more important are:

10.1.2.1 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. 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; but it increases rapidly at lower pH values, and decreases slowly at higher pH values. Aluminum alloys, however, may show increasing corrosion rates at pH values greater than 8.5.

10.1.2.2 Temperature

In general, corrosion rates increase with increasing temperature.

10.1.2.3 Velocity

In general, corrosion rates increase with higher rates of flow.

10.1.2.4 Heterogeneity

Localized variations in composition or microstructure may increase corrosion rates. Ringworm corrosion, that is 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 nonuniform grain structure.

10.1.2.5 High Stresses

Highly stressed areas may 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.

10.1.3 Corrosion Damage (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 may 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:

10.1.3.1 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.

10.1.3.2 Localized Attack (Pitting)

Corrosion may be localized in small, well-defined areas, causing pits. Their number, depth, and size may vary considerably; and they may be obscured by corrosion products. Pitting is difficult to detect and evaluate, since it may occur under corrosion products, mill scale and other deposits, in crevices or other stagnant areas, in highly stressed areas, etc. Pits can cause washouts and can serve as points of origin for fatigue cracks. Chlorides, oxygen, carbon dioxide, and hydrogen sulfide, and especially combinations of them, are major contributors to pitting corrosion.

10.1.3.3 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 may cause pitting, extensive damage, and failure.

10.1.3.4 Fatigue in a Corrosive Environment (Corrosion Fatigue)

Metals subjected to cyclic stresses of sufficient magnitude will develop fatigue cracks that may 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 Section 8.

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.

10.1.4 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. Various instruments and devices such as pH meters, oxygen meters, corrosion meters, hydrogen probes, chemical test kits, test coupons, etc. are available for field monitoring of corrosion agents and their effects.

The monitoring system described in Appendix A of API Recommended Practice 13B-1, *Recommended Practice Standard Procedure for Field Testing Water-Based Drilling Fluids,* can be used to evaluate corrosive conditions and to follow the effect of remedial actions taken to correct undesirable conditions. Preweighed test rings are placed in 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, cleaned, and reweighed. The degree and severity of pitting observed may be of greater significance than the weight loss measurement.

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, carbon dioxide, and hydrogen sulfide).

10.1.5 Procurement of Samples for Laboratory Testing

When laboratory examination of drilling fluid is desired, representative samples should be collected in a $^1/_2$ to 1 gallon (2 to 4 liter) clean container, allowing an air space of approximately 1 percent 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 must 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.

10.1.6 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 drill stem OD may be increased. 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.

10.1.7 Corrective Measures to Minimize Corrosion in Water-Base Drilling Fluids

The selection and control of appropriate corrective measures is usually performed by competent corrosion 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, 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 aluminum drill pipe increases at pH values higher than 8.5.

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 must 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. The use of the wrong type of inhibitor, or the wrong amount, may actually increase corrosion.

c. Use plastic coated drill pipe. Care must be exercised to prevent damage to the coating.

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.

f. Limit gas-cutting and formation fluid inflow by maintaining proper drilling fluid weight.

g. 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 Recommended Practice 5C1, *Recommended Practice for Care and Use of Casing and Tubing*).

10.1.8 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.

d. Use quenched and tempered components.

10.2 SULFIDE STRESS CRACKING

10.2.1 Mechanism of Sulfide Stress Cracking (SSC)

In the presence of hydrogen sulfide (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 sulfide stress cracking (SSC).

10.2.2 Materials Resistant to SSC

The latest revision of NACE Standard MR-01-75, *Sulfide 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:

10.2.2.1 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) must be used; and, the higher the strength required, the greater the necessity for the corrective measures.

10.2.2.2 Total Tensile Load (Stress) on the Steel

The higher the total tensile load 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.

10.2.2.3 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.

10.2.2.4 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 (see Figure 45).

10.2.2.5 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°C), 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 must be exercised, however—SSC failure may occur when the material returns to normal temperature after it is removed from the hole.

10.2.3 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.

Note: In some drilling fluids, aluminum 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 aluminum 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 10.1.7.

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 sulfide inflows, preferably prior to encountering the sulfide.

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, 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 400° to 600°F (204° to 316°C) 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.

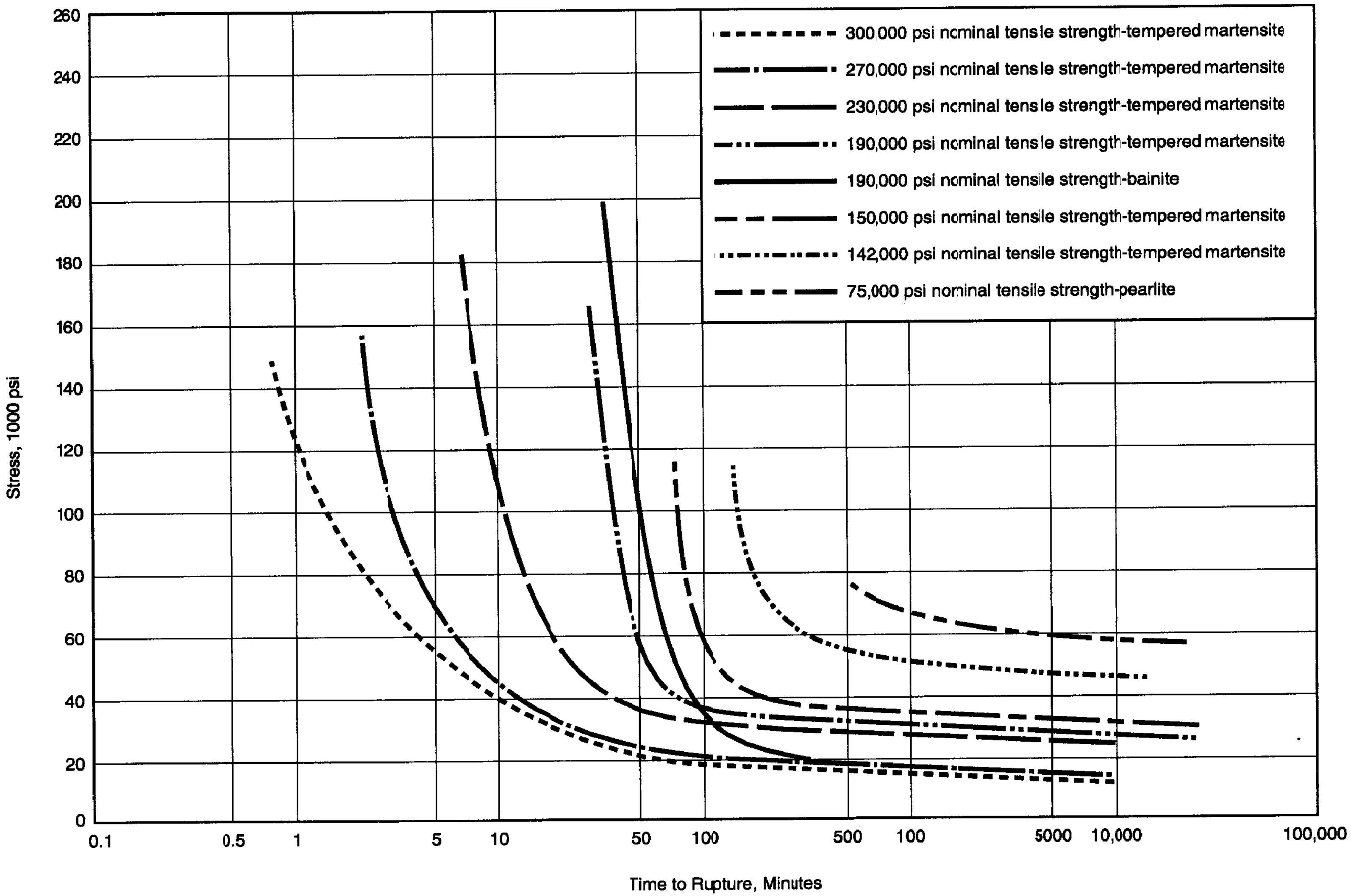
k. Limit drill stem testing in sour environments to as brief a period as possible, using operating procedures that will minimize exposure to SSC conditions.

10.3 DRILLING FLUIDS CONTAINING OIL

10.3.1 Use of Oil Muds for Drill Stem Protection

Corrosion and SSC can be minimized by the use of drilling fluids having oil as the continuous phase. Corrosion does not occur if metal is completely enveloped and wet by an oil environment that is electrically nonconductive.

Oil systems used for drilling (oil-base 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 mud.



Note:
Batelle Charging Condition A:
Electrolyte: 4 percent by weight of H_2SO_4 in water.
Poison: 5 drops per liter of cathodic poison composed of 2_8 phosphorous dissolved in 40 ml CS_2.
Current Density: 8 ma/in².

Figure 45—Delayed-Failure Characteristics of Unnotched Specimens of an SAE 4340 Steel During Cathodic Charging with Hydrogen Under Standardized Conditions

10.3.2 Monitoring Oil Mude for Drill Stem Protection

An oil mud must 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 mud include:

10.3.2.1 Electrical Stability

This test measures the voltage required to cause current to flow between electrodes immersed in the oil mud (see API Recommended Practice 13B-2, *Recommended Practice Standard Procedure for Field Testing Oil-Based Drilling Fluids*, for details). The higher the voltage, the greater the stability of the emulsion, and the better the protection provided to the drill stem.

10.3.2.2 Alkalinity

The acidic dissolved gases (carbon dioxide and hydrogen sulfide) are harmful contaminants for most oil muds. Monitoring the alkalinity of an oil mud can indicate when acidic gases are being encountered so that corrective treatment can be instituted.

10.3.2.3 Corrosion Test Rings

Test rings placed in the drill stem bore are used to monitor the corrosion protection afforded by oil muds (see API Recommended Practice 13B-2 for details). A properly functioning oil mud should show little or no visual evidence of corrosion on the test ring.

11 Compressive Service Limits for Drill Pipe (see also Appendices A.14 and A.15)

11.1 COMPRESSIVE SERVICE APPLICATIONS

11.1.1 Whenever drilling high angle, extended reach, or horizontal well bores it is desirable to use compressively loaded portions of the drill string. 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.

11.1.2 Drill strings are subject to compressive service conditions whenever significant portions of the borehole exceed the critical sliding hole angle as defined as follows:

$$\theta_c = \arctan\left(\frac{1}{f}\right),$$

where

θ_c = critical sliding hole angle,
f = coefficient of friction.

11.1.3 The critical sliding hole angle is the angle above which drill string components must be pushed into the hole. Although many factors affect the coefficient of friction between the drill string components and the wall of the hole, the type of drilling fluid used has the greatest impact (see Table 19). Water-based drilling fluids generate the highest coefficient to friction and produce critical hole angles of about 71 degrees. Synthetic-based drilling fluids provide the lowest coefficients of friction and produce critical hole angles of about 80 degrees.

11.1.4 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 string that can withstand the operating environment. Using heavier components or thicker wall tubulars often increases the operating loads without reducing the bending stresses.

Table 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

11.2 DRILL PIPE BUCKLING IN STRAIGHT, INCLINED WELL BORES

11.2.1 Overview

11.2.1.1 The curves shown in Figures 46 through 66 give the approximate axial compressive loads at which sinusoidal buckling is expected to occur in drill pipe in straight, inclined wellbores. If the drill pipe is being rotated, limiting the drill pipe compressive load to below the estimated buckling load shown in these curves will significantly reduce the danger of fatigue damage to the pipe. Conversely, rotating drill pipe while it is buckled can lead to rapid fatigue damage and failure.

11.2.1.2 These curves are based on the equations of Dawson and Paslay.[27] They are reproduced here with permission from Standard DS-1, *Drill Stem Design and Inspection*.[28] The assumptions behind these curves include:

a. Pipe weight is new nominal with X-grade tool joint dimensions (where applicable). If more than one tool joint is

(Text continued on page 78.)

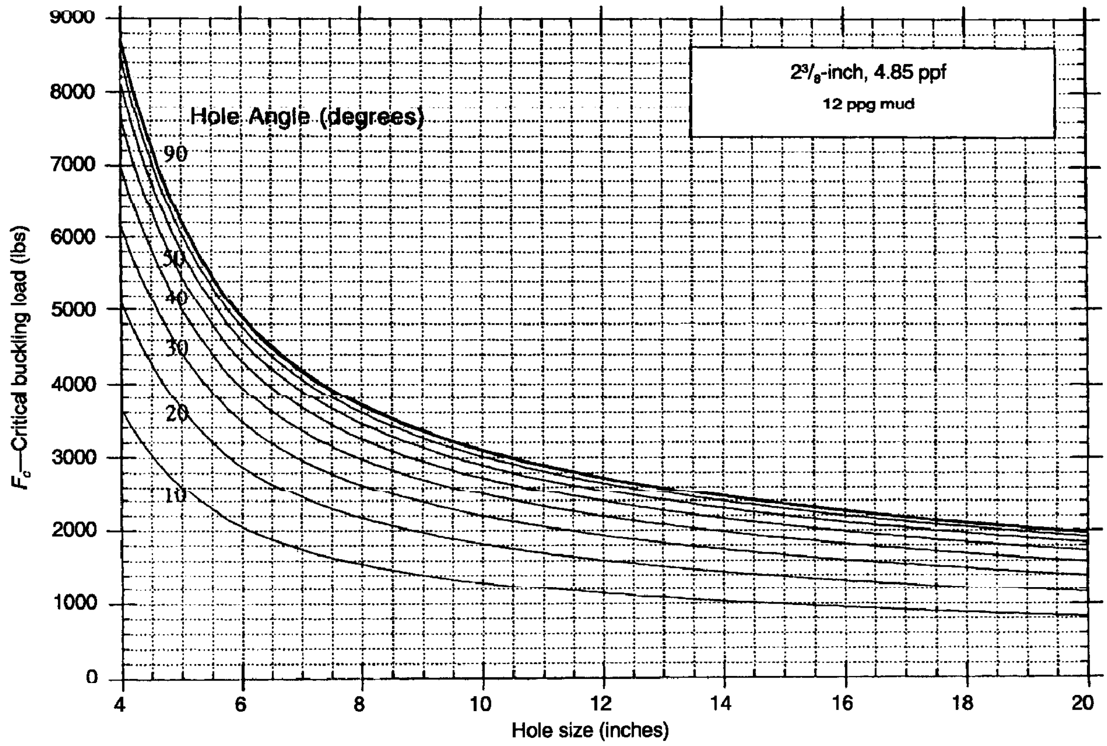


Figure 46—Approximate Axial Compressive Loads at which Sinusoidal Buckling is Expected to Occur

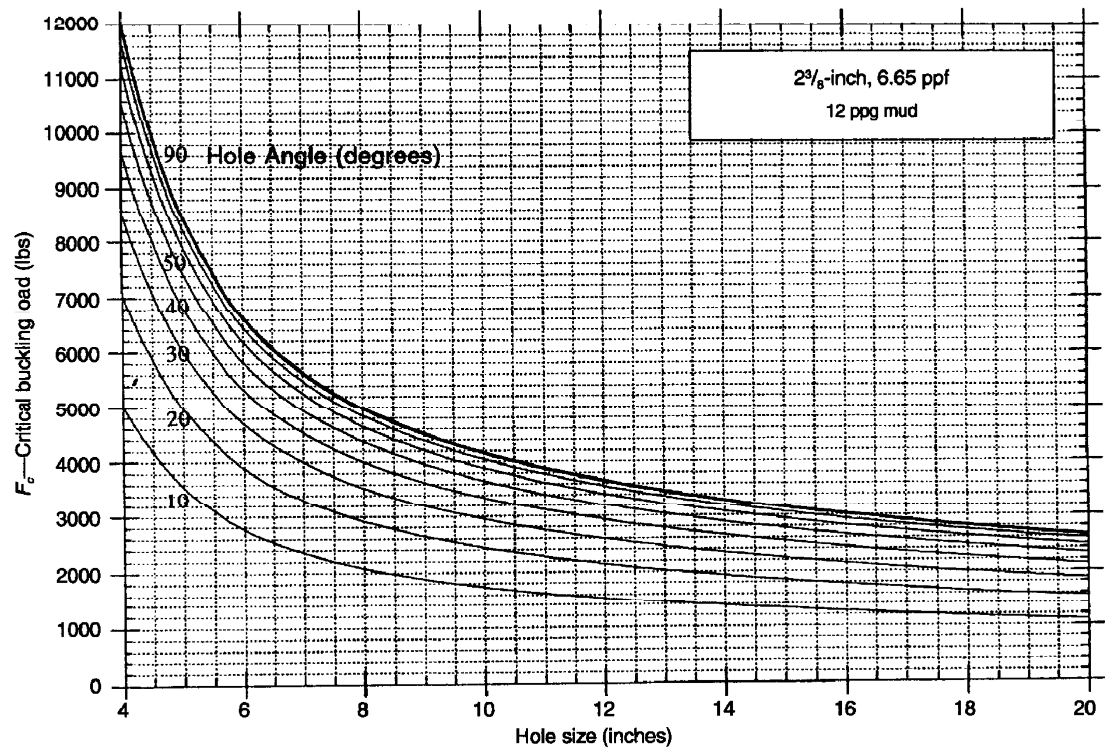


Figure 47—Approximate Axial Compressive Loads at which Sinusoidal Buckling is Expected to Occur

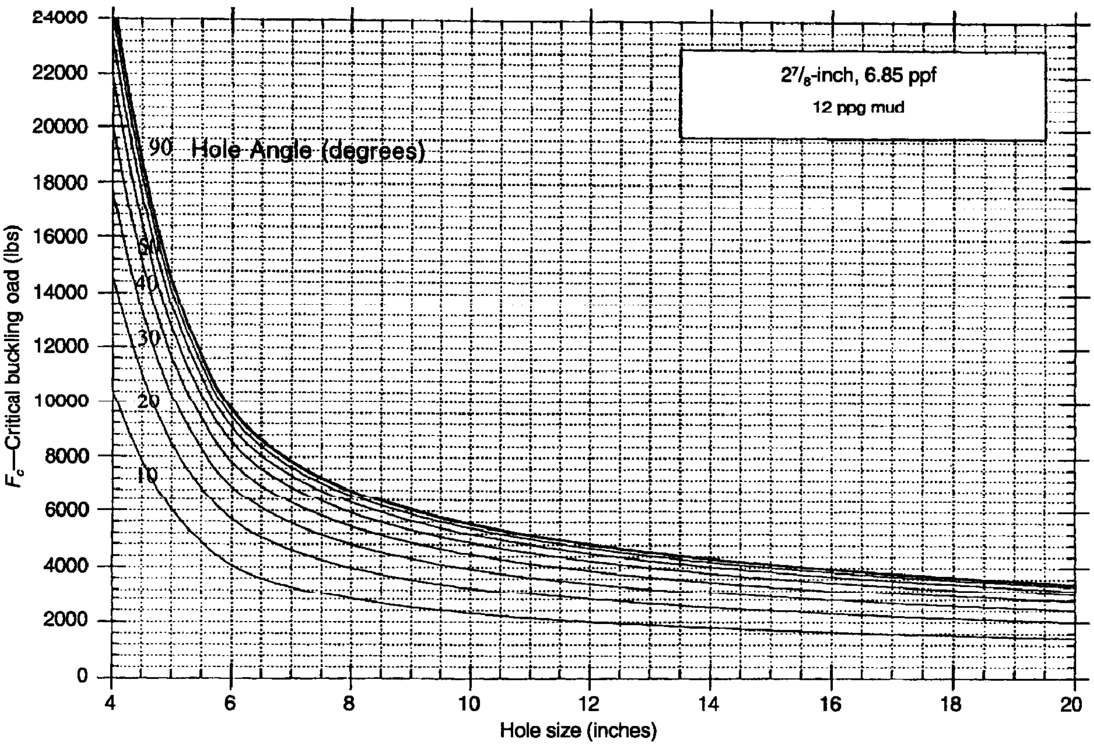


Figure 48—Approximate Axial Compressive Loads at which Sinusoidal Buckling is Expected to Occur

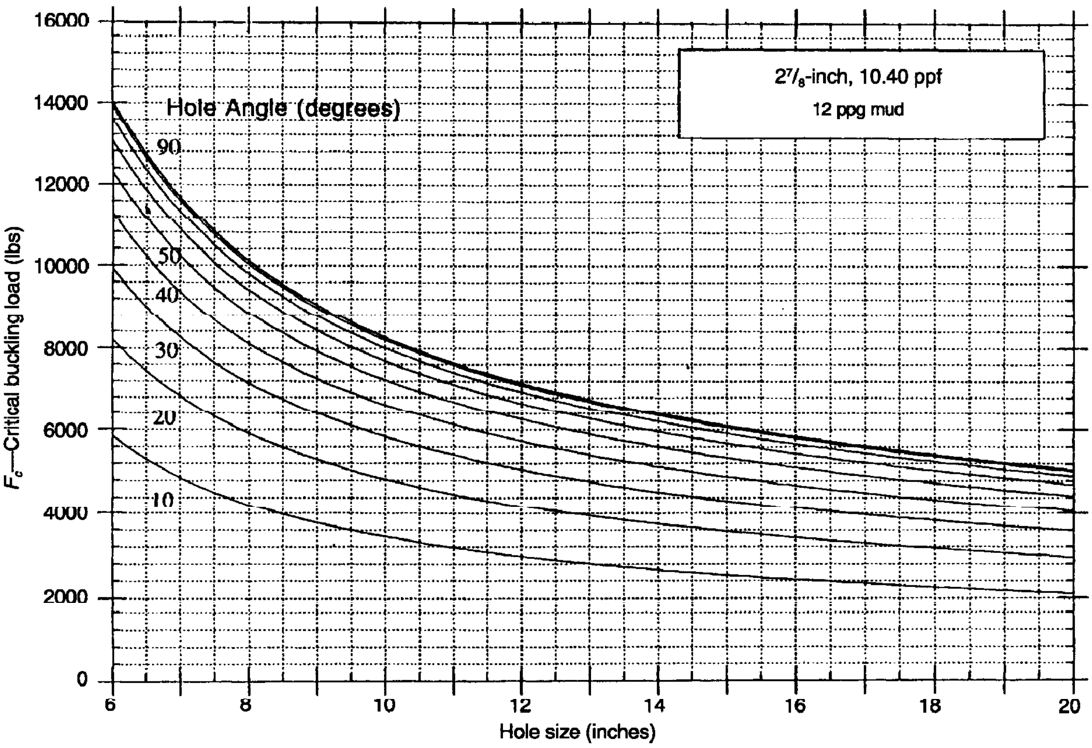


Figure 49—Approximate Axial Compressive Loads at which Sinusoidal Buckling is Expected to Occur

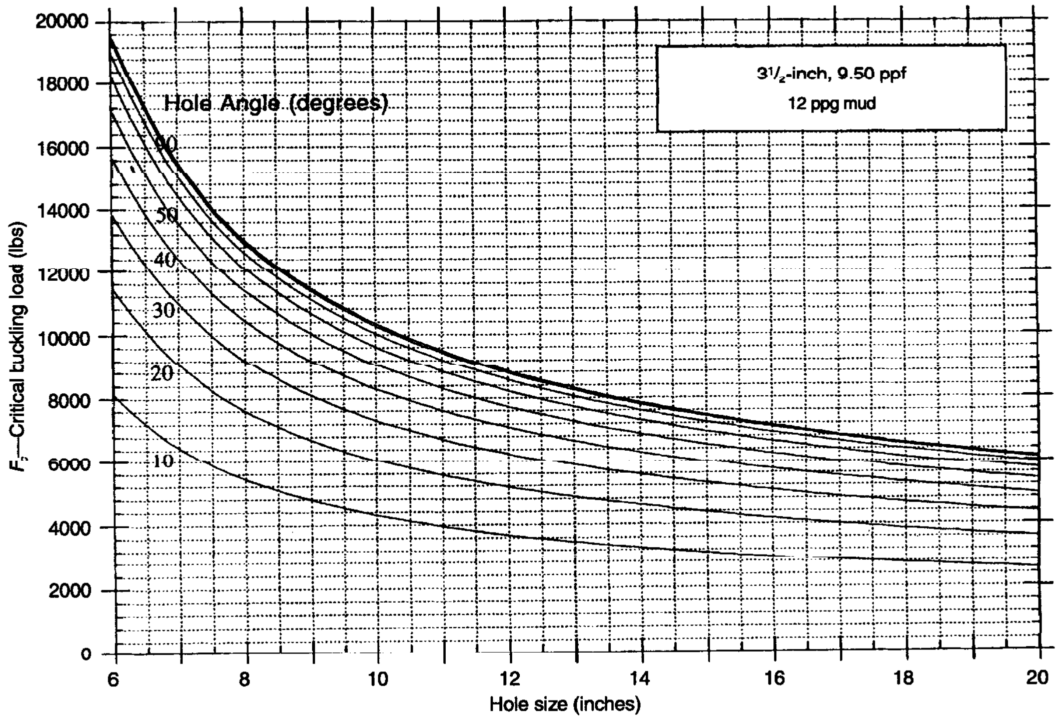


Figure 50—Approximate Axial Compressive Loads at which Sinusoidal Buckling is Expected to Occur

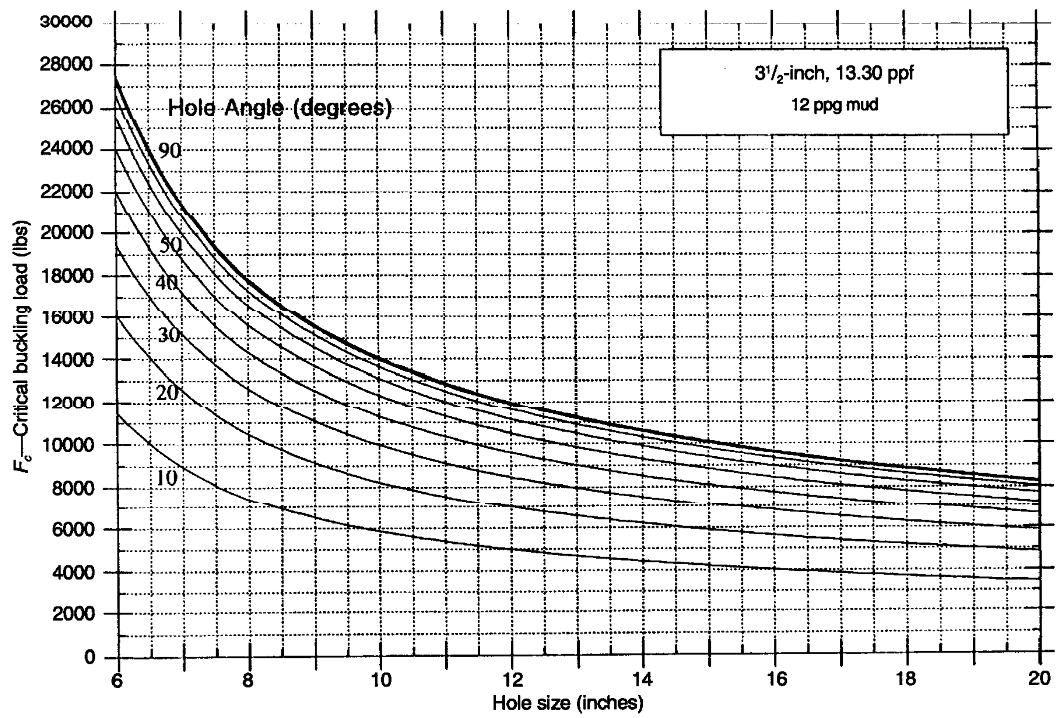


Figure 51—Approximate Axial Compressive Loads at which Sinusoidal Buckling is Expected to Occur

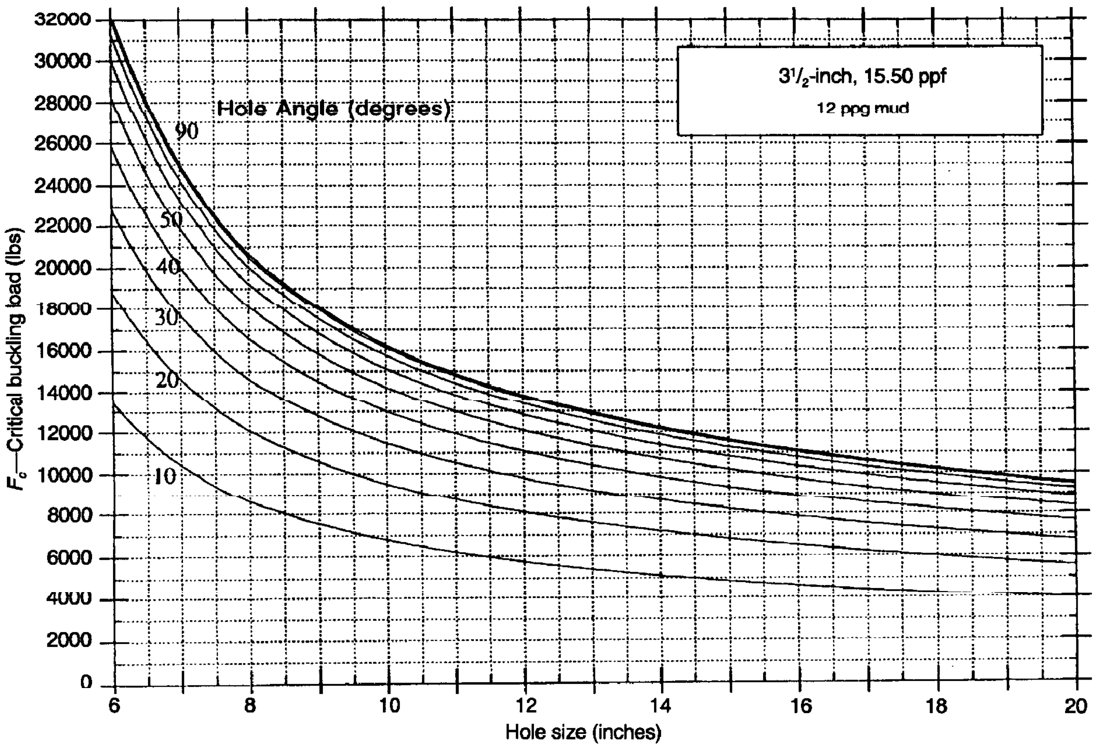


Figure 52—Approximate Axial Compressive Loads at which Sinusoidal Buckling is Expected to Occur

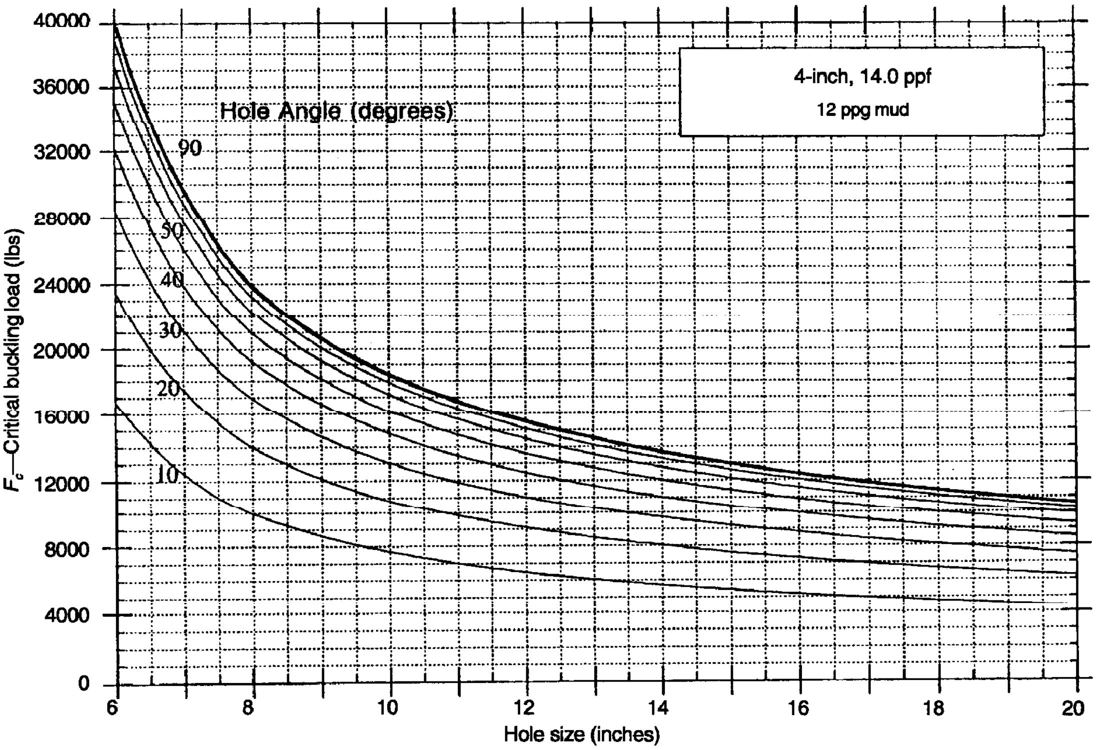


Figure 53—Approximate Axial Compressive Loads at which Sinusoidal Buckling is Expected to Occur

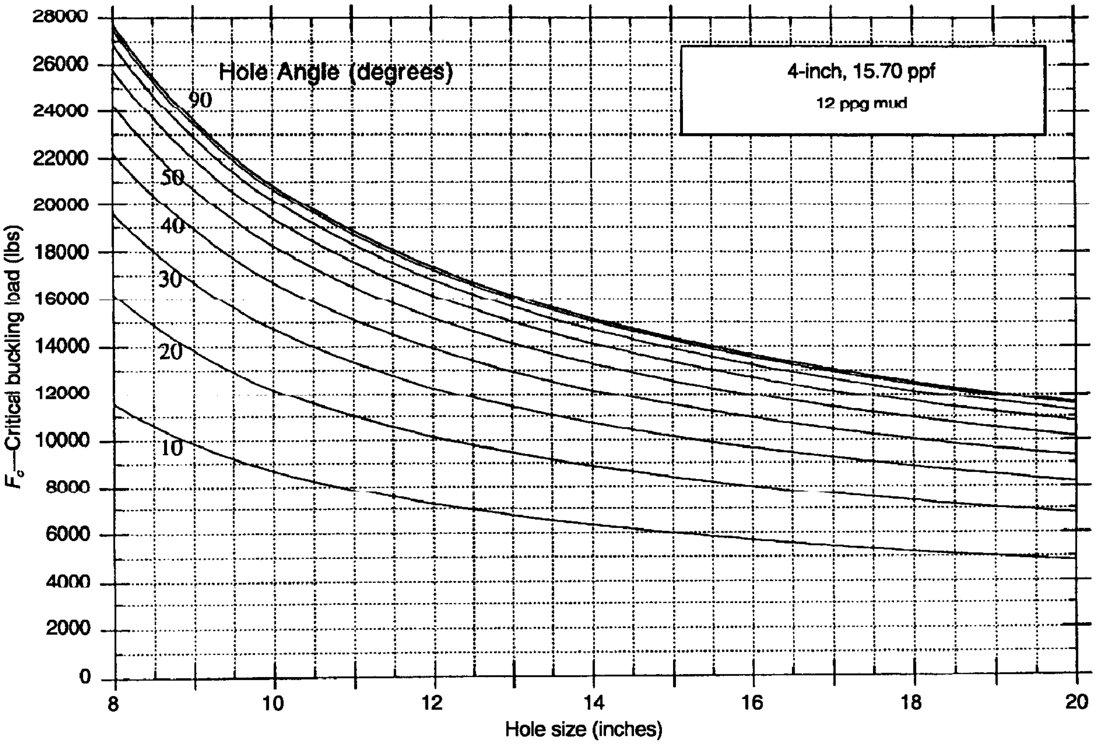


Figure 54—Approximate Axial Compressive Loads at which Sinusoidal Buckling is Expected to Occur

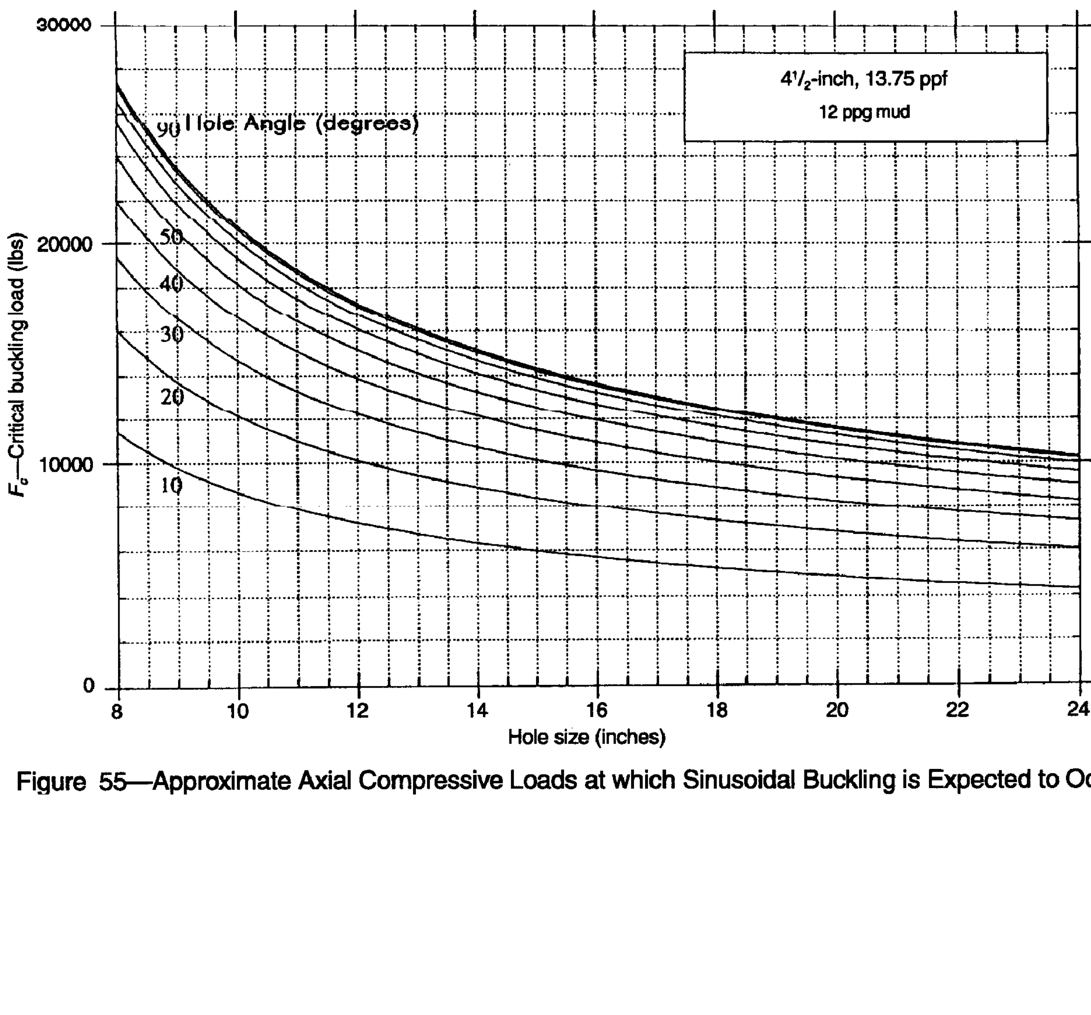


Figure 55—Approximate Axial Compressive Loads at which Sinusoidal Buckling is Expected to Occur

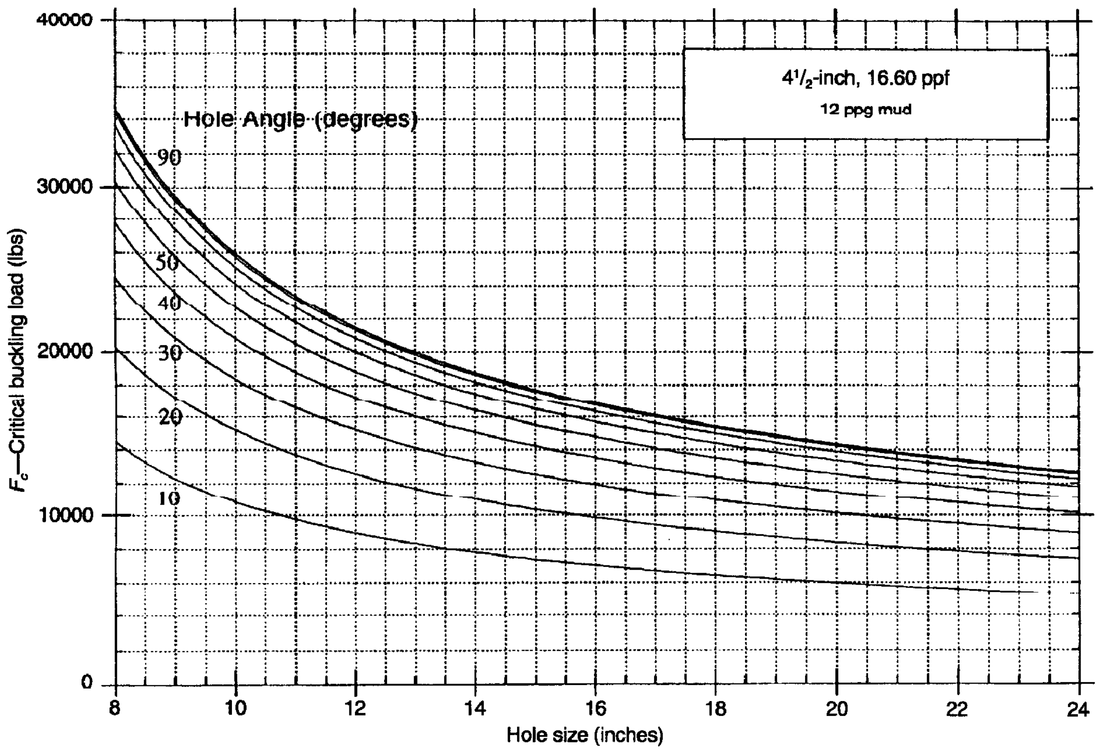


Figure 56—Approximate Axial Compressive Loads at which Sinusoidal Buckling is Expected to Occur

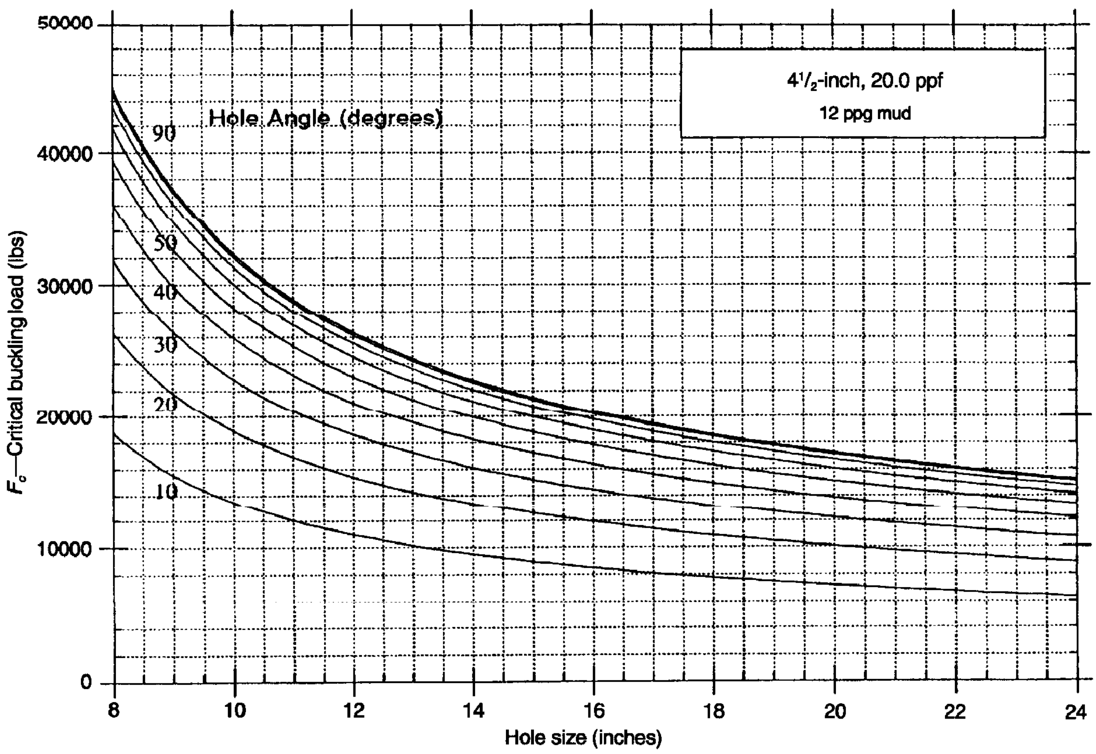


Figure 57—Approximate Axial Compressive Loads at which Sinusoidal Buckling is Expected to Occur

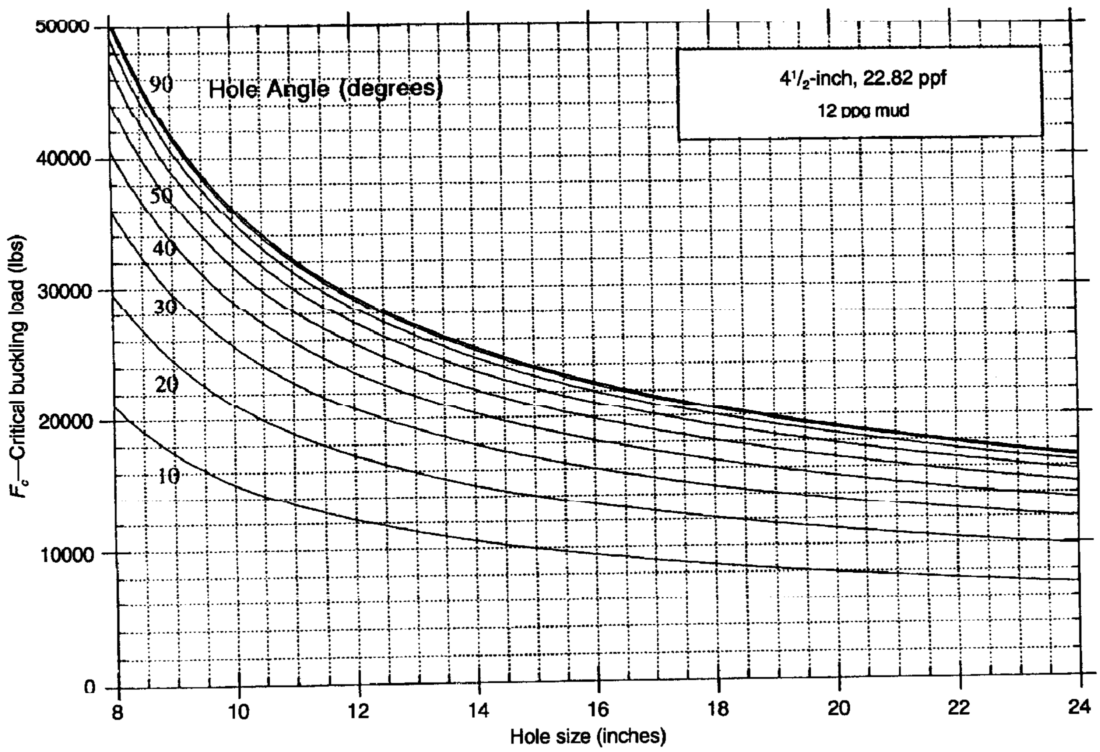


Figure 58—Approximate Axial Compressive Loads at which Sinusoidal Buckling is Expected to Occur

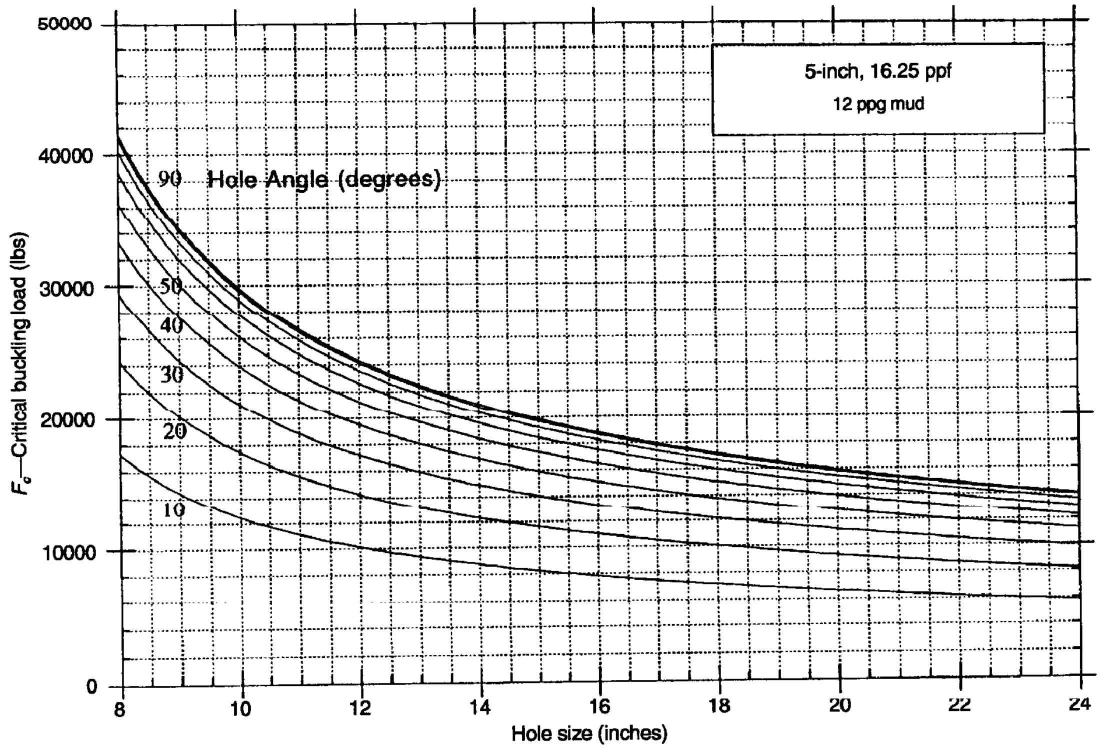


Figure 59—Approximate Axial Compressive Loads at which Sinusoidal Buckling is Expected to Occur

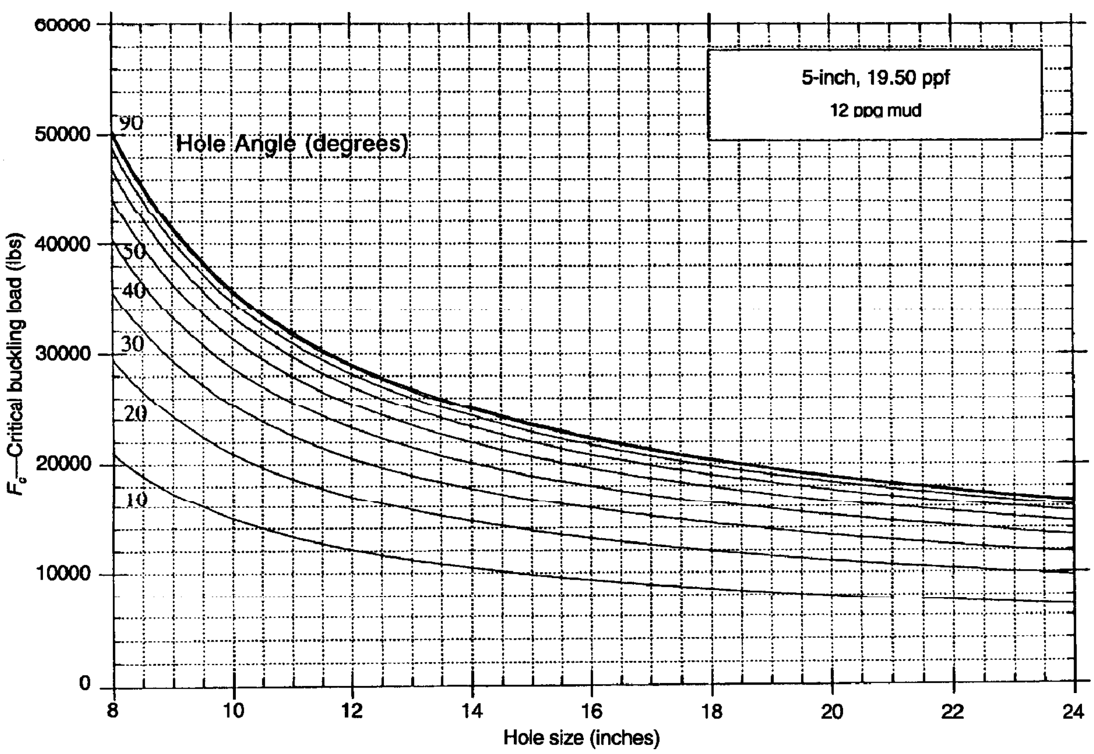


Figure 60—Approximate Axial Compressive Loads at which Sinusoidal Buckling is Expected to Occur

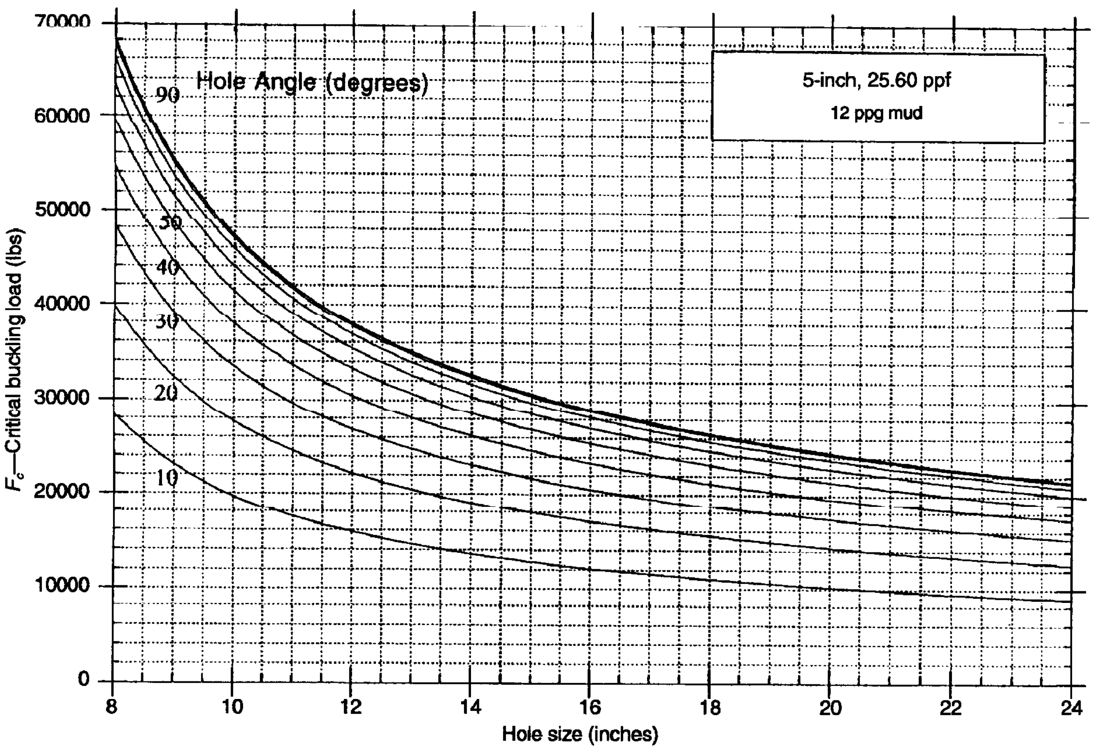


Figure 61—Approximate Axial Compressive Loads at which Sinusoidal Buckling is Expected to Occur

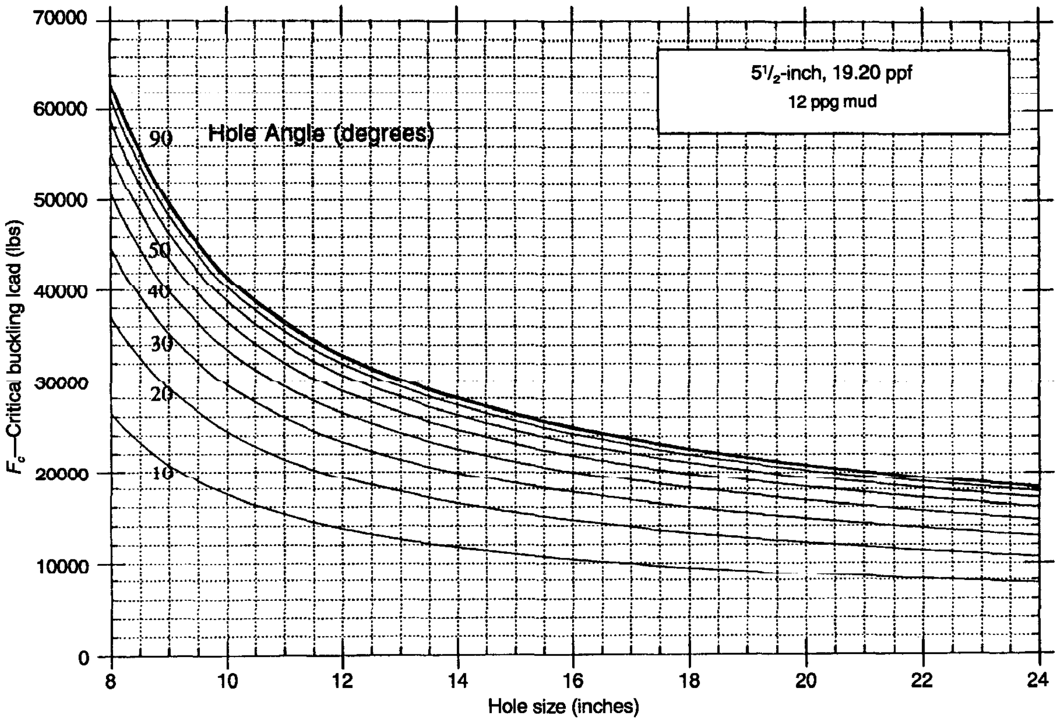


Figure 62—Approximate Axial Compressive Loads at which Sinusoidal Buckling is Expected to Occur

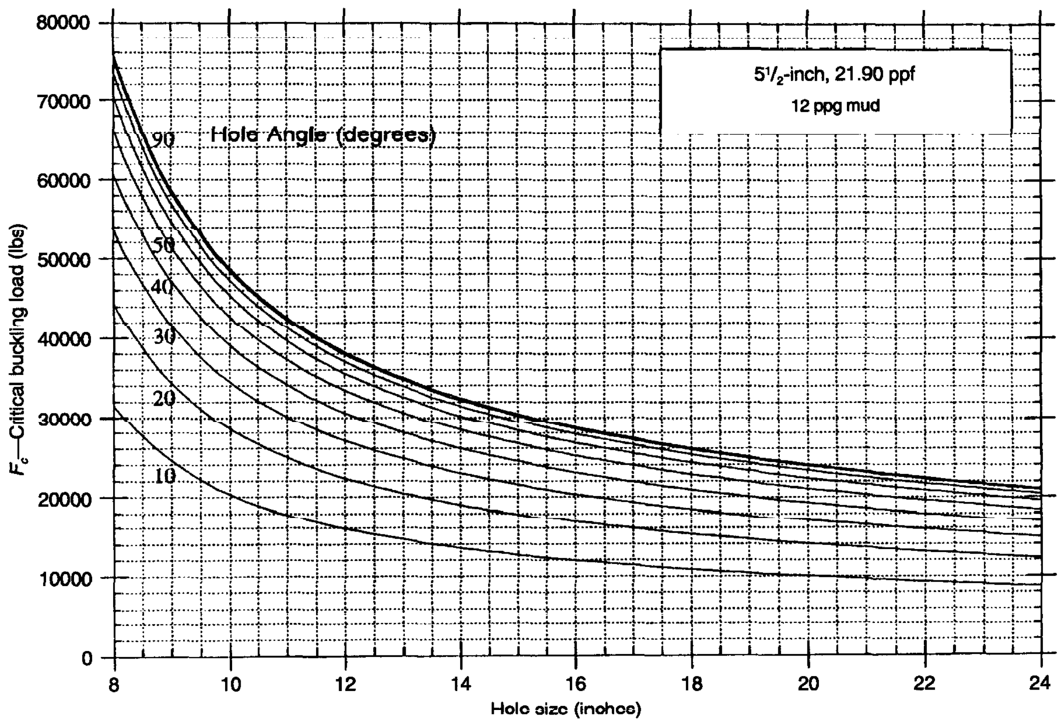


Figure 63—Approximate Axial Compressive Loads at which Sinusoidal Buckling is Expected to Occur

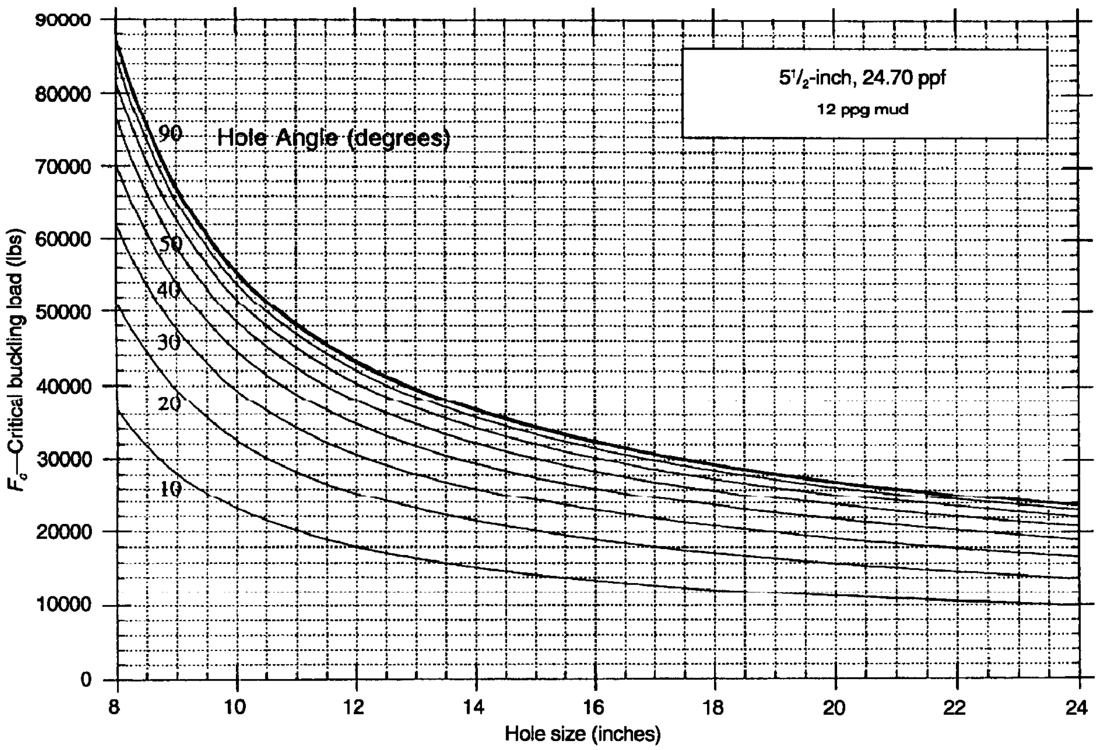


Figure 64—Approximate Axial Compressive Loads at which Sinusoidal Buckling is Expected to Occur

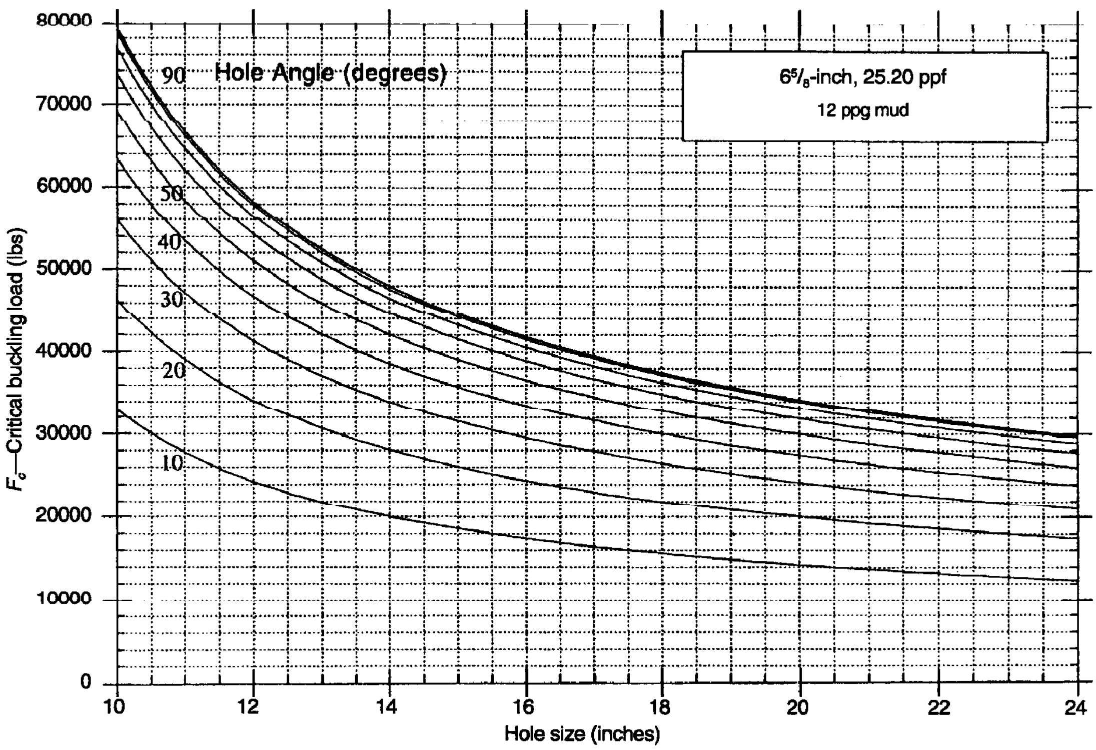


Figure 65—Approximate Axial Compressive Loads at which Sinusoidal Buckling is Expected to Occur

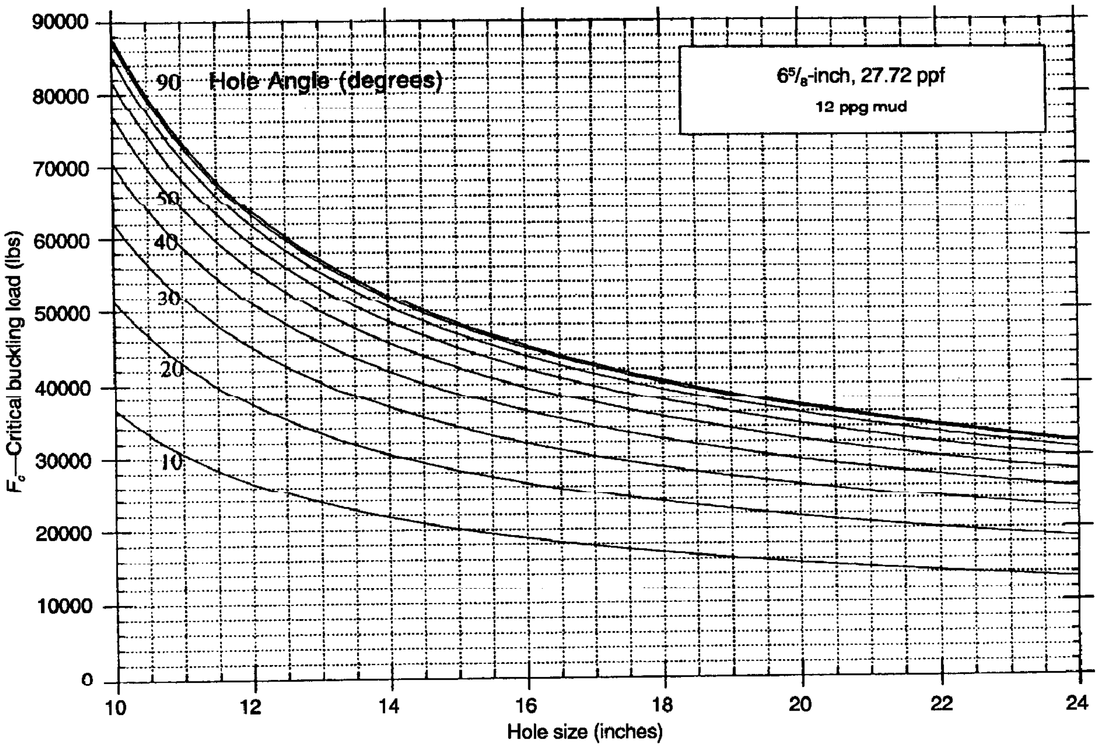


Figure 66—Approximate Axial Compressive Loads at which Sinusoidal Buckling is Expected to Occur

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.

b. Pipe wall thickness is new nominal.

c. The well bore is straight.

d. The effects of torque are neglected.

e. Mud weight is 12.0 lb/gal.

11.2.2 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:

$$(F_{crit\text{-}adj}) = (F_{crit})(f_{mw})$$

where

$F_{crit\text{-}adj}$ = adjusted critical buckling load (lbs),

F_{crit} = critical buckling load from curve (lbs),

f_{mw} = (buoyancy factor/0.817)$^{0.5}$ (see below),

Mud Weight (lb/gal)	f_{mw}	Mud Weight (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

11.2.3 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.

11.2.3.1 Example

How much compressive load may be applied to 5-inch, 19.50 lb/ft drill pipe in a 12¹/₄-inch horizontal hole before the drill pipe buckles? Mud weight is 9.0 lb/gal.

11.2.3.2 Solution:

Reading from the figure for the drill pipe in question (Figure 60), the critical buckling load is about 28,200 pounds. Adjusting to 9.0 lb/gal mud:

$$F_{crit\text{-}adj} = 28{,}200 \text{ lbs } (1.03) = 29{,}000 \text{ lbs}$$

11.3 CRITICAL BUCKLING FORCE FOR CURVED BOREHOLES[27,29,30,31,32]

The critical buckling force of compressively loaded drill pipe is also significantly influenced by the curvature of the borehole. In angle building intervals the upward curvature of the borehole increases the critical buckling force. In turning intervals the hole curvature also increases the buckling force of the drill pipe. Table 20 shows the hole curvature rates that prevent buckling for a range of pipe and hole sizes.

Table 20—Hole Curvatures that Prevent Buckling

Hole Size	Drill Pipe OD	Drill Pipe ID	Nominal Weight	Tool Joint OD	Axial Load							
					5 Mlbs	10 Mlbs	15Mlbs	20Mlbs	25Mlbs	30Mlbs	40Mlbs	50Mlbs
in.	in.	in.	lb/ft	in.	°/100ft.	°/100ft.	°/100ft.	°/100ft.	°/100ft.	°/100ft.	°/100ft.	°/100ft.
4.000	2.375	1.815	6.7	3.125	1.2	2.5	3.7	4.9	6.2	7.4	9.9	12.3
4.750	2.875	2.151	10.4	4.125	0.4	0.8	1.2	1.6	2.0	2.4	3.2	3.9
6.000	2.875	2.151	10.4	4.125	1.2	2.4	3.5	4.7	5.9	7.1	9.5	11.8
6.000	3.500	2.764	13.3	5.000	0.3	0.6	1.0	1.3	1.6	1.9	2.6	3.2
6.750	3.500	2.764	13.3	5.000	0.6	1.1	1.7	2.3	2.8	3.4	4.5	5.6
6.750	4.000	3.340	14.0	5.250	0.3	0.7	1.0	1.3	1.7	2.0	2.7	3.4
7.875	4.000	3.340	14.0	5.250	0.6	1.2	1.8	2.4	3.0	3.5	4.7	5.9
7.875	4.500	3.826	16.6	6.250	0.2	0.5	0.7	1.0	1.2	1.5	2.0	2.5
8.750	4.500	3.826	16.6	6.250	0.4	0.8	1.1	1.5	1.9	2.3	3.0	3.8
8.750	5.000	4.276	19.5	6.375	0.2	0.5	0.7	1.0	1.2	1.4	1.9	2.4
8.750	5.500	4.778	21.9	7.500	0.1	0.2	0.3	0.4	0.5	0.6	0.8	0.9
8.750	5.500	4.670	24.7	7.250	0.1	0.2	0.3	0.4	0.5	0.6	0.8	1.0
9.875	5.000	4.276	19.5	6.375	0.4	0.7	1.1	1.4	1.8	2.1	2.8	3.6
9.875	5.500	4.778	21.9	7.500	0.2	0.4	0.5	0.7	0.9	1.1	1.4	1.8
9.875	5.500	4.670	24.7	7.250	0.2	0.4	0.5	0.7	0.9	1.1	1.4	1.8
9.875	6.625	5.965	25.2	8.000	0.1	0.2	0.3	0.3	0.4	0.5	0.7	0.8
12.250	5.000	4.276	19.5	6.375	0.6	1.2	1.8	2.4	3.0	3.6	4.8	6.0
12.250	5.500	4.778	21.9	7.500	0.4	0.7	1.1	1.4	1.8	2.1	2.9	3.6
12.250	5.500	4.670	24.7	7.250	0.3	0.7	1.0	1.3	1.7	2.0	2.7	3.4
12.250	6.625	5.965	25.2	8.000	0.2	0.4	0.6	0.8	1.0	1.1	1.5	1.9

11.3.1 Example

Will 5-inch drill pipe buckle in a 10-degrees-per-100-foot build curve of a horizontal well? The hole size is 8.75 inches and the maximum required bit load is 30,000 pounds.

11.3.2 Solution

Table 20 shows that 5-inch, 19.5 lb/ft drill pipe in an 8.75-inch hole will not buckle in hole curvatures greater than 1.4 degrees per 100 feet with a 30,000 pound load.

11.4 BENDING STRESSES ON COMPRESSIVELY LOADED DRILL PIPE IN CURVED BOREHOLES[33,34]

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 compressive 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 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 $6^5/_8$-inch to $2^3/_8$-inch 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 relation is shown as a dashed line and for wrap contact the

(Text continued on page 96.)

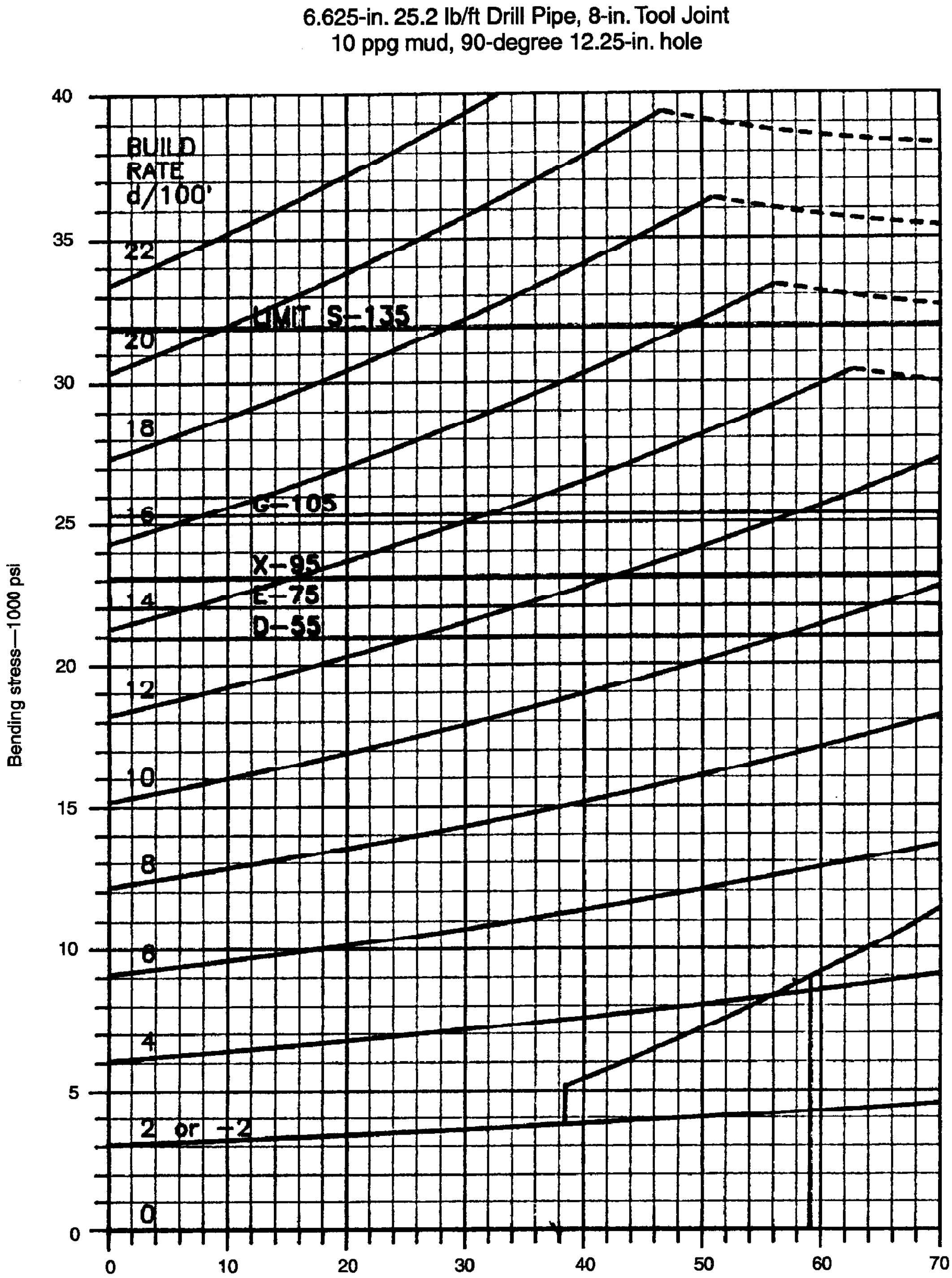


Figure 67a—Bending Stress and Fatigue Limits

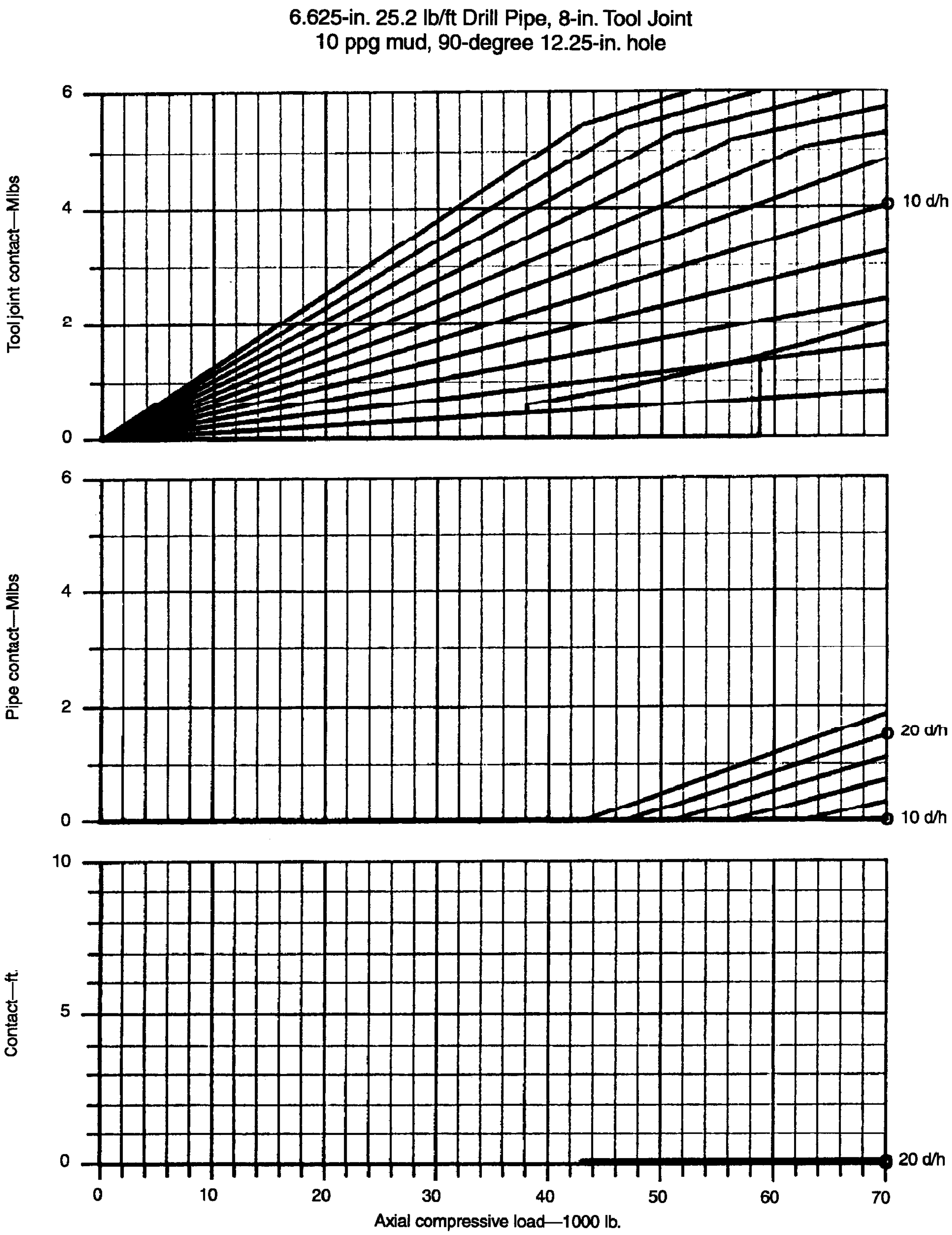


Figure 67b—Lateral Contact Forces and Length

5.5-in. 21.9 lb/ft Drill Pipe, 7.5-in. Tool Joint
10 ppg mud, 90-degree 9.875-in. hole

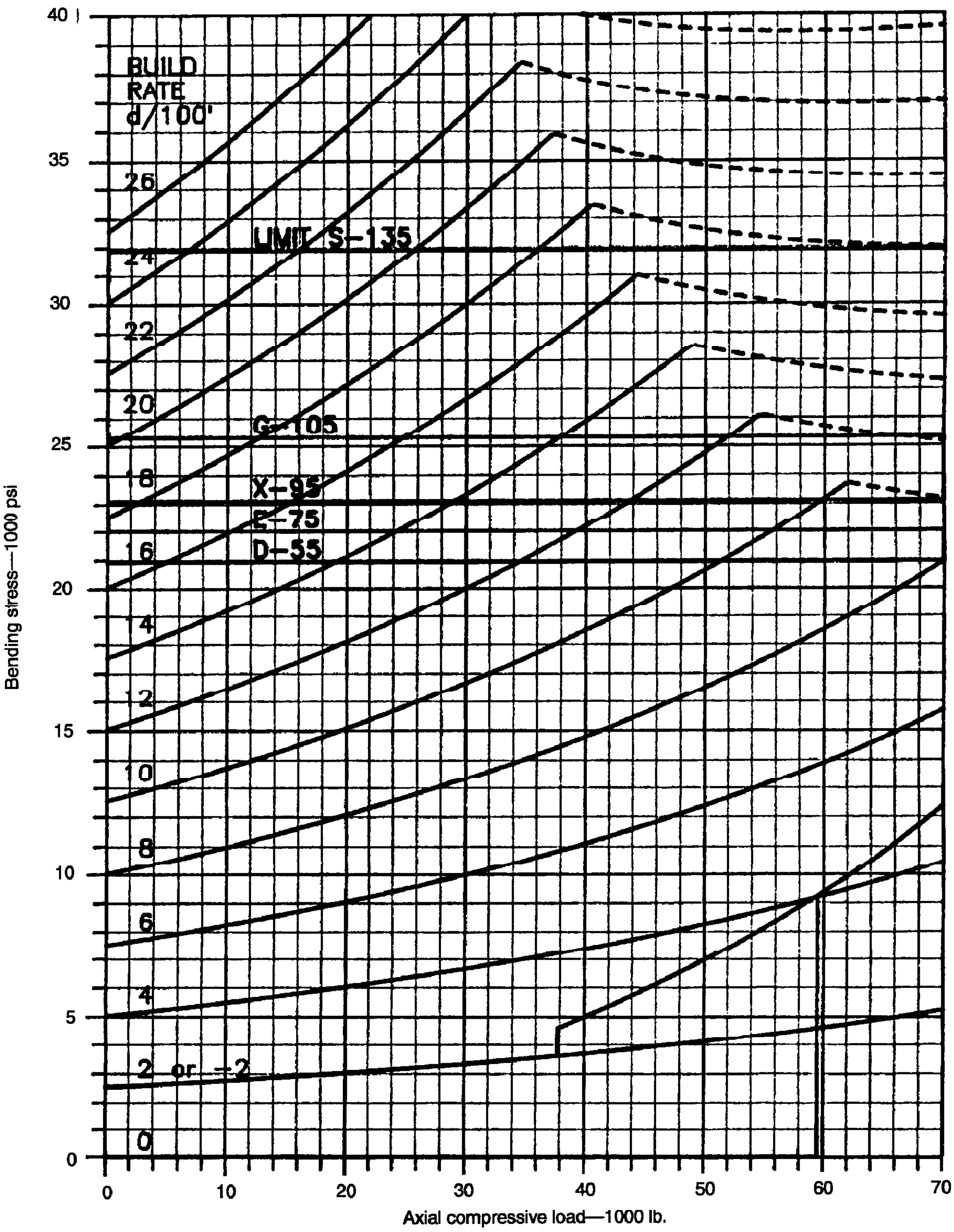


Figure 68a—Bending Stress and Fatigue Limits

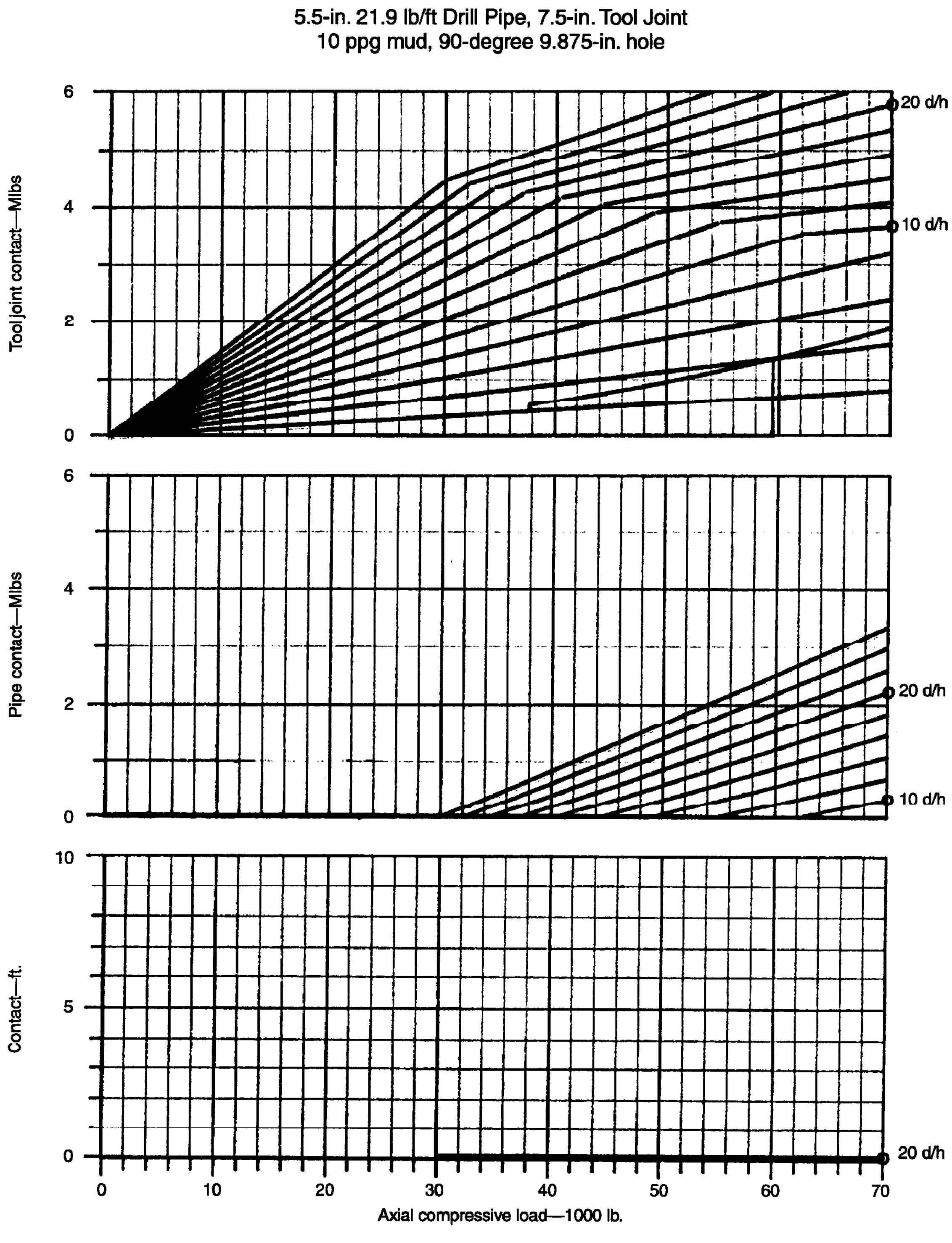


Figure 68b—Lateral Contact Forces and Length

5-in. 19.5 lb/ft Drill Pipe, 6.375-in. Tool Joint
10 ppg mud, 90-degree 8.50-in. hole

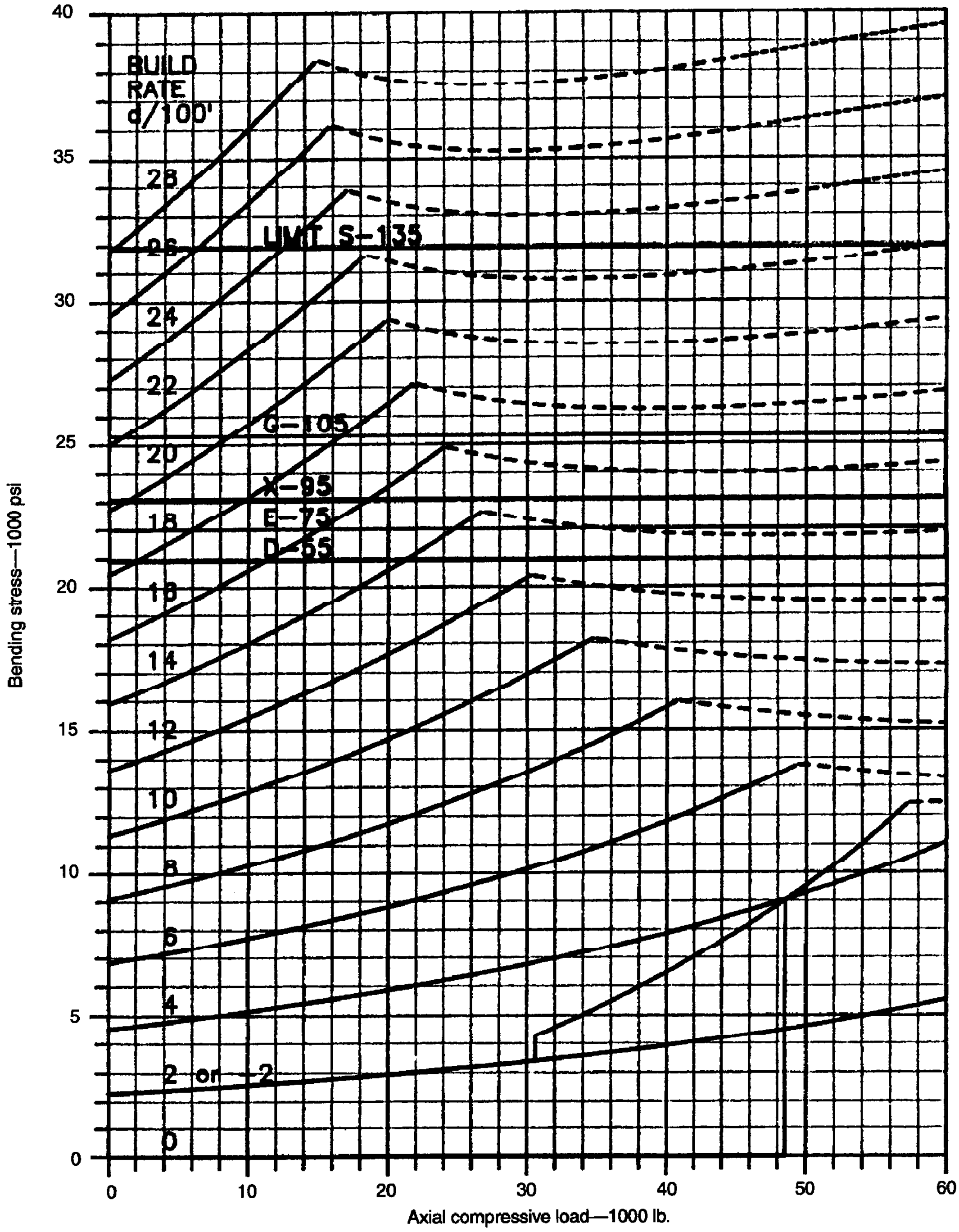


Figure 69a—Bending Stress and Fatigue Limits

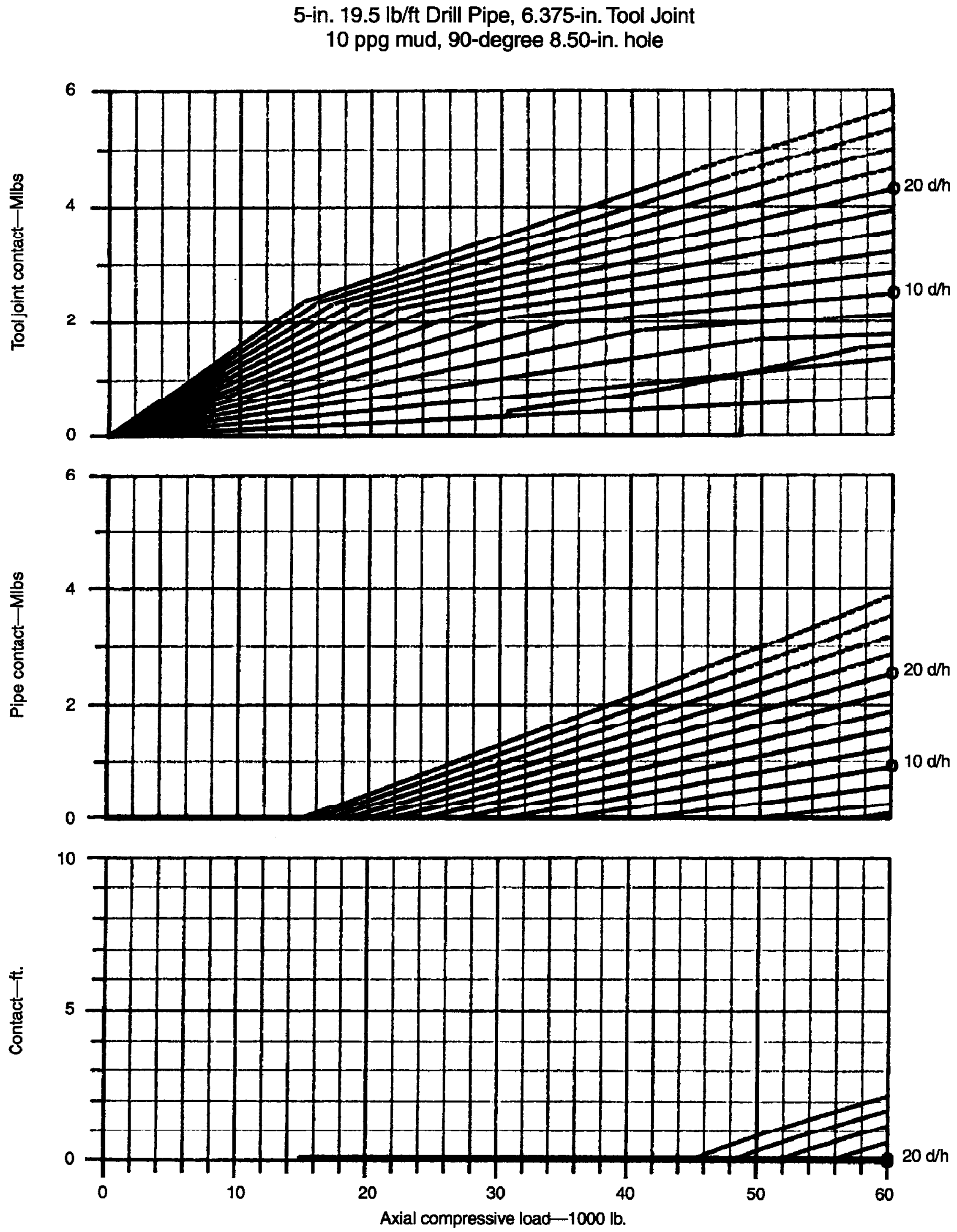


Figure 69b—Lateral Contact Forces and Length

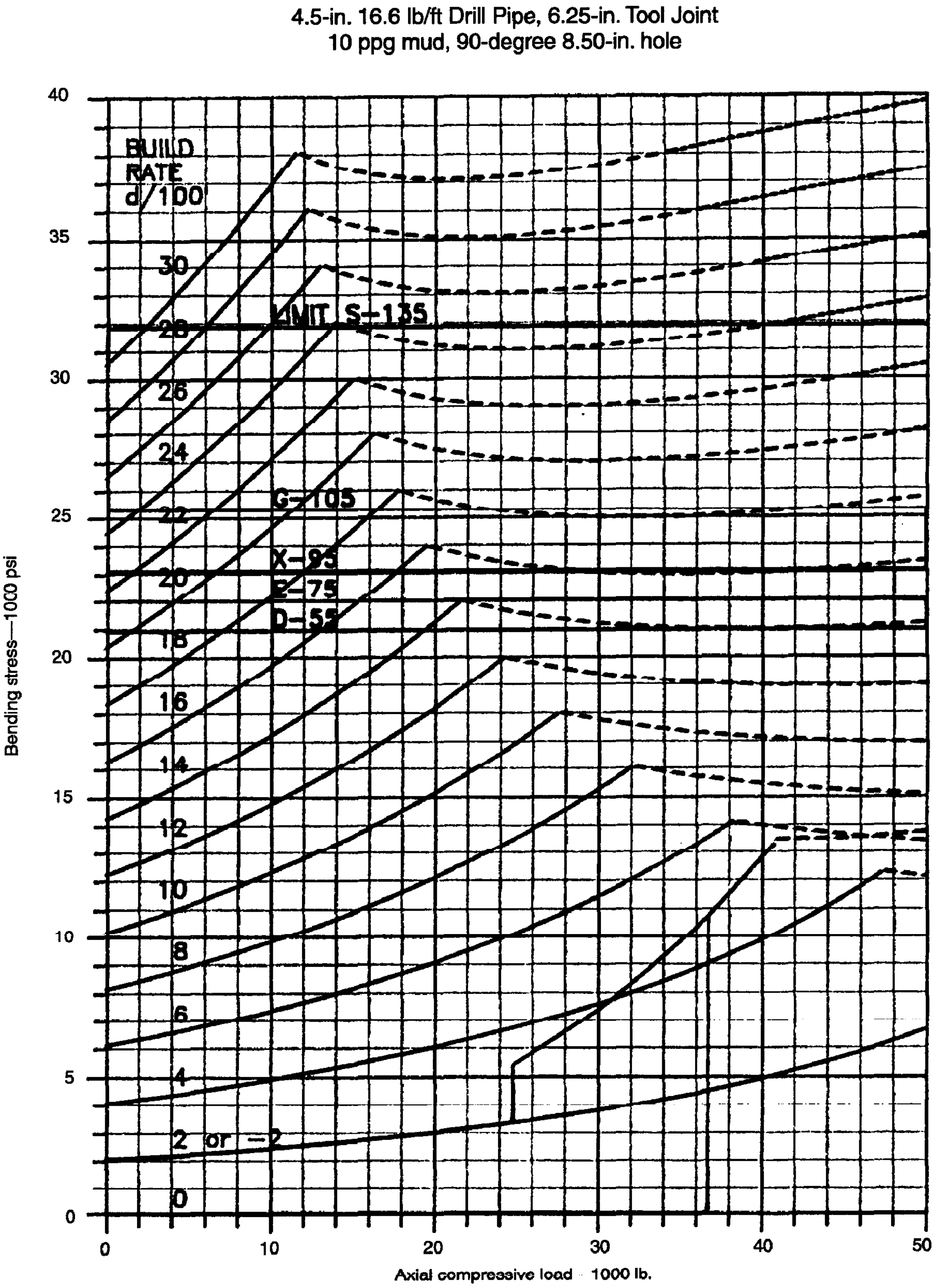


Figure 70a—Bending Stress and Fatigue Limits

4.5-in. 16.6 lb/ft Drill Pipe, 6.25-in. Tool Joint
10 ppg mud, 90-degree 8.50-in. hole

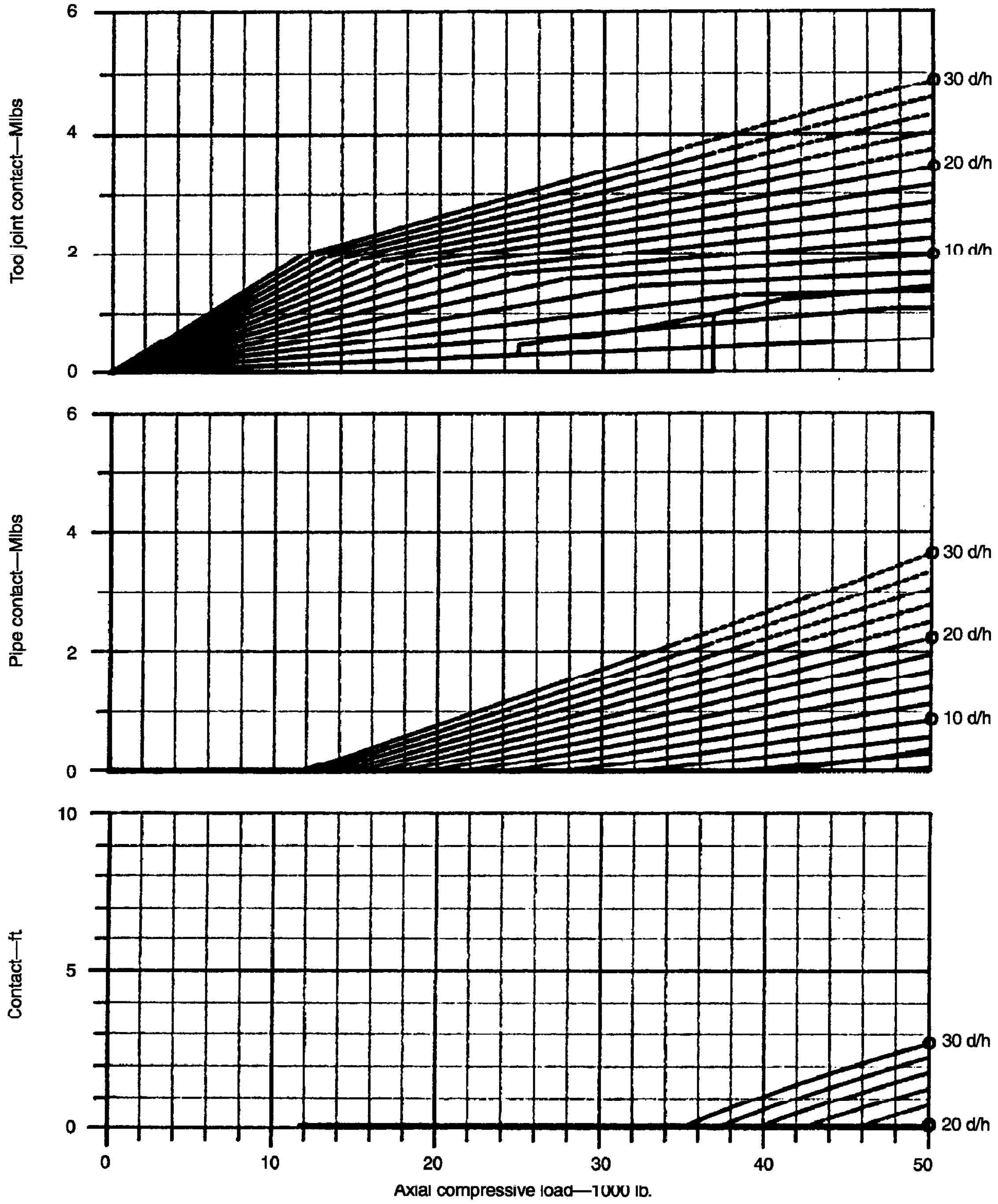


Figure 70b—Lateral Contact Forces and Length

4.0-in. 14.0 lb/ft Drill Pipe, 5.25-in. Tool Joint
with 10 lb/gal mud in a 6.75-in. hole

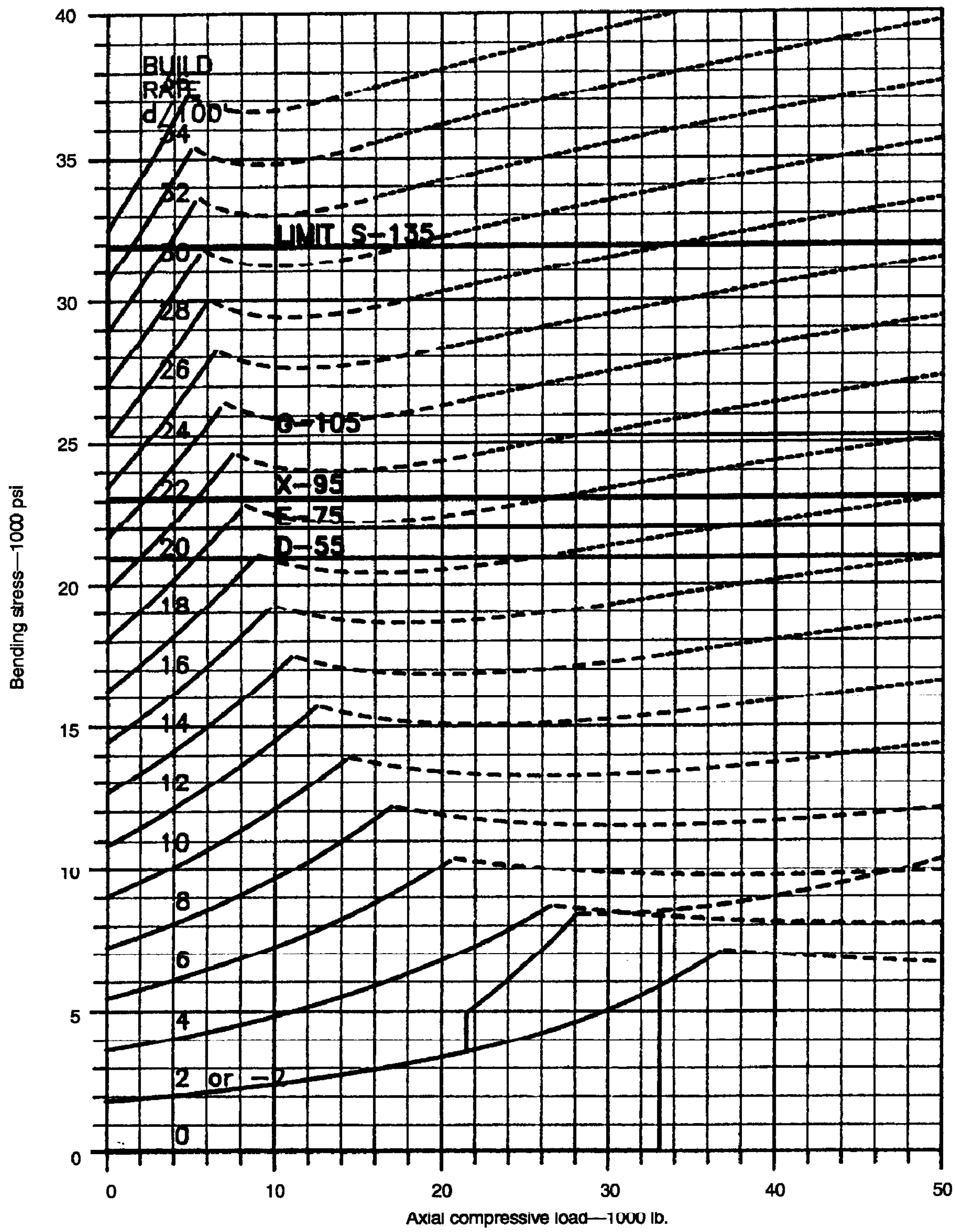


Figure 71a—Bending Stress and Fatigue Limits

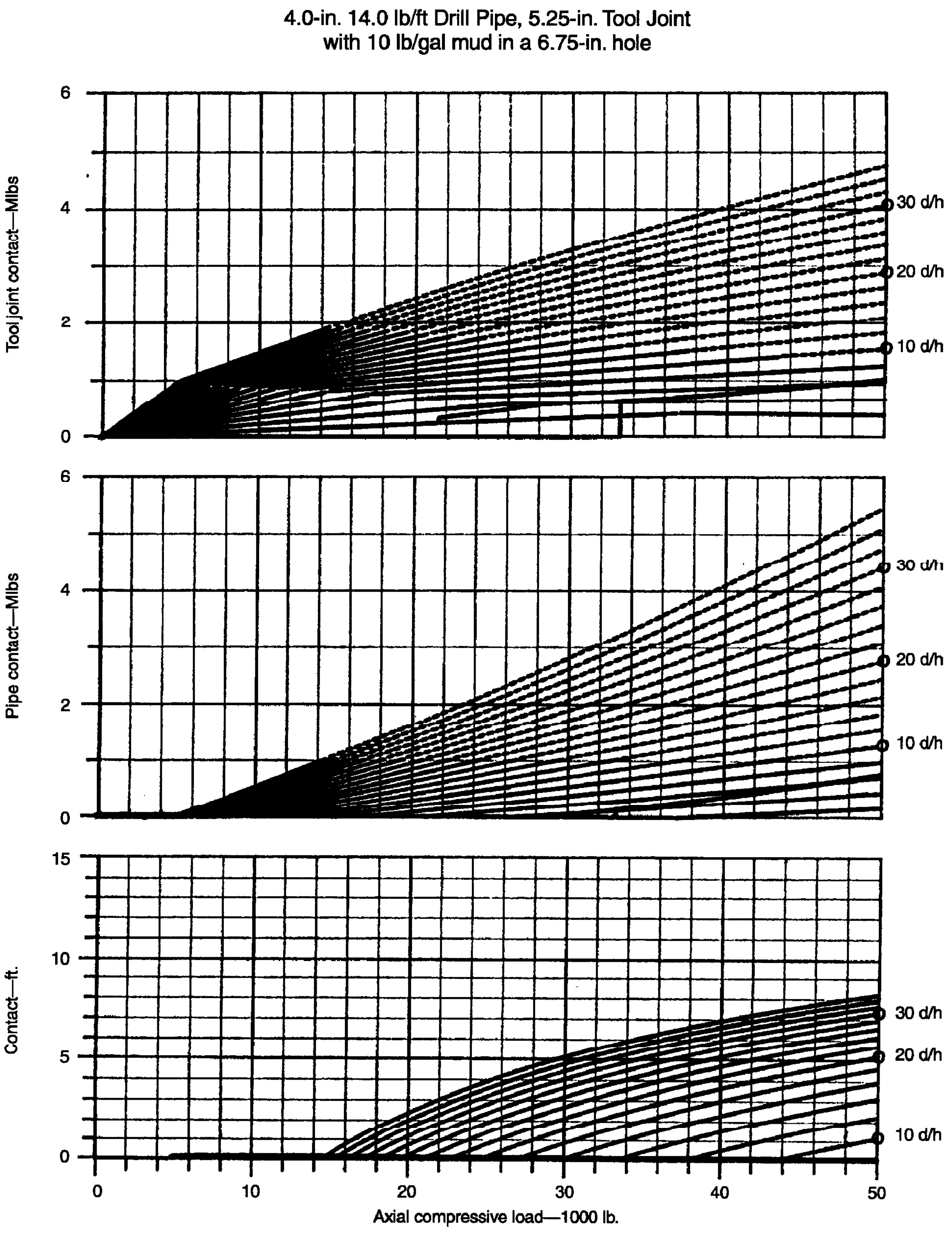


Figure 71b—Lateral Contact Forces and Length

3.5-in. 13.3 lb/ft Drill Pipe, 4.75-in. Tool Joint
with 10 lb/gal mud in a 6-in. hole

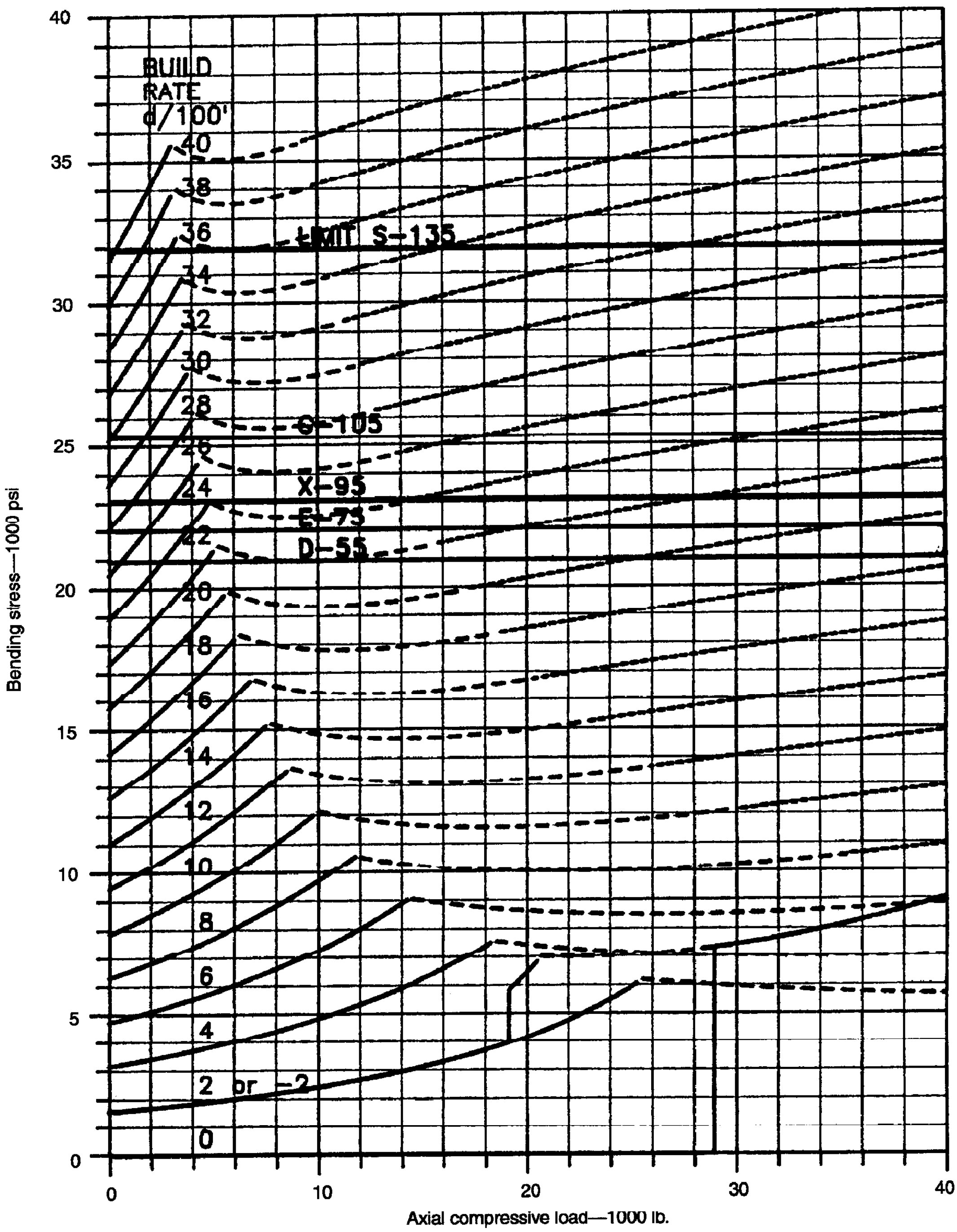


Figure 72a—Bending Stress and Fatigue Limits

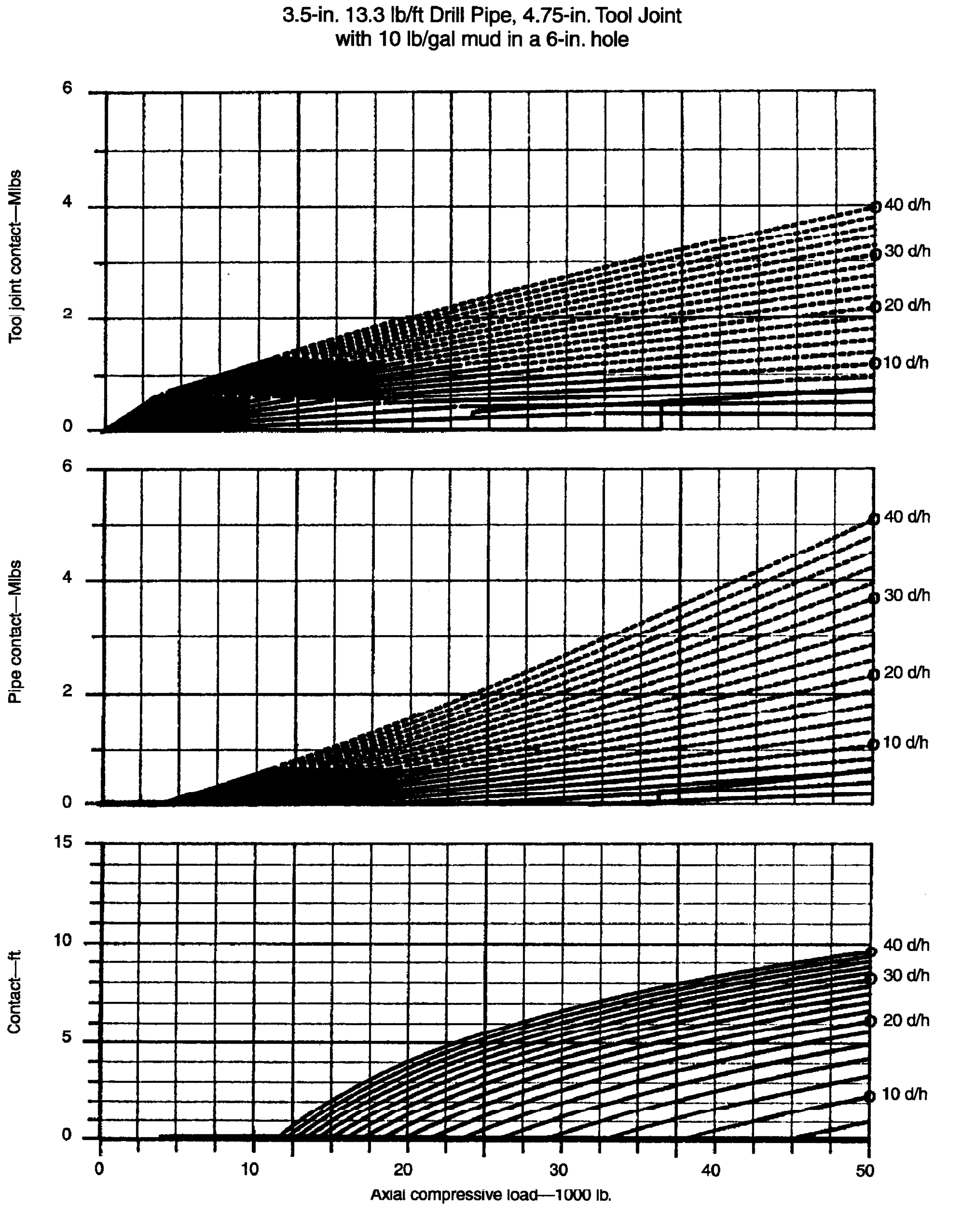


Figure 72b—Lateral Contact Forces and Length

2.875-in. 10.4 lb/ft Drill Pipe, 4.125-in. Tool Joint
with 10 lb/gal mud in a 4.75-in. hole

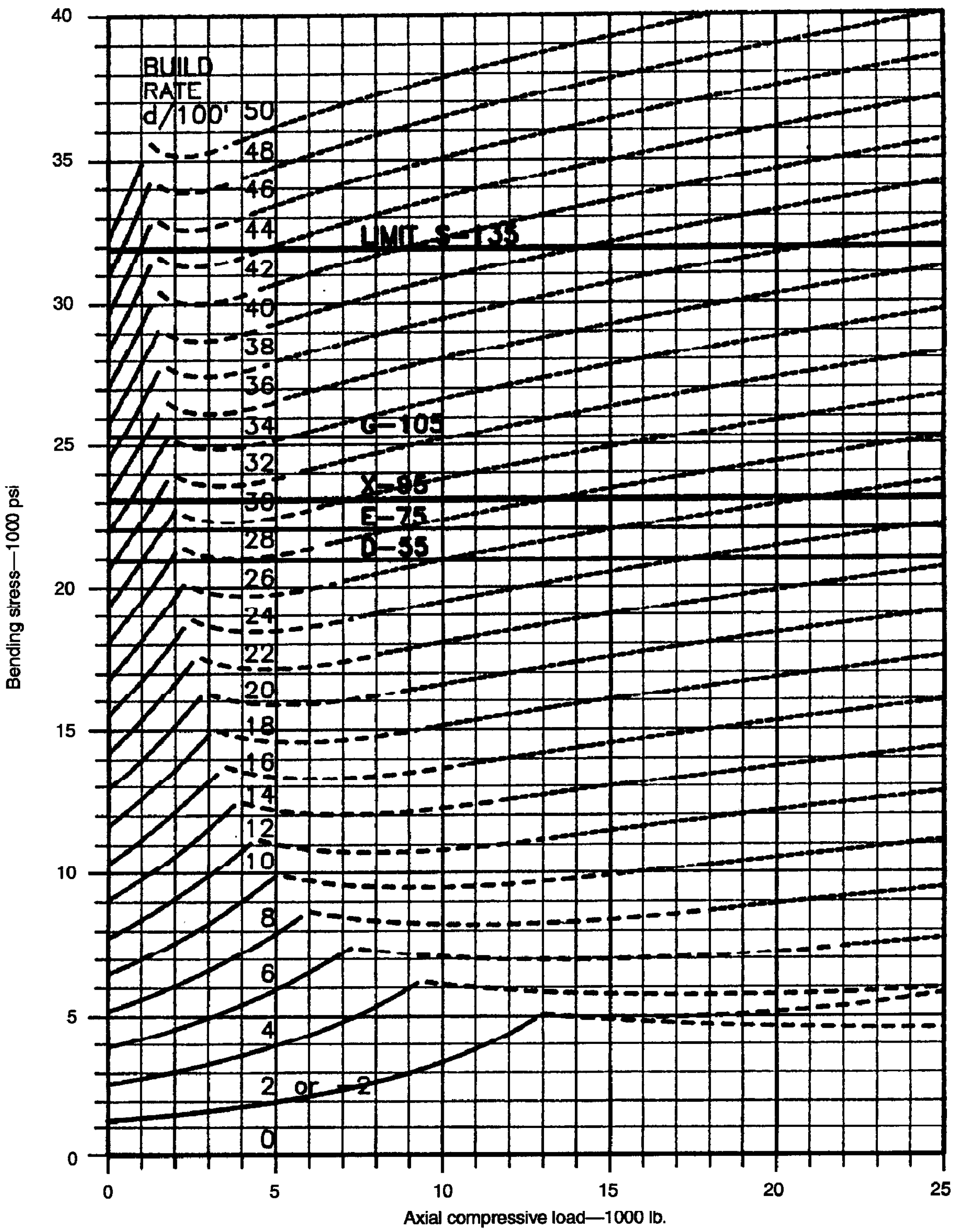


Figure 73a—Bending Stress and Fatigue Limits

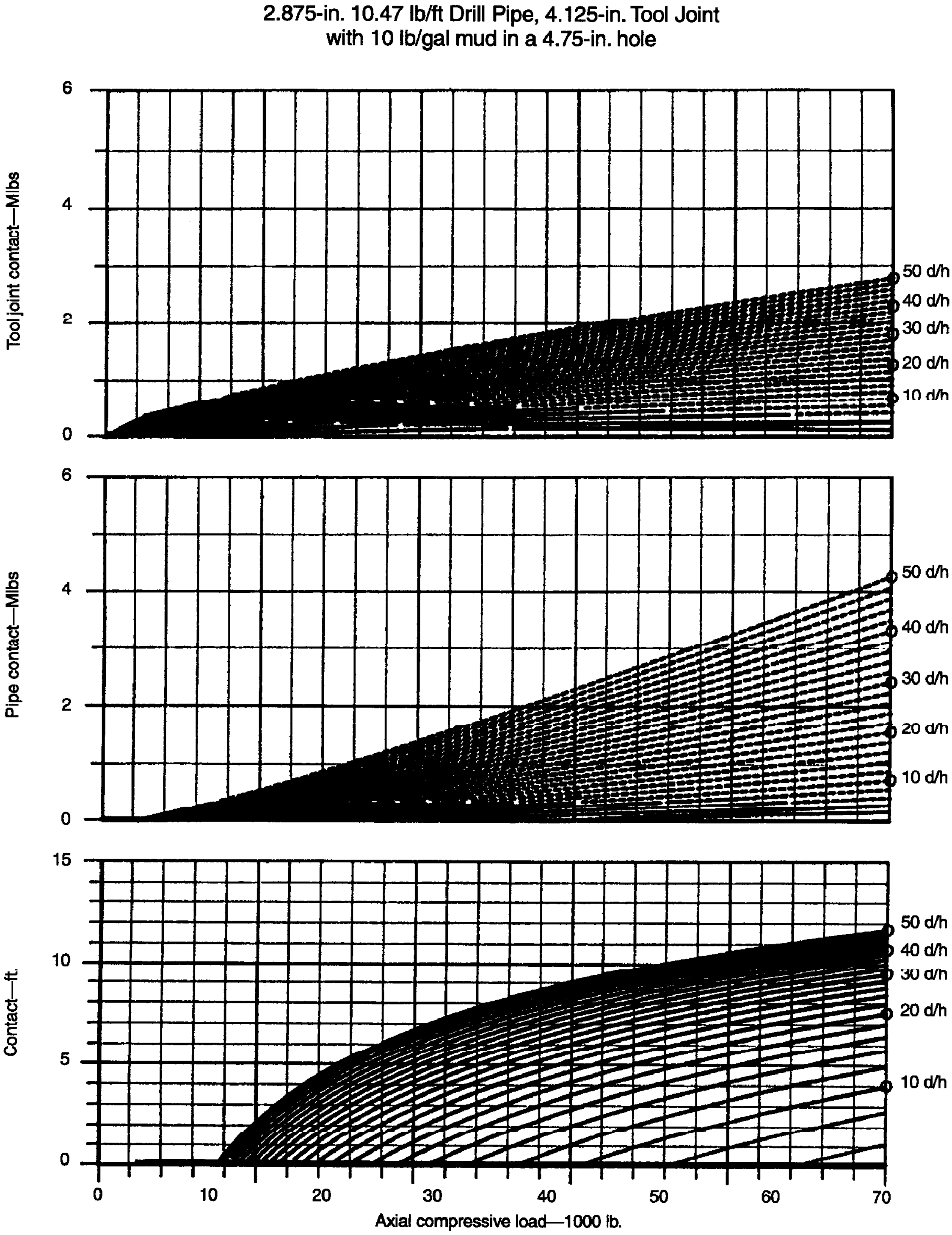


Figure 73b—Lateral Contact Forces and Length

2.375-in. 6.65 lb/ft Drill Pipe, 3.125-in. Tool Joint
with 10 lb/gal mud in a 3.875-in. hole

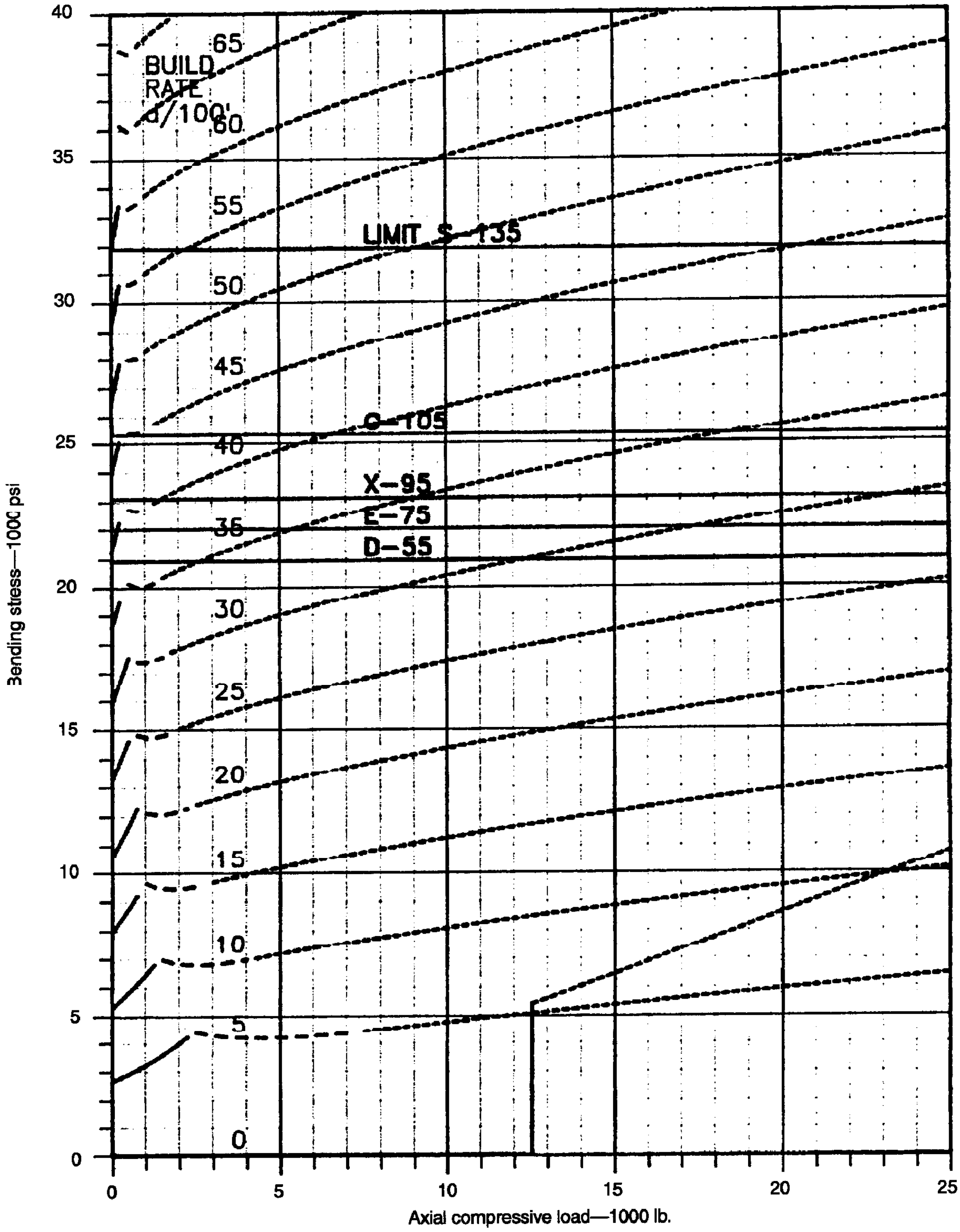


Figure 74a—Bending Stress and Fatigue Limits

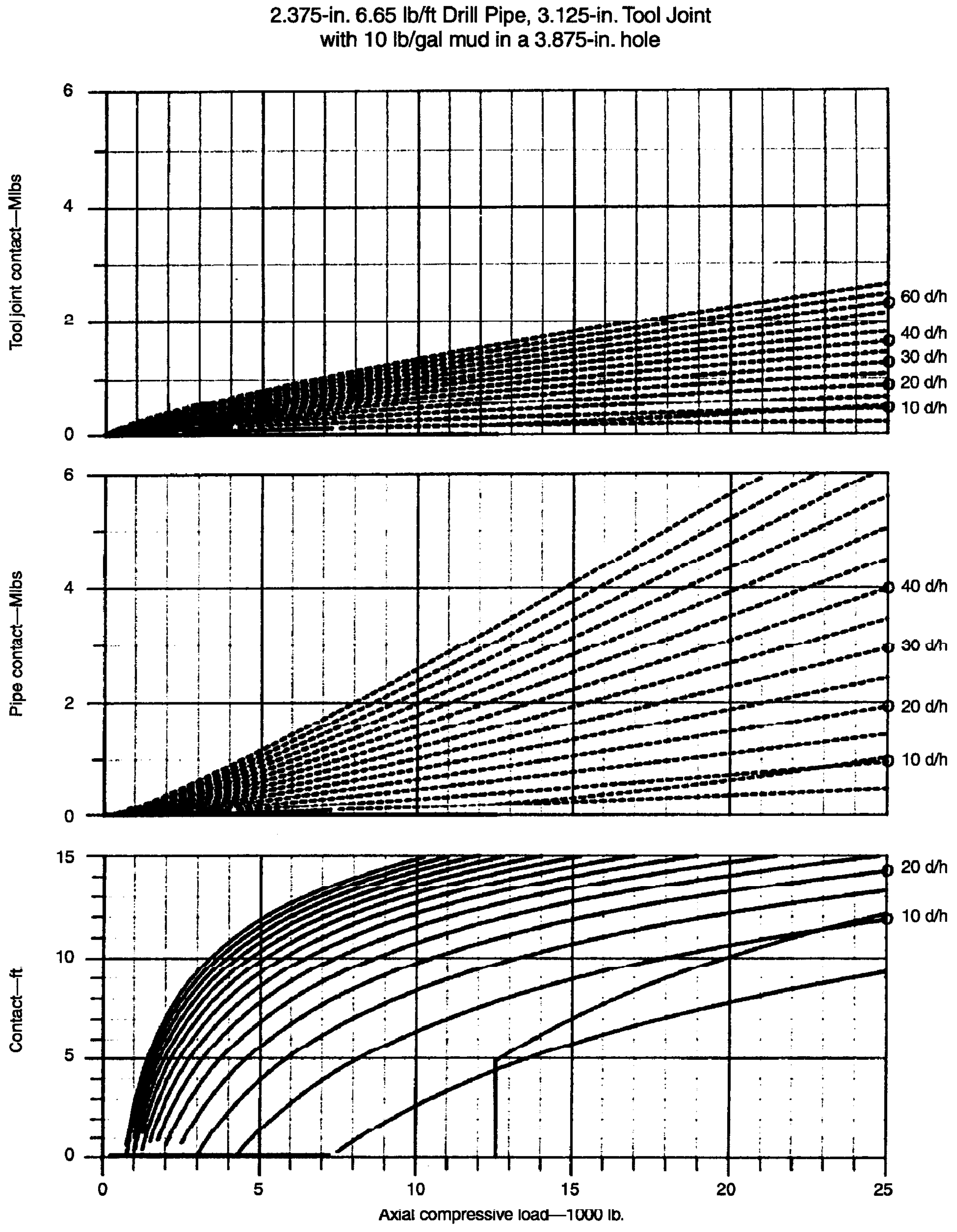


Figure 74b—Lateral Contact Forces and Length

plotted stress is shown as a dotted line. The plots also include the fatigue endurance bending stress limits from Section 11.5. The plots assume a 10 lb/gal mud, 90-degree 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.

11.4.1 Example

An $8^1/_2$-inch horizontal well will be drilled with 5-inch 19.50 lb/ft drill pipe with $6^3/_8$-inch tool joints in an $8^1/_2$-inch hole with 10 lb/gal mud. The maximum hole curvature will be 16 degrees per 100 feet. The horizontal interval will be drilled with surface rotation with loads up to 35,000 pounds. What grade of drill pipe is required for this example?

11.4.2 Solution

11.4.2.1 Figure 69a shows the bending stresses and fatigue limits for 5-inch, 19.5 lb/ft drill pipe, with $6^3/_8$-inch tool joints in a 90-degree, $8^1/_2$-inch hole with 10 lb/gal mud. For a hole curvature of 16-degrees-per-100-feet and an axial compressive load of 35,000 pounds, the maximum bending stress is 24,000 psi. A slightly higher bending stress of 25,000 psi is produced by a 24,000 pound axial load. The maximum bending stresses 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 pound axial load, the tool joint contact forces would be about 2,600 pounds with a 16 degrees per 100 foot curvature rate. The pipe body contact force will be about 600 pounds under a 35,000 pound axial load. For this curvature rate the contact will be at the center of the span.

11.4.2.2 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 condition produces point or wrap contact. If this is the case, we can compute an adjustment factor from the actual size of the tool joint and use that to compute an equivalent hole curvature to determine the correct bending stress.

11.4.3 Example

Determine the maximum bending stresses for 4-inch, 14 lb/ft drill pipe with $5^3/_8$-inch tool joints in 20-degrees-per-100-foot hole curvature, and a 20,000 pound axial compressive load.

11.4.4 Solution

Figure 71a shows the bending stresses for 4-inch, 14 lb/ft drill pipe, with $5^1/_4$-inch tool joints. The actual tool joint outside diameter is $5^3/_8$-inch or $^1/_8$-inch 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, $^1/_8$-inch larger than nominal, Figure 75 shows that with 4-inch drill pipe the curvature adjustment factor is 1.1. To determine the maximum bending stress for the $^1/_8$ inch larger tool joint in a 20-degrees-per-100-foot hole curvature at 20,000 pound load, multiply the actual hole curvature by the adjustment factor, 20-degrees-per-100-feet times 1.1 = 22-degrees-per-100-feet. Using 22-degrees-per-100-feet at 20,000 pounds on Figure 71a, the bending stress for $5^3/_8$-inch tool joints is 24,500 psi.

11.5 FATIGUE LIMITS FOR API DRILL PIPE

R. P. Morgan and M. J. Roblin[35] developed a method of measuring fatigue limits from small drill pipe samples. Their technique preserves the effect of the "as produced" drill pipe hot finish surfaces. They reported on the testing of API Grades E75, X95, G105, and S135 as well as an experimental high strength tubular identified as V180. Their test program showed that the fatigue endurance limits for drillpipe correlate well with the tensile strength of the pipe. Table 21 shows some of the results of their test program.

Table 21—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.

Casner[36] has utilized their test results to determine the minimum fatigue endurance limits for API drillpipe that meets the API minimum strength requirements. See Table 22.

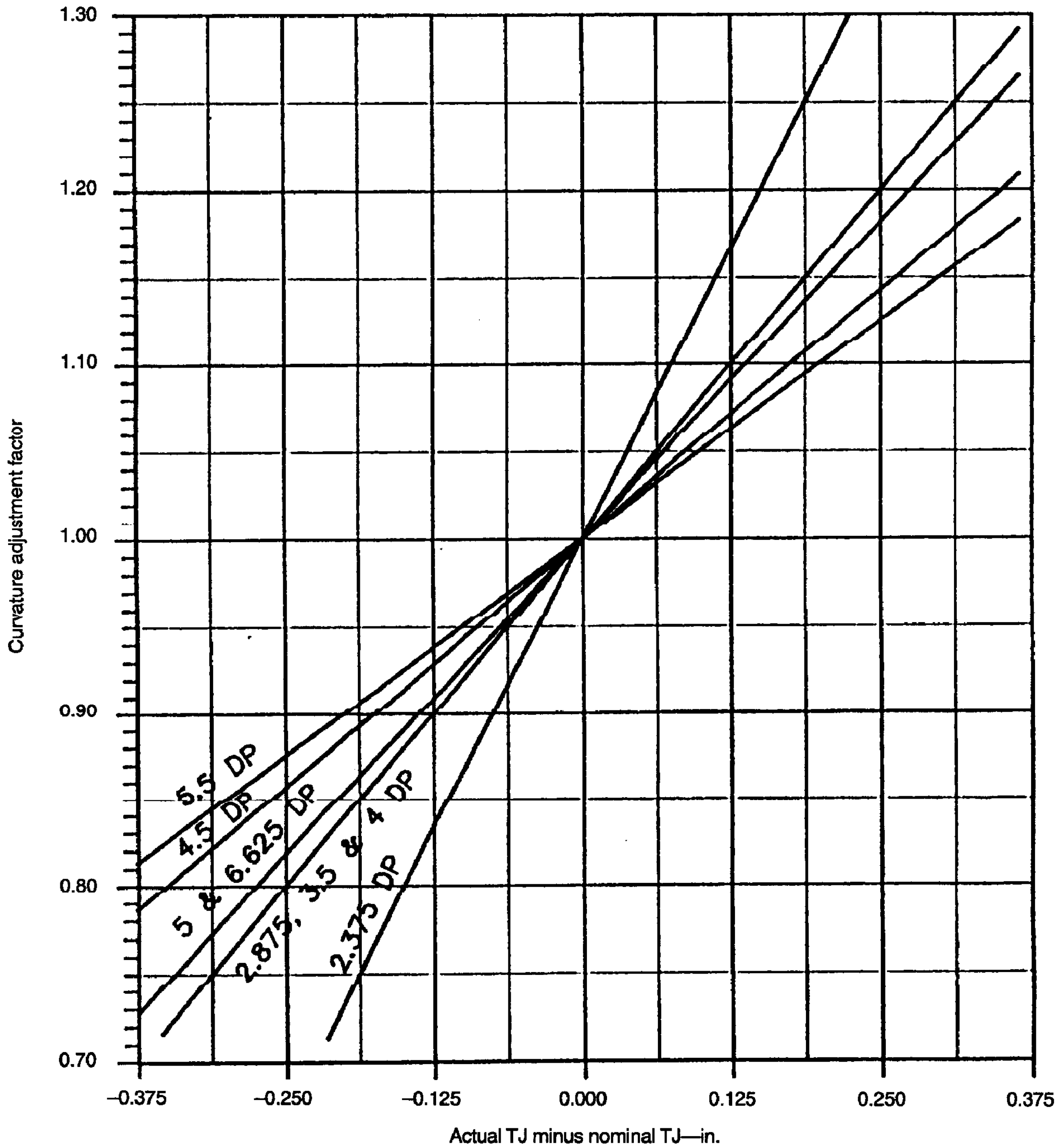


Figure 75—Hole Curvature Adjustment Factor To Allow for Differences in Tooljoint ODs

Table 22—Fatigue Endurance Limits Compressively Loaded Drill Pipe

Grade	API Minimum Yield Strength	API Minimum Tensile Strength	Minimum 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 23—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

11.6 ESTIMATING CUMULATIVE FATIGUE DAMAGE

An interesting alternative to operating below the fatigue endurance limits for drill pipe is to monitor the cumulative fatigue damage caused by rotating in high curvature intervals of the borehole and retire the pipe before failures occur. The concepts for tracking the cumulative fatigue damage have been well developed by Hansford and Lubinski.[37,38] 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. Figures 76 and 77 provide estimates of the median expected failure limits found by Morgan and Roblin and an estimate of the minimum failure limits of API drill pipe that has been manufactured to average API properties. We would expect that half of the drill pipe exposed to the limits shown in Figure 76 would fail. The minimum failure limits in Figure 77 should avoid fatigue failures on typical API drill pipe. 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 fatigues endurance limits from Table 21 were used to develop the stress versus revolutions to failure curves shown in Figure 76.

The equation is of the form: $S = \frac{TS}{N^x}$

where

S = bending stress limit, psi,
TS = tensile strength of pipe, psi,
N = revolutions to failure,
x = a fractional exponent of about D.1.

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 we computed the fatigued endurance limits for these tensile strengths. The SN curve was 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. The values used to prepare Figure 77 are summarized in Table 23.

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 limits. If, for example, the revolutions in a particular section of hole represent 20 percent of the predicted revolutions to failure for that dogleg severity it is judged that 20 percent of the fatigue life has been consumed. Figures 76 and 77 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 76. The appropriate minimum inspection levels can be judged from the minimum fatigue limits of Figure 77. 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.

Figures 78 through 80 are plots of the bending stresses for $3^1/_2$-inch, $2^7/_8$-inch, and $2^3/_8$-inch drill pipe in highly curved boreholes. The bending stresses determined from these plots are used to determine the total number of revolutions that the pipe can withstand from Figures 76 and 77. These are then compared to the observed number of revolutions to determine the level of fatigue damage. The ratio of the revolutions to the predicted median revolutions to failure defines the fraction of the drillpipe fatigue life consumed in drilling through the high curvature hole.

11.6.1 Example

An example of the cumulative fatigue damage calculations is given by the following. Consider drilling a 500-foot horizontal hole below a 100-foot radius build curve with $3^1/_2$-inch S135 drill pipe. Drill pipe will be rotated at 30 RPM with 10,000 WOB, and is expected to drill at 15 feet per hour. The equivalent build rate is given by:

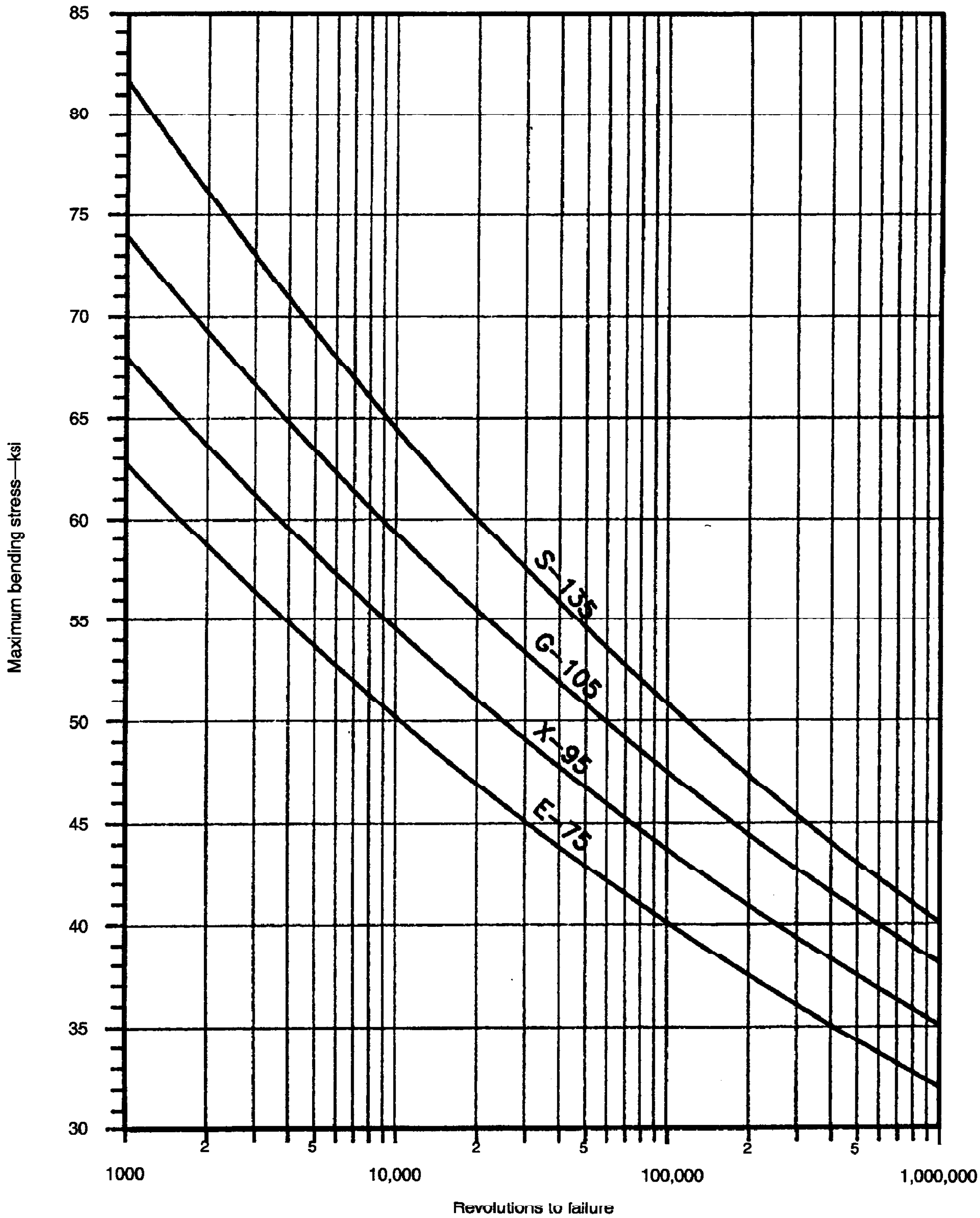


Figure 76—Median Failure Limits for API Drillpipe Noncorrosive Service

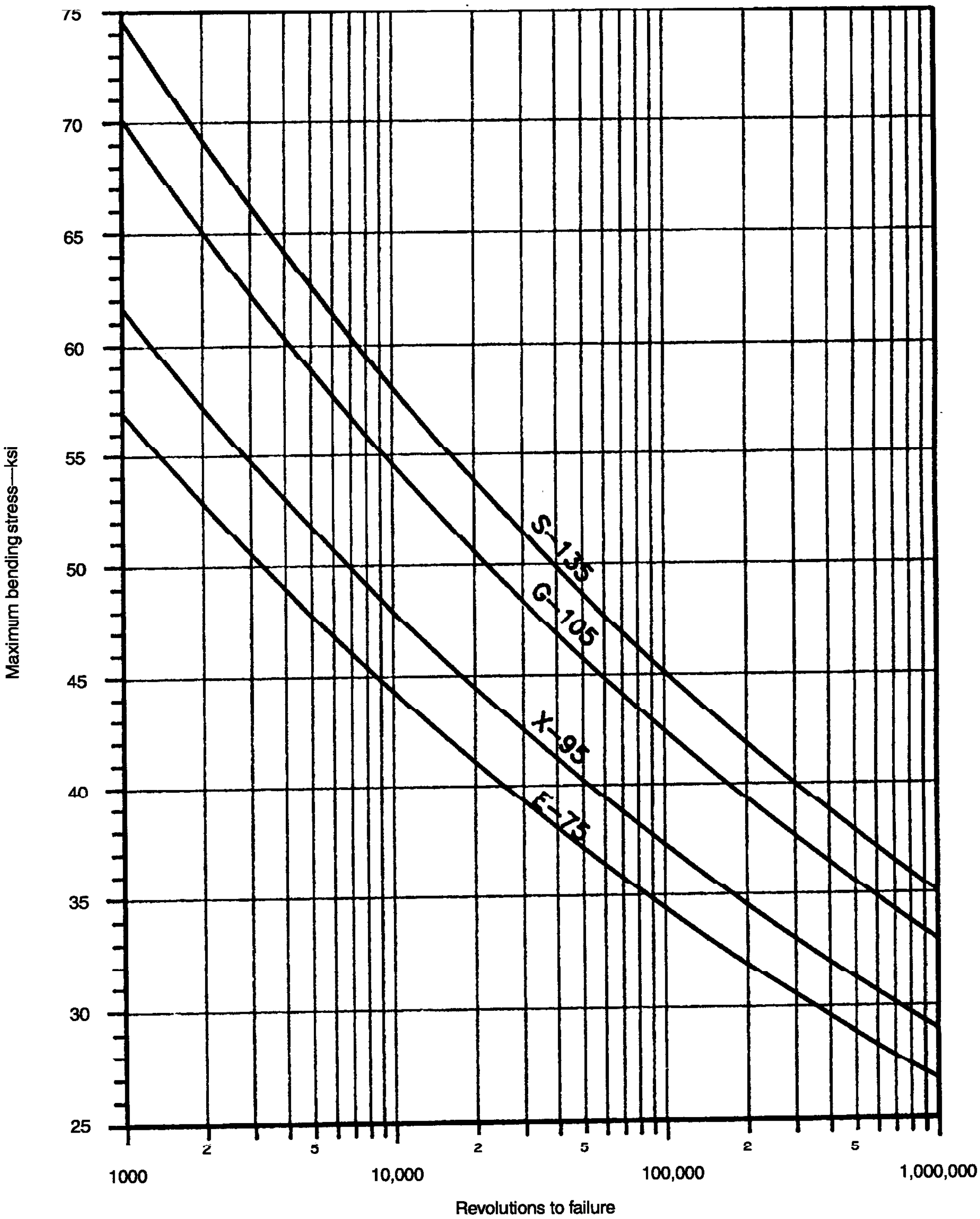


Figure 77—Minimum Failure Limits for API Drillpipe Noncorrosive Service

$$B = \frac{5730}{R}$$

where

B = build rate, degrees/100 ft.,
R = build radius, ft.

For this case: $B = \frac{5730}{100} = 57.3$ degrees/100ft.

11.6.2 Solution

Figure 78a shows that at 10,000 pound axial compressive load and a 57-degrees-per-100-foot build rate the maximum bending stress is 50,000 psi. Figure 76 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 77 predict that S135 pipe can be rotated 39,000 revolutions without failure.

The number of revolutions of exposure is given by:

$$N = \frac{60 \times L \times RPM}{ROP}$$

where

N = revolutions of exposure,
L = length of high curvature hole, ft.,
RPM = rotary speed, rev/min,
ROP = penetration rate, ft/hr.

The length of L of our 90 degree build curve is equal to:

$$L = \frac{\pi}{2} \times R = \frac{\pi}{2} \times 100 = 157 \text{ ft.}$$

Therefore, the number of revolutions of exposure for the pipe that is rotated through the build curve is equal to:

$$N = \frac{60 \times 157 \times 30}{15} = 18{,}850 \text{ revolutions}$$

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 percent 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 represents 48 percent 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.

11.7 BENDING STRESSES ON BUCKLED DRILL PIPE

The bending stresses on buckled drill pipe must 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:

$$B_{buc} = \frac{F \times h_c \times 57.3 \times 12 \times 100}{2 \times E \times I}$$

$$B_{buc} = \frac{17190 \times F(D_h - D_{tj})}{E \times I}$$

where

B_{buc} = curvature of buckled pipe, °/100 ft.,
F = axial load, lb.,
D_h = hole diameter, in.,
D_{tj} = tool joint OD, in.,
h_c = radial clearance
$= \frac{D_h - D_{tj}}{2}$, in.,
I = area moment of inertial of pipe, in.^{4},
E = Young's modulus, psi
= 30×10^6 psi for steel.

This curvature can be used in place of the hole curvature in the equations covered in Section 11.4.

12 Special Service Problems

12.1 SEVERE DOWNHOLE VIBRATION

Downhole vibration is inevitable. In many cases, low levels of vibration go undetected and are harmless. However, severe downhole vibration can cause drillstring fatigue failure (washout/twist-off), crooked drillstrings, 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 drillstring with the wellbore. The drillstring response to these excitation sources is very complex.

Vibration can induce three components of motion in the drillstring and the bit, namely: axial (motion along drillstring 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

(Text continued on page 108.)

3.5 in. 13.3 lb/ft Drill Pipe, 4.75 in. Tool Joint
with 10 lb/gal mud in a 6 in. hole

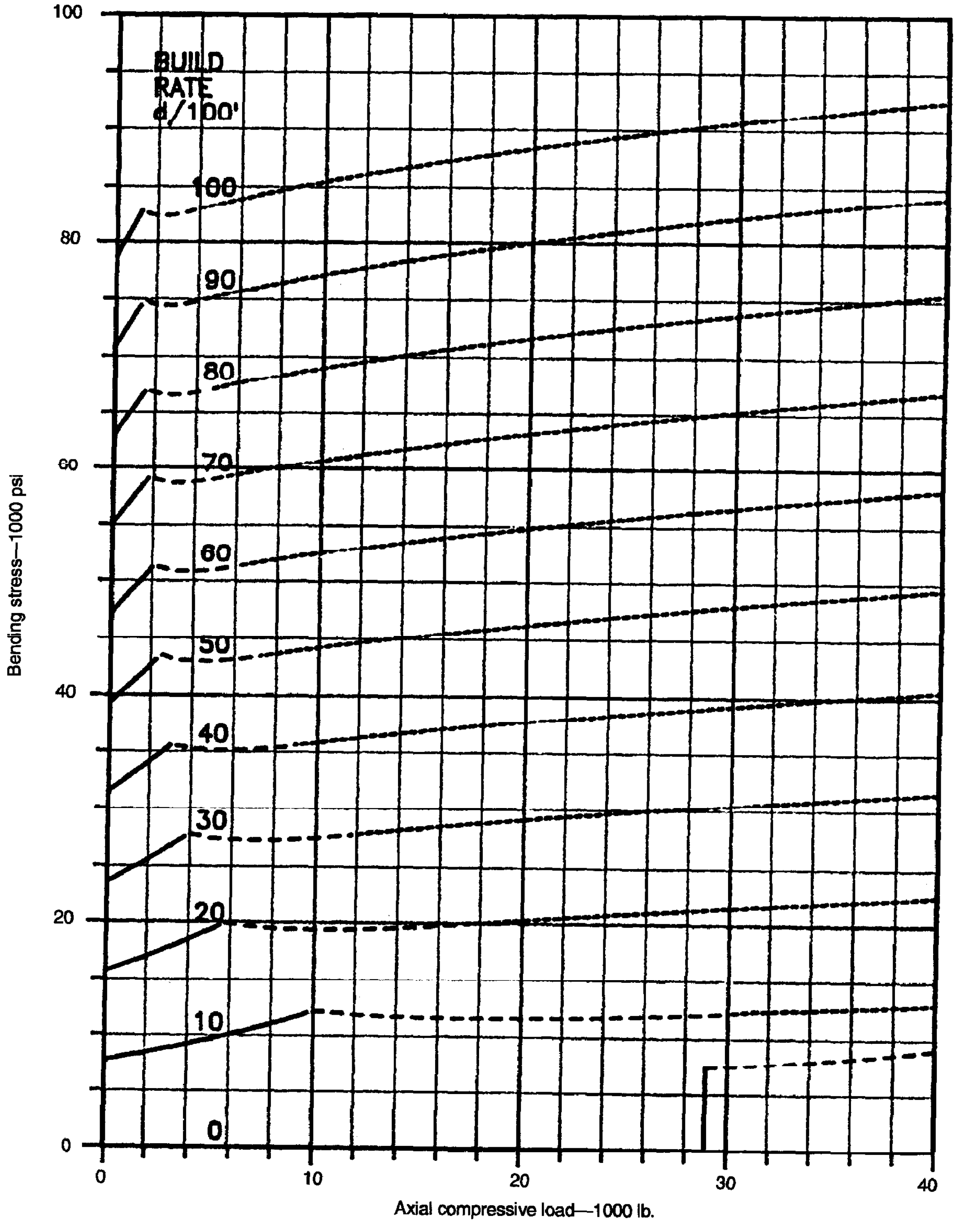


Figure 78a—Bending Stress for High Curvatures

3.5 in. 13.3 lb/ft Drill Pipe, 4.75 in. Tool Joint
with 10 lb/gal mud in a 6 in. hole

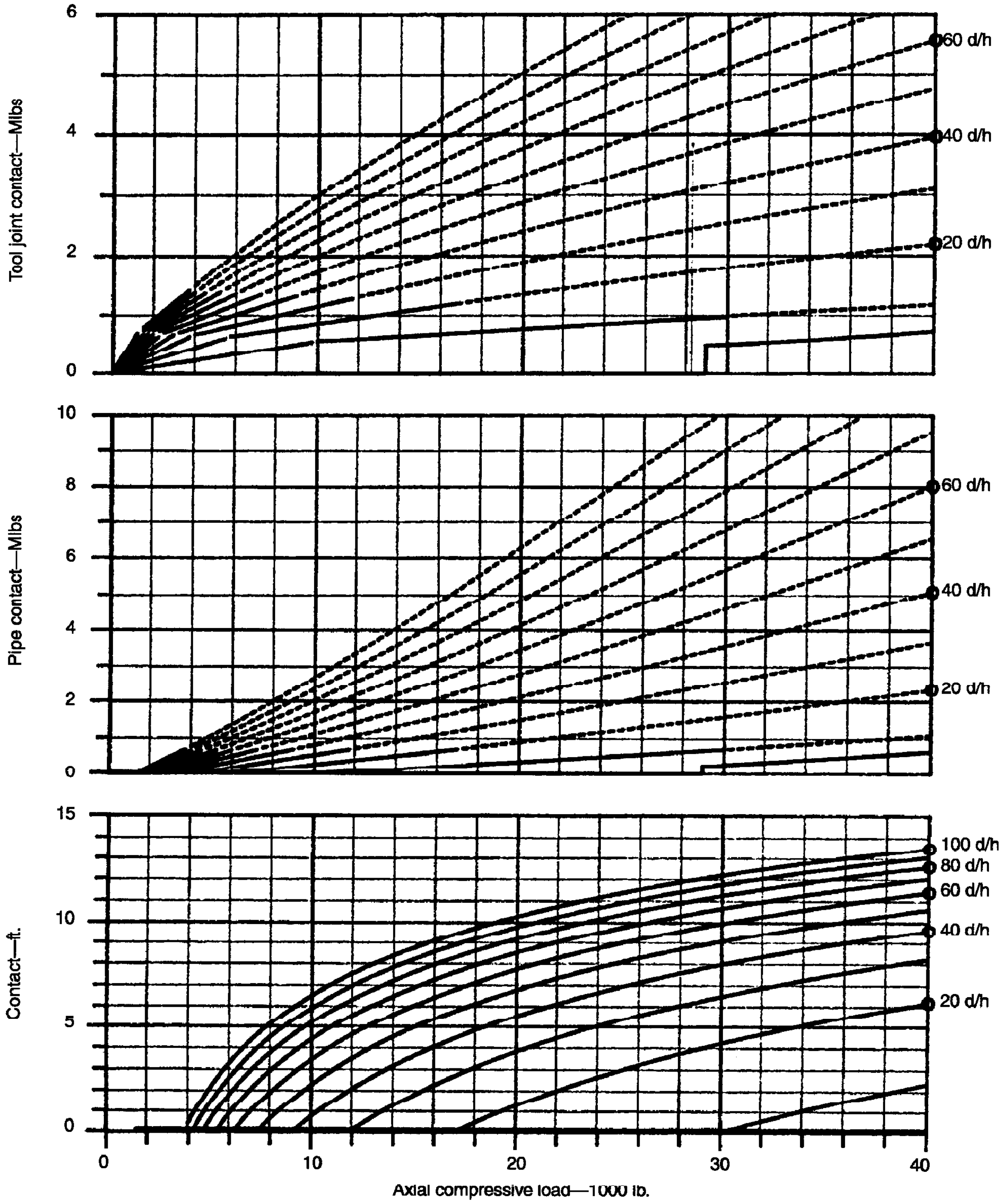


Figure 78b—Lateral Contact Forces and Length

2.875 in. 10.4 lb/ft Drill Pipe, 4.125 in. Tool Joint
with 10 lb/gal mud in a 4.75 in. hole

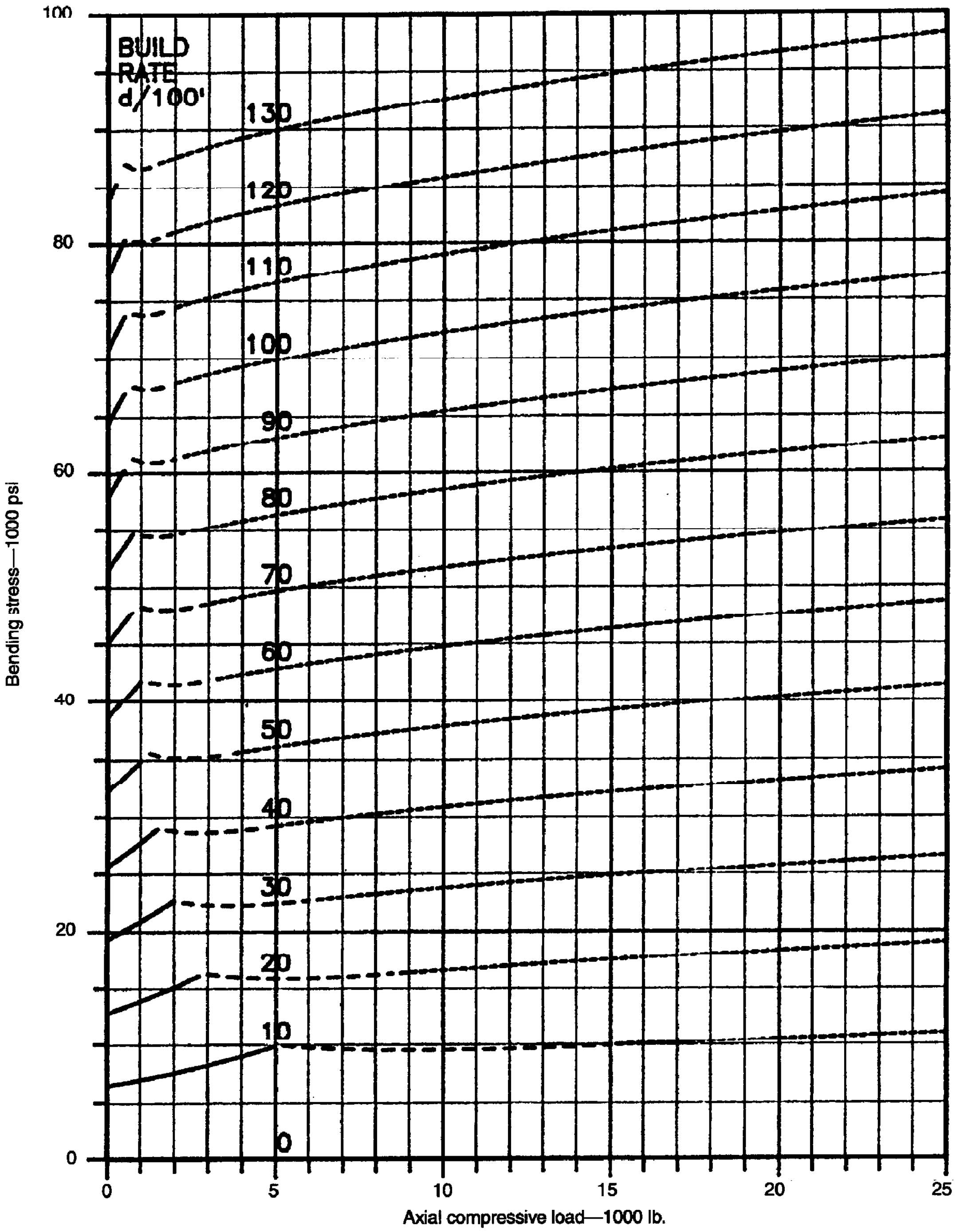


Figure 79a—Bending Stress for High Curvatures

2.875 in. 10.4 lb/ft Drill Pipe, 4.125 in. Tool Joint
with 10 lb/gal mud in a 4.75 in. hole

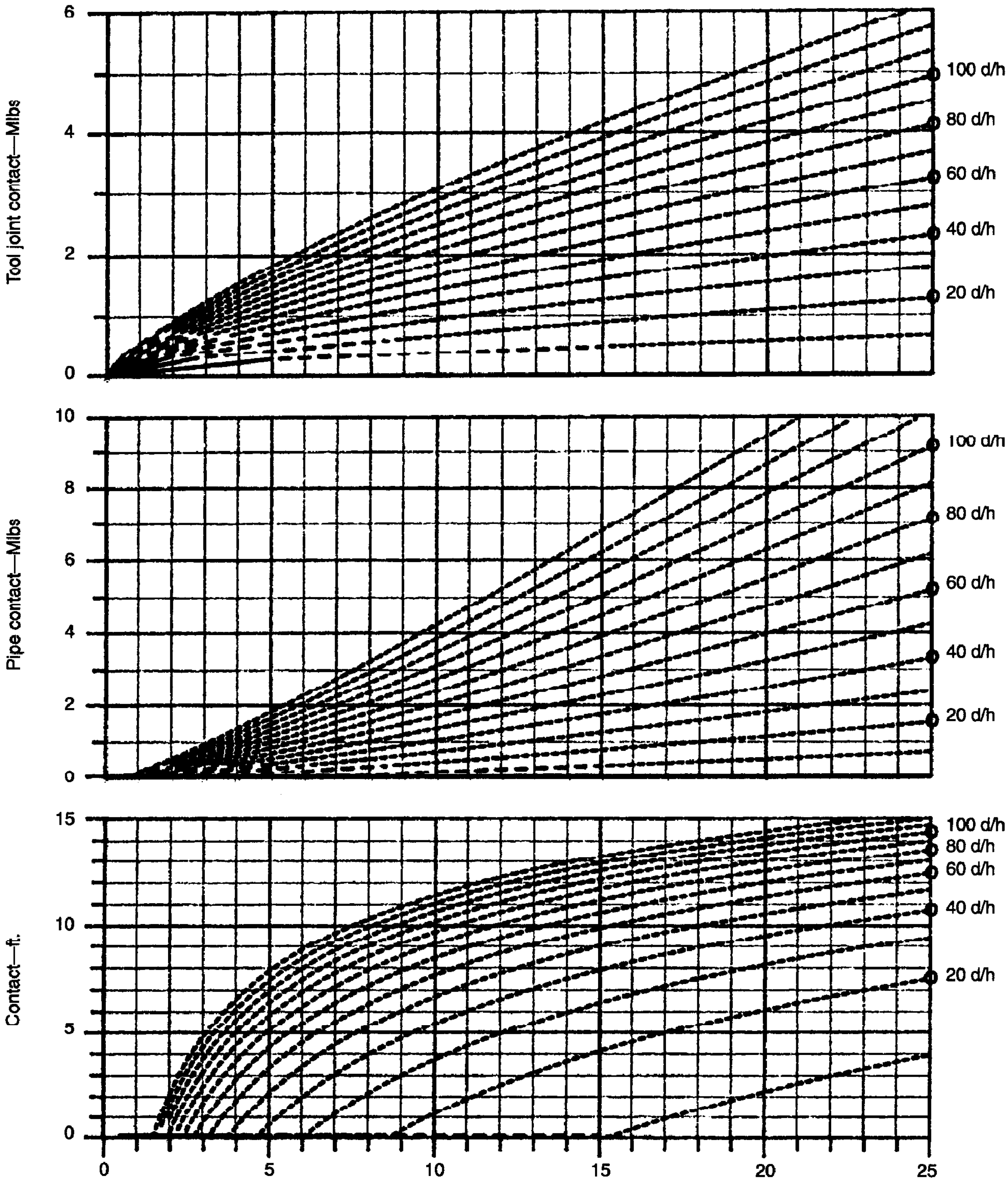


Figure 79b—Lateral Contact Forces and Length

2.375 in. 6.65 lb/ft Drill Pipe, 3.250 in. Tool Joint
with 10 lb/gal mud in a 4.00 in. hole

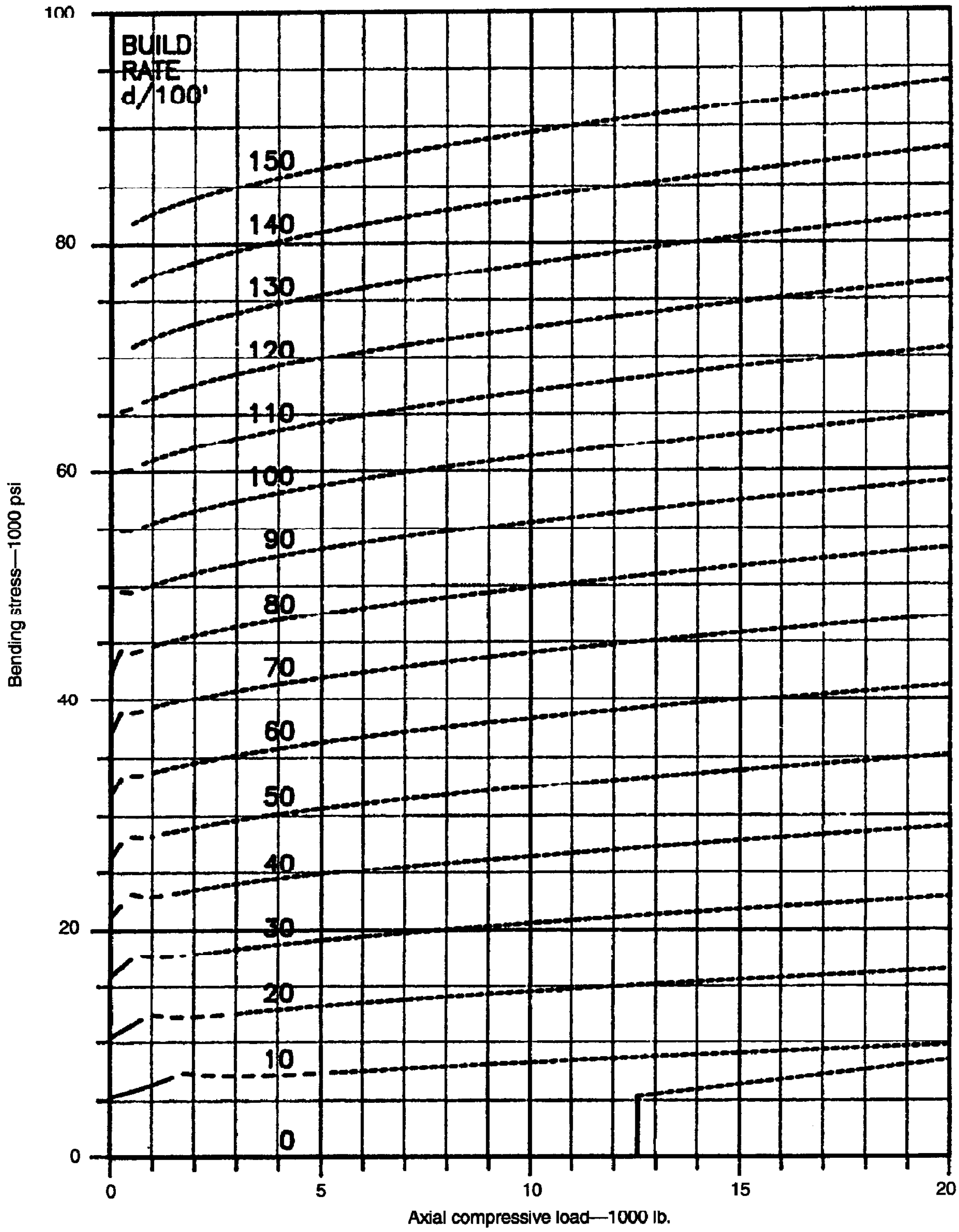


Figure 80a—Bending Stress for High Curvatures

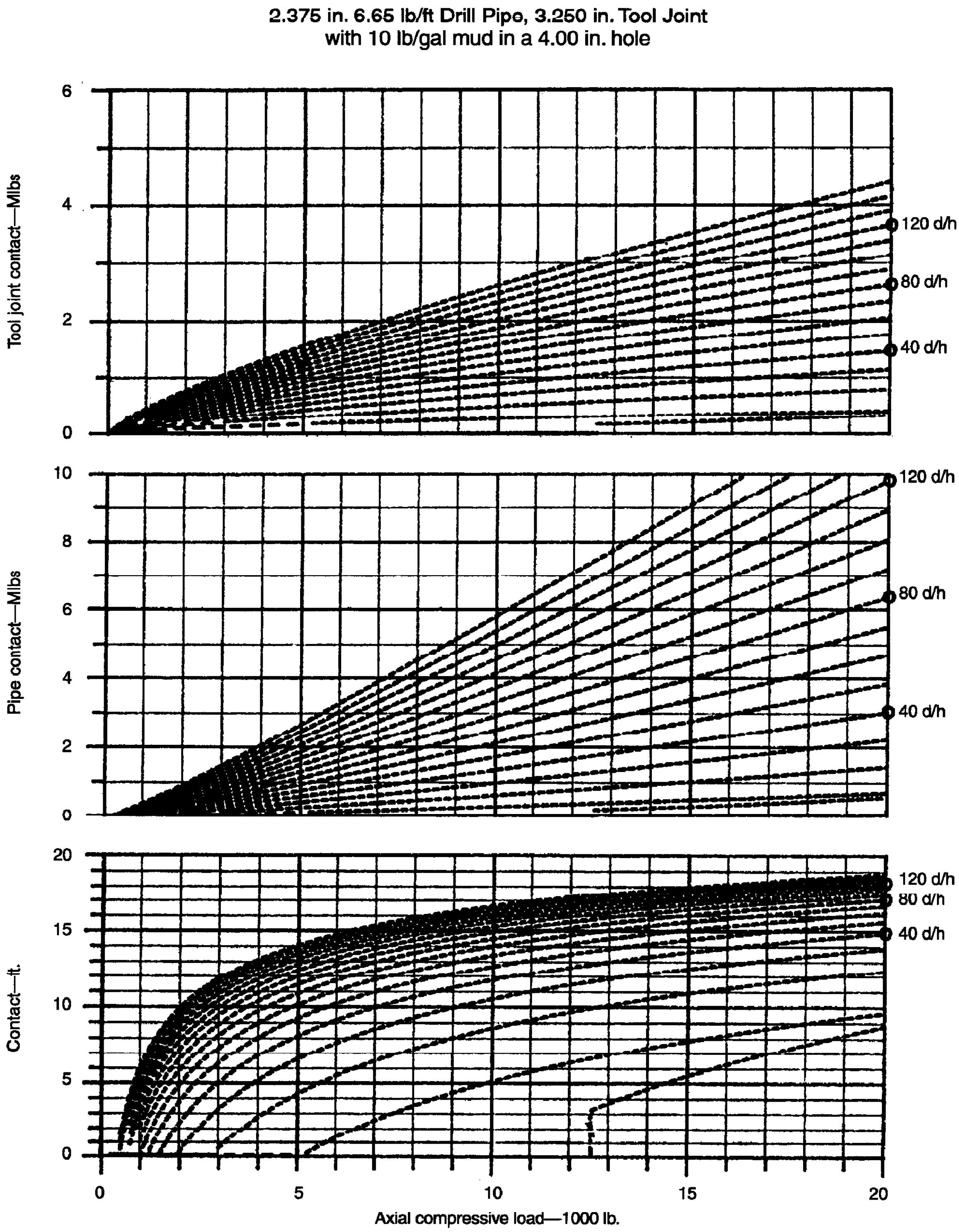


Figure 80b—Lateral Contact Forces and Length

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.

There are a number of mechanisms which can cause severe downhole vibration. For mechanisms, their symptoms, and methods of control are described below:

a. Slap stick:

1. Mechanism—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 the primary torsional vibrations in the string.

2. Symptoms—surface torque fluctuations >15 percent of average (below 1 Hz or stalling). increased MWD shock counts, cutter impact damage, drillstring washouts/twist-offs, connection over-torque or back-off.

Note: 1 Hz is one cycle per second.

3. Actions—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.

b. Drillstring whirl:

1. Mechanism—the BHA (or drillpipe) gears around the borehole. The violent whirling motion slams the drillstring against the borehole. The mechanism can cause torsional and lateral vibrations.

2. Symptoms—drillstring washouts/twist-offs, localized tool joint and/or stabilizer wear, increased average torque, 5 to 20Hz lateral vibrations even if bit off-bottom.

3. Actions—lift bit off bottom and stop rotation, then reduce RPM, avoid drill collar weight in excess of 1.15 to 1.25 times WOB, use packed hole assembly, reduce stabilizer rotational drag, adjust stabilizer placement, modify mud properties, consider drilling with a downhole motor.

c. Bit whirl:

1. Mechanism—eccentric rotation of the bit about a point other than its geometric center caused by bit/wellbore gearing (analogous to a planetary gear). This mechanism induces high frequency lateral and torsional vibration of the bit and drillstring.

2. Symptoms—cutter impact damage. uneven bit gauge wear, over-gauge hole, reduce ROP, 10 to 50 Hz lateral/torsional vibrations.

3. Actions—lift bit off bottom and stop rotation, then reduce RPM and increase WOB, consider changing bit (flatter profile, anti-whirl), use slow RPM when tagging bottom and when reaming. pickoff bottom before stopping rotation, use stabilized BHA with full gauge near-bit stabilizer or reamer.

d. Bit bounce:

1. Mechanism—large weight-on-bit fluctuations causing the bit to repeatedly lift off and impact the formation. This mechanism often occurs when drilling with roller cone bits in hard formations.

2. Symptoms—large axial 1 to 10 Hz vibrations (shaking of hoisting equipment), large WOB fluctuations, cutter and/or bearing impact damage, fatigue cracks, reduced ROP.

3. Actions—run shock sub or hydraulic thruster, adjust WOB/RPM, consider changing bit style, change length of BHA.

e. Other mechanisms—some field data and theoretical studies indicate that certain "critical speeds" exist which excite resonant vibrations. Previous editions of Recommended Practice 7G gave formulas and graphs for predicting these critical speeds. However, severe vibrations have been routinely measured at RPM other than those give by these simple calculations.

12.2 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 sections, 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.

12.3 PULLING ON STUCK PIPE

It is normally not considered good practice to pull on stuck drill pipe beyond the minimum tensile yield strength for the size, grade, weight, and classification of the pipe in use (see Tables 4, 6, and 21). 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.

12.3.1 Example I (see A.6 for derivation):

Determine the stretch in a 10,000 foot string of drill pipe freely suspended in 10 lb/gal drilling fluid.

$$e = \frac{L_1^2}{9.625 \times 10^7} [65.44 - 1.44\ W_g] \quad (22)$$

$$= \frac{10{,}000^2}{9.625 \times 10^7} [65.44 - 1.44\ \times 10]$$

$$= 53.03 \text{ in.},$$

where

L_1 = length of free drill pipe, feet,
W_g = weight of drilling fluid, lb/gal,
e = stretch, inches.

12.3.2 Example II (see A.4 for derivation):

Determine the free length in a 10,000 foot string of 4½-in. OD 16.60 lb/ft drill pipe which is stuck, and which stretches 49 in. due to a differential pull of 80,000 lbs.

$$L_1 = \frac{735{,}294 \times e \times W_{dp}}{P} \quad (23)$$

$$= \frac{735{,}294 \times 49 \times 16.60}{80{,}000}$$

$$= 7476 \text{ ft.},$$

where

L_1 = length of free drill pipe, feet,
e = differential stretch, inches,
W_{dp} = weight of drill pipe, pounds per foot,
P = differential pull, pounds.

12.4 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.

12.5 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.)

12.6 ALLOWABLE HOOKLOAD AND TORQUE COMBINATIONS

Allowable hookloads and torque combinations for stuck drill strings may be determined by use of the following formula:

$$Q_T = \frac{0.096167 J}{D} \sqrt{Y_m^2 - \frac{P^2}{A^2}}, \quad (24)$$

where

Q_T = minimum torsional yield strength under tension, lb-ft.,
J = polar moment of inertia:
= $\frac{\pi}{32}(D_4 - d_4)$ for tubes,
D = outside diameter, inches,
d = inside diameter, inches,
Y_m = minimum unit yield strength, psi,
S_s = minimum unit shear strength, psi: ($S_s = 0.577\ Y_m$),
P = total load in tension, pounds,
A = cross section area.

An example of the torque which may be applied to the pipe which is stuck while imposing a tensile load is as follows:

Assume:

a. 3½-in. OD 13.30 lb Grade E drill pipe.
b. 3½-in. IF tool joints.
c. Stuck point: 4000 feet.
d. Tensile pull: 100,000 pounds.
e. New drill pipe.

Then:

$$Q_T = \frac{0.096167 \times 9.00}{3.5} \sqrt{(75{,}000)^2 - \frac{(100{,}000)^2}{(3.62)^2}}$$

Q_T = 17,216 lb-ft.

For further information on allowable hookloads, torque application, and pump pressure use, see Stall and Blenkarn: *Allowable Hook Load and Torque Combinations For Stuck Drill Strings.*[12]

12.7 BIAXIAL LOADING OF DRILL PIPE

The collapse resistance of drill pipe corrected for the effect of tension loading may be calculated by reference to Figure 81 and the use of formulas and physical constants contained in 12.8, 12.9, 12.10, and 12.11.

12.8 FORMULAS AND PHYSICAL CONSTANTS

The ellipse of biaxial yield stress shown in Figure 46 is for use in the range of plastic collapse only, and gives the relation between axial stress (psi) in terms of average yield stress (psi) and effective collapse resistance in terms of nominal plastic collapse resistance. This relationship is depicted in the following formula:

$r^2 + rz + z^2 = 1$, having solutions as follows:

$$z = -\frac{r + \sqrt{4 - 3r^2}}{2}, \text{ and} \tag{1}$$

$$r = -\frac{z + \sqrt{4 - 3z^2}}{2}, \tag{2}$$

where

$$r = \frac{\text{Effective collapse resistance under tension (psi)}}{\text{Nominal plastic collapse resistance (psi)}}, \tag{3}$$

$$z = \frac{\text{Total tensile loading (pounds)}}{\text{Cross section area} \times \text{Average yield strength}}, \tag{4}$$

Average yield strengths in psi are as follows:

Grade E75	85,000
Grade X95	110,000
Grade G105	120,000
Grade S135	145,000

12.9 TRANSITION FROM ELASTIC TO PLASTIC COLLAPSE

Material in the elastic range when under no tensile load, transfers to the plastic range when subjected to sufficient axial load. Axial loading, below the transition load, has no effect on elastic collapse. At transition point, the collapse resistance under tension equals the nominal elastic collapse, and also equals a tension factor (r) times collapse resistance as calculated from the nominal plastic formula.

Method: Determine values for both elastic and plastic collapse from applicable formulas in Appendix A, substitute in formula (3), 12.8 and solve for r. Then, solve formula (1), 12.8, for z. For the total tension (transition) load, substitute value of z in formula (4), 12.8.

12.10 EFFECT OF TENSILE LOAD ON COLLAPSE RESISTANCE

The effect of tensile load applies only to greater than transition load on normally elastic items, and to any load on plastic collapse items. In either case, the value determined from the plastic collapse formula (Appendix A) is to be modified.

Method: Substitute the tensile load value in formula (4), 12.8, to find a value for z. Substitute this value in formula (2), 12.8, to permit solution for r. Next, substitute the value of r in formula (3), 12.8, to obtain the effective collapse resistance under tension.

12.11 EXAMPLE CALCULATION OF BIAXIAL LOADING

An example of the calculation of drill pipe collapse resistance, corrected for the effect of tensile load is as follows:

Given: String of 5-inch OD, 19.50 lb per ft, Grade E Premium Class drill pipe.

Required: Determine the collapse resistance corrected for tension loading during drill stem test, with drill pipe empty and 15 lb per gal. mud behind the drill pipe. Tension of 50,000 lb on the joint above the packer.

Solution: Find reduced cross section area of Premium Class drill pipe as follows:

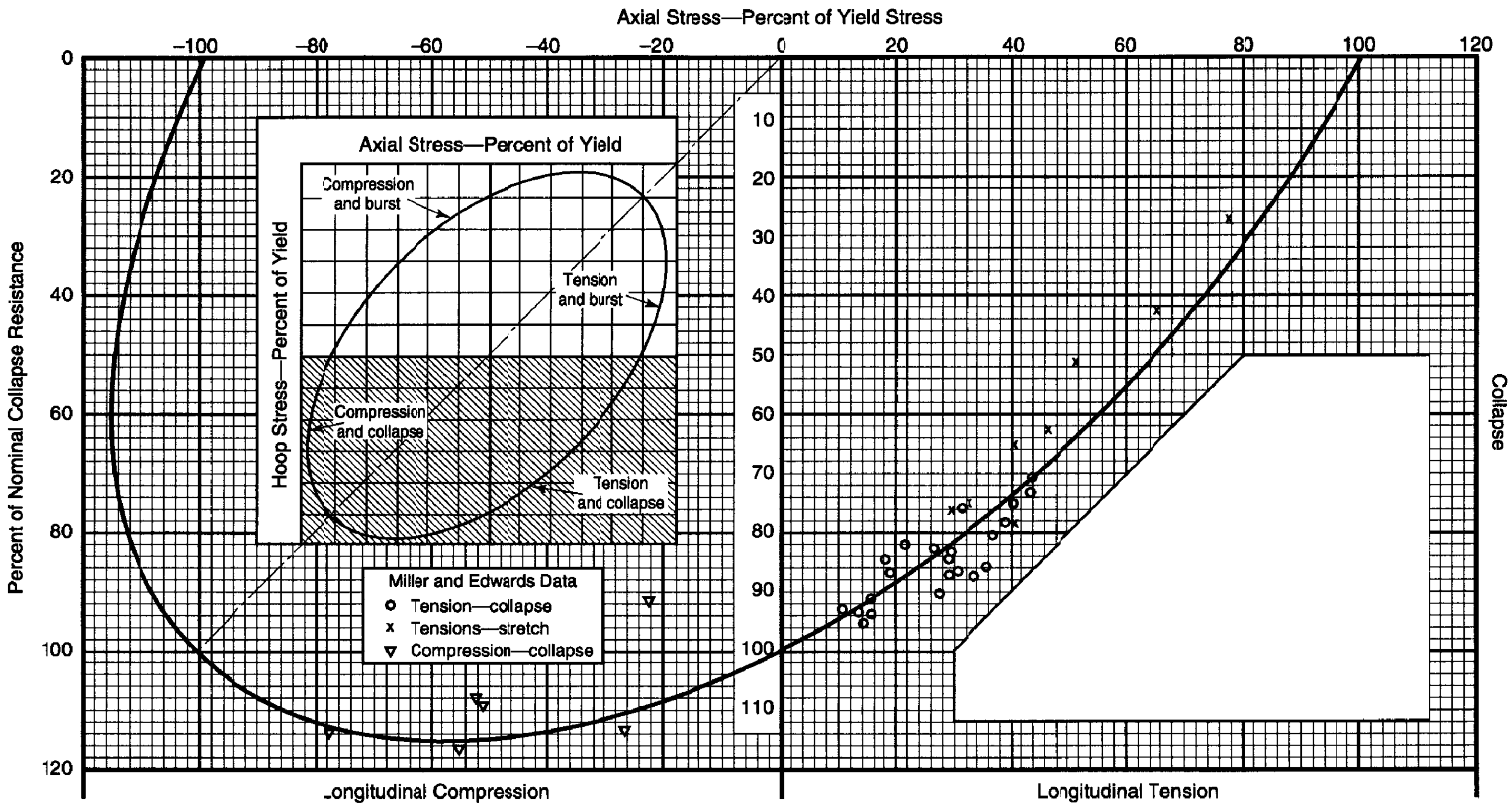
a. Nominal OD = 5 inches, nominal wall thickness = 0.362 inches.
b. Nominal ID = 4.276 inches.
c. Reduced wall thickness for premium.
d. Class = (0.8)(0.362) = 0.2896 inches.
e. Reduced OD for premium class = 4.8552 inches.
f. Cross-sectional area for premium.
g. Class = reduced OD area – nominal ID area
= 18.5141 – 14.3603 = 4.1538 sq. inches.
h. Tension load on bottom joint = 50,000 ÷ 4.1538
= 12037 psi.
i. Average yield strength for Grade E drill pipe = 85,000 psi.
j. Percent tensile stress to average yield strength

$$= \frac{12{,}037}{85{,}000} \times 100 = 14.16 \text{ percent}$$

k. Enter Figure 81 at 14.16 percent on upper right horizontal scale and drop vertically to intersect right-hand portion of the ellipse. Proceed horizontally to the left and intersect Nominal Collapse Resistance (center vertical scale) at 92 percent.
l. Minimum collapse resistance for premium class (Table 5) = 7041 psi.
m. Corrected collapse resistance for effect of tension = (7041)(.92) = 6478 psi.

CAUTION: No safety factors are included in this example calculation.

Note: Use reduced values for cross sectional area, tension and collapse rating for the appropriate class (Premium, Class 2) of used drill pipe being considered.



Note: Applies only in a noncorrosive environment.

Figure 81—Ellipse of Biaxial Yield Stress[12] or Maximum Shear-Strain Energy Diagram
After Holmquist and Nadai, Collapse of Deep Well Casing, API Drilling and Production Practice (1939)

13 Identification, Inspection and Classification of Drill Stem Components

13.1 DRILL STRING MARKING AND IDENTIFICATION

Sections of drill string manufactured in accordance with API Specification 7 are identified with the markings shown in Figure 82. It is recommended that drill string members not covered by Specification 7 also be stencilled at the base of the pin as shown in Figure 82. It is also recommended that drill string members be marked using the mill slot and groove method as shown in Figure 83.

13.2 INSPECTION STANDARDS—DRILL PIPE AND TUBING WORK STRINGS

Through efforts of joint committees of API and IADC, inspection standards for the classification of used drill pipe have been established. The procedure outlined in Table 24 was adopted as tentative at the 1964 Standardization Conference and was revised and approved as standard at the 1968 Standardization Conference. Additional revisions were made at the 1970 Standardization Conference to add Premium Class. At the 1971 Conference, it was determined that the drill pipe classification procedure be removed from an appendix to API Specification 7 and placed in API Recommended Practice 7G. At the 1979 Standardization Conference, Table 25 was revised to also cover classification of used tubing work strings.

The guidelines established in this recommended practice have been in use for several years. Use of the practice and classification guide have apparently been successful when applied in general application. There may be situations where additional inspections are required.

13.2.1 Limitations of Inspection Capability

Most failures of drill pipe result from some form of metal fatigue. A fatigue failure is one which originates as a result of repeated or fluctuating stresses having maximum values less than the tensile strength of the material. Fatigue fractures are progressive, beginning as minute cracks that grow under the action of the fluctuating stress. The rate of propagation is related to the applied cyclic loads and under certain conditions may be extremely rapid. The failure does not normally exhibit extensive plastic deformation and is therefore difficult to detect until such time as considerable damage has occurred. There is no accepted means of inspecting to determine the amount of accumulated fatigue damage or the remaining life in the pipe at a given stress level.

Presently accepted means of inspection are limited to location of cracks, pits, and other surface marks; measurement of remaining wall thickness; measurement of outside diameter; and calculation of remaining cross sectional area. Recent industry statistics confirm that a major percentage of tube body in-service failures occur near the upset runout or within the slip area. Special attention to these critical failure areas should be performed during inspection to facilitate crack detection in drill strings which have been subjected to abnormally high bending stresses. Drill pipe which has just been inspected and found free of cracks may develop cracks after very short additional service through the addition of damage to previously accumulated fatigue damage.

13.2.2 Drill String Cracks

A crack is a single line rupture of the pipe surface. The rupture shall (a) be of sufficient length to be shown by magnetic iron particles used in magnetic particle inspection or (b) be identifiable by visual inspection of the outside of the tube and/or optical or ultrasonic shear-wave inspection of the inside of the tube. Drill pipe tubes, tool joints, and drill collars found to contain cracks should be considered unfit for further drilling service. Shop repair of some tool joints and drill collars, containing cracks, may be possible if the unaffected area of the tool joint body or drill collar permits.

13.2.3 Measurement of Pipe Wall

Tube body conditions will be classified on the basis of the lowest wall thickness measurement obtained and the remaining wall requirements contained in Table 24. The only acceptable wall thickness measurements are those made with pipe-wall micrometers, ultrasonic instruments, or gamma-ray devices that the operator can demonstrate to be within 2 percent accuracy by use of test blocks sized to approximate pipe wall thickness. When using a highly sensitive ultrasonic instrument, care must be taken to ensure that detection of an inclusion or lamination is not interpreted as a wall thickness measurement.

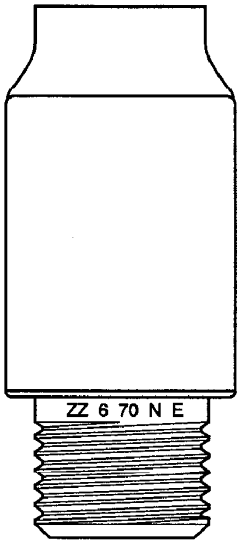
13.2.4 Determination of Cross Sectional Area (Optional)

Determine cross sectional area by use of a direct indicating instrument that the operator can demonstrate to be within 2 percent accuracy by use of a pipe section approximately the same as the pipe being inspected. In the absence of such an instrument, integrate wall thickness measurements taken at 1-inch intervals around the tube.

13.2.5 Procedure

Used drill pipe should be classified according to the procedure of Table 24 and as illustrated in Figure 84, dimension A. Hook loads at minimum yield strength for New, Premium and Class 2 drill pipe are listed in Table 26. Values recommended for minimum OD and make-up torque of weld-on tool joints used with the New, Premium and Class 2 drill pipe are listed in Table 10. Maximum allowable hook loads for New, Premium and Class 2 tubing work strings (also classified in accordance with Table 24) are listed in Table 27.

(Text continued on page 122.)



Sample markings at base of pin

1	2	3	4	5
ZZ	6	70	N	E

1 Tool Joint Manufacturer's Symbol:
ZZ Company (fictional for example only)

2 Month Welded:
6—June

3 Year Welded:
70—1970

4 Pipe Manufacturer's Symbol:
N—United States Steel Company

5 Drill Pipe Grade:
E—Grade E75 drill pipe

Notes:

Tool joint manufacturer's symbol, month welded, year welded, pipe manufacturer, and drill pipe grade symbol shall be stenciled at the base of the pin as shown above. Pipe manufacturer symbol and drill pipe grade symbol applied shall be as represented by manufacturer. Supplier, owner, or user shall be indicated on documents such as mill certification papers or purchase orders.

TOOL JOINT MANUFACTURER'S SYMBOL

Refer to the eleventh edition 1993 of the IADC Drilling Manual* (Section B-1-9) for a list of Tool Joint Manufacturer's symbols.

*Available from: International Association of Drilling Contractors (IADC) P.O. Box 4287, Houston, TX 77210.

Month and Year Welded

Month	Year
1 through 12	Last two digits of year

Drill Pipe Grade

Grade	Symbol
E75	E
X95	X
G105	G
S135	S

Heavy Weight Drill Pipe

(Double stencil pipe grade symbol.)

The "manufacturer" may be either a pipe mill or processor. See API Specification 5D, *Specification for Drill Pipe.*

These symbols are provided for pipe manufacturer identification and have been assigned at pipe manufacturers' requests. Manufacturers included in this list may not be current API Specification 5D licensed pipe manufacturers. A list of current licensed pipe manufacturers is available in the *Composite List of Manufacturers, (Licensed for Use of the API Monogram).*

Pipe mills may upset and heat treat their own drill pipe, or they may have this done according to their own specifications. In either case, the mill's assigned symbol should be used on each drill string assembly since they are the pipe manufacturer.

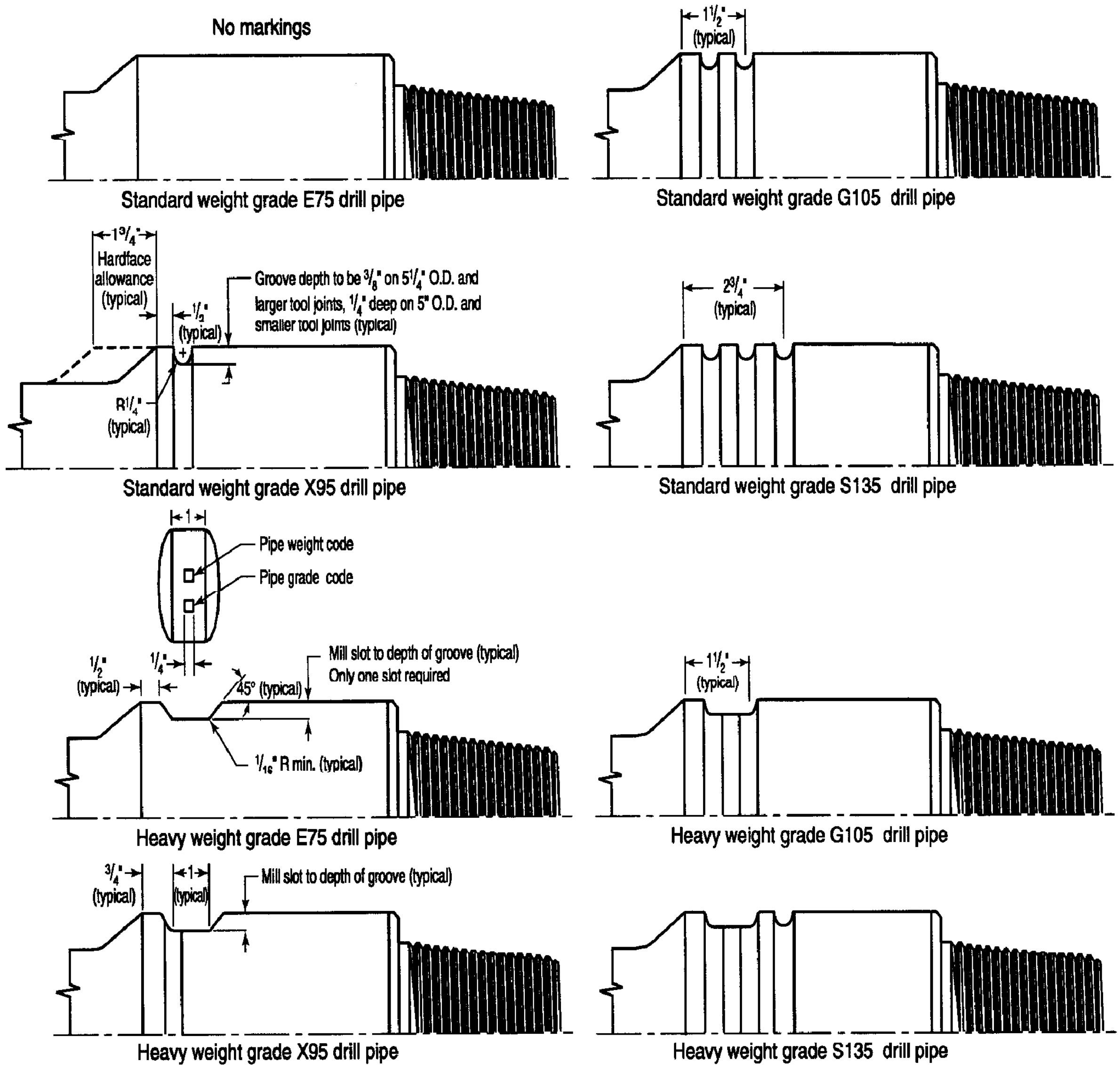
Pipe processors may buy "green" tubes and upset and heat treat these according to their own specifications. In this case, the processor's assigned symbol should be used on each drill string assembly since they are the pipe manufacturer.

Pipe Manufacturers
(Pipe Mills or Processors)

Active		Inactive	
Mill	Symbol	Mill	Symbol
Algoma	X	Armco	A
British Steel		American Seamless	AI
Seamless Tubes LTD	B	B&W	W
Dalmine	D	CF&I	C
Kawasaki	H	J&L	J
Nippon	I	Lone Star	L
NKK	K	Ohio	O
Mannesmann	M	Republic	R
Reynolds Aluminum	RA	TI	Z
Sumitomo	S	Tubemuse	TU
Siderca	SD	Voest	VA
Tamsa	T	Wheeling Pittsburgh	P
US Steel	N	Youngstown	Y
Vallourec	V		
Used	U		

Processor	Symbol
Grant TFW	TFW
Omsco	OMS
Prideco	PI

Figure 82—Marking on Tool Joints for Identification of Drill String Components



Drill Pipe Weight Code

(1) Size OD inches	(2) Nominal Weight lb per ft	(3) Wall Thickness inches	(4) Weight Code Number
$2^3/_8$	4.85	.190	1
	6.65*	.280	2
$2^7/_8$	6.85	.217	1
	10.40*	.362	2
$3^1/_2$	9.50	.254	1
	13.30*	.368	2
	15.50	.449	3
4	11.85	.262	1
	14.00*	.330	2
	15.70	.380	3
$4^1/_2$	13.75	.271	1
	16.60*	.337	2
$4^1/_2$	20.00	.430	3
	22.82	.500	4
	24.66	.550	5
	25.50	.575	6
5	16.25	.296	1
	19.50	.362	2
	25.60	.500	3
$5^1/_2$	19.20	.304	1
	21.90*	.361	2
	24.70	.415	3
$6^5/_8$	25.20*	.330	2
	27.70	.362	3

*Designates standard weight for drill pipe size.

Figure 83—Recommended Practice for Mill Slot and Groove Method of Drill String Identification

Table 24—Classification of Used Drill Pipe

(All sizes, weights and grades. Nominal dimension is basis for all calculations.)

(1)	(2)	(3)	(4)
Pipe Condition	Premium Class[1] Two White Bands One Center Punch Mark	Class 2 Yellow Bands Two Center Punch Marks	Class 3 Orange Bands Three Center Punch Marks
I. EXTERIOR CONDITIONS[4]			
A. OD wear			
Wall	Remaining wall not less than 80%	Remaining wall not less than 70%	Any imperfections or damages exceeding CLASS 2
B. Dents and mashes	Diameter reduction not over 3% of OD	Diameter reduction not over 4% of OD	
Crushing, necking	Diameter reduction not over 3% of OD	Diameter reduction not over 4% of OD	
C. Slip area			
Mechanical damage			
Cuts[2], gouges[2]	Depth not to exceed 10% of the average adjacent wall[5]	Depth not to exceed 20% of the average adjacent wall[5]	
D. Stress induced diameter variations			
1. Stretched	Diameter reduction not over 3% of OD	Diameter reduction not over 4% of OD	
2. String shot	Diameter increase not over 3% of OD	Diameter increase not over 4% of OD	
E. Corrosion, cuts, and gouges			
1. Corrosion	Remaining wall not less than 80%	Remaining wall not less than 70%	
2. Cuts and gouges			
Longitudinal	Remaining wall not less than 80%	Remaining wall not less than 70%	
Transverse	Remaining wall not less than 80%	Remaining wall not less than 80%	
F. Cracks[3]	None	None	None
II. INTERIOR CONDITIONS			
A. Corrosive pitting			
Wall	Remaining wall not less than 80% measured from base of deepest pit	Remaining wall not less than 70% measured from base of deepest pit	
B. Erosion and wear			
Wall	Remaining wall not less than 80%	Remaining wall not less than 70%	
C. Cracks[3]	None	None	None

[1]The premium classification is recommended for service where it is anticipated that torsional or tensile limits for Class 2 drill pipe and tubing work strings will be exceeded. These limits for Premium Class and Class 2 drill pipe are specified in Tables 4 and 6, respectively. Premium Class shall be identified with two white bands, plus one center punch mark on the 35 degree or 18 degree shoulder of the pin end tool joint.

[2]Remaining wall shall not be less than the value in I.E.2, defects may be ground out providing the remaining wall is not reduced below the value shown in I.E.1 of this table and such grinding to be approximately faired into outer contour of the pipe.

[3]In any classification where cracks or washouts appear, the pipe will be identified with the red band and considered unfit for further drilling service.

[4]An API Recommended Practice 7G inspection cannot be made with drill pipe rubbers on the pipe.

[5]Average adjacent wall is determined by measuring the wall thickness on each side of the cut or gouge adjacent to the deepest penetration.

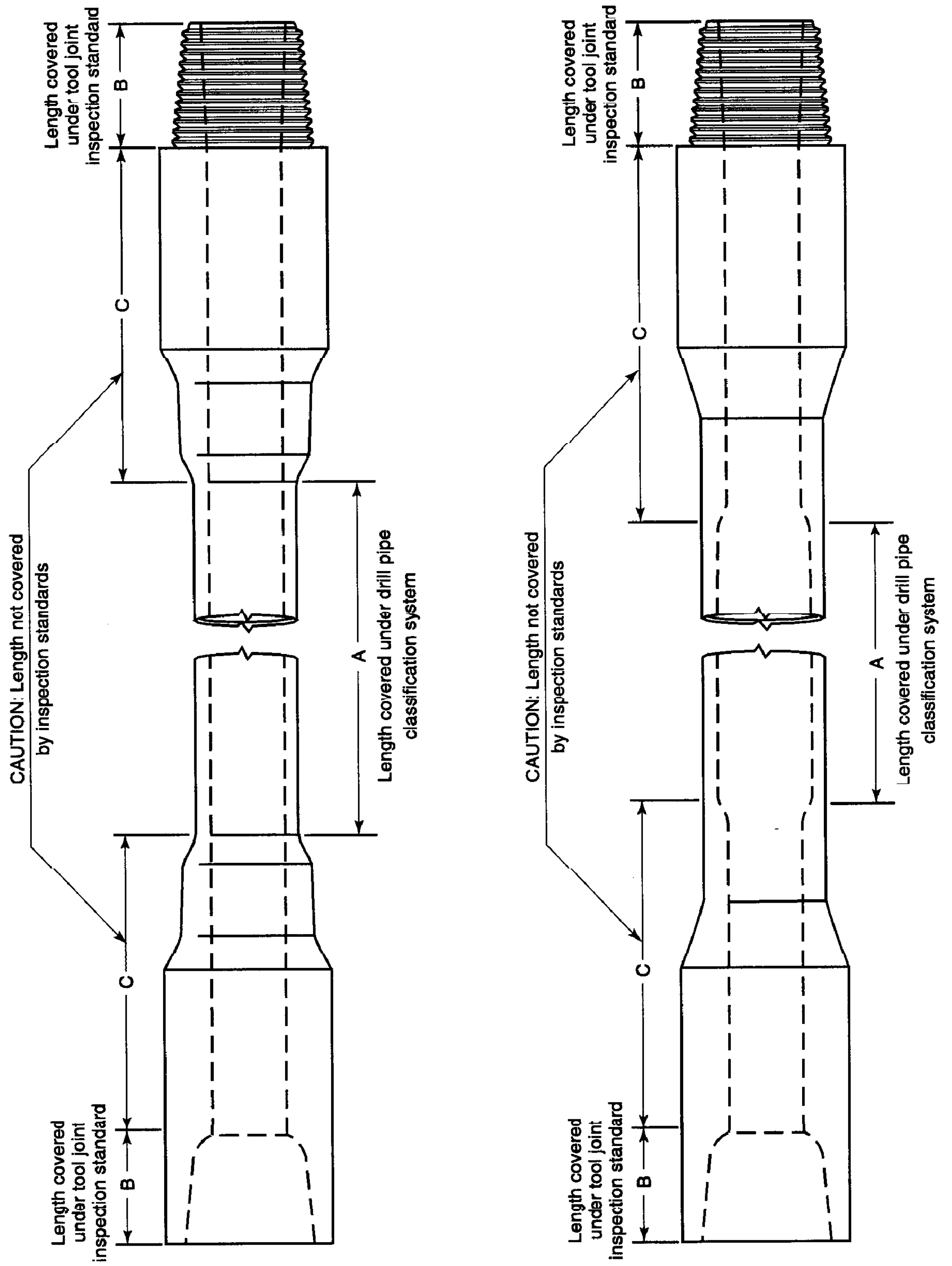


Figure 84—Identification of Lengths Covered by Inspection Standards

Table 26—Hook-Load at Minimum Yield Strength for New, Premium Class (Used), and Class 2 (Used) Drill Pipe

(Hook load values in this table vary slightly from tensile data for the same pipe size and class listed in Tables 2, 4, 6, 8, and 9 because of differences in rounding procedures used in calculations.)

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13	(14)	(15)	(16)	(17)	(18)
		New						Premium Class					Class 2				
Size OD	Weight	Original OD	Wall Thickness	ID	Original Cross-Section Area	Yield	Hook Load	OD with 20% Wall Reduction	Minimum Remaining Wall (80%)	Reduced Cross-Section Area	Cross-Section Area Ratio	Hook Load	OD with 30% Wall Reduction	Minimum Remaining Wall (70%)	Reduced Cross-Section Area	Cross Section Area Ratio	Hook Load
in.	lb/ft	in.	in.	in.	sq. in.	psi	lb	in.	in.	sq. in.	%	lb	in.	in.	sq. in.	%	lb
2 3/8	4.85	2.375	0.190	1.995	1.3042	75000.	97817.	2.2990	0.152	1.0252	78.61	76893.	2.2610	0.133	0.8891	68.17	66686.
						95000.	123902.					97398.					84469.
						105000.	136944.					107650.					93360.
						135000.	176071.					138407.					120035.
2 3/8	6.65	2.375	0.280	1.815	1.8429	75000.	138214.	2.2630	0.224	1.4349	77.86	107616.	2.2070	0.196	1.2383	67.19	92871.
						95000.	175072.					136313.					117636.
						105000.	193500.					150662.					130019.
						135000.	248786.					193709.					167167.
2 7/8	6.85	2.875	0.217	2.441	1.8120	75000.	135902.	2.7882	0.174	1.4260	78.69	106946.	2.7448	0.152	1.2374	68.29	92801.
						95000.	172143.					135465.					117549.
						105000.	190263.					149725.					129922.
						135000.	244624.					192503.					167043.
2 7/8	10.40	2.875	0.362	2.151	2.8579	75000.	214344.	2.7302	0.290	2.2205	77.70	166535.	2.6578	0.253	1.9141	66.97	143557.
						95000.	271503.					210945.					181839.
						105000.	300082.					233149.					200980.
						135000.	385820.					299764.					258403.
3 1/2	9.50	3.500	0.254	2.992	2.5902	75000.	194264.	3.3984	0.203	2.0397	78.75	152979.	3.3476	0.178	1.7706	68.36	132793.
						95000.	246068.					193774.					168204.
						105000.	271970.					214171.					185910.
						135000.	349676.					275363.					239027.
3 1/2	13.30	3.500	0.368	2.764	3.6209	75000.	271569.	3.3528	0.294	2.8287	78.12	212150.	3.2792	0.258	2.4453	67.53	183398.
						95000.	343988.					268723.					232304.
						105000.	380197.					297010.					256757.
						135000.	488825.					381870.					330116.
3 1/2	15.50	3.500	0.449	2.602	4.3037	75000.	322775.	3.3204	0.359	3.3416	77.65	250620.	3.2306	0.314	2.8796	66.91	215967.
						95000.	408848.					317452.					273558.
						105000.	451885.					350868.					302354.
						135000.	580995.					451115.					388741.
4	11.85	4.000	0.262	3.476	3.0767	75000.	230755.	3.8952	0.210	2.4269	78.88	182016.	3.8428	0.183	2.1084	68.53	158132.
						95000.	292290.					230554.					200301.
						105000.	323057.					254823.					221385.
						135000.	415360.					327630.					284638.
4	14.00	4.000	0.330	3.340	3.8048	75000.	285359.	3.8680	0.264	2.9891	78.56	224182.	3.8020	0.231	2.5915	68.11	194363.
						95000.	361454.					283963.					246193.
						105000.	399502.					313854.					272108.
						135000.	513646.					403527.					349853.
4	15.70	4.000	0.380	3.240	4.3216	75000.	324118.	3.8480	0.304	3.3847	78.32	253851.	3.7720	0.266	2.9298	67.80	219738.
						95000.	410550.					321544.					278335.
						105000.	453765.					355391.					307633.
						135000.	583413.					456931.					395526.
4 1/2	13.75	4.500	0.271	3.958	3.6005	75000.	270034.	4.3916	0.217	2.8434	78.97	213258.	4.3374	0.190	2.4719	68.65	185390.
						95000.	342043.					270127.					234827.
						105000.	378047.					298562.					259546.
						135000.	486061.					383865.					333702.

Table 25—Classification of Used Tubing Work Strings

(1) Pipe Body Condition	(2) Critical Service Class[1] One White Band	(3) Premium Class[2] Two White Bands	(4) Class 2 Blue Bands
I. EXTERIOR CONDITIONS	(Tube only)		
A. OD wear			
Wall	Remaining wall not less than 87$^1/_2$%	Remaining wall not less than 80%	Remaining wall not less than 70%
B. Dents and mashes	Diameter reduction not over 2% of OD	Diameter reduction not over 3% of OD	Diameter reduction not over 4% of OD
Crushing, necking	Diameter reduction not over 2% of OD	Diameter reduction not over 3% of OD	Diameter reduction not over 4% of OD
C. Slip area, tong area mechanical damage			
Cuts[3], gouges[3]	Depth not to exceed 10% of the average adjacent wall[5]	Depth not to exceed 10% of the average adjacent wall[5]	Depth not to exceed 20% of the average adjacent wall[5]
D. Stress induced diameter variations			
1. Stretched	Diameter reduction not over 2% of OD	Diameter reduction not over 3% of OD	Diameter reduction not over 4% of OD
2. String shot	Diameter increase not over 2% of OD	Diameter increase not over 3% of OD	Diameter increase not over 4% of OD
E. Corrosion, cuts, and gouges			
1. Corrosion	Remaining wall not less than 87$^1/_2$%	Remaining wall not less than 80%	Remaining wall not less than 70%
2. Cuts and gouges			
Longitudinal	Remaining wall not less than 87$^1/_2$%	Remaining wall not less than 80%	Remaining wall not less than 70%
Transverse	Remaining wall not less than 87$^1/_2$%	Remaining wall not less than 80%	Remaining wall not less than 80%
F. Cracks[4]	None	None	None
II. INTERIOR CONDITIONS (Tube and Upset)			
A. Corrosive pitting			
Wall	Remaining wall not less than 87$^1/_2$% measured from base of deepest pit	Remaining wall not less than 80% measured from base of deepest pit	Remaining wall not less than 70% measured from base of deepest pit
B. Erosion and wear			
Wall	Remaining wall not less than 87$^1/_2$%	Remaining wall not less than 80%	Remaining wall not less than 70%
C. Drift			
External upset Internal upset[6]	API dimensions $^1/_{16}$ inch less than specified bored ID	API dimensions $^1/_{16}$ inch less than specified bored ID	API dimensions $^1/_{16}$ inch less than specified bored ID
D. Cracks[4]	None	None	None

[1]The critical service classification is recommended for service where new or like new specifications apply. Critical service classification tubing work strings shall be identified with one white band.
[2]The premium classification is recommended for service where it is anticipated that torsion or tensile limits for Class 2 tubing work strings will be exceeded. Premium classification tubing work strings shall be identified with two white bands.
[3]Remaining wall shall not be less than the value in I.E.2. Defects may be ground out providing the remaining wall is not reduced below the value shown in I.E.1 of this table and such grinding to be approximately faired into outer contour of the tubing.
[4]In any classification where cracks or washouts appear, the tubing will be identified with the red band and considered unfit for further service.
[5]Average adjacent wall is determined by measuring the wall thickness on each side of the cut or gouge adjacent to the deepest penetration.
[6]Applicable to Internal Upsets which have been bored.

Table 26—Hook-Load at Minimum Yield Strength for New, Premium Class (Used), and Class 2 (Used) Drill Pipe (Continued)

(Hook load values in this table vary slightly from tensile data for the same pipe size and class listed in Tables 2, 4, 6, 8, and 9 because of differences in rounding procedures used in calculations.)

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13	(14)	(15)	(16)	(17)	(18)
		New						Premium Class					Class 2				
Size OD	Weight	Original OD	Wall Thickness	ID	Original Cross-Section Area	Yield	Hook Load	OD with 20% Wall Reduction	Minimum Remaining Wall (80%)	Reduced Cross-Section Area	Cross-Section Area Ratio	Hook Load	OD with 30% Wall Reduction	Minimum Remaining Wall (70%)	Reduced Cross-Section Area	Cross Section Area Ratio	Hook Load
in.	lb/ft	in.	in.	in.	sq. in.	psi	lb	in.	in.	sq. in.	%	lb	in.	in.	sq. in.	%	lb
4 1/2	16.60	4.500	0.337	3.826	4.4074	75000.	330558.	4.3652	0.270	3.4689	78.70	260165.	4.2978	0.236	3.0103	68.30	225771.
						95000.	418707.					329542.					285977.
						105000.	462781.					364231.					316080.
						135000.	595004.					468297.					406388.
4 1/2	20.00	4.500	0.430	3.640	5.4981	75000.	412358.	4.3280	0.344	4.3055	78.31	322916.	4.2420	0.301	3.7267	67.78	279502.
						95000.	522320.					409026.					354035.
						105000.	577301.					452082.					391302.
						135000.	742244.					581248.					503103.
4 1/2	22.82	4.500	0.500	3.500	6.2832	75000.	471239.	4.3000	0.400	4.9009	78.00	367566.	4.2000	0.350	4.2333	67.37	317497.
						95000.	596903.					465584.					402163.
						105000.	659735.					514593.					444496.
						135000.	848230.					661620.					571495.
5	16.25	5.000	0.296	4.408	4.3743	75000.	328073.	4.8816	0.237	3.4554	78.99	259155.	4.8224	0.207	3.0042	68.68	225316.
						95000.	415559.					328263.					285400.
						105000.	459302.					362817.					315442.
						135000.	590531.					466479.					405568.
5	19.50	5.000	0.362	4.276	5.2746	75000.	395595.	4.8552	0.290	4.1538	78.75	311535.	4.7828	0.253	3.6058	68.36	270432.
						95000	501087.					394612.					342548.
						105000.	553833.					436150.					378605.
						135000.	712070.					560764.					486778.
5	25.60	5.000	0.500	4.000	7.0686	75000.	530144.	4.8000	0.400	5.5292	78.22	414690.	4.7000	0.350	4.7831	67.67	358731.
						95000.	671515.					525274.					454392.
						105000.	742201.					580566.					502223.
						135000.	954259.					746443.					645715.
5 1/2	19.20	5.500	0.304	4.892	4.9624	75000.	372181.	5.3784	0.243	3.9235	79.06	294260.	5.3176	0.213	3.4127	68.77	255954.
						95000.	471429.					372730.					324208.
						105000.	521053.					411965.					358335.
						135000.	669925.					529669.					460717.
5 1/2	21.90	5.500	0.361	4.778	5.8282	75000.	437116.	5.3556	0.289	4.5971	78.88	344780.	5.2834	0.253	3.9938	68.52	299533.
						95000.	553681.					436721.					379409.
						105000.	611963.					482692.					419346.
						135000.	786809.					620604.					539160.
5 1/2	24.70	5.500	0.415	4.670	6.6296	75000.	497222.	5.3340	0.332	5.2171	78.69	391285.	5.2510	0.290	4.5271	68.29	339533.
						95000.	629814.					495627.					430076.
						105000.	696111.					547799.					475347.
						135000.	894999.					704313.					611160.
6 5/8	25.20	6.625	0.330	5.965	6.5262	75000.	489464.	6.4930	0.264	5.1662	79.16	387466.	6.4270	0.231	4.4965	68.90	337236.
						95000.	619988.					490790.					427166.
						105000.	685250.					542452.					472131.
						135000.	881035.					697438.					607026.
6 5/8	27.70	6.625	0.362	5.901	7.1227	75000.	534199.	6.4802	0.290	5.6323	79.08	422419.	6.4078	0.253	4.8994	68.79	367455.
						95000.	676652.					535064.					465443.
						105000.	747879.					591387.					514437.
						135000.	961558.					760354.					661419.

Table 27—Hook-Load at Minimum Yield Strength for New, Premium Class (Used), and Class 2 (Used) Tubing Work Strings

(Hook load values in this table vary slightly from tensile data for the same pipe size and class listed in Tables 2, 4, 6, 8, and 9 because of differences in rounding procedures used in calculations.)

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13	(14)	(15)	(16)	(17)	(18)
		New						Premium Class					Class 2				
Size OD	Weight	Original OD	Wall Thickness	ID	Original Cross-Section Area	Yield	Hook Load	OD with 20% Wall Reduction	Minimum Remaining Wall (80%)	Reduced Cross-Section Area	Cross-Section Area Ratio	Hook Load	OD with 30% Wall Reduction	Minimum Remaining Wall (70%)	Reduced Cross-Section Area	Cross Section Area Ratio	Hook Load
in.	lb/ft	in.	in.	in.	sq. in.	psi	lb	in.	in.	sq. in.	%	lb	in.	in.	sq. in.	%	lb
3/4	1.20	1.050	0.113	0.824	0.3326	55000.	18295.	1.0048	0.090	0.2597	78.07	14283.	0.9822	0.079	0.2244	67.47	12343.
						75000.	24948.					19477.					16832.
						80000.	26611.					20775.					17954.
						105000.	34927.					27267.					23564.
3/4	1.50	1.050	0.154	0.742	0.4335	55000.	23842.	0.9884	0.123	0.3349	77.25	18418.	0.9576	0.108	0.2878	66.39	15829.
						75000.	32512.					25115.					21585.
						80000.	34679.					26790.					23024.
						105000.	45516.					35161.					30219.
1	1.80	1.315	0.133	1.049	0.4939	55000.	27163.	1.2618	0.106	0.3862	78.20	21242.	1.2352	0.093	0.3340	67.64	18372.
						75000.	37041.					28966.					25053.
						80000.	39510.					30897.					26724.
						105000.	51857.					40552.					35075.
1	2.25	1.315	0.179	0.957	0.6388	55000.	35135.	1.2434	0.143	0.4950	77.48	27222.	1.2076	0.125	0.4260	66.69	23432.
						75000.	47912.					37122.					31953.
						80000.	51106.					39596.					34083.
						105000.	67077.					51970.					44734.
1 1/4	2.40	1.660	0.140	1.380	0.6685	55000.	36769.	1.6040	0.112	0.5250	78.53	28873.	1.5760	0.098	0.4550	68.07	25027.
						75000.	50140.					39373.					34128.
						80000.	53482.					41998.					36403.
						105000.	70196.					55122.					47779.
1 1/4	3.02	1.660	0.191	1.278	0.8815	55000.	48481.	1.5836	0.153	0.6868	77.92	37776.	1.5454	0.134	0.5930	67.27	32613.
						75000.	66110.					51513.					44472.
						80000.	70517.					54947.					47437.
						105000.	92554.					72118.					62261.
1 1/4	3.20	1.660	0.198	1.264	0.9094	55000.	50018.	1.5808	0.158	0.7078	77.83	38930.	1.5412	0.139	0.6107	67.16	33590.
						75000.	68206.					53087.					45805.
						80000.	72753.					56626.					48858.
						105000.	95489.					74322.					64126.
1 1/2	2.90	1.900	0.145	1.610	0.7995	55000.	43970.	1.8420	0.116	0.6290	78.68	34595.	1.8130	0.101	0.5457	68.26	30016.
						75000.	59959.					47175.					40931.
						80000.	63957.					50320.					43660.
						105000.	83943.					66045.					57304.
1 1/2	4.19	1.900	0.219	1.462	1.1565	55000.	63610.	1.8124	0.175	0.9011	77.92	49562.	1.7686	0.153	0.7779	67.26	42787.
						75000.	86741.					67584.					58345.
						80000.	92523.					72090.					62235.
						105000.	121437.					94618.					81684.
2 1/16	3.25	2.063	0.156	1.751	0.9346	55000.	51403.	2.0006	0.125	0.7354	78.69	40450.	1.9694	0.109	0.6382	68.28	35099.
						75000.	70095.					55158.					47862.
						80000.	74768.					58836.					51053.
						105000.	98133.					77222.					67007.
2 3/8	4.70	2.375	0.190	1.995	1.3042	55000.	71733.	2.2990	0.152	1.0252	78.61	56388.	2.2610	0.133	0.8891	68.17	48903.
						75000.	97817.					76893.					66686.
						80000.	104339.					82019.					71132.
						105000.	136944.					107650.					93360.
2 3/8	5.30	2.375	0.218	1.939	1.4773	55000.	81249.	2.2878	0.174	1.1579	78.38	63686.	2.2442	0.153	1.0027	67.88	55150.
						75000.	110794.					86844.					75205.
						80000.	118181.					92634.					80218.
						105000.	155112.					121581.					105286.

Table 27—Hook-Load at Minimum Yield Strength for New, Premium Class (Used), and Class 2 (Used) Tubing Work Strings (Continued)

(Hook load values in this table vary slightly from tensile data for the same pipe size and class listed in Tables 2, 4, 6, 8, and 9 because of differences in rounding procedures used in calculations.)

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13	(14)	(15)	(16)	(17)	(18)
		New						Premium Class					Class 2				
Size OD	Weight	Original OD	Wall Thickness	ID	Original Cross-Section Area	Yield	Hook Load	OD with 20% Wall Reduction	Minimum Remaining Wall (80%)	Reduced Cross-Section Area	Cross-Section Area Ratio	Hook Load	OD with 30% Wall Reduction	Minimum Remaining Wall (70%)	Reduced Cross-Section Area	Cross Section Area Ratio	Hook Load
in.	lb/ft	in.	in.	in.	sq. in.	psi	lb	in.	in.	sq. in.	%	lb	in.	in.	sq. in.	%	lb
$2^{7}/_{8}$	5.95	2.375	0.254	1.867	1.6925	55000.	93087.	2.2734	0.203	1.3216	78.08	72686.	2.2226	0.178	1.1422	67.49	62820.
						75000.	126936.					99117.					85663.
						80000.	135399.					105725.					91374.
						105000.	177711.					138764.					119928.
$2^{3}/_{8}$	6.50	2.875	0.217	2.441	1.8120	55000.	99661.	2.7882	0.174	1.4260	78.69	76427.	2.7448	0.152	1.2374	68.29	88054.
						75000.	135902.					106946.					92801.
						80000.	144962.					114076.					98988.
						105000.	190263.					149725.					129922.
$2^{7}/_{8}$	8.70	2.875	0.308	2.259	2.4839	55000.	136612.	2.7518	0.246	1.9394	78.08	106667.	2.6902	0.216	1.6761	67.48	92186.
						75000.	186289.					145455.					125709.
						80000.	198708.					155152.					134089.
						105000.	260805.					203637.					175992.
$2^{7}/_{8}$	9.50	2.875	0.340	2.195	2.7077	55000.	148926.	2.7390	0.272	2.1081	77.85	115945.	2.6710	0.238	1.8192	67.18	100053.
						75000.	203080.					158106.					136436.
						80000.	216619.					168647.					145532.
						105000.	284313.					221349.					191011.
$2^{7}/_{8}$	10.70	2.875	0.392	2.091	3.0578	55000.	168180.	2.7182	0.314	2.3690	77.47	130296.	2.6398	0.274	2.0391	66.68	112151.
						75000.	229337.					177676.					152933.
						80000.	244626.					189522.					163128.
						105000.	321072.					248747.					214106.
$2^{7}/_{8}$	11.00	2.875	0.405	2.065	3.1427	55000.	172848.	2.7130	0.324	2.4317	77.38	133744.	2.6320	0.283	2.0917	66.56	115042.
						75000.	235702.					182378.					156875.
						80000.	251415.					194536.					167334.
						105000.	329983.					255329.					219626.
$3^{1}/_{2}$	12.80	3.500	0.368	2.764	3.6209	55000.	199151.	3.3528	0.294	2.8287	78.12	155577.	3.2792	0.258	2.4453	67.53	134492.
						75000.	271569.					212150.					183398.
						80000.	289674.					226293.					195624.
						105000.	380197.					297010.					256757.
$3^{1}/_{2}$	12.95	3.500	0.375	2.750	3.6816	55000.	202485.	3.3500	0.300	2.8746	78.08	158101.	3.2750	0.263	2.4843	67.48	136637.
						75000.	276117.					215592.					186323.
						80000.	294524.					229965.					198745.
						105000.	386563.					301828.					260853.
$3^{1}/_{2}$	15.80	3.500	0.476	2.548	4.5221	55000.	248715.	3.3096	0.381	3.5038	77.48	192708.	3.2144	0.333	3.0160	66.69	165879.
						75000.	339156.					262783.					226198.
						80000.	361767.					280302.					241278.
						105000.	474819.					367897.					316678.
$3^{1}/_{2}$	16.70	3.500	0.510	2.480	4.7906	55000.	263484.	3.2960	0.408	3.7018	77.27	203596.	3.1940	0.357	3.1818	66.42	175001.
						75000.	359296.					277631.					238638.
						80000.	383249.					296140.					254547.
						105000.	503015.					388684.					334093.
$4^{1}/_{2}$	15.50	4.500	0.337	3.826	4.4074	55000.	242409.	4.3652	0.270	3.4689	78.70	190788.	4.2978	0.236	3.0103	68.30	165565.
						75000.	330558.					260165.					225771.
						80000.	352595.					277509.					240823.
						105000.	462781.					364231.					316080.
$4^{1}/_{2}$	19.20	4.500	0.430	3.640	5.4981	55000.	302396.	4.3280	0.344	4.3055	78.31	236805.	4.2420	0.301	3.7267	67.78	204968.
						75000.	412358.					322916.					279502.
						80000.	439848.					344443.					298135.
						105000.	577301.					452082.					391302.

13.2.6 Inspection Classification Marking

A permanent mark or marks signifying the classification of the pipe (for example, refer to Table 24, Note 1) should be stamped as follows:

a. On the 35-degree or 18-degree sloping shoulder of the pin end tool joint.

b. Or in some other *low-stressed* section of the tool joint where the marking will normally carry through operations.

c. Cold steel stenciling should be avoided on outer surface of tube body.

d. One center punch denotes Premium, two denote Class 2 and three denote Class 3.

13.3 TOOL JOINTS

13.3.1 Color Coding

The classification system for used drill pipe outlined in Table 24 includes a color code designation to identify the drill pipe class. The same system is recommended for tool joint class identification. In addition, it is recommended that the tool joint be identified as (1) field repairable, or (2) scrap or shop repairable. This color code system for tool joints and for drill pipe is shown in Figure 85.

13.3.2 Inspection Standard

The following recommended inspection standard for used tool joints was initially included as an appendix to API Specification 7. It was moved to API Recommended Practice 7G by committee action at the 1971 Standardization Conference.

13.3.2.1 Required Inspections

Following are required inspections:

a. Outside diameter measurement—measure tool joint outside diameter at a distance of 1 inch from the shoulder and determine classification from data in Table 10. Minimum shoulder width should be used when tool joints are worn eccentrically.

b. Shoulder condition—check shoulders for galls, nicks, washes, fins, or any other matter which would affect the pressure holding capacity of the joint and conditions which may affect joint stability. Make certain joint has a minimum $^{1}/_{32}$ in. x 45 degree OD shoulder bevel.

13.3.2.2 Optional Inspections

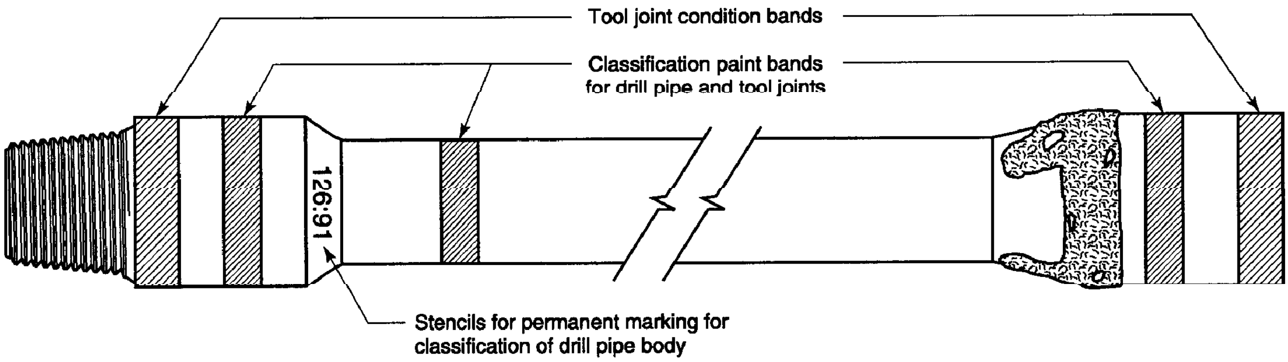
Following are optional inspections:

a. Shoulder width—using data in Table 10, determine minimum shoulder width acceptable for tool joint in class as governed by the outside diameter.

b. Visual thread inspection—the thread profile is checked to detect over-torque, insufficient torque, lapped threads, and stretching. Threads are visually inspected to detect handling damage, corrosion damage and galling.

c. Box swell and/or pin stretch—these are indications of over-torquing and their presence greatly affects the future performance of the tool joint. The lead gauge is the only standard method for measuring pin stretch. On used tool joints, it is recommended that pins having stretch which exceeds 0.006 inch in 2 inches should be recut. All pins which have been stretched should be inspected for cracks.

It is recommended that box counterbores (Qc), API Specification 7, Table 25, be checked. If the Qc diameter is more than 0.031 inch ($^{1}/_{32}$ inch) outside the allowed tolerance, then the box should be recut.



Tool Joint and Drill Pipe Classification	Number and Color of Bands
Premium Class	Two White
Class 2	One Yellow
Class 3	One Orange
Scrap	One Red

Tool Joint Condition	Color of Bands
Scrap or Shop Repairable	Red
Field Repairable	Green

Figure 85—Drill Pipe and Tool Joint Color Code Identification

d. Magnetic particle inspection—if evidence of pin stretching is found, magnetic particle inspection should be made of the entire pin threaded area, especially the last engaged thread area, to determine if transverse cracks are present.

In highly stressed drilling environments or if evidence of damage, such as cracking is noted, magnetic particle inspection should be made of the entire box threaded area, especially the last engaged thread area, to determine if transverse cracks are present.

Longitudinal or irregular orientation of cracking may occur as a result of friction heat checking (see 8.6). In that case magnetic particle inspection of both box and pin tool joint surfaces, excluding any hardband area, should be performed, with an emphasis on detection of longitudinal cracks.

For crack detection, the wet fluorescent magnetic particle method is preferred for tool joint inspections. Tool joints found to contain cracks in the threaded areas or within the tool joint body, excluding any hardband area, should be considered unfit for further drilling service. Shop repair of some cracked tool joints may be possible if the unaffected area of the tool joint body permits.

e. Minimum tong space—refer to Figure 86. The criteria for determining the minimum tong space for tool joints on used drill pipe should be based on safe and efficient tonging operations on the rig floor, primarily when manual tongs are in use. In this regard, there should be sufficient tong space to allow full engagement of the tong dies, plus an adequate amount of tong space remaining to allow the driller and/or floorhand to visually verify that the mating shoulders of the connection are unencumbered to allow proper make-up or break-out of the connection without damage.

It is also recommended that any hard banded surfaces of the pin or box tool joint tong space be excluded from the area of tong die engagement as stated above when minimum tong space is determined. This practice will ensure that optimum gripping of the tongs is achieved and that damage to tong dies is minimized. In the case where tool joint diameters have been worn to the extent that the original hard banding has been substantially removed, the user may include this area in determining the minimum tong space.

The use of other types of tongs, or devices designed for the purpose of making and breaking connections may require a different minimum tong space than what would be determined for manual tongs. In this case, the user should apply the criteria necessary to ensure that the intent of this recommendation is satisfied.

Such minimum tong space should not be construed as a means by which tool joints are acceptable or not with regard to the strength or integrity of the connection as otherwise specified in this recommended practice.

13.3.3 General

a. Gauging—thread wear, plastic deformation, mechanical damage and lack of cleanliness may all contribute to erroneous figures when plug and ring gauges are applied to used connections. Therefore, ring and plug standoffs should not be used to determine rejection or continued use of rotary shouldered connections.

b. Repair of damaged shoulders—when refacing a damaged tool joint shoulder, a minimum of material should be removed. It is a good practice to remove not more than $^1/_{32}$-inch from a box or pin shoulder at any one refacing and not more than $^1/_{16}$-inch cumulatively.

It is suggested that a benchmark be provided for the determination of the amount of material which may be removed from the tool joint makeup shoulder. This benchmark should be applied to new or recut tool joints after facing to gauge. The form of the benchmark may be a $^3/_{16}$-inch diameter circle with a bar tangent to the circle parallel to the makeup shoulder, as shown in Figure 86. The distance from the shoulder to the bar should be $^1/_8$ inch. Variations of this benchmark or other type benchmarks may be available from tool joint manufacturers or machine shops.

13.3.4 Coverage

Figure 84, dimension A, indicates the length covered under the drill pipe classification system recommended in 13.2.5. Figure 84, dimension B, indicates the length covered under the tool joint inspection standard in 13.3.2. The length not covered by inspection standards is indicated under a *CAUTION* heading by dimension C, Figure 84.

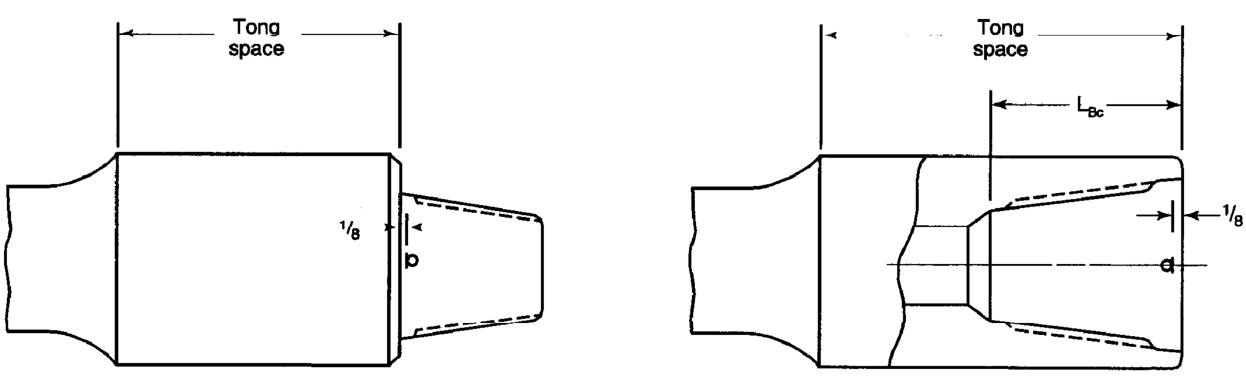


Figure 86—Tong Space and Bench Mark Position

13.4 DRILL COLLAR INSPECTION PROCEDURE

The following inspection procedure for used drill collars is recommended:

a. Visually inspect full length to determine obvious damage and overall condition.
b. Measure OD and ID of both ends.
c. Thoroughly clean box and pin threads. Follow immediately with wet fluorescent magnetic particle inspection for detection of cracks. A magnifying mirror may be used in crack detection of the box threads. Drill collars found to contain cracks should be considered unfit for further drilling service. Shop repair of cracked drill collars is typically possible if the unaffected area of the drill collar permits.
d. Use a profile gauge to check thread form and to check for stretched pins.
e. Check box counterbore diameter for swelling. In addition, use a straight edge on the crests of the threads in the box checking for rocking due to swelling of the box. Some machine shops may cut box counterbores larger than API standards, therefore, a check of the diameter of the counterbore may give a misleading result.
f. Check box and pin shoulders for damage. All field repairable damage shall be repaired by refacing and beveling. Excessive damage to shoulders should be repaired in reputable machine shops with API standard gauges.

13.5 DRILL COLLAR HANDLING SYSTEMS

13.5.1 While the recent increased usage of grooved drill collars has helped save time in tripping, it has introduced some potential dangers to rig floor operations. These problems can be minimized by strict adherence to regularly scheduled inspections.

13.5.2 When the elevator shoulder on a drill collar is new it is square and has sufficient area in contact with the elevator. (See Figures 87 and 88, and Table 28 for suggested dimensions on new drill collars and elevators.) As the collar is used for drilling, however, it wears as shown in Figure 89. Elevator contact area is decreased by collar OD wear and elevator spreading load is increased by angle and radius buildup on the collar and corresponding wear on the elevator seat. Elevator capacity is drastically reduced by spreading action as most all drill collar elevators are intended for use with square shoulders only. As an example, with $^1/_{16}$-inch wear on the collar OD, $^1/_{32}$-inch radius worn on the corner, and a 5-degree angle on the shoulder, elevator capacity can be reduced by as much as 40 to 60 percent, depending on collar size and elevator design.

13.5.3 Before this danger point is reached, the collar and elevator should be shopped and the shoulders brought back to a square condition. Be very sure the elevator shoulder radius on the drill collar is cold worked when shoulder is reworked. (See Section 14 for welding procedure limitations.) The top bore of the elevator should also be checked and corrected at this time as excessive slack between collar groove diameter and elevator bore decreases shoulder support area and can also let too much load shift to the elevator door.

13.5.4 The following inspection procedure for drill collar handling systems is recommended:

a. Thoroughly clean and examine elevator adapter for cracks.
b. Check links (commonly called bails) for cracks and measure them eye to eye to be certain they are within 1 inch of being the same length.
c. Thoroughly clean and examine drill collar elevator for cracks with magnetic particle inspection. Make certain that elevator safety latch works easily and works every time. Check top seat of elevator to be certain it is square. Check elevator top bore as follows:
   1. Center-latch elevator—latch elevator, then wedge front and back of elevator open and measure at largest part of top bore straight across between link arms. This method will measure total wear in bore (of which there will be very little), and wear on hinge pin and latch surfaces. Wear should not be allowed to go above $^1/_{32}$-inch on elevators for $5^5/_8$-inches and smaller drill collars, and $^1/_{16}$-inch for drill collars larger than $5^5/_8$-inches.
   2. Side-door elevators—latch elevator, then wedge latch open. Measure top bore from front to back. Use same wear allowance as for center-latch elevators.
d. Check elevator shoulder on drill collar to be certain it is square. (See 13.4 for inspection procedure for the drill collar.)
e. Examine drill collar slips for general condition and for correct size range for the collars being run. Look for cracks, missing cotter keys, loose liners, dull liner teeth, bent back tapers (from catching on drill collar shoulder), and bent handles.
f. Examine safety clamp for general condition. Look for cracks, missing cotter keys, galled or stripped threads, rounded-off nuts or wrenches, dull teeth, broken slip springs, and slips that do not move up and down easily.

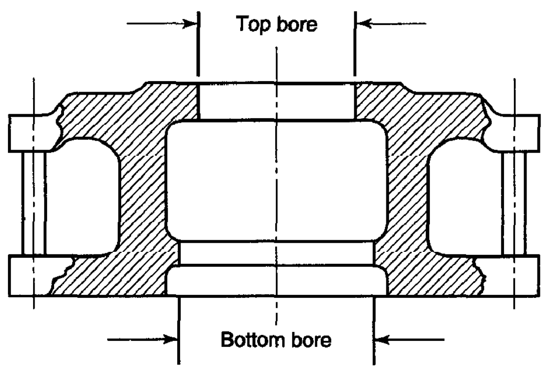


Figure 87—Drill Collar Elevator

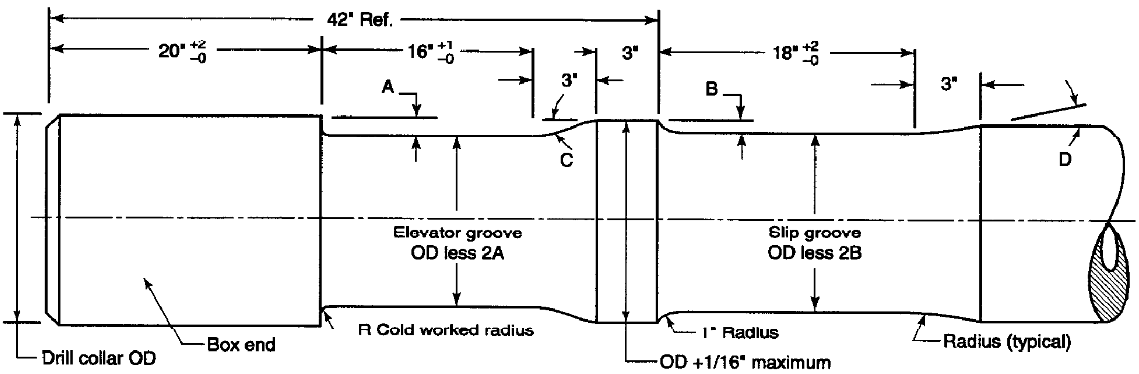


Figure 88—Drill Collar Grooves for Elevators and Slips

Table 28—Drill Collar Groove and Elevator Bore Dimensions

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)
	Groove Dimensions Based on Drill Collar OD					Elevator Bores Based on Drill Collar OD	
Drill Collar OD Ranges	Elev. Groove Depth A[1]	R	C[2]	Slip Groove Depth B[1]	D[2]	+0 Top Bore $-^1/_{32}$	$+^1/_{16}$ Bottom Bore $-c$
4 to $4^5/_8$	$^7/_{32}$	$^1/_8$	4°	$^3/_{16}$	$3^1/_2$°	OD minus $^5/_{16}$	OD plus $^1/_8$
$4^3/_4$ to $5^5/_8$	$^1/_4$	$^1/_8$	5°	$^3/_{16}$	$3^1/_2$°	OD minus $^3/_8$	OD plus $^1/_8$
$5^3/_4$ to $6^5/_8$	$^5/_{16}$	$^1/_8$	6°	$^1/_4$	5°	OD minus $^1/_2$	OD plus $^1/_8$
$6^3/_4$ to $8^5/_8$	$^3/_8$	$^3/_{16}$	$7^1/_2$°	$^1/_4$	5°	OD minus $^9/_{16}$	OD plus $^1/_8$
$8^3/_4$ and Larger	$^7/_{16}$	$^1/_4$	9°	$^1/_4$	5°	OD minus $^5/_8$	OD plus $^1/_8$

Notes:
All dimensions in inches unless otherwise specified.
These dimensions are not to be construed as being API standard.
[1]A and B dimensions are from nominal OD of new drill collar.
[2]Angle C and D dimensions are reference and approximate.

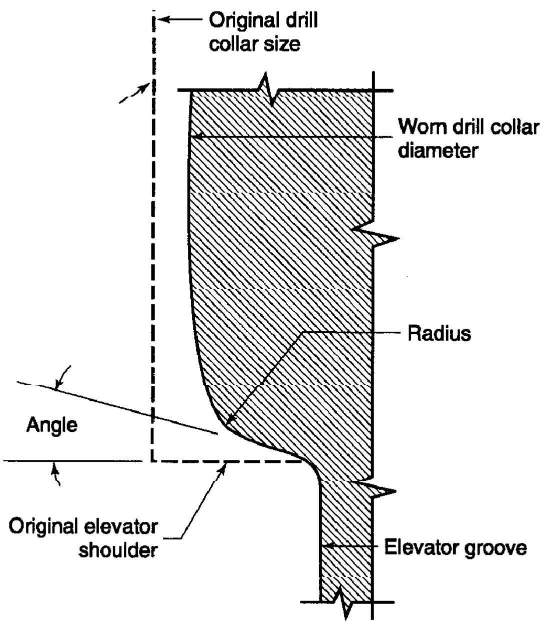


Figure 89—Drill Collar Wear

13.6 KELLYS

The following inspection procedure is recommended for used kellys:

a. Follow all steps listed in 13.4 for drill collar inspection procedure.

b. Examine junction between upsets and drive section for cracks.

c. Check corners of drive section for narrow wear surface particularly on hexagonal kellys. If wear surface does not extend at least $^1/_3$ across flat, the kelly drive bushings should be adjusted if possible and/or examined for wear.

d. Kelly straightness can be checked either of two ways:

1. By watching for excessive swing of the swivel and traveling block while drilling, or

2. By placing square kellys on level supports (one at each end of drive section), stretching a heavy cord from one end of a vertical face of the square to the other, measuring deflection, rolling kelly 90 degrees, and repeating procedure. On hexagon kellys, use the same method except kelly will need to be placed in 120-degree V-blocks so side face of drive section is vertical and deflection measurements taken on three successive sides (turning kelly through 60 degrees each time).

13.7 RECUT CONNECTIONS

The following is recommended for recut connections:

a. When rotary shouldered connections are inspected and found to require recutting, the recut connection should comply with the requirements of the section entitled "Gauging Practice, Rotary Shouldered Connections" of API Specification 7.

b. It is recommended that a benchmark be applied to the recut connection as suggested in 13.3.3.b.

13.8 PIN STRESS RELIEF GROOVES FOR RENTAL TOOLS AND OTHER SHORT TERM USAGE TOOLS

Following are recommendations for pin stress relief grooves for rental and other short term usage tools:

a. Laboratory fatigue tests and tests under actual service conditions have demonstrated the beneficial effects of stress relief contours at the pin shoulder. It is recommended that, where fatigue failures at points of high stress are a problem, relief grooves be provided. 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. Connection repairs are made primarily because of galling, shoulder leaks and handling damages while repairs caused by fatigue failures are secondary in occurrence.

Providers of short term usage rental tools have been reluctant to take advantage of the benefits of stress relief grooves because of the material loss in repairing shoulder and thread damage. Refacing the pin shoulder and reconditioning the threads is restricted by the tolerance on the groove width per API Specification 7.

b. 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 90. Dimensions for D_{RG} are found in Table 16 of API Specification 7.

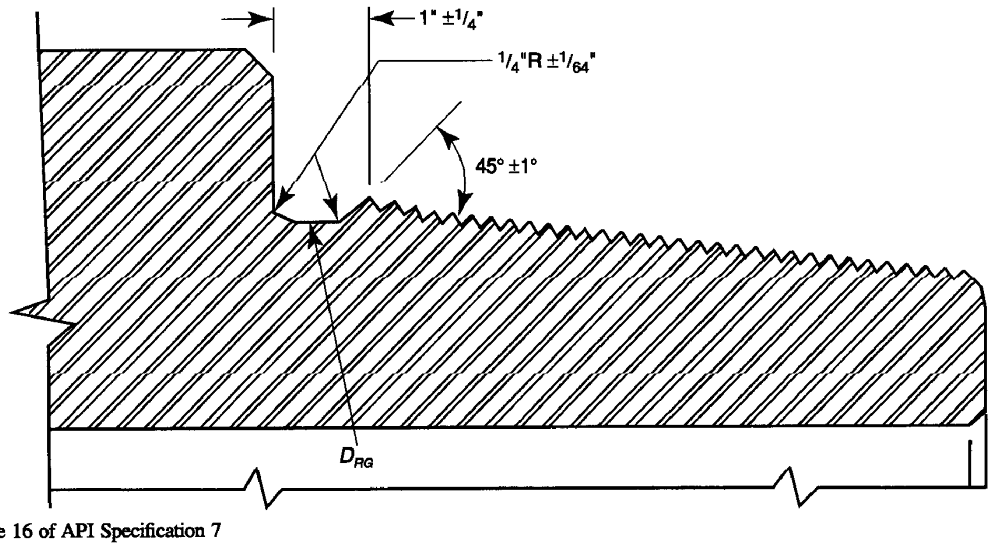
c. Recommended initial width of the modified stress relief groove is $^3/_4$ ($+^1/_{16}$, -0) inches. After reworking for damaged threads and shoulders, the width of the modified stress relief groove should not exceed $1^1/_4$ inches.

d. 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 29 shows calculated relative stresses for an NC50 axisymmetric finite element model with $6^1/_2$-inch box OD and 3-inch pin ID. A pin with no stress relief groove is the basis for comparison.

Table 29—Maximum Stress at Root of Last Engaged Thread for the Pin of an NC50 Axisymmetric Model

Load Condition	Note 1		Note 2	
SRG Width, inches	Maximum Equivalent Stress	Maximum Axial Stress	Maximum Equivalent Stress	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%

Notes:
1. Make-up only at 562,000 pounds axial force on shoulder.
2. 1,125,000 pounds axial tension applied to connection which causes shoulder separation.
3. Equivalent stress is equal to 0.707 $[(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2]^{1/2}$ where σ_1, σ_2, and σ_3 are principle stresses.
4. 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.



Note: See Table 16 of API Specification 7 for Dimension D_{RG}.

Figure 90—Modified Pin Stress-Relief Groove

Note that the least stress is expected for a groove width of 1 inch. Consequently, in operations where fatigue failures are a problem, a groove width of 1 inch is recommended. (See Figure 16 of API Specification 7.)

14 Welding on Down Hole Drilling Tools

14.1 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.

14.2 These are alloy steels and are normally in the heat treated state, these materials are not weldable unless proper procedures are used to prevent cracking and to recondition the sections where welding has been performed.

14.3 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.

15 Dynamic Loading Of Drill Pipe

Note: For quantitative results, see Reference 15, Appendix C.

15.1 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.

15.2 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.

15.3 Dynamic loading may exceed yield in long strings, such as 10,000 feet.

15.4 Dynamic loading increases with the length of drill collar string.

15.5 In the event the smallest value of the dynamic load tries to become negative, the pipe is kicked off the slips, and the string may be dropped into the hole.

15.6 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.

15.7 For a long drill pipe string, such as 10,000 feet, a jumpoff 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.

15.8 Dynamic phenomena are severe only when damping is small, which may 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.

15.9 In case of small damping, the running time of a stand of drill pipe should not be less than 15 seconds.

16 Classification Size and Make-Up Torque for Rock Bits

16.1 A classification system 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 was accepted by IADC in March 1987 and approval by API was proposed by the IADC subcommittee. Following API task group review it was recommended that API accept this system of classification for bit designation. The task group recommendation was adopted by the Committee on Standardization of Drilling and Servicing Equipment and subsequently approved by letter ballot. It was further determined that the approved form be included in API Recommended Practice 7G.

16.2 Bits may be classified and designated by an alphanumeric code on the bit carton with a series of three numbers and one letter keyed to the classification system shown in Tables 30 and 31.

16.3 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.

16.4 Type numbers 1 through 4 designate formation hardness subclassification from softest to hardest within each series classification.

16.5 The seven column listings under the heading, Features, include seven features common to the milled tooth and insert bits of most manufacturers. Columns 8 and 9 have been removed and reserved for future bit development.

16.6 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.

16.7 In using the form shown in Table 30, in conjunction with Table 31, 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 31.

Table 30—IADC Roller Bit Classification Chart

Manufacturer ____________ Revised: ____________

	Series	Formations	Type	Features						
				Standard Roller Bearing (1)	Roller Bearing, Air Cooled (2)	Roller Bearing, Gauge Protected (3)	Sealed Roller Bearing (4)	Sealed Roller Bearing, Gauge Protected (5)	Sealed Friction Bearing (6)	Sealed Friction Bearing, Gauge Protected (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							

Notes:
Bit classifications are general guidelines only. All bit types may drill effectively in formations other than these specified.

Table 31—IADC Bit Classification Codes Fourth Position

The following codes are used in the 4th position of the 4-character IADC bit classification code to indicate additional design features:

Code	Feature	Code	Feature
A	Air application[1]	N	
B		O	
C	Center Jet	P	
D	Deviation Control	Q	
E	Extended Jets[2]	R	Reinforced Welds[3]
F		S	Standard Steel Tooth Model[4]
G	Extra Gauge/Body Protection	T	
H		U	
I		V	
J	Jet Deflection	W	
K		X	Chisel Inserts
L		Y	Conical Insert
M		Z	Other Insert Shapes

[1]Journal bearing bits with air circulation nozzles.
[2]Full extension (welded tubes with nozzles). Partial extensions should be noted elsewhere.
[3]For percussion applications.
[4]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, will be designated 1-1-1-S on the bit carton. The manufacturer will also list this bit designation in this block on the form.

Table 32—Recommended Make-up Torque Ranges for Roller Cone Drill Bits

Connection	Minimum Make-up Torque ft/lb	Maximum Make-up Torque ft/lb
$2^3/_8$ API REG	3000	3500
$2^7/_8$ API REG	4500	5500
$3^1/_2$ API REG	7000	9000
$4^1/_2$ API REG	12000	16000
$6^5/_8$ API REG	28000	32000
$7^5/_8$ API REG	34000	40000
$8^5/_8$ API REG	40000	60000

Note:
Basis of calculations for recommended make-up torque assumed the use of a thread compound containing 40 to 60 percent by weight of finely powdered metallic zinc or 60 percent by weight of finely powdered metallic lead, with not more than 0.3 percent 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). Due to the irregular geometry of the ID bore in roller cone bits, torque valves are based on estimated cross-sectional areas and have been proven by field experience.

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

16.9 Recommended torque for roller cone bits is shown in Table 32. Recommended torque for diamond drill bits is shown in Table 33.

16.10 Common sizes for roller bits are listed in Table 34. Sizes other than those shown may be available in limited cutting structure types.

Common sizes for fixed cutter bits are listed in Table 35. Sizes other than those shown may be available in limited cutting structure types.

Table 33—Recommended Minimum Make-up Torques for Diamond Drill Bits

Connection	Maximum Pin ID in.	Bit Sub OD in.	Minimum Make-Up Torque ft/lb
$2^3/_8$ API REG	1	3	1791*
		$3^1/_8$	2419*
		$3^1/_4$	3085*
$2^7/_8$ API REG	$1^1/_4$	$3^1/_2$	3073*
		$3^3/_4$	4617
		$3^7/_8$	4658
$3^1/_2$ API REG	$1^1/_2$	$4^1/_8$	5171*
		$4^1/_4$	6306*
		$4^1/_2$	7660
$4^1/_2$ API REG	$2^1/_4$	$5^1/_2$	12451*
		$5^3/_4$	16476*
		6	17551
		$6^1/_4$	17757
$6^5/_8$ API REG	$3^1/_4$	$7^1/_2$	37100*
		$7^3/_4$	37857
		8	38193
		$8^1/_4$	38527
$7^5/_8$ API REG	$3^3/_4$	$8^1/_2$	48296*
		$8^3/_4$	57704*
		9	59966
		$9^1/_4$	60430
		$9^1/_2$	60895

*Note:

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.

Basis of calculations for recommended make-up torque assumed the use of a thread compound containing 40 to 60 percent by weight of finely powdered metallic zinc or 60 percent by weight of finely powdered metallic lead, with not more than 0.3 percent total active sulfur, applied thoroughly to all threads and shoulders (reference the *CAUTION* regarding the use of hazardous materials in Appendix G of API Specification 7) and using the modified Screw Jack formula in A.8 and a unit stress of 50,000 psi in the box or pin, whichever is weaker.

Normal torque range is tabulated value plus 10 percent. Higher torque values may be used under extreme conditions.

Table 34—Common Roller Bit Sizes

Size of Bit, in.	Size of Bit, in.
$3^3/_4$	$9^1/_2$
$3^7/_8$	$9^7/_8$
$4^3/_4$	$10^5/_8$
$5^7/_8$	11
6	$12^1/_4$
$6^1/_8$	$13^1/_2$
$6^1/_4$	$14^3/_4$
$6^1/_2$	16
$6^3/_4$	$17^1/_2$
$7^7/_8$	20
$8^3/_8$	22
$8^1/_2$	24
$8^3/_4$	26

Table 35—Common Fixed Cutter Bit Sizes

Size of Bit, in.	Size of Bit, in.
$3^7/_8$	$8^1/_2$
$4^1/_2$	$8^3/_4$
$4^3/_4$	$9^1/_2$
$5^7/_8$	$9^7/_8$
6	$10^5/_8$
$6^1/_8$	$12^1/_4$
$6^1/_4$	$14^3/_4$
$6^1/_2$	16
$6^3/_4$	$17^1/_2$
$7^7/_8$	

APPENDIX A—STRENGTH AND DESIGN FORMULAS

A.1 Torsional Strength of Eccentrically Worn Drill Pipe

Assume 1: Eccentric hollow circular section (see Figure A-1).
Reference: *Formulas for Stress & Strain*, Roark, 3rd Edition.

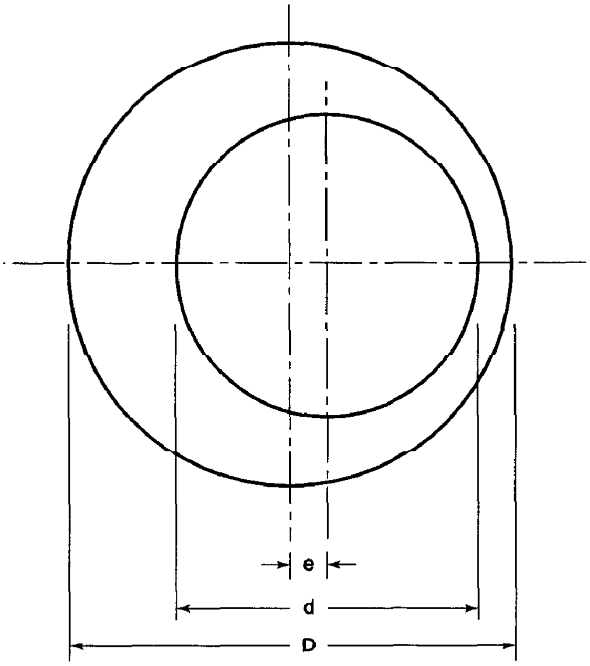


Figure A-1—Eccentric Hollow Section of Drill Pipe

$$T = \frac{\pi S_s(D^4 - d^4)}{12 \times 16 \times D \times F}, \tag{A.1}$$

where

$$F = 1 + \frac{4N^2\phi}{(1-N^2)} + \frac{32N^2\phi^2}{(1-N^2)\ (1-N^4)} + \frac{48N^2(1+2N^2+3N^4+2N^6)\phi^3}{(1-N^2)\ (1-N^4)\ (1-N^6)},$$

$N = d/D$,

$\phi = \dfrac{e}{D}$,

T = torque, ft-lbs.,
S_s = minimum shear strength, psi,
D = outside diameter, in.,
d = inside diameter, 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 4, and Class 2, Table 6 were calculated from Equation A.1, using the assumption that wear is uniform on the external surface.

A.2 Safety Factors

Values for various performance properties of drill pipe are given in Tables 2 through 7. The values shown are minimum values and do not include factors of safety. In the design of drill pipe strings, factors of safety should be used as are considered necessary for the particular application.

A.3 Collapse Pressure for Drill Pipe

Note: See API Bulletin 5C3 for derivation of equations in A.3.

The minimum collapse pressures given in Tables 3, 5, and 7 are calculated values determined from equations in API Bulletin 5C3. Equations A.2 through A.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.

$$P_c = 2Y_m\left[\frac{(D/t)-1}{(D/t)^2}\right] \tag{A.2}$$

Applicable D/t ratios for application of Equation A.2 are as follows:

Grade	*D/t* Ratio
E75	13.60 and less
X95	12.85 and less
G105	12.57 and less
S135	11.92 and less

For minimum collapse failure in the plastic range:

$$P_c = Y_m\left[\left(\frac{A'}{D/t}\right) - B'\right] - C \tag{A.3}$$

Factors and applicable D/t ratios for application of Equation A.3 are as follows:

	Formula Factors			
Grade	A'	B'	C	*D/t* Ratio
E75	3.054	0.0642	1806	13.60 to 22.91
X95	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:

$$P_c = Y_m\left[\left(\frac{A}{D/t}\right) - B\right] \qquad \text{(A.4)}$$

Factors and applicable D/t ratios for application of Equation A.4 are as follows:

	Formula Factors		
Grade	A	B	D/t Ratio
E75	1.990	0.0418	22.91 to 32.05
X95	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:

$$P_c = \frac{46.95 \times 10^6}{(D/t)[(D/t)-1]^2} \qquad \text{(A.5)}$$

Applicable D/t ratios for application of Equation A.5 are as follows:

Grade	D/t Ratio
E75	32.05 and greater
X95	28.36 and greater
G105	26.89 and greater
S135	23.44 and greater

where

P_c = minimum collapse pressure, psi,

$*D$ = nominal outside diameter, in.,

$*t$ = nominal wall thickness, in.,

Y_m = material minimum yield strength, psi.

Notes:
*Collapse pressures for used drill pipe are determined by adjusting the nominal outside diameter, D, and wall thickness, t, as if the 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 used drill pipe follow. These values are to be used in applicable Equation A.2, A.3, A.4, or A.5, depending on the D/t ratio, to determine collapse pressure.

Premium Class: t = (0.80) (nominal wall), D = nominal OD – (0.40) (nominal wall)

Class 2: t = (0.70) (nominal wall), D = nominal OD – (0.60) (nominal wall)

A.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:

$$L_1 = \frac{E \times e \times W_{dp}}{40.8\,P}, \qquad \text{(A.6)}$$

where

L_1 = length of free drill pipe, ft.,

E = modulus of elasticity, lb/in.,[2]

e = differential stretch, in.,

W_{dP} = weight per foot of pipe, lbs/ft.,

P = differential pull, lbs.

Where $E = 30 \times 10^6$, this formula becomes:

$$L_1 = \frac{735{,}294 \times e \times W_{dP}}{P} \qquad \text{(A.7)}$$

A.5 Internal Pressure

A.5.1 DRILL PIPE

$$P_i = \frac{2Y_m t}{D}, \qquad \text{(A.8)}$$

where

P_i = internal pressure, psi,

Y_m = material minimum yield strength, psi,

t = remaining wall thickness of tube, in.,

D = nominal outside diameter of tube, in.

Notes:
1. Internal pressures for new drill pipe in Table 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 12½ percent.
2. Internal pressures for used drill pipe were determined by adjusting the nominal wall thickness according to footnotes below Table 5 and 7 and using the nominal outside diameter, in the above Equation A.8.

A.5.2 KELLYS

$$P_i = \frac{Y_m[D_{FL}^2 - (D_{FL}-2t)^2]}{\sqrt{3(D_{FL})^4 + (D_{FL}-2t)^4}}, \qquad \text{(A.9)}$$

where

P_i = internal pressure, psi,

Y_m = material minimum yield strength, psi,

D_{FL} = distance across drive section flats, in.,

t = minimum wall, in.

Note: The dimension t is the minimum wall thickness of the drive section and must be determined in each case through the use of an ultrasonic thickness gauge or similar device.

A.6 Stretch of Suspended Drill Pipe

When pipe is freely suspended in a fluid, the stretch due to its own weight is:

$$e = \frac{L_1^2}{24E}\,[W_a - 2W_f(1-\mu)], \qquad \text{(A.10)}$$

where

e = stretch, in.,

L_1 = length of free drill pipe, ft.,

E = modulus of elasticity, psi,
W_a = weight of pipe material, lb/cu ft.,
W_f = weight of fluid, lb/cu ft.,
m = Poisson's ratio.

For steel pipe where W_s = 489.5 lb/cu ft, $E = 30 \times 10^6$ psi and $\mu = 0.28$, this formula will be:

$$e = \frac{L_1^2}{72 \times 10^7}\,[489.5 - 1.44W_f]\,, \qquad \text{(A.11)}$$

or

$$e = \frac{L_1^2}{9.625 \times 10^7}\,[65.44 - 1.44W_g]\,, \qquad \text{(A.12)}$$

where
W_f = weight of fluid, lb/cu ft.,
W_g = weight of fluid, lb/gal.

A.7 Tension

$$P = Y_m A, \qquad \text{(A.13)}$$

where
P = minimum tensile strength, lbs.,
Y_m = material minimum yield strength, psi,
A = cross-section area, sq. in. (Table 1, Column 6, for drill pipe).

A.8 Torque Calculations for Rotary Shouldered Connections (see Table A-1 and Figure A-2)

A.8.1 TORQUE TO YIELD A ROTARY SHOULDERED CONNECTION

$$T_y = \frac{Y_m A}{12}\left(\frac{p}{2\pi} + \frac{R_t f}{\cos\theta} + R_s f\right),$$

where
T_y = turning moment or torque required to yield, ft-lbs.,
Y_m = material minimum yield strength, psi,
p = lead of thread, in.,
f = coefficient of friction on mating surfaces, threads and shoulders, assumed 0.08 for thread compounds containing 40 to 60 percent by weight of finely powdered metallic zinc. (Reference the caution regarding the use of hazardous materials in Appendix G of Specification 7.),
q = $\frac{1}{2}$ included angle of thread (Figures 21 or 22, Specification 7), degrees,

$$R_t = \frac{C + [C - (L_{pc} - .625) \times tpr \times \frac{1}{12}\,]}{4},$$

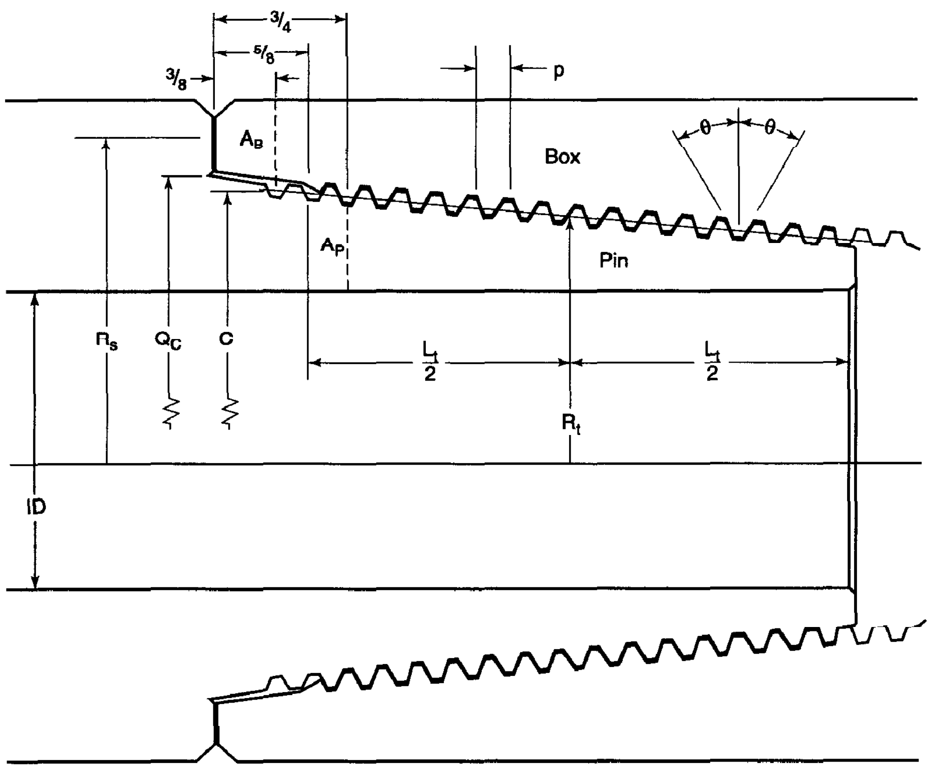


Figure A-2—Rotary Shouldered Connection

L_{pc} = length of pin (Specification 7, Table 25, Column 9), in.,

R_s = $^1/_4$ (OD + Q_c), in.

The maximum value of R_s is limited to the value obtained from the calculated OD where $A_p = A_b$,

A = cross-section area A_b or A_p whichever is smaller, sq. in.

where

$$A_p = \frac{\pi}{4}\left[(C - B)^2 - ID^2\right] = \text{without relief grooves,}$$

or

$$A_p = \frac{\pi}{4}\left[(D_{RG}{}^2 - ID^2\right] = \text{with relief grooves,}$$

where

D_{RG} = diameter of relief groove (Specification 7, Table 16, Column 5), in.,

C = pitch diameter of thread at gauge point (Specification 7, Table 25, Column 5), in.,

ID = inside diameter, in.,

$$B = 2\left(\frac{H}{2} - S_{rs}\right) + tpr \times {}^1/_8 \times {}^1/_{12},$$

H = thread height not truncated (Specification 7, Table 26, Column 3), in.,

S_{rs} = root truncation (Specification 7, Table 26, Column 5), in.,

tpr = taper (Specification 7, Table 25, Column 4), in./ft.,

$$A_b = \frac{\pi}{4}\left[OD^2 - (Q_c - E)^2\right],$$

where

OD = outside diameter, in.,

Q_c = box counterbore (Specification 7, Table 25, Column 11), in.,

E = $tpr \times {}^3/_8 \times {}^1/_{12}$

A.8.2 MAKE-UP TORQUE FOR ROTARY SHOULDERED CONNECTIONS

$$T = \frac{SA}{12}\left(\frac{p}{2\pi} + \frac{R_t f}{\cos\theta} + R_s f\right),$$

where

A = A_b or A_P whichever is smaller; A_P shall be based on pin connections without relief grooves, sq. in.,

S = recommended make-up stress level, psi.

Note: For values of S, see 4.8.1 for Tool Joints and 5.2 for Drill Collars.

A.8.3 COMBINED TORSION AND TENSION TO YIELD A ROTARY SHOULDERED CONNECTION

Figure A-3 shows the limits for combined torsion and tension for a rotary shouldered connection. The connection nomenclature is defined in A.8.1. The loads considered in this simplified approach are torsion and tension. Bending and internal pressure are not included, nor is the contribution of shear stress due to torsion. A design factor of 1.1 should be used to provide some safety margin. This safety margin may not be sufficient for cases involving severe bending or elevated temperature. (See 4.5.)

The failure criteria is either torsional yield or shoulder separation.

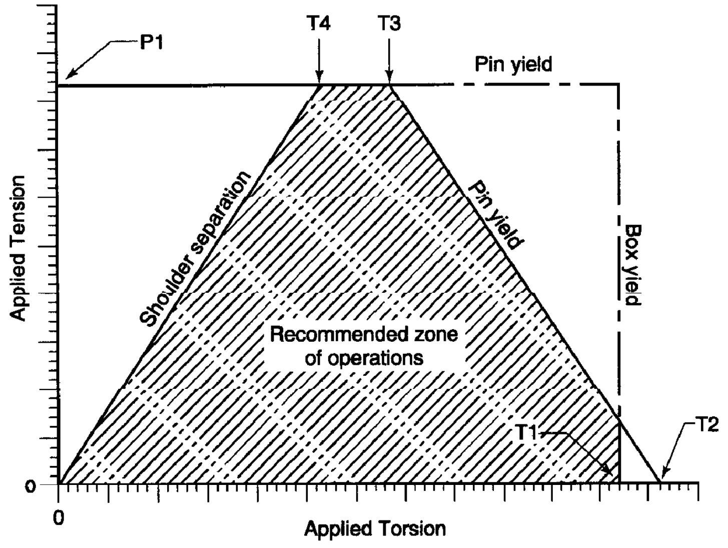


Figure A-3—Limits for Combined Torsion and Tension for a Rotary Shouldered Connection

The end points for the limits lines are defined by five equations:

$$P1 = \left(\frac{Y_m}{1.1}\right)A_p$$

$$T1 = \left(\frac{Y_m}{1.1 \times 12}\right)A_b\left(\frac{p}{2\pi} + \frac{R_t f}{\cos\theta} + R_s f\right)$$

$$T2 = \left(\frac{Y_m}{1.1 \times 12}\right)A_p\left(\frac{p}{2\pi} + \frac{R_t f}{\cos\theta} + R_s f\right)$$

$$T3 = \left(\frac{Y_m}{1.1 \times 12}\right)A_p\left(\frac{p}{2\pi} + \frac{R_t f}{\cos\theta}\right)$$

$$T4 = \left(\frac{Y_m}{1.1 \times 12}\right)\left(\frac{A_p A_b}{A_p + A_b}\right)\left(\frac{p}{2\pi} + \frac{R_t f}{\cos\theta} + R_s f\right)$$

Depending on the connection geometry, $T3$ may be greater or smaller than $T4$. The same is true for $T1$ and $T2$.

The line (0,0) to ($T4$, $P1$) represents shoulder separation for low makeup torque. The line ($T2$, 0) to ($T3$, $P1$) represents pin yield under the combination of torque and tension. The line ($T1$, 0) to ($T1$, $P1$) represents box yield due to torsion. The horizontal line from $P1$ represents maximum tension load on the pin.

A.9 Drill Pipe Torsional Yield Strength

A.9.1 PURE TORSION ONLY

$$Q = \frac{0.096167 J Y_m}{D}, \qquad \text{(A.15)}$$

where

Q = minimum torsional yield strength, ft-lb.,

Y_m = material minimum yield strength, psi,

J = polar moment of inertia

$= \frac{\pi}{32}(D^4 - d^4)$ for tubes

$= 0.098175\,(D^4 - d^4)$,

D = outside diameter, in.,

d = inside diameter, in.

A.9.2 TORSION AND TENSION

$$Q_T = \frac{0.096167 J}{D}\sqrt{Y_m^2 - \frac{P^2}{A^2}}, \qquad \text{(A.16)}$$

where

Q_T = minimum torsional yield strength under tension, ft-lb.,

J = polar moment of inertia

$= \frac{\pi}{32}(D^4 - d^4)$ for tubes

$= 0.098175\,(D^4 - d^4)$,

D = outside diameter, in.,

d = inside diameter, in.,

Y_m = material minimum yield strength, psi,

P = total load in tension, lbs.,

A = cross section area, sq. in.

A.10 Drill Collar Bending Strength Ratio

The bending strength ratios in Figures 26 through 32 were determined by application of Equation A.17. The effect of stress-relief features was disregarded.

$$BSR = \frac{Z_B}{Z_P} = \frac{0.098\,\frac{(D^4 - b^4)}{D}}{0.098\,\frac{(R^4 - d^4)}{R}} = \frac{\frac{D^4 - b^4}{D}}{\frac{R^4 - d^4}{R}}, \qquad \text{(A.17)}$$

where

BSR = bending strength ratio,

Z_B = box section modulus, cu. in.,

Z_P = pin section modulus, cu. in.,

D = outside diameter of pin and box (Figure A-4), in.,

d = inside diameter or bore (Figure A-4), in.,

b = thread root diameter of box threads at end of pin (Figure A-4), in.,

R = thread root diameter of pin threads $^3/_4$ inch from shoulder of pin (Figure A-4), in.

To use Equation A.17, first calculate:

Dedendum, b, and R

$$Dedendum = \frac{H}{2} - f_m, \qquad \text{(A.18)}$$

where

H = thread height not truncated, in.,

f_m = root truncation, in.

$$b = C - \frac{tpr(L_{pc} - .625)}{12} + (2 \times dedendum), \qquad \text{(A.19)}$$

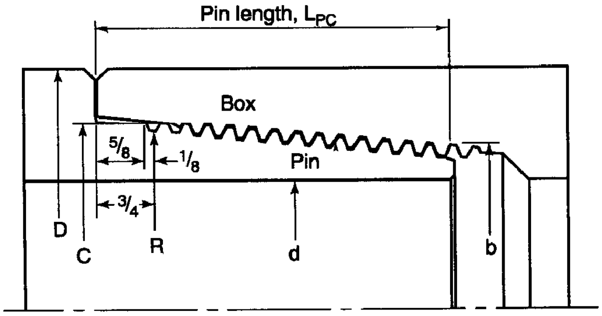


Figure A-4—Rotary Shouldered Connection Location of Dimensions for Bending Strength Ratio Calculations

where

C = pitch diameter at gauge point, in.,
tpr = taper, in./ft.

$$R = C - (2 \times dedendum) - (tpr \times {}^1/_8 \times {}^1/_{12}) \qquad (A.20)$$

An example of the use of Equation A.17 in determining the bending strength of a typical drill collar connection is as follows:

Determine the bending strength ratio of drill collar NC46-62 ($6^1/_4$ OD x $2^{13}/_{16}$) ID connection.

D = 6.25 (Specification 7, Table 13, Column 2),
d = $2^{13}/_{16}$ = 2.8125 (Specification 7, Table 13, Column 3),
C = 4.626 (Specification 7, Table 25, Column 5),
Taper = 2 (Specification 7, Table 25, Column 4),
L_{pc} = 4.5 (Specification 7, Table 25, Column 9),
H = 0.216005 (Specification 7, Table 26, Column 3),
f_m = 0.038000 (Specification 7, Table 26, Column 5).

First calculate dedendum, b, and R

$$Dedendum = \frac{H}{2} - f_m = \frac{.216005}{2} - .038000 = .0700025$$

$$b = \frac{C - tpr(L_{pc} - .625)}{12} + (2 \times dedendum)$$

$$b = 4.626 - \frac{2(4.5 - .625)}{12} + (2 \times .0700025)$$

$$b = 4.120,$$

$$R = C - (2 \times dedendum) - (tpr \times {}^1/_8 \times {}^1/_{12})$$

$$R = 4.626 - (2 \times .0700025) - (2 \times {}^1/_8 \times {}^1/_{12})$$

$$R = 4.465.$$

Substituting these values in Equation A.16 determines the bending strength ratio as follows:

$$BSR\ (NC46\text{-}62) = \frac{\frac{D^4 - b^4}{D}}{\frac{R^4 - d^4}{R}}$$

$$= \frac{\frac{(6.25)^4 - (4.120)^4}{6.25}}{\frac{(4.465)^4 - (2.8125)^4}{4.465}}$$

$$= 2.64:1$$

A.11 Torsional Yield Strength of Kelly Drive Section

The torsional yield strength of the kelly drive section values listed in Tables 15 and 17 were derived from the following equation:

$$Y = \frac{0.577 Y_m [0.200(a^3 - b^3)]}{12},$$

where

Y_m = tensile yield, psi,
a = distance across flats, in.,
b = kelly bore, in.

A.12 Bending Strength, Kelly Drive Section

The yield in bending values of the kelly drive section listed in Tables 15 and 17 were determined by one of the following equations:

a. Yield in bending through corners of the square drive section, Y_{BC}, ft-lb:

$$Y_{BC} = \frac{Y_m (0.118 a^4 - 0.069 b^4)}{12a}$$

b. Yield in bending through the faces of the hexagonal drive section Y_{BF}, ft-lb:

$$Y_{BF} = \frac{Y_m (0.104 a^4 - 0.085 b^4)}{12a}$$

A.13 Approximate Weight of Tool Joint Plus Drill Pipe

Approximate Weight of Tool Joint Plus Drill Pipe Assembly, lb/ft =

$$\frac{\left(\begin{array}{c}\text{Approximate Adjusted} \\ \text{Weight of Drill Pipe x 29.4}\end{array} + \begin{array}{c}\text{Approximate Weight} \\ \text{of Tool Joint}\end{array}\right)}{\text{Tool Joint Adjusted Length + 29.4}}, \qquad (A.21)$$

where

Approximate Adjusted Weight of Drill Pipe, lb/ft =

$$\text{Plain End Weight} + \frac{\text{Upset Weight}}{29.4} \qquad \text{(A.22)}$$

Plain end weight and upset weight are found in API Specification 5D.

Approximate Weight of Tool Joint, lbs =

$$0.222\,L(D^2 - d^2) + 0.167\,(D^3 - D_{TE}^3) - 0.501\,d^2(D - D_{aE}) \qquad \text{(A.23)}$$

Dimensions for L, D, d, and D_{TE} are in API Specification 7, Figure 6 and Table 7.

Adjusted Length of Tool Joint, ft =

$$\frac{L + 2.253\,(D - D_{TE})}{12} \qquad \text{(A.24)}$$

A.14 Critical Buckling Force for Curved Boreholes[27,29,30,31,32]

A.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.

$$\text{if } F_b < \frac{4 \times E \times I}{12 \times h_c \times R_L} \text{ pipe not buckled,}$$

$$\text{if } F_b \geq \frac{4 \times E \times I}{12 \times h_c \times R_L},$$

$$W_{eq} = \frac{12 \times h_c \times F_b^2}{4 \times E \times I},$$

$$B_{Vmin} = \frac{-5730}{F_b}\left[\left(W_{eq}^2 - \left(\frac{F_b}{R_L}\right)^2\right)^{1/2} + W_m \times \sin\theta\right],$$

$$B_{Vmax} = \frac{5730}{F_b}\left[\left(W_{eq}^2 - \left(\frac{F_b}{R_L}\right)^2\right)^{1/2} - W_m \times \sin\theta\right],$$

where

F_b = critical buckling force (+ compressive) (lb),
B_{Vmin} = minimum vertical curvature rate to cause buckling (+ building, – dropping) (°/100 ft),
B_{Vmax} = maximum vertical curvature rate that buckles pipe (+ building, – dropping) (°/100 ft),
W_{eq} = equivalent pipe weight required to buckle pipe at F_b axial load,
E = 29.6×10^6 psi,
$I = \dfrac{0.7854(OD^4 - ID^4)}{16}$,
$W_m = W_a\left(\dfrac{65.5 - MW}{65.5}\right)$ buoyant weight of pipe (lb/ft),
W_a = actual weight in air (lb/ft),
MW = mud density (lb/gal),
$h_c = \left(\dfrac{D_H - TJOD}{2}\right)$ radial clearance of tool joint to hole (in.),
D_H = diameter of hole (in.),
$TJOD$ = OD tool joints (in.),
$B_L = \sqrt{B_T^2 - B_V^2}$ lateral curvature rate (°/100 ft),
B_T = total curvature rate (°/100 ft),
$R_L = \dfrac{5730}{B_L}$ lateral build radius (ft),
θ = inclination angle (deg).

A.14.2 If the hole curvature is limited to the vertical plane, the buckling equations simplify to the following:

$$W_{eq} = \frac{12 \times h_c \times F_b^2}{4 \times E \times I},$$

$$B_{Vmin} = \frac{-5730 \times (W_{eq} + W_m \times \sin\theta)}{F_b},$$

$$B_{Vmax} = \frac{5730 \times (W_{eq} - W_m \times \sin\theta)}{F_b}.$$

where

B_{Vmin} = minimum vertical curvature rate for buckling (+ building, – dropping) (°/100 ft),
B_{Vmax} = maximum vertical curvature rate for buckling (+ building, – dropping) (°/100 ft),
F_b = buckling force (lb),
E = 29.6×10^6 (psi),
$I = \dfrac{\pi}{64}(OD^4 - ID^4)$,
W_{eq} = buoyant weight equivalent for pipe in curved borehole (lb/ft),
$W_m = W_a\left(\dfrac{65.5 - MW}{65.5}\right)$ buoyant weight of pipe (lb/ft),
W_a = actual weight of pipe in air (lb/ft),
MW = mud density (lb/gal),
$h_c = \left(\dfrac{DH - TJOD}{2}\right)$ radial clearance of tool joint to hole (in.),
DH = diameter of hole (in.),
$TJOD$ = OD of tool joint (in.),
θ = inclination angle (deg).

A.14.3 Figures A-5 and A-6 show the effect of hole curvature on the buckling force for 5-inch and 3½-inch drillpipe. Figure A-7 shows the effect of lateral curvatures on the buckling force of 5-inch drillpipe. For lateral and upward curvatures, the critical buckling force increases with the total curvature rate.

A.15 Bending Stresses on Compressively Loaded Drillpipe in Curved Boreholes[33,34]

A.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.

$$B_c = \frac{57.3 \times 100 \times 12 \times \Delta D}{J \times L\left[\tan\left(\frac{57.3 \times L}{4 \times J}\right) - \frac{L}{4 \times J}\right]},$$

where

B_c = the critical hole curvature that defines the transition from no pipe body contact to point contact (°/100 ft),

ΔD = $(TJOD - OD)$,

$TJOD$ = tool joint OD (in.),

OD = pipe body OD (in.),

$J = \left(\frac{E \times I}{F}\right)^{1/2}$ (in.),

L = length of one joint of pipe (in.),

E = Young's modulus 30 x 10^6 for steel (psi),

I = moment of inertia of pipe body (in.)

$= \frac{\pi(OD^4 - ID^4)}{64},$

F = axial compressive load on pipe (lb),

ID = pipe body ID (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 = the critical curvature that defines the transition from point contact to wrap contact (°/100 ft).

ΔD = $(TJOD - OD)$,

$TJOD$ = tool joint OD (in.),

OD = pipe body OD (in.),

$J = \left(\frac{E \times I}{F}\right)^{1/2}$ (in.),

L = length of one joint of pipe (in.),

E = Young's modulus 30 x 10^6 for steel (psi),

I = moment of inertia of pipe ($in.^4$)

$= \frac{\pi(OD^4 - ID^4)}{64},$

F = axial compressive load on pipe (lb),

ID = pipe body ID (in.).

A.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:

$$S_b = \frac{B \times OD \times F \times J \times L}{57.3 \times 100 \times 12 \times 4 \times I \times \sin\left(\frac{57.3L}{2J}\right)},$$

where

S_b = maximum bending stress (psi),

B = hole curvature,

F = axial compressive load on pipe (lb),

$J = \left(\frac{E \times I}{F}\right)^{1/2}$ (in.),

E = Young's modulus 30 x 10^6 for steel (psi),

I = moment of inertia (in.)

$= \frac{\pi(OD^4 - ID^4)}{64},$

OD = pipe body OD (in.),

ID = pipe body ID (in.).

L = length of one joint of pipe (in.),

A.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:

$$S_b = \frac{E \times OD \times U^2}{4R}\left[\frac{A \times \sin\theta + B \times \cos\theta}{U \times \sin U - 4 \times \sin^2\left(\frac{U}{2}\right)}\right],$$

where

E = Young's modulus, 29.6 x 10^6 for steel (psi),

OD = pipe body OD (in.),

ID = pipe body ID (in.),

$U = \frac{L}{2J},$

L = length of one joint of drillpipe for point contact of pipe body (in.),

L = L_c for wrap contact (in.),

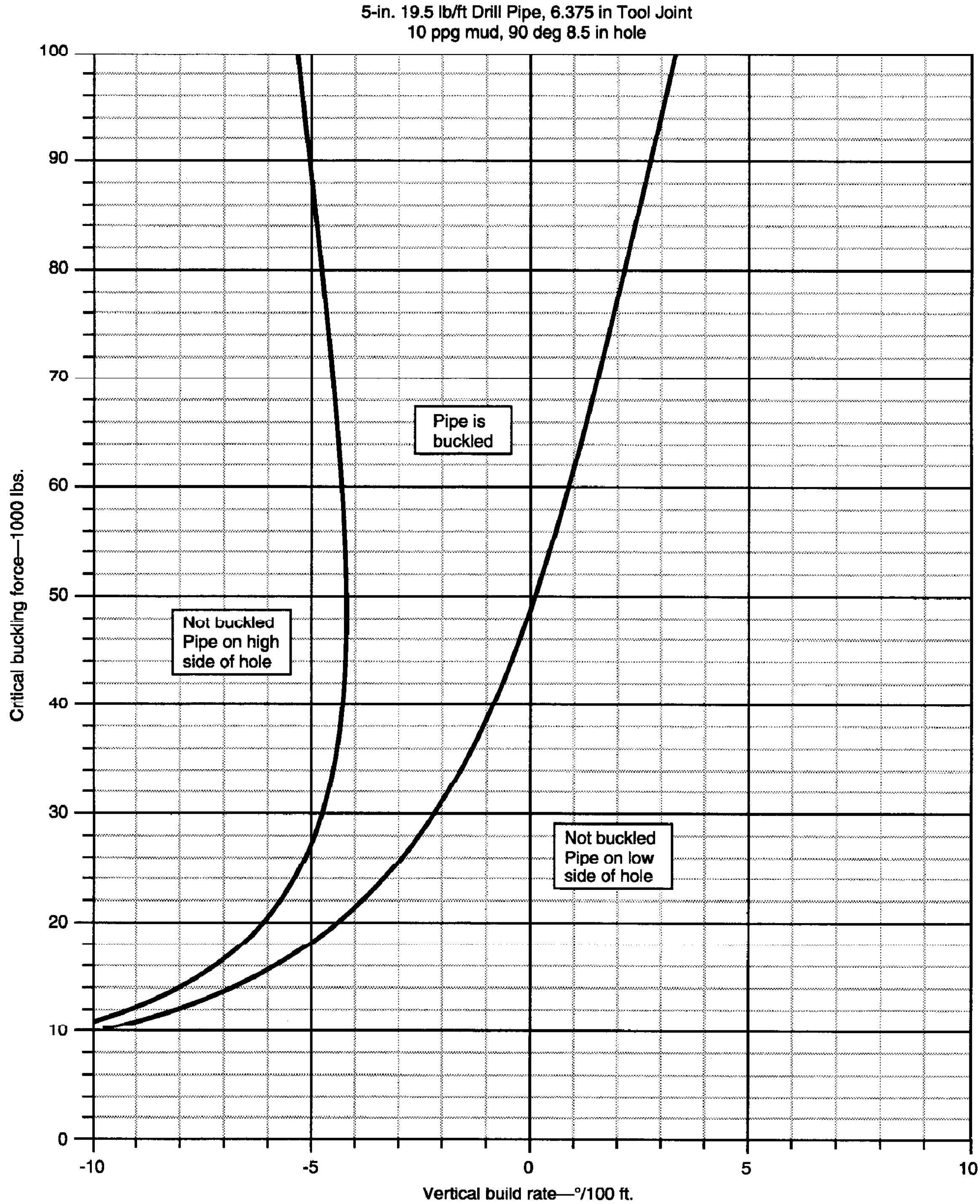


Figure A-5—Buckling Force vs Hole Curvature

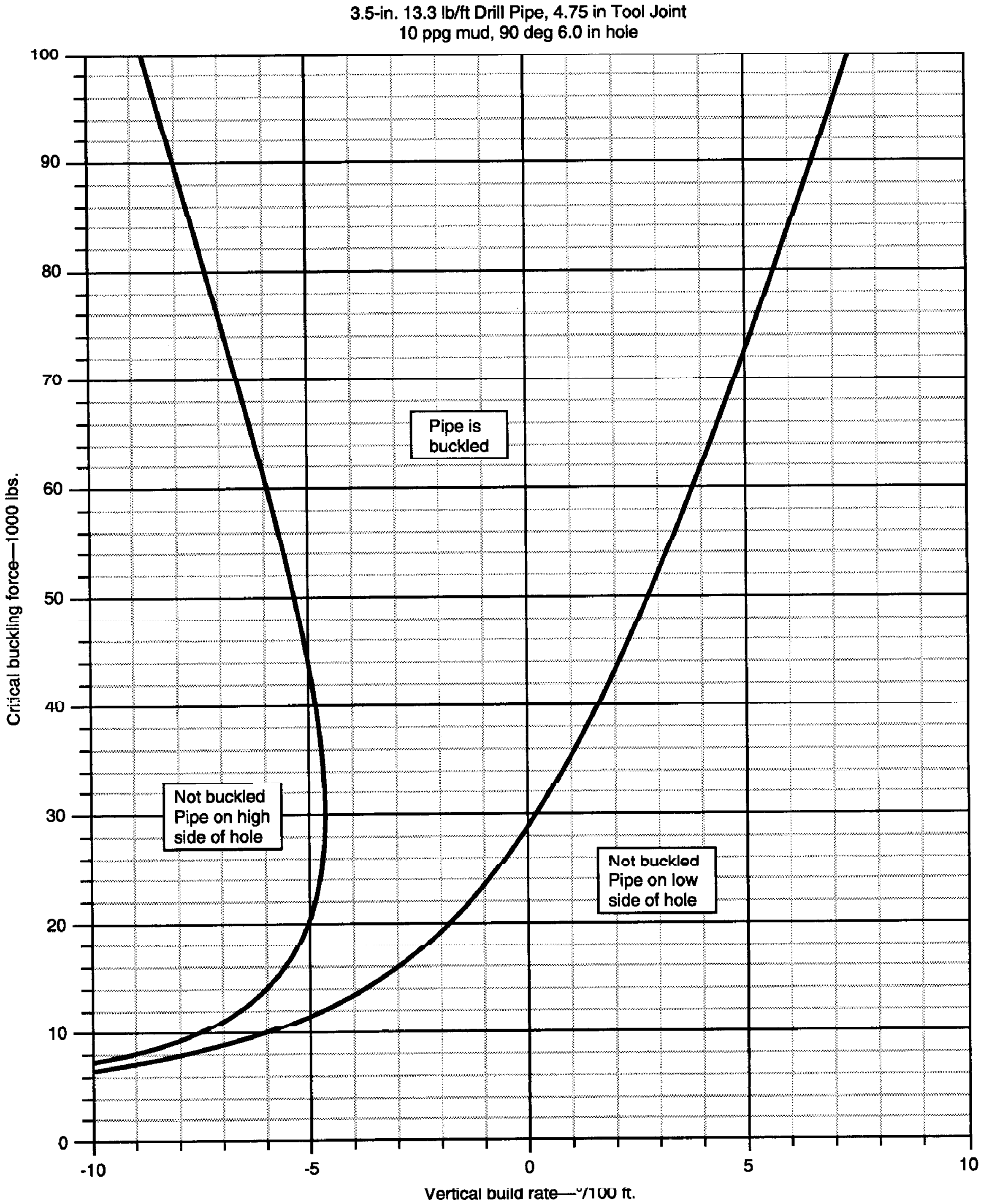


Figure A-6—Buckling Force vs Hole Curvature

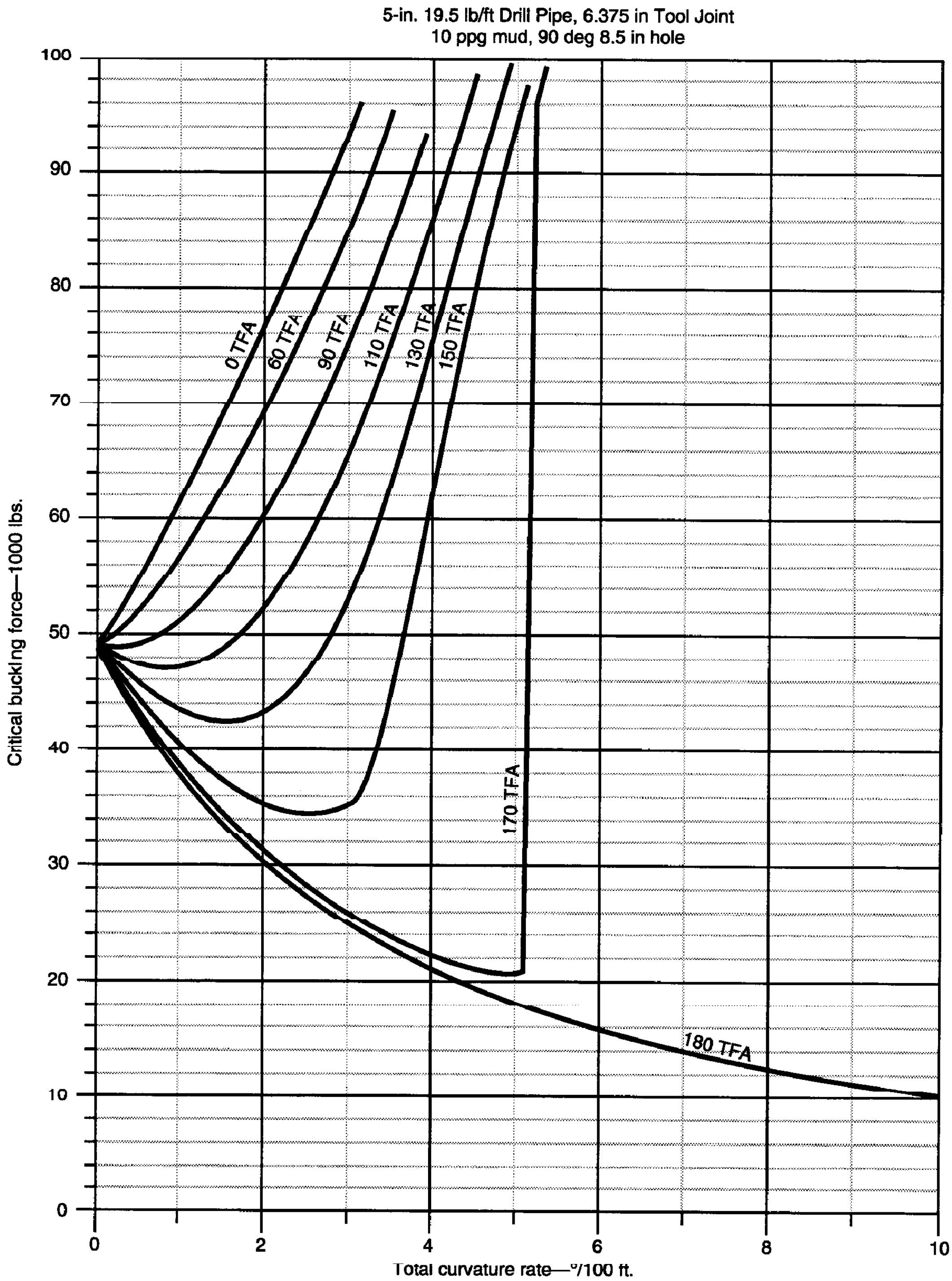


Figure A-7—Buckling Force vs Hole Curvature

$$A = \left(1 + \frac{4 \times R \times \Delta D}{L^2}\right)\sin U - \frac{4}{U} \times \sin^2\left(\frac{U}{2}\right),$$

$$B = 2\left[1 - \frac{\sin U}{U}\left(1 + \frac{4 \times R \times \Delta D}{L^2}\right)\sin^2\left(\frac{U}{2}\right)\right],$$

$$\theta = \arctan\left(\frac{A}{B}\right),$$

$$J = \left(\frac{E \times I}{F}\right)^{1/2} \text{ (in.)},$$

$$I = \frac{\pi}{64}(OD^4 - ID^4) \text{ (in.}^4\text{)},$$

ΔD = diameter difference tool joint minus pipe body OD,
ΔD = $(TJOD - OD)$ (in.),
$TJOD$ = tool joint OD (in.),
R = 57.3 x 100 x 12B,
B = hole curvature (°/100 ft.).

A.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:

$$B = \frac{57.3 \times 100 \times 12 \times \Delta D}{J \times L_e\left[\frac{4J}{L_e} + \frac{L_e}{4J \times \tan^2\left(\frac{57.3 L_e}{4J}\right)} - \frac{2}{\tan\left(\frac{57.3 L_e}{4J}\right)}\right]},$$

where
L_e = effective span length (in.),
B = hole curvature (°/100 ft.),
ΔD = diameter difference between tool joint and pipe body,
ΔD = $TJOD - OD$ (in.),
$TJOD$ = tool joint OD (in.),
OD = pipe body OD (in.),
ID = pipe body ID (in.).

$$J = \left(\frac{E \times I}{F}\right)^{1/2} \text{ (in.)},$$

E = Young's modulus 29.6 x 10^6 for steel (psi),

$$I = \frac{\pi}{64}(OD^4 - ID^4),$$

F = axial compressive load (lb),
L_w = length of pipe body touching hole (in.),
L_w = $L - L_e$,
L = length of one joint of pipe (in.).

A.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.

A.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:

$$LF_{TJ} = \frac{F \times L \times B}{57.3 \times 100 \times 12},$$

where
F_{TJ} = lateral force on tool joint (lb),
L = length of one joint of pipe (in.),
B = hole curvature (°/100 ft.).

A.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:

$$LF_{TJ} = \frac{2 \times E \times I \times U^2}{R \times L_e}\left[\frac{\left(1 - \frac{4R \times \Delta D}{L_e^2}\right)\sin U - \frac{4}{U}\sin^2\left(\frac{U}{2}\right)}{\sin U - \frac{4}{U}\sin^2\left(\frac{U}{2}\right)}\right],$$

where

$$LF_{pipe} = \frac{F \times L}{R} - LF_{tj}$$

LF_{tj} = lateral force on tool joint (lb),
LF_{pipe} = lateral force on pipe body (lb),
L_w = $L - L_e$,
L_w = length of pipe (in.),
L_e = L for point contact (in.),
L_e = effective span length for wrap contact,

$$R = \frac{57.3 \times 100 \times 12}{B}$$

B = hole curvature (°/100 ft.),

$$U = \frac{L_e}{2J}$$

$$J = \left(\frac{E \times I}{F}\right)^{1/2} \text{ (in.)}^{1/2},$$

ΔD = diameter difference tool joint minus pipe OD (in.),
ΔD = $TJOD - OD$ (in.),

$$I = \frac{\pi}{64}(OD^4 - ID^4).$$

Table A-1—Rotary Shouldered Connection Thread Element Information

(1) Connection Type	(2) Pitch Diameter C	(3) Taper	(4) Pin Length	(5) Thread Height Not Truncated	(6) Root Truncation	(7) Nominal Counterbore	(8) Thread Pitch	(9) Thread Angle θ	(10) Stress Relief Groove Diameter	(11) Bore-Back Cylinder Diameter	(12) Low Torque Counterbore	(13) Low Torque Bevel Diameter
NC10	1.063000	1.500000	1.500000	0.144100	0.040600	1.204000	.166667	30	—	—	—	—
NC12	1.265000	1.500000	1.750000	0.144100	0.040600	1.406000	.166667	30	—	—	—	—
NC13	1.391000	1.500000	1.750000	0.144100	0.040600	1.532000	.166667	30	—	—	—	—
NC16	1.609000	1.500000	1.750000	0.144100	0.040600	1.751000	.166667	30	—	—	—	—
NC23	2.355000	2.000000	3.000000	0.216005	0.038000	2.625000	.250000	30	2.140625	2.218750	—	—
NC26	2.668000	2.000000	3.000000	0.216005	0.038000	2.937500	.250000	30	2.375000	2.531250	—	—
NC31	3.183000	2.000000	3.500000	0.216005	0.038000	3.453125	.250000	30	2.890625	2.953125	—	—
NC35	3.531000	2.000000	3.750000	0.216005	0.038000	3.812500	.250000	30	3.234375	3.234375	—	—
NC38	3.808000	2.000000	4.000000	0.216005	0.038000	4.078125	.250000	30	3.515625	3.468750	—	—
NC40	4.072000	2.000000	4.500000	0.216005	0.038000	4.343750	.250000	30	3.781250	3.656250	—	—
NC44	4.417000	2.000000	4.500000	0.216005	0.038000	4.687500	.250000	30	4.187500	4.000000	—	—
NC46	4.626000	2.000000	4.500000	0.216005	0.038000	4.906250	.250000	30	4.328125	4.203125	—	—
NC50	5.041700	2.000000	4.500000	0.216005	0.038000	5.312500	.250000	30	4.750000	4.625000	—	—
NC56	5.616000	3.000000	5.000000	0.215379	0.038000	5.937500	.250000	30	5.296875	4.796875	—	—
NC61	6.178000	3.000000	5.500000	0.215379	0.038000	6.500000	.250000	30	5.859375	5.234375	—	—
NC70	7.053000	3.000000	6.000000	0.215379	0.038000	7.375000	.250000	30	6.734375	5.984375	—	—
NC77	7.741000	3.000000	6.500000	0.215379	0.038000	8.062500	.250000	30	7.421875	6.546875	—	—
5 1/2 IF	6.189000	2.000000	5.000000	0.216005	0.038000	6.453125	.250000	30	5.890625	5.687500	—	—
6 5/8 IF	7.251000	2.000000	5.000000	0.216005	0.038000	7.515625	.250000	30	6.953125	6.750000	—	—
2 3/8 REG	2.365370	3.000000	3.000000	0.172303	0.020000	2.687500	.200000	30	2.015625	2.062500	—	—
2 7/8 REG	2.740370	3.000000	3.500000	0.172303	0.020000	3.062500	.200000	30	2.390625	2.312500	—	—
3 1/2 REG	3.239870	3.000000	3.750000	0.172303	0.020000	3.562500	.200000	30	2.906250	2.718750	—	—
4 1/2 REG	4.364870	3.000000	4.250000	0.172303	0.020000	4.687500	.200000	30	4.015625	3.718750	—	—
5 1/2 REG	5.234020	3.000000	4.750000	0.215379	0.025000	5.578125	.250000	30	4.859375	4.500000	—	—
6 5/8 REG	5.757800	2.000000	5.000000	0.216005	0.025000	6.062500	.250000	30	5.421875	5.281250	—	—
7 5/8 REG	6.714530	3.000000	5.250000	0.215379	0.025000	7.093750	.250000	30	6.406250	5.859375	7.750000	9.250000
8 5/8 REG	7.666580	3.000000	5.375000	0.215379	0.025000	8.046875	.250000	30	7.296875	6.781250	9.000000	10.500000
2 7/8 FH	3.364000	3.000000	3.500000	0.172303	0.020000	3.687500	.200000	30	—	—	—	—
3 1/2 FH	3.734000	3.000000	3.750000	0.172303	0.020000	4.046875	.200000	30	3.421875	3.218750	—	—
4 1/2 FH	4.532000	3.000000	4.000000	0.172303	0.020000	4.875000	.200000	30	4.203125	3.953125	—	—
5 1/2 FH	5.591000	2.000000	5.000000	0.216005	0.025000	5.906250	.250000	30	5.250000	5.109375	—	—
6 5/8 FH	6.519600	2.000000	5.000000	0.216005	0.025000	6.843750.	.250000	30	6.171875	6.046875	—	—
2 3/8 SL-H90	2.578000	1.250000	2.812500	0.166215	0.034300	2.765625	.333333	45	2.328125	2.531250	—	—
2 7/8 SL-H90	3.049000	1.250000	2.937500	0.166215	0.034300	3.234375	.333333	45	2.671875	2.984375	—	—
3 1/2 SL-H90	3.688000	1.250000	3.187500	0.166215	0.034300	3.875000	.333333	45	3.312500	3.593750	—	—

Table A-1—Rotary Shouldered Connection Thread Element Information

(1) Connection Type	(2) Pitch Diameter C	(3) Taper	(4) Pin Length	(5) Thread Height Not Truncated	(6) Root Truncation	(7) Nominal Counterbore	(8) Thread Pitch	(9) Thread Angle θ	(10) Stress Relief Groove Diameter	(11) Bore-Back Cylinder Diameter	(12) Low Torque Counterbore	(13) Low Torque Bevel Diameter
3½ H-90	3.929860	2.000000	4.000000	0.141865	0.017042	4.187500	.285710	45	3.656250	3.562500	—	—
4 H-90	4.303600	2.000000	4.250000	0.141865	0.017042	4.562500	.285710	45	4.031250	3.875000	—	—
4½ H-90	4.637600	2.000000	4.500000	0.141865	0.017042	4.890625	.285710	45	4.359375	4.187500	—	—
5 H-90	4.908100	2.000000	4.750000	0.141865	0.017042	5.171875	.285710	45	4.625000	4.406250	—	—
5½ H-90	5.178600	2.000000	4.750000	0.141865	0.017042	5.437500	.285710	45	4.906250	4.687500	—	—
6⅝ H-90	5.803600	2.000000	5.000000	0.141865	0.017042	6.062500	.285710	45	5.531250	5.265625	—	—
7 H-90	6.252300	3.000000	5.500000	0.140625	0.016733	6.562500	.285710	45	6.031250	5.265625	7.125000	8.250000
7⅝ H-90	7.141100	3.000000	6.125000	0.140625	0.016733	7.453125	.285710	45	6.906250	6.000000	8.000000	9.250000
8⅝ H-90	8.016100	3.000000	6.625000	0.140625	0.016733	8.328125	.285710	45	7.781250	6.750000	9.375000	10.500000
2⅜ PAC	2.203000	1.500000	2.375000	0.216224	0.057948	2.406250	.250000	30	1.984375	2.171875	—	—
2⅞ PAC	2.369000	1.500000	2.375000	0.216224	0.057948	2.578125	.250000	30	2.156250	2.343750	—	—
3½ PAC	2.884000	1.500000	3.250000	0.216224	0.057948	3.109375	.250000	30	—	—	—	—
2⅜ SH	2.230000	2.000000	2.875000	0.216005	0.038000	2.500000	.250000	30	1.937500	2.093750	—	—
2⅞ SH	2.668000	2.000000	3.000000	0.216005	0.038000	2.937500	.250000	30	2.375000	2.531250	—	—
3½ SH	3.183000	2.000000	3.500000	0.216005	0.038000	3.453125	.250000	30	2.890625	2.953125	—	—
4 SH	3.604000	2.000000	3.500000	0.216005	0.038000	3.875000	.250000	30	3.312500	3.375000	—	—
4½ SH	3.808000	2.000000	4.000000	0.216005	0.038000	4.078125	.250000	30	3.515625	3.468750	—	—
2⅞ XH	3.119000	2.000000	4.000000	0.216005	0.038000	3.359375	.250000	30	2.828125	2.781250	—	—
3½ XH	3.604000	2.000000	3.500000	0.216005	0.038000	3.875000	.250000	30	3.312500	3.375000	—	—
5 XH	5.041700	2.000000	4.500000	0.216005	0.038000	5.312500	.250000	30	4.750000	4.625000	—	—
2⅜ OH SW	2.588000	1.500000	2.375000	0.216224	0.057948	2.796875	.250000	30	—	—	—	—
2⅜ OH LW	2.588000	1.500000	2.375000	0.216224	0.057948	2.796875	.250000	30	—	—	—	—
2⅞ OH SW	2.984000	1.500000	2.875000	0.216224	0.057948	3.203125	.250000	30	—	—	—	—
2⅞ OH LW	2.984000	1.500000	2.500000	0.216224	0.057948	3.203125	.250000	30	—	—	—	—
3½ OH SW	3.728000	1.500000	3.250000	0.216224	0.057948	3.953125	.250000	30	—	—	—	—
3½ OH LW	3.728000	1.500000	3.250000	0.216224	0.057948	3.953125	.250000	30	—	—	—	—
4 OH SW	4.416000	1.500000	4.000000	0.216224	0.057948	4.640625	.250000	30	—	—	—	—
4 OH LW	4.416000	1.500000	3.500000	0.216224	0.057948	4.640625	.250000	30	—	—	—	—
4½ OH SW	4.752000	1.500000	3.750000	0.216224	0.057948	4.953125	.250000	30	—	—	—	—
4½ OH LW	4.752000	1.500000	3.750000	0.216224	0.057948	4.953125	.250000	30	—	—	—	—
2⅜ WO	2.605000	2.000000	2.375000	0.216005	0.038000	2.859375	.250000	30	—	—	—	—
2⅞ WO	3.121000	2.000000	3.000000	0.216005	0.038000	3.375000	.250000	30	—	—	—	—
3½ WO	3.808000	2.000000	3.500000	0.216005	0.038000	4.078125	.250000	30	—	—	—	—
4 WO	4.626000	2.000000	4.500000	0.216005	0.038000	4.906250	.250000	30	—	—	—	—
4½ WO	5.041700	2.000000	4.500000	0.216005	0.038000	5.312500	.250000	30	—	—	—	—

APPENDIX B—ARTICLES AND TECHNICAL PAPERS

Many of the topics covered in Recommended Practice 7G have been the subject of significant investigation by various parties in the industry. Following is a partial list of articles and technical papers on topics related to items found in Recommended Practice 7G.

1. Arthur Lubinski, "Maximum Permissible Dog-Legs in Rotary Boreholes," *Journal of Petroleum Technology*, February 1961.

2. Robert W. Nicholson, "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.*

3. K. D. Schenk. "Calco Learns About Drilling Through Excessive Doglegs," *Oil and Gas Journal*, October 12, 1964.

4. G. J. Wilson, "Dogleg Control In Directionally Drilled Wells," *Transactions of The American Institute of Mining, Metallurgical, and Petroleum Engineers (AIME)*, 1967, Vol. 240.

5. Hansford, John E. and Lubinski, Arthur, "Cumulative Fatigue Damage of Drill Pipe in Dog Legs," *Journal of Petroleum Technology*, March 1966.

6. Hansford, John E. and Lubinski, Arthur: "The Effect of Drilling Vessel Pitch or Roll on Kelly and Drill Pipe Fatigue," *Transactions of AIME*, 1964, Vol. 231.

7. Thad Vreeland, Jr., "Dynamic Stresses In Long Drill Pipe Strings," *The Petroleum Engineer*, May 1961.

8. Henry Bourne, Continental Oil Co., Ponca City, Oklahoma, *Drilling Fluid Corrosion*, Unpublished.

9. Edward R. Slaughter, E. Ellis Fletcher, Arthur R. Elsear, and George K. Manning, "An Investigation of the Effects of Hydrogen on the Brittle Failure of High Strength Steels," WADC TR 56-83, June 1955.

10. H. M. Rollins, "Drill Stem Failures Due to H_2S," *Oil and Gas Journal*. January 24. 1966.

11. Walter Main, Discussion of Paper by Grant and Texter, "Causes and Prevention of Drill Pipe and Tool Joint Troubles," *World Oil*, October 1948.

12. J. C. Stall and K. A. Blenkarn, "Allowable Hook Load and Torque Combinations For Stuck Drill String," Mid-Continent API District Meeting, Paper No. 851-36-M, April 6, 1962.

13. Armco Steel Corporation, "Oil Country Tubular Products—Engineering Data," 1966.

14. Arthur Lubinski, "Fatigue of Range 3 Drill Pipe," *Revue de L'Institut Francais du Petrole*, November 2, 1977 (English translation).

15. Arthur Lubinski, "Dynamic Loading of Drill Pipe During Tripping," *Journal of Petroleum Technology*, August 1988.

16. Huang T., Dareing D.W., "Buckling and Lateral Vibration of Drill Pipe," *Journal of Engineering for Industry*, November 1968, pp. 613-619.

17. Dareing D. W., Livesay B. J., "Longitudinal and Angular Drillstring Vibrations With Damping," *Journal of Engineering for Industry*, November 1968, pp. 671-679.

18. Combes J. D., Baxter V. R., "Critical Rotary Speeds Can Be Costly," *Petroleum Engineer*, September 1969, pp. 60-63.

19. Besaisow A. A., et. al., "Development of a Surface Drillstring Vibration Measurement System, " Society of Petroleum Engineers (SPE) SPE 14327, presented at the 1985 Technical Conference, Las Vegas, September 22-25, 1985.

20. Burgess T. M., McDaniel G. L., Das P. K., "Improving Tool Reliability with Drillstring Vibration Models—Field Experience and Limitations," SPE/IADC 16109, presented at the 1987 SPE/IADC Drilling Conference, New Orleans, March 1987.

21. Halsey G. W., et. al., "Torque Feedback Used to Cure Slip-Stick Motion," SPE 18049, presented at the 1988 Technical Conference, New Orleans, March 1987.

22. Cook R. L., Nicholson J. W., Sheppard M. C., and Westlake W., "First Real Time Measurements of Downhole Vibrations, Forces, and Pressures Used to Monitor Directional Drilling Operations," SPE/IADC 19651, presented at the 1989 SPE/IADC Drilling Conference, New Orleans, February 28 to March 3, 1989.

23. Warren T. M., Brett J. F., and Sinor L. A., "Development of a Whirl-Resistant Bit," SPE 19572, presented at 1989 SPE Annual Technical Conference and Exhibition, San Antonio, October 8-11.

24. Yanglie Zhang, Zaiyang Xiao, "Motion of Reviewing the Section 9 of API RP 7G," presented at the API Standardization Conference, June 1993.

25. Nicholson J. W., "An Integrated Approach to Drilling Dynamics Planning, Identification, and Control," SPE/IADC 27537, presented at the SPE/IADC Drilling Conference, Dallas, February 15-18, 1994.

26. Fereidoun Abbassian, "Drillstring Vibration Primer," *BP Exploration*, Unpublished.

27. Dawson, Rapier, and Paslay, P.R., "Drillpipe Buckling in Inclined Holes," *JPT*, October 1984.

28. *Standard DS-1, Drill Stem Design and Inspection, Second Edition*, T.H. Hill Associates, Inc., Houston, Texas, August 1997.

29. Mitchell, R.F., "Effects of Well Deviation on Helical Buckling," SPE 29462, SPE Production Operations Symposium, Oklahoma City, April 1995.

30. Schuh, F.J., "The Critical Buckling Force and Stresses for Pipe in Inclined Curved Boreholes," SPE/IADC 21942, SPE/IADC Drilling Conference, Amsterdam, March 11–14, 1991.

31. Kyllingstad, Aage, "Buckling of Tubular Strings in Curved Wells," *Journal of Petroleum Science and Engineering, 12*, 1995, pp. 209–218.

32. Suryanarayana, P.V.R.; McCann, R.C., "An Experimental Study of Buckling and Post-Buckling of Laterally Constrained Rods," *Journal of Energy Resources Technology, Vol. 177*, June 1995, pp. 115–124.

33. Paslay, P.R.; Cernocky, E.P., "Bending Stress Magnification in Constant Curvature Doglegs with Impact on Drilling and Casing," SPE 22547, the SPE 66th Annual Technical Conference and Exposition, Dallas, Texas, October 6–9, 1991.

34. Acknowledgment: Subcommittee appreciates the use of Shell Exploration and Production Company proprietary BSM Bending Stress Code.

35. Morgan, R.P.; Roblin, M.J., "A Method for the Investigation of Fatigue Strength in Seamless Drillpipe," ASME Conference, Tulsa, Oklahoma, September 22, 1969.

36. Casner, John. A., "Endurance Limit of Drill Pipe." letter 1/21/95 to John Altermann.

37. Hansford, J.E.; Lubinski, A., "Cumulative Fatigue Damage of Drillpipe in Doglegs," *Journal of Petroleum Technology*, March 1966, pp. 359–363.

38. Lubinski, Arthur A., "Maximum Permissible Dog-Legs in Rotary Boreholes," *Journal of Petroleum Technology*, February 1961.

39. Acknowledgment: Subcommittee appreciates the release by Mobil Exploration and Production Technical Center, Dallas, Texas of the report, "Buckling of a Rod Confined to be in Contact with a Toroidal Surface, Parts I and II," by Paul R. Paslay, October 15, 1993.

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