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<b>Submitter Name(s)</b>	David Pattillo
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**Work Item Charge:** To update API 5C3 by (1) incorporating all balloted and approved Work Items since the 7<sup>th</sup> Edition, (2) removing Extreme-Line references, (3) revising language to facilitate the transition from Technical Report to Recommended Practice, and (4) incorporating various editorial clean-ups collected since the 7<sup>th</sup> Edition.

**Ballot Rationale:** Implementation of the changes as discussed above. The focus of this ballot is to ensure proper integration of previously balloted and approved Work Items since the 7<sup>th</sup> Edition. Any technical or substantive comments are welcome but may be addressed as New Work Items for the 9<sup>th</sup> Edition.

**Ballot Text:**

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# **Calculating Performance Properties of Pipe Used as Casing or Tubing**

API RECOMMENDED PRACTICE 5C3  
EIGHTH EDITION, XXXX 202X

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Suggested revisions are invited and should be submitted to the Standards Department, API, 200 Massachusetts Avenue, Suite 1100, Washington, DC 20001, [standards@api.org](mailto:standards@api.org).

## Introduction

Performance design of tubulars for the petroleum and natural gas industries, whether it is formulated by deterministic or probabilistic calculations, compares anticipated loads to which the tubular may be subjected to the anticipated resistance of the tubular to each load. The load and/or resistance may be modified by a design factor.

Both deterministic and probabilistic approaches to performance properties are addressed in this document. The deterministic approach uses specific geometric and material property values to calculate a single performance property value. The probabilistic method treats the same variables as random and thus arrives at a statistical distribution of a performance property. A performance distribution in combination with a defined lower percentile determines the final design equation.

Both the well design process itself and the definition of anticipated loads are currently outside the scope of standardization for the petroleum and natural gas industries. Neither of these aspects is addressed in this document. Rather, it serves to identify useful equations for obtaining the resistance of a tubular to specified loads, independent of their origin. It provides limit state equations (see annexes) that are useful for determining the resistance of an individual sample whose geometric and material properties are given, and design equations that are useful for well design based on conservative geometric and material parameters.

Whenever possible, decisions on specific constants to use in a design equation are left to the discretion of the user.

For equations related to performance properties, extensive background information is also provided regarding their development and use.

Equations presented here are intended for use with pipe manufactured in accordance with API 5CT, API 5DP, and API 5L, as applicable. These equations and templates may be extended to other pipe with due caution. Pipe cold-worked during production is included in the scope of this document (e.g. cold rotary straightened [CRS] pipe). Pipe modified by cold working after production, such as expandable tubulars and coiled tubing, is beyond the scope of this document.

Application of performance property equations in this document to line pipe and other pipe is restricted to their use as casing/tubing in a well or laboratory test, and requires due caution to match the heat-treat process, straightening process, yield strength, and so forth, with the closest appropriate casing/tubing product. Similar caution should be exercised when using the performance equations for drill pipe or for collapse cold-expanded API 5L pipe.

This document and the equations contained herein relate the input pipe manufacturing parameters in API 5CT, API 5DP, and API 5L to expected pipe performance. The design equations in this document do not imply a warranty of performance. Manufacturers are typically licensed to produce tubular products in accordance with manufacturing specifications that control the dimensions and physical properties of their product. Design equations, on the other hand, are a reference point for users to characterize tubular performance and begin their own well design or research of pipe input properties.

This document is not a design code. It only provides equations and templates for calculating the properties of tubulars intended for use in downhole applications. This document does not provide any guidance about loads that can be encountered by tubulars or about safety margins needed for acceptable design. Users are responsible for defining appropriate design loads and selecting adequate safety factors to develop safe and efficient designs. The design loads and safety factors will likely be selected based on historical practice, local regulatory requirements, and specific well conditions.

All equations and listed values for performance properties in this document assume a benign environment and material properties conforming to API 5CT, API 5DP, and API 5L. Other environments may require additional analyses, such as that outlined in Annex D.

In the interests of worldwide application of this document, the API Subcommittee on Tubular Goods (SC5) has decided, after detailed technical analysis, that certain documents listed in the Bibliography and prepared by API-SC5 or other technical committees are interchangeable in the context of the relevant requirement with the relevant document prepared by the International Organization for Standardization (ISO) or ASTM International. These latter documents are cited in the running text following the API reference and preceded by or, for example, "API XXXX or ISO YYYY" or "ISO YYYY or ASTM ZZZZ." Application of an alternative document cited in this manner may lead to technical results different from the use of the preceding API reference. However, both results are acceptable and these documents are thus considered interchangeable in practice.

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



# Calculating Performance Properties of Pipe used as Casing or Tubing

## 1 Scope

This **document** illustrates the equations and templates necessary to calculate the various pipe properties, including the following:

- pipe performance properties, such as axial strength, internal pressure resistance, and collapse resistance;
- minimum physical properties;
- product assembly force (torque);
- product test pressures;
- critical product dimensions related to testing criteria;
- critical dimensions of testing equipment;
- critical dimensions of test samples.

~~For equations related to performance properties, extensive background information is also provided regarding their development and use.~~

~~Equations presented here are intended for use with pipe manufactured in accordance with API 5CT, API 5DP, and API 5L, as applicable. These equations and templates may be extended to other pipe with due caution. Pipe cold worked during production is included in the scope of this document (e.g. cold rotary straightened [CRS] pipe). Pipe modified by cold working after production, such as expandable tubulars and coiled tubing, is beyond the scope of this document.~~

~~Application of performance property equations in this document to line pipe and other pipe is restricted to their use as casing/tubing in a well or laboratory test, and requires due caution to match the heat treat process, straightening process, yield strength, and so forth, with the closest appropriate casing/tubing product. Similar caution should be exercised when using the performance equations for drill pipe or for collapse cold-expanded API 5L pipe.~~

~~This document and the equations contained herein relate the input pipe manufacturing parameters in API 5CT, API 5DP, and API 5L to expected pipe performance. The design equations in this document do not imply a warranty of performance. Manufacturers are typically licensed to produce tubular products in accordance with manufacturing specifications that control the dimensions and physical properties of their product. Design equations, on the other hand, are a reference point for users to characterize tubular performance and begin their own well design or research of pipe input properties.~~

~~This document is not a design code. It only provides equations and templates for calculating the properties of tubulars intended for use in downhole applications. This document does not provide any guidance about loads that can be encountered by tubulars or about safety margins needed for acceptable design. Users are responsible for defining appropriate design loads and selecting adequate safety factors to develop safe and efficient designs. The design loads and safety factors will likely be selected based on historical practice, local regulatory requirements, and specific well conditions.~~

~~All equations and listed values for performance properties in this document assume a benign environment and material properties conforming to API 5CT, API 5DP, and API 5L. Other environments may require additional analyses, such as that outlined in Annex D.~~

Pipe performance properties under dynamic loads and pipe connection sealing resistance are excluded from the scope of this document.

Throughout this document tensile stresses are positive.

## 2 Normative References

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

ASTM E1152-95, Test Method for Determining J-R Curves

ASTM E1928, Standard Practice for Estimating the Approximate Residual Circumferential Stress in Straight Thin-walled Tubing

## 3 Terms, Definitions, and Abbreviations

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

### 3.1.1

#### **Cauchy stress**

#### **true stress**

Force applied to the surface of a body divided by the current area of that surface.

### 3.1.2

#### **coefficient of variance**

Dimensionless measure of the dispersion of a random variable, calculated by dividing the standard deviation by the mean.

### 3.1.3

#### **design equation**

Equation that, based on production measurements or specifications, provides a performance property useful in design calculations.

**NOTE** A design equation can be defined by applying reasonable extremes to the variables in a limit state equation to arrive at a conservative value of expected performance; when statistically derived, the design equation corresponds to a defined lower percentile of the resistance probability distribution curve.

### 3.1.4

#### **deterministic**

Approach that assumes that all variables controlling a performance property are known with certainty.

**NOTE** Pipe performance properties generally depend on one or more controlling parameters; a deterministic equation uses specific geometric and material property values to calculate a single performance property value; for design formulations, this value is the expected minimum.

### 3.1.5

#### **ductile rupture**

Failure of a tube due to internal pressure and/or axial force in the plastic deformation range.

### 3.1.6

#### **effective axial force**

Material axial force (pipe wall axial stress times cross-sectional area) adjusted for the effect of internal and external pressure.

NOTE See I.4.4.3.

### 3.1.7

#### **engineering strain**

Dimensionless measure of the stretch of a deforming line element, defined as the change in length of the line element divided by its original length.

### 3.1.8

#### **engineering stress**

Force applied to the surface of a body divided by the original area of that surface.

### 3.1.9

#### **fracture pressure**

Internal pressure at which a tube fails due to propagation of an imperfection.

### 3.1.10

#### **inspection threshold**

Maximum size of a crack-like imperfection that is defined to be acceptable by the inspection system.

### 3.1.11

#### **J-integral**

Measure of the intensity of the stress-strain field near the tip of a crack.

### 3.1.xxx

#### **JCOE**

An acronym referring to the process of forming pipe in which plate steel edges are formed into a “J”, then the full plate is shaped into a “C”, then an “O”, then welded together and finally expanded.

### 3.1.12

#### **Label 1**

Dimensionless designation for the size or specified outside diameter that may be used when ordering pipe.

### 3.1.13

#### **Label 2**

Dimensionless designation for the mass per unit length or wall thickness that may be used when ordering pipe.

### 3.1.14

#### **limit state equation**

Equation that, when used with the measured geometry and material properties of a sample, produces an estimate of the failure value of that sample.

NOTE A limit state equation describes the performance of an individual sample as closely as possible, without regard for the tolerances to which the sample was built.

### 3.1.15

#### **logarithmic strain**

Dimensionless measure of the stretch of a deforming line element, defined as the natural logarithm of the ratio of the current length of the line element to its original length.

**NOTE** Alternatively, the logarithmic strain can be estimated as the natural logarithm of one plus the engineering strain.

### **3.1.16**

#### **mass**

Label used to represent wall thickness of tube cross section for a given pipe size.

### **3.1.17**

#### **pipe body yield**

Stress state necessary to initiate yield at any location in the pipe body.

### **3.1.18**

#### **principal stress**

Stress on a principal plane for which the shear stress is zero.

**NOTE** For any general state of stress at any point, there exist three mutually perpendicular planes at that point on which shearing stresses are zero; the remaining normal stress components on these three planes are principal stresses; the largest of these three stresses is called the maximum principal stress.

### **3.1.19**

#### **probabilistic method**

Approach that uses distributions of geometric and material property values to calculate a distribution of performance property values.

### **3.1.20**

#### **true stress-strain curve**

Plot of Cauchy stress (ordinate) vs. logarithmic strain (abscissa).

### **3.1.21**

#### **UOE**

An acronym referring to the process of forming pipe in which plate steel is shaped into a "U", then an "O", then welded together and finally expanded.

### **3.1.22**

#### **yield stress bias**

Ratio of actual yield stress to specified minimum yield stress.

## **3.1 Abbreviations**

CRS	cold rotary straightened
CVN	Charpy V-Notch
DCB	double cantilever beam
FAD	failure assessment diagram
FORM	first-order reliability method
HC	high-collapse
HRS	hot rotary straightened

JCOE—acronym term referring to the process of forming pipe where steel plate edges are formed into a "J," the full plate is then shaped into a "C," then an "O," welded together and finally expanded "E"

OCTG oil country tubular goods

SENT single edge notch tension

SORM second-order reliability method

TPI threads per inch,  $1 \text{ TPI} = 0.0394 \text{ threads per mm}$ ,  $1 \text{ thread per mm} = 25.4 \text{ TPI}$

TRL target reliability level

ULS ultimate limit state

UOE acronym term referring to the process of forming pipe where steel plate is shaped into a "U," then an "O," welded together and finally expanded "E"

### 3.2 Symbols

$A$  hand-tight standoff, in turns

$A_c$  empirical constant in historical API collapse equation

$A_{crit}$  area of the weaker connection component at the critical cross section

$A_{gbtj}$  critical dimension on guided bend test jig, denoted as dimension  $A$  in API 5L

$A_i$  area to pipe inside diameter;  $A_i = \pi d^2/4$

$A_{jc}$  area of the coupling cross section,  $A_{jc} = \pi/4 (W^2 - d_1^2)$

$A_{jp}$  area of the pipe cross section under the last perfect thread

$A_o$  area to pipe outside diameter;  $A_o = \pi D^2/4$

$A_p$  area of the pipe cross section,  $A_p = A_o - A_i$

$A_{p \text{ ave}}$  average area of the pipe cross section,  $A_{p \text{ ave}} = \pi/4 [D_{ave}^2 - (D_{ave} - 2 t_{c \text{ ave}})^2]$

$A_s$  cross-sectional area of the tensile test specimen in square millimeters (square inches), based on specified outside diameter or nominal specimen width and specified wall thickness, rounded to the nearest 10 mm<sup>2</sup> (0.01 in.<sup>2</sup>) or 490 mm<sup>2</sup> (0.75 in.<sup>2</sup>), whichever is smaller

$a$  for a limit state equation, the maximum actual depth of a crack-like imperfection; for a design equation, the maximum depth of a crack-like imperfection that could likely pass the manufacturer's inspection system

$a_N$  imperfection depth associated with a specified inspection threshold, i.e. the maximum depth of a crack-like imperfection that could reasonably be missed by the pipe inspection system; e.g. for a 5 % imperfection threshold inspection in a 12.7 mm (0.500 in.) wall thickness pipe,  $a_N = 0.635 \text{ mm}$  (0.025 in.)

$a_{i/D}$  average value of  $t/D$  ratios used in the regression

$B_c$  empirical constant in historical API collapse equation

$B_f$  maximum bearing face diameter, special bevel, in accordance with API 5CT

$b$  Weibull shape parameter

$C_c$  empirical constant in historical API collapse equation

$C_{iR}$  random variable that represents model uncertainty

$c$	tube curvature, the inverse of the radius of curvature to the centerline of the pipe
$D$	specified pipe outside diameter
$D_{ac}$	average outside diameter after cutting
$D_{ave}$	average pipe outside diameter
$D_{bc}$	average outside diameter before cutting
$D_{max}$	maximum pipe outside diameter
$D_{min}$	minimum pipe outside diameter
$D_4$	major diameter, in accordance with API 5B
$d$	pipe inside diameter, $d = D - 2t$
$d_{iu}$	inside diameter of pin upset, in accordance with API 5CT
$d_{ou}$	inside diameter at end of upset pipe
$d_{wall}$	inside diameter based on $k_{wall} t$ , $d_{wall} = D - 2k_{wall} t$
$d_1$	diameter at the root of the coupling thread at the end of the pipe in the power-tight position
<b>e</b>	<b>Euler's constant, equal to 2.718281828</b>
$E$	Young's modulus
$E_c$	pitch diameter, at center of coupling
$E_{ec}$	pitch diameter, at end of coupling
$E_s$	pitch diameter, at plane of seal
$E_0$	pitch diameter, at end of pipe
$E_1$	pitch diameter at the hand-tight plane, in accordance with API 5B
$E_7$	pitch diameter, in accordance with API 5B
$ec$	eccentricity
$e_m$	mass gain due to end finishing
$F_a$	material axial force
$F_{eff}$	effective axial force; $F_{eff} = F_a - p_i A_i + p_o A_o$
$F_c$	empirical constant in historical API collapse equation
$F_{YAPI}$	material axial force at yield, historical API equation
$f$	degrees of freedom = $N_t - 1$
$f(\vec{x})$	joint probability density function of the variables in $\vec{x}$
$f_m$	root truncation of the pipe thread of API line pipe threads, as follows: 0.030 mm (0.0012 in.) for 27 TPI 0.046 mm (0.0018 in.) for 18 TPI 0.061 mm (0.0024 in.) for 14 TPI 0.074 mm (0.0029 in.) for 11-1/2 TPI 0.104 mm (0.0041 in.) for 8 TPI
$f_u$	tensile strength of a representative tensile specimen
$f_{uc}$	tensile strength of a representative tensile specimen from the coupling

$f_{umn}$	specified minimum tensile strength
$f_{umnc}$	specified minimum tensile strength of the coupling
$f_{umnp}$	specified minimum tensile strength of the pipe body
$f_{up}$	tensile strength of a representative tensile specimen from the pipe body
$f_y$	yield strength of a representative tensile specimen
$f_{ycom}$	combined loading equivalent grade
$f_{ye}$	equivalent yield stress in the presence of axial stress
$f_{ymn}$	specified minimum yield strength
$f_{ymnc}$	specified minimum yield strength of the coupling
$f_{ymnp}$	specified minimum yield strength of the pipe body
$f_{ymx}$	specified maximum yield strength
$f_{yp}$	yield strength of a representative tensile specimen from the pipe body
$G_c$	empirical constant in historical API collapse equation
$G_0$	influence coefficient for fracture limit state FAD curve
$G_1$	influence coefficient for fracture limit state FAD curve
$G_2$	influence coefficient for fracture limit state FAD curve
$G_3$	influence coefficient for fracture limit state FAD curve
$G_4$	influence coefficient for fracture limit state FAD curve
$g$	length of imperfect threads, in accordance with API 5B
$g(\bar{x})$	limit state function
$H$	thread height of a round-thread equivalent Vee thread, as follows: 0.815 mm (0.0321 in.) for 27 TPI 1.222 mm (0.0481 in.) for 18 TPI 1.755 mm (0.0691 in.) for 14 TPI 1.913 mm (0.0753 in.) for 11-1/2 TPI 2.1996 mm (0.08660 in.) for 10 TPI 2.7496 mm (0.10825 in.) for 8 TPI
$H_{t_{des}}$	decrement factor for design collapse strength, as given in Table F.9
$H_{t_{ult}}$	a decrement factor for ultimate collapse strength, as defined in Equation (F.4)
$h_B$	buttress thread height, equal to: 1.575 for SI units 0.062 for USC units
$h_n$	stress-strain curve shape factor
$h_s$	round thread height
$I$	moment of inertia of the pipe cross section, $I = \pi/64 (D^4 - d^4)$
$I_{ave}$	average moment of inertia of the pipe cross section, $I = \pi/64 [D_{ave}^4 - (D_{ave} - 2 t_{c ave})^4]$
$I_B$	length from the face of the buttress thread coupling to the base of the triangle in the hand-tight position, as follows: 0.16 mm (0.400 in.) for Label 1: 4-1/2 12.70 mm (0.500 in.) for sizes between Label 1: 5 and Label 1: 13-3/8, inclusive

	9.52 mm (0.375 in.) for sizes greater than Label 1: 13-3/8
$J$	distance from end of pipe to center of coupling in power-tight position, in accordance with API 5B
$J_{lc}$	fracture resistance of the material
$J_{lmat}$	fracture resistance of the material in a particular environment
$J_p$	polar moment of inertia of the pipe cross section, $J_p = \pi/32 (D^4 - d^4)$
$J_r$	stress intensity ratio based on the J-integral
$K$	stress intensity factor at the crack tip
$K_{lmat}$	fracture toughness of a material in a particular environment
$K_p$	ratio of internal pressure stress to yield strength, or $p_i D/(2 f_{ymnp} t)$
$K_r$	stress intensity ratio
$k_A$	variable intermediate term in API 5C5 representation of von Mises yield criterion
$k_a$	burst strength factor, having the numerical value 1.0 for quenched and tempered (martensitic structure) or 13Cr products and 2.0 for as-rolled and normalized products based on available test data, and the default value set to 2.0 where the value has not been measured
	NOTE The value of $k_a$ can be established for a specific pipe material based on testing.
$k_B$	variable intermediate term in API 5C5 representation of von Mises yield criterion
$k_C$	variable intermediate term in API 5C5 representation of von Mises yield criterion
$k_c$	constant used in elastic collapse equation
$k_{dr}$	correction factor based on pipe deformation and material strain hardening, having the numerical value $[(1/2)^{n+1} + (1/\sqrt{3})^{n+1}]$
$k_e$	bias factor for elastic collapse
$k_{e\ des}$	down-rating factor for design elastic collapse
$k_{el}$	elongation constant, equal to: 1942.57 for SI units 625,000 for USC units
$k_{e\ uls}$	calibration factor for ultimate elastic collapse, equal to 1.089
$k_i$	factor used to determine minimum wall thickness for transverse impact specimens, as follows: 1.00 for full-size specimens 0.75 for three-quarter-size specimens 0.50 for one-half-size specimens
$k_{lsl}$	length conversion factor, equal to: 0.001 for SI units 1/12 for USC units
$k_m$	mass correction factor, equal to: 1.000 for carbon steel 0.989 for martensitic chromium steel
$k_n$	stress conversion factor, equal to: $1.18 \times 10^{-4} \text{ MPa}^{-1}$ for SI units $8.12 \times 10^{-7} \text{ psi}^{-1}$ for USC units
$k_{pi}$	upper quadrant geometry factor in API 5C5 representation of von Mises yield criterion



$k_{po}$	lower quadrant geometry factor in API 5C5 representation of von Mises yield criterion
$k_{wall}$	factor to account for the specified manufacturing tolerance of the pipe wall; e.g. for a tolerance of $-12.5\%$ , $k_{wall} = 0.875$
$k_{wpe}$	mass per unit length conversion factor, equal to: 0.0246615 for SI units 10.69 for USC units
$k_y$	bias factor for yield collapse
$k_{y\ des}$	down-rating factor for design yield collapse
$k_{y\ uls}$	calibration factor for ultimate yield collapse, equal to 0.9911
$L$	length
$L_c$	minimum length of full crest threads from end of pipe, in accordance with API 5B
$L_{ef}$	length of pipe including end finish
$L_{et}$	engaged thread length, $[= L_4 - M]$ for nominal make-up, in accordance with API 5B
$L_{eu}$	length from end of pipe to start of taper, in accordance with API 5CT
$L_{iu}$	length of pin upset, in accordance with API 5CT
$L_j$	length of a standard piece of pipe
$L_r$	load ratio
$L_1$	length from the end of the pipe to the hand-tight plane, in accordance with API 5B
$L_7$	length of perfect threads, in accordance with API 5B
$M$	length from the face of the coupling to the hand-tight plane for line pipe and for round thread casing and tubing, in accordance with API 5B
$M_b$	bending moment
$m_c$	coupling mass
$m_{cB}$	coupling mass of buttress thread casing
$m_{crsb}$	coupling mass removed by special bevel
$m_{csb}$	coupling mass with special bevel
$m_{eu}$	length of box upset taper, in accordance with API 5CT
$m_{exu}$	external upset mass
$m_{eiu}$	external-internal upset mass
$m_{irt}$	integral joint mass removed by threading and recessing
$m_{inu}$	internal upset mass
$m_{iu}$	length of pin upset taper, in accordance with API 5CT
$m_{prt}$	pin mass removed by threading
$m_{rt}$	mass removed by threading
$\mu$	model uncertainty
$N$	number of thread turns make-up
$N_L$	coupling length, in accordance with API 5CT

$N_t$	number of tests
$n$	dimensionless hardening index used to obtain a curve fit (see B.2.3.3) of the true stress-strain curve derived from the uniaxial tensile test
$ov$	ovality
$P_j$	joint strength
$p$	thread pitch, as follows: 3.175 mm (0.125 in.) for round thread casing 5.080 mm (0.200 in.) for buttress thread casing
$p_c$	collapse resistance
$p_{des}$	design collapse pressure
$p_{desi}$	design collapse pressure corrected for internal pressure
$p_{des e}$	collapse pressure corrected for axial stress and internal pressure
$p_e$	elastic collapse term
$p_{ec}$	elastic collapse pressure difference
$p_{e des}$	design elastic collapse term
$p_{e ult}$	ultimate elastic collapse term
$p_E$	pressure for elastic collapse
$p_{ht}$	hydrostatic test pressure
$p_i$	internal pressure
$p_{iF}$	internal pressure at fracture
$p_{iS}$	internal pressure at thread separation
$p_{iR}$	internal pressure at ductile rupture of an end-capped pipe
$p_{iRa}$	$p_{iR}$ adjusted for axial load and external pressure
$p_{iYAPI}$	internal pressure at yield for a thin tube
$p_{iYc}$	internal pressure at yield for coupling
$p_{iYLC}$	internal pressure at yield for a capped-end thick tube
$p_{iYLo}$	internal pressure at yield for an open-ended thick tube
$p_o$	external pressure
$p_{o ult}$	ultimate external pressure for collapse
$p_P$	pressure for plastic collapse
$p_{Pav}$	pressure for average plastic collapse
$p_T$	pressure for transition collapse
$p_{ult}$	ultimate collapse pressure
$p_y$	yield collapse term
$p_{yc}$	yield collapse pressure difference
$p_{y des}$	design yield collapse term
$p_{yM}$	through-wall von Mises yield pressure difference

$p_{Yp}$	pressure for yield strength collapse
$p_{y \text{ Tresca}}$	Tresca yield pressure for collapse
$p_{y \text{ ult}}$	ultimate yield collapse term
$p_{y \text{ vme}}$	von Mises yield pressure for collapse
$Q$	diameter of coupling recess, in accordance with API 5B
$r$	radial coordinate, $(d/2) \leq r \leq (D/2)$
$rs$	residual stress (compression at ID face is negative)
$S$	distance between flattening plates
$S_p$	standard error of estimate of the regression equation
$s_{rn}$	root truncation of the pipe thread of round threads, as follows: 0.36 mm (0.014 in.) for 10 TPI 0.43 mm (0.017 in.) for 8 TPI
$s_{t/D}$	standard deviation of $t/D$ ratios used in the regression
$T$	applied torque
$T_d$	taper (on diameter)
$t$	specified pipe wall thickness
$t_{ave}$	actual average pipe wall thickness disregarding crack-like imperfections
$t_{c \text{ ave}}$	actual average pipe wall thickness
$t_{c \text{ max}}$	maximum pipe wall thickness
$t_{c \text{ min}}$	minimum pipe wall thickness
<u><math>t_{eff}</math></u>	effective thickness as per ASTM E1928
$t_{max}$	actual maximum pipe wall thickness disregarding crack-like imperfections
$t_{min}$	actual minimum pipe wall thickness disregarding crack-like imperfections
$t_p$	tolerance interval corresponding to a confidence level of $p$ that the proportion of the population not included does not exceed
$W$	specified coupling outside diameter, in accordance with API 5CT
$u_p$	fractile corresponding to confidence level $p$
$u_{1-\theta}$	fractile, the deviation from the mean of a standardized normal cumulative distribution that includes the fraction $1 - \theta_p$ of the population
$W_L$	calculated mass of a piece of pipe of length $L$
$w_{ij}$	upset and threaded mass per unit length
$w_{pe}$	plain-end mass per unit length
$w_{tc}$	threaded and coupled mass per unit length
$w_u$	upset mass per unit length
$\bar{x}$	vector of random variables
$Z_p$	correction factor for variation in $t/D$ from average
$\alpha$	sensitivity factor

$\beta$	first-order reliability index
$\varepsilon_{\text{eng}}$	engineering strain
$\varepsilon_{\text{el}}$	minimum gauge length extension in 50.8 mm (2.0 in.), in percent, rounded to the nearest 0.5 % below 10 % and to the nearest unit percent for 10 % and larger
$\varepsilon_{\text{ln}}$	logarithmic strain
$\varepsilon_{\text{ymn}}$	strain at which specified minimum yield strength is determined
$\theta_p$	the proportion of the population not included
$\mu$	mean
$\mu_{\text{ec}}$	mean calculated eccentricity as a percent, [eccentricity = $100 (t_{\text{c max}} - t_{\text{c min}})/t_{\text{c ave}}$ ]
$\mu_{\text{fy}}$	mean calculated $f_y$ as a percent
$\mu_{\text{ov}}$	mean calculated ovality as a percent, [ovality = $100 (D_{\text{max}} - D_{\text{min}})/D_{\text{ave}}$ ]
$\mu_{\text{rs}}$	mean calculated residual stress (compression at ID face is negative)
$\nu$	Poisson's ratio
$\pi$	circumference of a circle divided by its diameter, assigned a value of 3.1416
$\phi_f$	probability of failure
$\sigma$	standard deviation
$\sigma_a$	component of axial stress not due to bending
$\sigma_b$	component of axial stress due to bending
$\sigma_c$	true (Cauchy) stress
$\sigma_e$	equivalent stress
$\sigma_{\text{eff}}$	effective axial stress; $\sigma_{\text{eff}} = F_{\text{eff}} / A_p$
$\sigma_f$	fiber stress corresponding to the percent of specified yield strength as given in Table 12
$\sigma_h$	circumferential or hoop stress
$\sigma_{\text{pmx}}$	maximum principal stress
$\sigma_r$	radial stress
$\sigma_{\text{res}}$	residual stress
$\sigma_{\text{th}}$	threshold stress
$\sigma_{\text{ye}}$	equivalent yield strength in the presence of axial tension
$\sigma_{\text{ymne}}$	equivalent minimum yield strength in the presence of axial tension
$\tau_{\text{ha}}$	torsional (shear) stress

#### 4 Conformance—Dual Referencing

In the interests of worldwide application of this document, the API Subcommittee on Tubular Goods (SC5) has decided, after detailed technical analysis, that certain documents listed in the Bibliography and prepared by API SC5 or other technical committees are interchangeable in the context of the relevant requirement with the relevant document prepared by the International Organization for Standardization (ISO) or ASTM International. These latter documents are cited in the running text following the API reference and preceded by or, for example, "API XXXX or ISO YYYY" or "ISO YYYY or ASTM ZZZZ." Application of an alternative document cited in this manner may lead to technical results different from the

~~use of the preceding API reference. However, both results are acceptable and these documents are thus considered interchangeable in practice.~~

## **5 Conformance—Units of Measurement**

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

## **4 Triaxial Yield of Pipe Body**

### **4.1 General**

The criterion for triaxial pipe body yield is that proposed by von Mises. The elastic state leading to incipient yield consists of the superposition of:

- radial and circumferential stress as determined by the Lamé equations for a thick cylinder;
- uniform axial stress due to all sources except bending;
- axial bending stress for a Timoshenko beam;
- torsional shear stress due to a moment aligned with the axis of the pipe.

Details of the derivation of the design equation can be found in Annex A.

### **4.2 Assumptions and Limitations**

#### **4.2.1 General**

Equations (1) to (7) are based on the assumptions given in 6.2.2 to 6.2.5.

#### **4.2.2 Concentric, Circular Cross-sectional Geometry**

The equations for radial stress, circumferential or hoop stress, bending and torsion presume the pipe cross section to consist of inner and outer surfaces that are circular and concentric.

#### **4.2.3 Isotropic Yield**

The yield strength of the material of which the pipe is composed is assumed to be independent of direction. An axial sample and a circumferential sample are assumed to possess identical elastic moduli and yield stress in both tension and compression.

#### **4.2.4 No Residual Stress**

For determination of the onset of yield, residual stresses due to manufacturing processes are assumed to be negligible and are ignored.

#### **4.2.5 Cross-sectional Instability (Collapse) and Axial Instability (Column Buckling)**

Particularly in instances where  $p_o > p_i$ , it is possible for the pipe cross section to collapse due to instability prior to yield. For external pressure greater than internal pressure, see Section 8 on collapse. Similarly, if

$\sigma_{\text{eff}} < 0$ , it is possible for the pipe to buckle as a column prior to yield, and the bending stress due to buckling should be included in the yield check.

### 4.3 Data Requirements

The following input data ~~are necessary to complete~~ shall be provided for the calculation for triaxial yield of the pipe body:

$c$	tube curvature, the inverse of the radius of curvature to the centerline of the pipe;
$D$	specified pipe outside diameter;
$E$	Young's modulus;
$F_a$	axial force;
$f_{\text{ymn}}$	specified minimum yield strength;
$k_{\text{wall}}$	factor to account for the specified manufacturing tolerance of the pipe wall; e.g. for a tolerance of $-12.5\%$ , $k_{\text{wall}} = 0.875$ ;
$p_i$	internal pressure;
$p_o$	external pressure;
$T$	applied torque;
$t$	specified pipe wall thickness.

### 4.4 Design Equation for Triaxial Yield of Pipe Body

**4.4.1** The onset of yield is defined as follows:

$$\sigma_e = f_{\text{ymn}} \quad (1)$$

where

$\sigma_e < f_{\text{ymn}}$  corresponds to elastic behavior;

$f_{\text{ymn}}$  is the specified minimum yield strength;

$\sigma_e$  is the equivalent stress.

**4.4.2** The equivalent stress is defined as follows:

$$\sigma_e = [\sigma_r^2 + \sigma_h^2 + (\sigma_a + \sigma_b)^2 - \sigma_r\sigma_h - \sigma_r(\sigma_a + \sigma_b) - \sigma_h(\sigma_a + \sigma_b) + 3\tau_{\text{ha}}^2]^{1/2} \quad (2)$$

with

$$\sigma_r = [(p_i d_{\text{wall}}^2 - p_o D^2) - (p_i - p_o) d_{\text{wall}}^2 D^2 / (4r^2)] / (D^2 - d_{\text{wall}}^2) \quad (3)$$

$$\sigma_h = [(p_i d_{\text{wall}}^2 - p_o D^2) + (p_i - p_o) d_{\text{wall}}^2 D^2 / (4r^2)] / (D^2 - d_{\text{wall}}^2) \quad (4)$$

$$\sigma_a = F_a / A_p \quad (5)$$

$$\sigma_b = \pm M_b r / I = \pm E c r \quad (6)$$

$$\tau_{ha} = Tr/J_p \quad (7)$$

where

- $A_p$  is the area of the pipe cross section,  $A_p = \pi/4 (D^2 - d^2)$ ;
- $c$  is the tube curvature, the inverse of the radius of curvature to the centerline of the pipe, in rad/m;
- $D$  is the specified pipe outside diameter;
- $d$  is the pipe inside diameter,  $d = D - 2t$ ;
- $d_{wall}$  is the inside diameter based on  $k_{wall} t$ ,  $d_{wall} = D - 2k_{wall} t$ ;
- $E$  is Young's modulus;
- $F_a$  is the axial force;
- $I$  is the moment of inertia of the pipe cross section,  $I = \pi/64 (D^4 - d^4)$ ;
- $J_p$  is the polar moment of inertia of the pipe cross section,  $J_p = \pi/32 (D^4 - d^4)$ ;
- $k_{wall}$  is the factor to account for the specified manufacturing tolerance of the pipe wall; e.g. for a tolerance of  $-12.5\%$ ,  $k_{wall} = 0.875$ ;
- $M_b$  is the bending moment;
- $p_i$  is the internal pressure;
- $p_o$  is the external pressure;
- $r$  is the radial coordinate, as follows:  
 $(d/2) \leq r \leq (D/2)$  for  $\sigma_a$ ,  $\sigma_b$ , and  $\tau_{ha}$   
 $(d_{wall}/2) \leq r \leq (D/2)$  for  $\sigma_r$  and  $\sigma_h$ ;
- $T$  is the applied torque;
- $t$  is the specified pipe wall thickness;
- $\sigma_a$  is the component of axial stress not due to bending;
- $\sigma_b$  is the component of axial stress due to bending;
- $\sigma_e$  is the equivalent stress;
- $\sigma_h$  is the circumferential or hoop stress;
- $\sigma_r$  is the radial stress;
- $\tau_{ha}$  is the torsional (shear) stress.

**4.4.3** The  $\pm$  sign in Equation (6) indicates that the component of axial stress due to bending can be positive (tension) or negative (compression), depending on the location of the point in the cross section. Points in the pipe cross section closer to the center of tube curvature than the centerline of the pipe experience compressive bending stress. Points in the pipe cross section farther from the center of tube curvature than the centerline of the pipe experience tensile bending stress.

**4.4.4** The variable  $c$  has units of radian/length, which are not the norm for the petroleum industry. The more common measure of  $c$  in the industry is  $^{\circ}/30 \text{ m}$  or  $^{\circ}/100 \text{ ft}$ . If, therefore, the necessary units for  $c$  are radians/meter, and  $c$  is expressed in  $^{\circ}/30 \text{ m}$  ( $^{\circ}/100 \text{ ft}$ ), the right-hand side of Equation (6) should be multiplied by the constant  $\pi/(180 \times 30) = 5.8178 \times 10^{-4} \text{ rad-m}/^{\circ} 30 \text{ m}$  [ $\pi/(180 \times 100) = 1.7453 \times 10^{-4} \text{ rad-ft}/^{\circ} 100 \text{ ft}$ ]. Similarly, if the units for  $c$  are radians/in., and  $c$  is expressed in  $^{\circ}/100 \text{ ft}$ , the right-hand side of Equation (6) should be multiplied by the constant  $\pi/(180 \times 100 \times 12) = 1.4544 \times 10^{-5}$ .

**4.4.5** In the presence of bending, Equation (2) should be evaluated four times, i.e. at the inner and outer radii on both the tensile and compressive sides of the cross section. In the presence of torsion, Equation (2) should be evaluated two times, i.e. at the inner and outer radii. In the absence of both bending and torsion, Equation (2) should be evaluated once, at the minimum value of the radius. In all cases, the maximum computed value of  $\sigma_e$  should be used in Equation (1).

**4.4.6** The purpose of the design equation is to determine the stress state that results in the onset of pipe yield when the properties of the pipe are at their worst-case, minimum allowable values. The wall thickness of the pipe at all times accounts for the extreme allowable thin-wall eccentricity that comes about naturally as part of the pipe manufacturing process.

## 4.5 Application of Design Equation for Triaxial Yield of Pipe Body to Line Pipe

The pipe body yield strength of line pipe can be calculated by means of equations presented in this section, minding the limitations given in 6.2.

## 4.6 Example Calculations

### 4.6.1 Initial Yield of Pipe Body, Lamé Equation for Pipe when External Pressure, Bending, and Torsion are Zero

#### 4.6.1.1 General

The Lamé equations for the radial and hoop stresses of the pipe are based on the three-dimensional equations of equilibrium for a linear elastic cross section. As such, these equations are triaxial equations and provide the most accurate calculation of pipe stresses. Two equations are provided: open-end with zero axial stress and closed-end with axial stress due to internal pressure acting on the end cap.

#### 4.6.1.2 Yield Design Equation, Special Case for Capped-end Conditions

**4.6.1.2.1** Initial yield of a capped-end thick tube is a special case of Equations (1) and (2) when external pressure, bending, and torsion are zero. The axial stress is generated solely by the action of internal pressure on the ends of the sample (e.g. the capped-end condition).

**4.6.1.2.2** A design equation for initial yield of the pipe body with capped-end conditions and using the Lamé equations for the radial and hoop stresses should be formulated from Equation (2), evaluated at the inner diameter. The resulting design equation is as follows:

$$P_{iYLC} = f_{ymn} / \{ (3D^4 + d_{wall}^4) / (D^2 - d_{wall}^2)^2 + d^4 / (D^2 - d^2)^2 - 2d^2 d_{wall}^2 / [(D^2 - d^2)(D^2 - d_{wall}^2)] \}^{1/2} \quad (8)$$

where

$D$  is the specified pipe outside diameter;

$d$  is the pipe inside diameter,  $d = D - 2t$ ;

$d_{wall}$  is the inside diameter based on  $k_{wall}$ ,  $d_{wall} = D - 2k_{wall} t$ ;



$f_{ymn}$  is the specified minimum yield strength;

$k_{wall}$  is the factor to account for the specified manufacturing tolerance of the pipe wall; e.g. for a tolerance of  $-12.5\%$ ,  $k_{wall} = 0.875$ ;

~~$p_{iYLo}$  is the internal pressure at yield for a capped-end thick tube;~~

$t$  is the specified pipe wall thickness.

**4.6.1.2.3** There is no adjustment to Equation (8) for axial tension, as all axial tension is generated by the action of internal pressure on the (closed) ends of the pipe. The more general case, where axial stress is generated by other than the action of internal pressure on the ends of the pipe, is addressed by the triaxial yield criterion, Equations (1) and (2).

#### **4.6.1.2.4.6.1.3 Yield Design Equation, Special Case for Internal Pressure Only and Zero Axial Load**

**4.6.1.3.1** Initial yield of an open-ended thick tube is a special case of Equations (1) and (2) when the uniform axial stress, external pressure, bending, and torsion are zero.

**4.6.1.3.2** A design equation for initial yield of the pipe body with open-end conditions and using the Lamé equations for the radial and hoop stresses should be formulated from Equation (2) evaluated at the inner diameter. The resulting design equation is as follows:

$$p_{iYLo} = f_{ymn} (D^2 - d_{wall}^2) / (3 D^4 + d_{wall}^4)^{1/2} \quad (9)$$

where

$D$  is the specified pipe outside diameter;

$d_{wall}$  is the inside diameter based on  $k_{wall}$ ,  $t$ ,  $d_{wall} = D - 2k_{wall} t$ ;

$f_{ymn}$  is the specified minimum yield strength;

$k_{wall}$  is the factor to account for the specified manufacturing tolerance of the pipe wall; e.g. for a tolerance of  $-12.5\%$ ,  $k_{wall} = 0.875$ ;

~~$p_{iYLo}$  is the internal pressure at yield for an open-ended thick tube;~~

$t$  is the specified pipe wall thickness.

The more general case, where axial stress is non-zero, is addressed by the triaxial yield criterion, in Equations (1) and (2).

### **4.6.2 Yield Design Equation, Special Case for Thin Wall Pipe with Internal Pressure Only and Zero Axial Load (Historical API Equation)**

#### **4.6.2.1 General**

The Barlow equation for pipe yield, which is the historical API equation, is based on a one-dimensional (not triaxial) approximate equation of the von Mises yield condition, combined with an approximate expression for the hoop stress in the pipe. In essence, the Barlow equation approximates the hoop stress and then equates this approximation to the yield strength. This approximation is less accurate than the Lamé equation of yield discussed in 6.6.1. Because the Barlow equation neglects axial stress, there is no distinction between pipe with capped ends, pipe with open ends, or pipe with tension end load. More detail on the Barlow equation may be found in A.2.6.

#### 4.6.2.2 Historical Yield Pressure Design Equation

Initial yield of a thin tube is defined by the following expression:

$$p_{iYAPI} = 2f_{ymn}(k_{wall}t)/D \quad (10)$$

where

$D$  is the specified pipe outside diameter;

$f_{ymn}$  is the specified minimum yield strength;

$k_{wall}$  is the factor to account for the specified manufacturing tolerance of the pipe wall; e.g. for a tolerance of  $-12.5\%$ ,  $k_{wall} = 0.875$ ;

~~$p_{iYAPI}$  is the internal pressure at yield for a thin tube;~~

$t$  is the specified pipe wall thickness.

This equation is subject to the same assumptions and limitations as the more general expressions from which it is derived (see A.1.3.3).

#### 4.6.3 Pipe Body Yield Strength, Historical API Equation

Pipe body yield strength is the axial load **necessary** to yield the pipe in the absence of internal and external pressure, bending, and torsion. It is taken as the product of the cross-sectional area and the specified minimum yield strength for the particular grade of pipe by:

$$F_{YAPI} = f_{ymn} A_p \quad (11)$$

where

$A_p$  is the area of the pipe cross section,  $A_p = \pi/4 (D^2 - d^2)$ ;

$D$  is the specified pipe outside diameter;

$d$  is the pipe inside diameter,  $d = D - 2t$ ;

~~$F_{YAPI}$  is the axial force at yield, historical API equation;~~

$f_{ymn}$  is the specified minimum yield strength;

$t$  is the specified pipe wall thickness.

#### 4.6.4 Yield in the Absence of Bending and Torsion

**4.6.4.1** If bending and torsion are zero, Equation (2) reduces to:

$$\sigma_e = [\sigma_r^2 + \sigma_h^2 + \sigma_a^2 - \sigma_r\sigma_h - \sigma_r\sigma_a - \sigma_h\sigma_a]^{1/2} \quad (12)$$

Consider, for example, the case of a tube being lowered open-ended into a vertical well full of fluid with mass density  $1.080 \text{ kg/dm}^3$  ( $9 \text{ lb/gal}$ ). The internal pressure and external pressure at any depth are equal, and the bottom of the tube is exposed to an axial compression equal to the product of the fluid pressure and the tube cross-sectional area. Ignoring connections, let the tube be of outside diameter  $244.48 \text{ mm}$  ( $9.625 \text{ in.}$ ), of wall thickness  $13.84 \text{ mm}$  ( $0.545 \text{ in.}$ ), and have  $k_{wall} = 0.875$ . Assume the mass density of steel is  $7.85 \text{ kg/dm}^3$  ( $0.2836 \text{ lb/in.}^3$ ). Compute the yield state of the uppermost cross section of the tube when the tube is at a depth of  $3000 \text{ m}$  ( $9842.5 \text{ ft}$ ). The tube has a minimum yield stress of  $551.6 \text{ MPa}$  ( $80,000 \text{ psi}$ ).

Table 1 presents the results of the calculation in both SI and USC units.

**Table 1—Example Calculations: Yield in the Absence of Bending and Torsion**

Term	SI		USC	
	Value	Units	Value	Units
Load				
$F_a$	1,997,797	N	449,205	lb
$p_i$	31.73	MPa	4602	psi
$p_o$	31.73	MPa	4602	psi
$c$	0	°/30 m	0	°/100 ft
$T$	0	N-m	0	ft-lb
Geometry				
$D$	244.48	mm	9.625	in.
$t$	13.84	mm	0.545	in.
$k_{wall}$	0.875		0.875	
Material				
$E$	206,842	MPa	30,000,000	psi
Calculations				
$d$	216.80	mm	8.535	in.
$d_{wall}$	220.26	mm	8.671	in.
$A_p$	10,028	mm <sup>2</sup>	15.55	in. <sup>2</sup>
$I$	66,920,762	mm <sup>4</sup>	160.8	in. <sup>4</sup>
$J_p$	133,841,524	mm <sup>4</sup>	321.6	in. <sup>4</sup>
$\sigma_a$	199.22	MPa	28,894	psi
Inner radius				
$\sigma_b$	0	MPa	0	psi
$\sigma_h$	-31.73	MPa	-4602	psi
$\sigma_r$	-31.73	MPa	-4602	psi
$\tau_{ha}$	0	MPa	0	psi
$\sigma_e, \sigma_b^+$	230.95	MPa	33,496	psi
$\sigma_e, \sigma_b^-$	230.95	MPa	33,496	psi
Outer radius				
$\sigma_b$	0	MPa	0	psi
$\sigma_h$	-31.73	MPa	-4602	psi
$\sigma_r$	-31.73	MPa	-4602	psi
$\tau_{ha}$	0	MPa	0	psi
$\sigma_e, \sigma_b^+$	230.95	MPa	33,496	psi
$\sigma_e, \sigma_b^-$	230.95	MPa	33,496	psi

**4.6.4.2** Since both bending and torsion are zero in this example, there is no need to calculate  $\sigma_e$  at both the inner and outer radii of the tube, as yield will always occur at the inner radius. The calculations have been included here for completeness.

**4.6.4.3** As the equivalent stress is less than the yield stress, the tube does not yield at its uppermost cross section.

## 5 Ductile Rupture of the Pipe Body

### 5.1 General

**5.5.1** The equations for ductile rupture pertain to the actual failure of the pipe body due to internal pressure. Whereas the yield equations of Section 6-4 are intended to describe the onset of permanent plastic deformation and not loss of pressure integrity, the rupture equations are intended to describe the ultimate pressure capacity of the pipe at a pressure that fails the pipe body with loss of internal pressure integrity.

**5.5.2** The equations for ductile rupture depend on the minimum physical wall thickness, the pipe outer diameter, the maximum depth of imperfections that have a reasonable probability of passing through the inspection process undetected, the fracture toughness of the material, the work hardening of the material, and the ultimate tensile strength of the pipe. Yield strength has no direct impact on the ductile rupture pressure except through the correlation of the work hardening parameter  $n$ .

**5.5.3** The ductile rupture equations can be derived from the mechanics of pipe equilibrium combined with a model of pipe plasticity and a model of the effect of imperfections. The selection of the basic equation, the application of the basic equation to API tubular products, and the verification of the equation with actual test data are discussed in detail in Annex B. The ductile rupture limit state and design equations consist of the following three interlinked concepts:

- equation for equilibrium-plasticity based rupture of a pipe with known physical wall thickness and diameter;
- reduction in performance due to wall loss in proportion to depths of imperfections that may not be detected by the manufacturing and inspection system;
- criterion for minimum toughness at which ductile rupture applies.

**5.5.4** These equations are applicable to direct pressure and axial loading, but do not describe the capacity of the pipe under fatigue loading. The subtraction to the pipe wall for the presence of imperfections and the interrelated role of pipe toughness are based on a fracture mechanics approach relating  $J_{Ic}$  toughness measurements of sample pipes to numerically calculated crack-tip intensities (J-integrals) as a function of imperfection depth. This is explained in detail in Annex B.

### 5.2 Assumptions and Limitations

**5.2.1** These equations are applicable only when the pipe material in its environment has sufficient toughness to meet a minimum criterion such that the deformation of the pipe, in its environment, through to rupture is ductile and not brittle, even in the presence of small imperfections.

**5.2.2** Bending stresses (for instance, due to buckling or bending due to curvature of the well trajectory) are not included in the equation for ductile rupture pressure. Hence, the ductile rupture equation may not apply to pipe that is buckled (nor to pipe in a dogleg).

### 5.3 Data Requirements

#### 5.3.1 General

The following input data are necessary to complete the calculation for ductile rupture of the pipe body:

- $a_N$  imperfection depth associated with a specified inspection threshold, i.e. the maximum depth of a crack-like imperfection that could reasonably be missed by the pipe inspection system; e.g. for a

5 % imperfection threshold inspection in a 12.7 mm (0.500 in.) wall thickness pipe,  $a_N = 0.635$  mm (0.025 in.);

$D$  specified pipe outside diameter;

$f_{umn}$  specified minimum tensile strength;

$k_a$  burst strength factor, having the numerical value 1.0 for quenched and tempered (martensitic structure) or 13Cr products and 2.0 for as-rolled and normalized products based on available test data, and the default value set to 20 where the value has not been measured;

NOTE The value of  $k_a$  can be established for a specific pipe material based on testing.

$k_{wall}$  factor to account for the specified manufacturing tolerance of the pipe wall; e.g. for a tolerance of -12.5 %,  $k_{wall} = 0.875$ ;

$n$  dimensionless hardening index used to obtain a curve fit (see B.2.3.3) of the true stress-strain curve derived from the uniaxial tensile test;

$t$  specified pipe wall thickness.

### 5.3.2 Determination of the Hardening Index

**5.3.2.1** In the absence of stress-strain information, the following values of  $n$  in Table 2 should be used.

**Table 2—Suggested Values for Hardening Index in Ductile Rupture Equation**

API Grade	$n$
H40	0.14
J55	0.12
K55	0.12
M65	0.12
N80	0.10
L80 Type 1	0.10
L80 Chrome	0.10
C90	0.10
R95	0.09
T95	0.09
P110	0.08
Q125	0.07

**5.3.2.2** If the grade of the material is unknown but is not high-hardening, the hardening index used to obtain a curve fit (see B.2.3.3) of the true stress-strain curve derived from the uniaxial tensile test can alternatively be determined from the following correlation:

$$n = 0.1693 - k_n f_y \quad (13)$$

where  $f_y$  has units of MPa (psi);

$f_y$  is the yield strength of a representative tensile specimen, in MPa (psi);

$k_n$  is the stress conversion factor, equal to:

$1.8 \times 10^{-4} \text{ MPa}^{-1}$  for SI units

$8.12 \times 10^{-7} \text{ psi}^{-1}$  for USC units;

$n$  is the dimensionless hardening index used to obtain a curve fit (see B.2.3.3) of the true stress-strain curve derived from the uniaxial tensile test.

**5.3.2.3** The effort expended to determine  $n$  should be weighed against the fact that the equation for ductile rupture is relatively insensitive to this quantity for commonly used OCTG. However, if a high-hardening material such as duplex steel is used, it is important to determine  $n$  to avoid a nonconservative rupture strength prediction. Values of  $n$  for these materials can be as high as 0.30.

### 5.3.3 Determination of the Burst Strength Factor, $k_a$

#### 5.3.3.1 General

**5.3.3.1.1** The burst strength factor  $k_a$  quantifies the impact that the material toughness has on ductile rupture when a crack of depth  $a_N$  is present. The value of  $k_a$  does not need to be determined for each pipe order; instead, it is recommended that  $k_a$  be determined for a fixed tubular product line with a fixed process control plan. For a material with high material toughness,  $k_a$  will be 1.0 or lower, and the influence the crack has on the ductile rupture pressure will be no greater than the depth of the crack. However, for a pipe material with lower material toughness,  $k_a$  can be larger, say 2.0, so that the penalty from having the crack is as though the crack had twice its actual depth.

**5.3.3.1.2** When  $k_a$  has not been determined for the pipe material, use a  $k_a$  value of 2.0. The value of  $k_a$  has been measured to be 1.0 for quenched and tempered (martensitic structure) pipe and for 13Cr products. The value of  $k_a$  can be established for a specific pipe material based on testing. One of two methods presented below can be used to calculate  $k_a$ .

#### 5.3.3.2 Analytical Method for Determining $k_a$

The following procedure should be used.

- 1) Construct a finite element model of a pipe with three separate cases of crack depth: zero, 5 %, and 12.5 %. Model the crack as an infinitely long, longitudinal crack on the ID of the pipe. In the finite element model, use the specified pipe wall thickness (do not reduce the pipe wall for manufacturing eccentricity) and a typical stress-strain curve for the material being analyzed.
- 2) Use the finite element model to simulate applying internal pressure to the pipe model to calculate the value of the J-integral of the pipe as a function of internal pressure (see B.7.2 for examples).
- 3) Experimentally measure the critical  $J_{1C}$  value of the pipe material in air.  $J_{1C}$  is a parameter similar to  $K_{1C}$ , but based on a different type of test. See ASTM E1152-95 for the precisely defined test methodology.

- 4) Terminate each of the finite element J-integral curves at the critical value of  $J_{1c}$  measured in the test. The internal pressure corresponding to this terminal point (where the J-integral equals  $J_{1c}$ ) represents the rupture pressure in the presence of the crack, for a material with toughness reflected by  $J_{1c}$ .
- 5) Next, divide the rupture pressure for pipe with a crack depth 5 % by the rupture pressure for pipe with no crack, using the finite element results combined with the  $J_{1c}$  measurement. Next, set this ratio equal to  $(1-k_a a/t)$  where  $a/t$  is the ratio of the crack depth to the specified pipe wall thickness (i.e. 5 % in this case). Solve this equation for the value of  $k_a$ .
- 6) Repeat the above calculation using the finite element model results for the crack with 12.5 % depth.
- 7) Average the two values of  $k_a$  from steps e) and f) and use the average in the rupture equation.

### 5.3.3.3 Experimental Method for Determining $k_a$

An alternative way to measure  $k_a$  in lieu of the  $J_{1c}$  measurement and the finite element calculation is to conduct full-scale pipe burst tests using a pipe with no measurable crack-like imperfection, another pipe with a crack-like imperfection of 5 % depth, and another pipe with a crack-like imperfection of 12.5 % depth. Then construct the ratios of the rupture pressures as described above and calculate  $k_a$ . The limitation and difficulty with this approach is that it is not valid unless the imperfection is sharp and crack-like. A mechanical or EDM notch is not adequate for this purpose. This means that the crack-like imperfection needs to be generated either as part of the pipe manufacturing process or else through fatigue pre-cracking. In general, the finite element approach combined with the  $J_{1c}$  measurement is easier to pursue.

### 5.3.3.4 Sensitivity of $k_a$

It turns out that  $k_a$  is a weakly sensitive parameter in regard to the grade of the pipe; that is,  $k_a$  primarily depends on the manufacturing process and does not vary much by grade within a fixed process. This is understandable since  $k_a$  only represents a potential amplification of the pipe's response to the presence of the crack while being loaded to rupture. Because of this, it is recommended that  $k_a$  only be determined for a specific grade of product and not for specific sizes or orders of product.

## 5.4 Design Equation for Capped-end Ductile Rupture

### 5.4.1 Minimum ductile rupture of a tube is defined by:

$$p_{IR} = 2k_{dr}f_{umN}(k_{wall}t - k_a a_N)/[D - (k_{wall}t - k_a a_N)] \quad (14)$$

where

$a_N$  is the imperfection depth associated with a specified inspection threshold, i.e. the maximum depth of a crack-like imperfection that could reasonably be missed by the pipe inspection system. For example, for a 5 % imperfection threshold inspection in a 12.7 mm (0.500 in.) wall thickness pipe,  $a_N = 0.635$  mm (0.025 in.);

$D$  is the specified pipe outside diameter;

$f_{umN}$  is the specified minimum tensile strength;



- $k_a$  is the burst strength factor, having the numerical value 1.0 for quenched and tempered (martensitic structure) or 13Cr products and 2.0 for as-rolled and normalized products based on available test data, and the default value set to 2.0 where the value has not been measured; the value of  $k_a$  can be established for a specific pipe material based on testing;
- $k_{dr}$  is the correction factor based on pipe deformation and material strain hardening, having the numerical value  $[(1/2)^{n+1} + (1/\sqrt{3})^{n+1}]$ ;
- $k_{wall}$  is the factor to account for the specified manufacturing tolerance of the pipe wall; e.g. for a tolerance of  $-12.5\%$ ,  $k_{wall} = 0.875$ ;
- $n$  is the dimensionless hardening index used to obtain a curve fit (see B.2.3.3) of the true stress-strain curve derived from the uniaxial tensile test;
- $p_{iR}$  internal pressure at ductile rupture of an end-capped pipe;
- $t$  specified pipe wall thickness.

**5.4.2** The value selected for  $k_{dr}$  renders  $p_{iR}$  the average of rupture pressures predicted using Tresca yield condition and von Mises yield condition.

**5.4.3** The factor  $k_{wall}$  addresses minimum pipe body wall thickness without considering imperfections. This value may be adjusted if minimum wall is guaranteed by a particular process or purchasing agreement.

**5.4.4** Crack-like imperfections are accounted for by  $a_N$ . The term  $k_a a_N$  represents a further reduction in minimum wall thickness associated with crack-like imperfections that are outside the sensitivity setting of the inspection equipment and assumed coincident with the location of minimum wall thickness. This stacking of minimum wall thickness and a crack-like imperfection depends on the frequency of occurrence of thin wall and the frequency of occurrence of sharp-bottomed imperfections approaching the depth of the inspection threshold.

**5.4.5** For the deterministic calculation of rupture pressure, it is necessary to calculate a conservative ductile rupture design pressure. In this case, the frequency of occurrence of the imperfection is set to 100 % and the imperfection depth equals the inspection threshold.

**5.4.6** For the probabilistic calculation of rupture pressure (see Annex B), the depth of the imperfection still equals the depth of the inspection threshold, but the calculation takes account of the actual frequency of occurrence of thin wall and the actual frequency of occurrence of sharp-bottomed imperfections with depth comparable to the inspection threshold.

## 5.5 Adjustment for the Effect of Axial Force and External Pressure

### 5.5.1 General

**5.5.1.1** The ductile rupture strength Equation (14) was developed for the situation of an end-capped pipe, where the axial tension is determined by the internal pressure acting on the closed inner pipe surface area. This is a special case of a more general situation where a pipe may reach a maximum internal pressure load, that is a rupture load, under the simultaneous action of arbitrary external pressure and arbitrary axial force. The combined loads together determine when the pipe is going to yield and how it will plastically deform toward the point of rupture. A fundamental criterion when this rupture load is attained can still be expressed, but this will now be a more involved equation governed by the formulation of the von Mises or Tresca yield surface in terms of axial stress, radial stress, and hoop stress.



**5.5.1.2** Moreover, rupture is the prevailing failure mechanism only if the axial force is not too large. For large axial tension and smaller internal overpressure, a maximum axial load (a precursor to necking and axial splitting of the pipe) is reached before the maximum pressure phenomenon occurs.

**5.5.1.3** Below, equations for both rupture and necking under combined loads are described, together with a criterion to identify which phenomenon occurs first. The equation is given in terms of “effective axial force” associated with “effective axial stress” defined in A.1.3.2.4. For effective axial force, these approximate equations are very accurate when compared to the exact theoretical equation [24]; performance against combined loading test data is given in B.6.2.

**5.5.1.4** For negative values of effective axial force, i.e. effective axial compression, the pipe can buckle as a column, depending on the quality of lateral support. If buckling is adequately suppressed, the equation for rupture under combined loads is valid also for effective axial compression. However, for higher values of effective axial compression, it is the phenomenon of local buckling of the pipe wall (“wrinkling”) that presents the governing failure mechanism. Therefore, there exists a value of effective axial compression that limits the validity of the exact combined loading rupture equation.

## 5.5.2 Design Equation for Ductile Rupture under Combined Loads

**5.5.2.1** In the presence of external pressure and axial force different from capped-end conditions, the general equation for ductile rupture is as follows:

$$p_{iRa} = p_o + \min[1/2(p_M + p_{ref T}), p_M] \quad (15)$$

with

$$p_M = p_{ref M} [1 - k_R (F_{eff}/F_{uts})^2]^{1/2} \quad (16)$$

where

$$F_a = \pi t (D - t) \sigma_a \quad (17)$$

$$F_{eff} = F_a + p_o \pi t (D - t) - p_M t (D - t) / [(k_{wall} t - k_a a_N) (D - k_{wall} t + k_a a_N)] \pi / 4 [D - 2 (k_{wall} t - k_a a_N)]^2 \quad (18)$$

$$F_{uts} = \pi t (D - t) f_{umN} \quad (19)$$

$$p_{uts} = 2 f_{umN} (k_{wall} t - k_a a_N) / [D - (k_{wall} t - k_a a_N)] \quad (20)$$

$$p_{ref} = 1/2 (p_{ref M} + p_{ref T}), \text{ used in Figure 1} \quad (21)$$

$$p_{ref M} = (2/\sqrt{3})^{1+n} (1/2)^n p_{uts} \quad (22)$$

$$p_{ref T} = (1/2)^n p_{uts} \quad (23)$$

$$k_R = (4^{1-n} - 1) 3^{1-n} \quad (24)$$

and

$a_N$  is the imperfection depth associated with a specified inspection threshold, i.e. the maximum depth of a crack-like imperfection that could reasonably be missed by the pipe inspection system, e.g. for a 5 % imperfection threshold inspection in a 12.7 mm (0.500 in.) wall thickness pipe,  $a_N = 0.635$  mm (0.025 in.);

$D$  is the specified pipe outside diameter;

$F_a$  is the axial force;

$f_{umn}$  is the specified minimum tensile strength;

$F_{eff}$  is the effective axial load, i.e. for perfect pipes the axial load additional to the end-cap forces induced by internal and external pressures;

$k_a$  is the burst strength factor, having the numerical value 1.0 for quenched and tempered (martensitic structure) or 13Cr products and 2.0 for as-rolled and normalized products based on available test data, and the default value set to 2.0 where the value has not been measured;

NOTE The value of  $k_a$  can be established for a specific pipe material based on testing.

$k_{wall}$  is the factor to account for the specified manufacturing tolerance of the pipe wall; e.g. for a tolerance of  $-12.5\%$ ,  $k_{wall} = 0.875$ ;

$n$  is the dimensionless hardening index used to obtain a curve fit (see B.2.3.3) of the true stress-strain curve derived from the uniaxial tensile test;

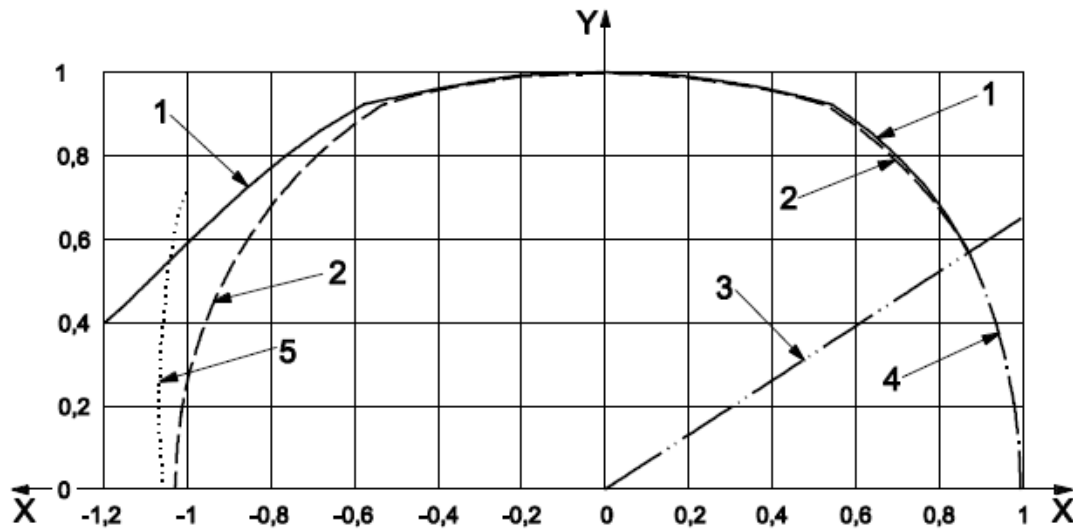
$p_{iR}$  is the internal pressure at ductile rupture of an end-capped pipe;

$p_{iRa}$  is  $p_{iR}$  adjusted for axial load and external pressure;

$p_o$  is the external pressure;

$t$  is the specified pipe wall thickness.

**5.5.2.2** Equation (15) is illustrated in Figure 1, together with the exact formulation.



#### Key

- X effective axial force ( $F_{\text{eff}}/F_{\text{uts}}$ )
- Y pressure differential ( $(p_i - p_o)/p_{\text{ref}}$ )
- 1 rupture (exact)
- 2 rupture [Equation (15)]
- 3 transition
- 4 necking [Equation (26)]
- 5 wrinkling

**Figure 1—Illustration of the Effect of Effective Axial Force and External Pressure on Ductile Rupture**

**5.5.2.3** Under capped-end conditions, the effective axial load is zero and Equation (15) reduces to Equation (14).

**5.5.2.4** The rupture equation is valid; i.e. rupture occurs before necking, when:

$$F_{\text{eff}}/F_{\text{uts}} \leq (\sqrt{3}/2)^{1-n} \quad (25)$$

### 5.5.3 Design Equation for Ductile Necking under Combined Loads

**5.5.3.1** In the presence of internal and external pressure, the general equation for ductile necking is as follows:

$$F_{\text{eff}} = F_{\text{uts}} \sqrt{[1 - k_N [(p_i - p_o)/p_{\text{ref}} M]^2]} \quad (26)$$

where

$$F_a = \pi t (D - t) \sigma_a \quad (27)$$

$$F_{\text{eff}} = F_a + p_o \pi t (D - t) - p_M t (D - t) / [(k_{\text{wall}} t - k_a a_N) (D - k_{\text{wall}} t + k_a a_N)] \pi / 4 [D - 2 (k_{\text{wall}} t - k_a a_N)]^2 \quad (28)$$

$$F_{uts} = \pi t (D - t) f_{umn} \quad (29)$$

$$p_{uts} = 2 f_{umn} (k_{wall} t - k_a a_N) / [D - (k_{wall} t - k_a a_N)] \quad (30)$$

$$p_{ref M} = (2/\sqrt{3})^{1+n} (1/2)^n p_{uts} \quad (31)$$

$$k_N = 4^{1-n} - 3^{1-n} \quad (32)$$

**5.5.3.2** Under zero pressure conditions, the effective axial load equals the true axial load, and Equation (26) for the maximum axial load reduces to the ultimate tensile strength.

**5.5.3.3** The necking equation is valid; i.e. necking occurs before rupture, when:

$$(p_i - p_o) / p_{ref M} \leq (1/2)^{1-n} \quad (33)$$

#### 5.5.4 Boundary between Rupture and Necking

Comparing Equations (15) and (26) reveals that the necking criterion is reached earlier than the rupture criterion when:

$$F_{eff} / F_{uts} \geq (3/2) (p_i - p_o) / p_{uts} \quad (34)$$

and this criterion (also shown in Figure 1) describes the boundary between rupture and necking.

#### 5.5.5 Axisymmetric Wrinkling under Combined Loads

Figure 1 shows that in the axial compression range, i.e. for negative values of the effective axial load, Equation (15) is conservative when compared to both the exact rupture equation and the local pipe wall buckling limit, called wrinkling. Although it would be easy to construct an equation such as Equation (16) with a different factor  $k_R$  that would better fit the exact rupture curve in the effective axial compression range, it is perceived such a separate equation would not have great practical impact.

### 5.6 Example Calculations

#### 5.6.1 Ductile Rupture of an End-capped Pipe

**5.6.1.1** For an end-capped pipe under pressure load, the effective axial load is zero and Equations (14) and (15) are the same. Moreover, if burst tests are performed on end-capped pipes with an additional axial load applied at the closed pipe ends, this is a situation where the effective axial load is given, and Equation (15) can be used directly to calculate the rupture pressure for any value of effective axial load.

**5.6.1.2** Compute the ductile rupture pressure of a 177.8 mm (7 in.) tube with wall thickness 11.51 mm (0.453 in.) made of P110 material. The tube is end-capped, and no additional axial load is present. Use the hardening index suggested in Table 2 and assume an inspection threshold of 5 % of the wall thickness.

Table 3 presents the results of the calculation in both SI and USC units.

**Table 3—Example Calculations: Ductile Rupture of an End-capped Pipe**

Term	SI		USC	
	Value	Units	Value	Units
Load				
$F_{\text{eff}}$	0	N	0	lb
Geometry				
$D$	177.8	mm	7	in.
$t$	11.51	mm	0.453	in.
$k_{\text{wall}}$	0.875		0.875	
Material				
$E$	206,842	MPa	30,000,000	psi
$f_{\text{um}}$	862	MPa	125,000	psi
Calculations				
$n$	0.08		0.08	
$a_N$	0.575	mm	0.0227	in.
$k_a$	1		1	
$p_{\text{IR}}$	99.7	MPa	14,460	psi

### 5.6.2 Ductile Rupture for a Given True Axial Load

**5.6.2.1** If the true axial load on the pipe is given, the pressure differential  $p_M$  cannot be calculated directly from Equation (15), because the effective axial load is a function of  $p_M$ . The solution can be found by solving for  $p_M$  iteratively, or by rewriting Equation (15) as a quadratic equation in the unknown  $p_M$ .

**5.6.2.2** Compute the ductile rupture pressure of a 177.8 mm (7 in.) tube with wall thickness 11.51 mm (0.453 in.) made of P110 material. The axial compressive load is 889,600 N (200,000 lb). Use the hardening index suggested in Table 2 and assume an inspection threshold of 5 % of the wall thickness.

Table 4 presents the results of the calculation in both SI and USC units.

**Table 4—Example Calculations: Ductile Rupture for a Given True Axial Load**

Term	SI		USC	
	Value	Units	Value	Units
Load				
$F_a$	−889,600	N	−200,000	lb
$p_o$	0	MPa	0	psi
Geometry				
$D$	177.8	mm	7	in.
$t$	11.51	mm	0.453	in.
$k_{wall}$	0.875		0.875	
$k_a$	1		1	
Material				
$E$	206,843	MPa	30,000,000	psi
$f_{umn}$	862	MPa	125,000	psi
Calculations				
$n$	0.08		0.08	
$a_N$	0.5766	mm	0.0227	in.
$F_{uts}$	5,180,423	N	1,164,663	lb
$p_{uts}$	97.20	MPa	14,098	psi
$P_{ref T}$	91.96	MPa	13,338	psi
$P_{ref M}$	107.42	MPa	15,579	psi
$k_R$	0.93904		0.93904	
$p_M$	88.92	MPa	12,896	psi
$p_{IRa}$	88.92	MPa	12,896	psi

## 6 External Pressure Resistance

### 6.1 General

**6.1.1** The collapse design equation is intended for load cases when the external fluid pressure exceeds internal fluid pressure. A convenient, theoretically rigorous equation for cross-sectional collapse of a tube that accounts for realistic imperfections does not currently exist. The approach taken here combines theoretical, numerical, and statistical tools.

**6.1.2** The equations in [Section 8](#) this section are taken directly from [API 5C3 \(6<sup>th</sup> Edition\)](#). The collapse equations presented here were originally developed in USC units and should only be used in these units.

### 6.2 Assumptions and Limitations

The following limitation applies to [For](#) the design equations for external pressure resistance.

the axial tension correction does not include the nonuniform axial stress component due to bending. Including bending is a design issue left to the individual user.

### 6.3 Data Requirements

The following input data are necessary to complete the calculation for collapse of the pipe body:

- $D$  specified pipe outside diameter;
- $f_{ymn}$  specified minimum yield strength;
- $t$  specified pipe wall thickness.

### 6.4 Design Equation for Collapse of Pipe Body

#### 6.4.1 General

The minimum collapse strength for pipe with no axial force or internal pressure is given by a series of equations, depending on the specified minimum yield strength and cross-sectional dimensions of the tube body.

#### 6.4.2 Yield Strength Collapse Pressure Equation

**6.4.2.1** The yield strength collapse pressure is not a true collapse pressure, but rather the external pressure,  $p_{Yp}$ , that generates minimum yield stress,  $f_{ymn}$ , on the inside wall of a tube as calculated by:

$$p_{Yp} = 2 f_{ymn} [(D/t) - 1] / [(D/t)^2] \quad (35)$$

where

- $D$  is the specified pipe outside diameter;
- $f_{ymn}$  is the specified minimum yield strength;
- $p_{Yp}$  is the pressure for yield strength collapse;
- $t$  is the specified pipe wall thickness.

**6.4.2.2** Equation (35) for yield strength collapse pressure is applicable for  $D/t$  values up to the value of  $D/t$  corresponding to the intersection with the plastic collapse [Equation (37)]. This intersection is calculated by:

$$(D/t)_{yp} = \{[(A_c - 2)^2 + 8(B_c + C_c/f_{ymn})]^{1/2} + (A_c - 2)\} / [2(B_c + C_c/f_{ymn})] \quad (36)$$

where

- $A_c$  is the empirical constant in historical API collapse equation;
- $B_c$  is the empirical constant in historical API collapse equation;
- $C_c$  is the empirical constant in historical API collapse equation;
- $f_{ymn}$  is the specified minimum yield strength.

**6.4.2.3** The parameters used to calculate collapse pressures depend on the pipe yield strength and on the axial load, as explained in later subsections.

The applicable  $D/t$  ratios for yield strength collapse are shown in Table 5.

**Table 5—Yield Collapse Pressure Equation Range**

Grade <sup>a</sup>	$D/t$ Range <sup>b</sup>
H40	16.40 and less
–50	15.24 and less
J55, K55	14.81 and less
–60	14.44 and less
–70	13.85 and less
C75, E75	13.60 and less
L-N-80	13.38 and less
C90	13.01 and less
R95, T95, X95	12.85 and less
–100	12.70 and less
P105, G105	12.57 and less
P110	12.44 and less
–120	12.21 and less
Q125	12.11 and less
–130	12.02 and less
S135	11.92 and less
–140	11.84 and less
–150	11.67 and less
–155	11.59 and less
–160	11.52 and less
–170	11.37 and less
–180	11.23 and less

<sup>a</sup> Grades indicated without letter designation are not API grades but are grades in use or grades being considered for use and are shown for information purposes.

<sup>b</sup> The  $D/t$  range values were calculated from Equations (36), (43) or (48), (44) or (49), and (45) or (50).

### 6.4.3 Plastic Collapse Pressure Equation

**6.4.3.1** The minimum collapse pressure for the plastic range of collapse is calculated by:

$$p_P = f_{ymn} [A_c/(D/t) - B_c] - C_c \quad (37)$$

where

$A_c$  is the empirical constant in historical API collapse equation;



$B_c$  is the empirical constant in historical API collapse equation;

$C_c$  is the empirical constant in historical API collapse equation;

$D$  is the specified pipe outside diameter;

$f_{ymn}$  is the specified minimum yield strength;

~~$p_p$  is the pressure for plastic collapse;~~

$t$  is the specified pipe wall thickness.

**6.4.3.2** The equation for minimum plastic collapse pressure is applicable for  $D/t$  values ranging from  $(D/t)_{yp}$ , Equation (36) for yield strength collapse pressure, to the intersection with Equation (39) for transition collapse pressure  $(D/t)_{pt}$ . Values for  $(D/t)_{pt}$  are calculated by means of:

$$(D/t)_{pt} = [f_{ymn} (A_c - F_c)] / [C_c + f_{ymn} (B_c - G_c)] \quad (38)$$

where

$A_c$  is the empirical constant in historical API collapse equation;

$B_c$  is the empirical constant in historical API collapse equation;

$C_c$  is the empirical constant in historical API collapse equation;

$F_c$  is the empirical constant in historical API collapse equation;

$f_{ymn}$  is the specified minimum yield strength;

$G_c$  is the empirical constant in historical API collapse equation.

The factors and applicable  $D/t$  range for the plastic collapse equation are shown in Table 6.

**Table 6—Equation Factors and  $D/t$  Range for Plastic Collapse**

Grade <sup>a</sup>	$A_c$	$B_c$	$C_c$ psi	$D/t$ Range <sup>b</sup>
H40	2.950	0.0465	754	16.40 to 27.01
–50	2.976	0.0515	1056	15.24 to 25.63
J55, K55	2.991	0.0541	1206	14.81 to 25.01
–60	3.005	0.0566	1356	14.44 to 24.42
–70	3.037	0.0617	1656	13.85 to 23.38
C75, E75	3.054	0.0642	1806	13.60 to 22.91
L-N-80	3.071	0.0667	1955	13.38 to 22.47
C90	3.106	0.0718	2254	13.01 to 21.69
R95, T95, X95	3.124	0.0743	2404	12.85 to 21.33
–100	3.143	0.0768	2553	12.70 to 21.00
P105, G105	3.162	0.0794	2702	12.57 to 20.70
P110	3.181	0.0819	2852	12.44 to 20.41
–120	3.219	0.0870	3151	12.21 to 19.88
Q125	3.239	0.0895	3301	12.11 to 19.63
–130	3.258	0.0920	3451	12.02 to 19.40
S135	3.278	0.0946	3601	11.92 to 19.18
–140	3.297	0.0971	3751	11.84 to 18.97
–150	3.336	0.1021	4053	11.67 to 18.57
–155	3.356	0.1047	4204	11.59 to 18.37
–160	3.375	0.1072	4356	11.52 to 18.19
–170	3.412	0.1123	4660	11.37 to 17.82
–180	3.449	0.1173	4966	11.21 to 17.47

<sup>a</sup> Grades indicated without letter designation are not API grades but are grades in use or grades being considered for use and are shown for information purposes.

<sup>b</sup> The  $D/t$  range values and equation factors were calculated from Equations (36), (38), (43) or (48), (44) or (49), (45) or (50), (46) or (51), and (47) or (52).

#### 6.4.4 Transition Collapse Pressure Equation

**6.4.4.1** The minimum collapse pressure for the plastic to elastic transition zone  $p_T$  is calculated by:

$$p_T = f_{ymn} [F_c / (D/t) - G_c] \quad (39)$$

where

$D$  is the specified pipe outside diameter;

$F_c$  is the empirical constant in historical API collapse equation;

$f_{ymn}$  is the specified minimum yield strength;

$G_c$  is the empirical constant in historical API collapse equation;

$p_T$  is the pressure for transition collapse;

$t$  is the specified pipe wall thickness.

**6.4.4.2** The equation for  $p_T$  is applicable for  $D/t$  values from  $(D/t)_{pt}$ , Equation (38) for plastic collapse pressure, to the intersection  $(D/t)_{te}$  with Equation (41) for elastic collapse. Values for  $(D/t)_{te}$  are calculated by:

$$(D/t)_{te} = [2 + B_c/A_c]/[3(B_c/A_c)] \quad (40)$$

where

$A_c$  is the empirical constant in historical API collapse equation;

$B_c$  is the empirical constant in historical API collapse equation.

The factors and applicable  $D/t$  range for the transition collapse pressure equation are shown in Table 7.

**Table 7—Equation Factors and  $D/t$  Range for Transition Collapse**

Grade <sup>a</sup>	$F_c$	$G_c$	$D/t$ Range <sup>b</sup>
H40	2.063	0.0325	27.01 to 42.64
-50	2.003	0.0347	25.63 to 38.83
J55, K55	1.989	0.0360	25.01 to 37.21
-60	1.983	0.0373	24.42 to 35.73
-70	1.984	0.0403	23.38 to 33.17
C75, E75	1.990	0.0418	22.91 to 32.05
L-N-80	1.998	0.0434	22.47 to 31.02
C90	2.017	0.0466	21.69 to 29.18
R95, T95, X95	2.029	0.0482	21.33 to 28.36
-100	2.040	0.0499	21.00 to 27.60
P105, G105	2.053	0.0515	20.70 to 26.89
P110	2.066	0.0532	20.41 to 26.22
-120	2.092	0.0565	19.88 to 25.01
Q125	2.106	0.0582	19.63 to 24.46
-130	2.119	0.0599	19.40 to 23.94
S135	2.133	0.0615	19.18 to 23.44
-140	2.146	0.0632	18.97 to 22.98
-150	2.174	0.0666	18.57 to 22.11
-155	2.188	0.0683	18.37 to 21.70
-160	2.202	0.0700	18.19 to 21.32
-170	2.231	0.0734	17.82 to 20.60
-180	2.261	0.0769	17.47 to 19.93

<sup>a</sup> Grades indicated without letter designation are not API grades but are grades in use or grades being considered for use and are shown for information purposes.

<sup>b</sup> The  $D/t$  range values and equation factors were calculated from Equations (36), (38), (43) or (48), (44) or (49), (45) or (50), (46) or (51), and (47) or (52).

#### 6.4.5 Elastic Collapse Pressure Equation

The minimum collapse pressure for the elastic range of collapse,  $p_E$ , is calculated by:

$$p_E = 46.95 \times 10^6 H C / [(D/t) (D/t - 1)^2] \quad (41)$$

where

$D$  is the specified pipe outside diameter;

$H C$  is the stress conservation factor,  $3.237 \times 10^5$  MPa ( $46.95 \times 10^6$  psi);

$p_E$  is the pressure for elastic collapse;

$t$  is the specified pipe wall thickness.

The applicable  $D/t$  range for elastic collapse is shown in Table 8.

**Table 8— $D/t$  Range for Elastic Collapse**

Grade <sup>a</sup>	$D/t$ Range <sup>b</sup>
H40	42.64 and greater
–50	38.83 and greater
J55, K55	37.21 and greater
–60	35.73 and greater
–70	33.17 and greater
C75, E75	32.05 and greater
L-N-80	31.02 and greater
C90	29.18 and greater
R95, T95, X95	28.36 and greater
–100	27.60 and greater
P105, G105	26.89 and greater
P110	26.22 and greater
–120	25.01 and greater
Q125	24.46 and greater
–130	23.94 and greater
S135	23.44 and greater
–140	22.98 and greater
–150	22.11 and greater
–155	21.70 and greater
–160	21.32 and greater
–170	20.60 and greater
–180	19.93 and greater
<sup>a</sup> Grades indicated without letter designation are not API grades but are grades in use or grades being considered for use and are shown for information purposes. <sup>b</sup> The $D/t$ range values were calculated from Equations (40), (43) or (48), and (44) or (49).	

## 6.4.6 Collapse Pressure under Axial Tension Stress and Internal Pressure

**6.4.6.1** The above equations for collapse resistance  $p_c$  are valid for zero axial stress and zero internal pressure. The external pressure at collapse under combined axial stress and internal pressure can be calculated by replacing the collapse resistance  $p_c$  by the collapse pressure differential  $p_c - p_i$  and at the same time modifying the yield stress to a combined loading equivalent grade according to Equation (42) for  $\sigma_a + p_i \geq 0$ :

$$f_{ycom} = \{[1 - 0.75((\sigma_a + p_i) / f_{ymn})^2]^{1/2} - 0.5 (\sigma_a + p_i) / f_{ymn}\} f_{ymn} \quad (42)$$

where

$f_{ycom}$  is the combined loading equivalent grade, the equivalent yield strength in the presence of axial stress and internal pressure;

$f_{ymn}$  is the specified minimum yield strength;

$\sigma_a$  is the component of axial stress not due to bending;

$p_i$  is the internal pressure.

**NOTE** Equation (42) is based on the Hencky-von Mises maximum strain energy of distortion theory of yielding.

**6.4.6.2** Collapse resistance equation factors and  $D/t$  ranges for the combined loading equivalent grade are then calculated by means of Equations (36), (38), (40), (43) or (48), (44) or (49), (45) or (50), (46) or (51), and (47) or (52). Using equation factors for the combined loading equivalent grade, collapse resistance under axial stress and internal pressure is calculated by means of Equations (35), (37), (39), (41), and (42).

**6.4.6.3** If the yield strength of combined loading equivalent grade ( $f_{ycom}$ ) is less than 24,000 psi, use the collapse equation determined for  $f_{ycom}$  equal to 24,000 psi, but limited by the value obtained from Equation (35). API collapse resistance equations are not valid for the yield strength of combined loading equivalent grade ( $f_{ycom}$ ) less than 24,000 psi.

Equation (42) is based on the Hencky-von Mises maximum strain energy of distortion theory of yielding.

## 6.5 Equations for Empirical Constants

The following equations may be used to calculate the empirical constants in the historical API collapse equations. There are two versions of each equation, one each for SI and USC units.

SI units:

$$A_c = 2.8762 + 0.15489 \times 10^{-3} f_{ymn} + 0.44809 \times 10^{-6} f_{ymn}^2 - 0.16211 \times 10^{-9} f_{ymn}^3 \quad (43)$$

$$B_c = 0.026233 + 0.73402 \times 10^{-4} f_{ymn} \quad (44)$$

$$C_c = -3.2125 + 0.030867 f_{ymn} - 0.15204 \times 10^{-5} f_{ymn}^2 + 0.778 \times 10^{-9} f_{ymn}^3 \quad (45)$$

$$F_c = 3.237 \times 10^5 [(3 B_c/A_c)/(2 + B_c/A_c)]^3 / \{f_{ymn} [(3 B_c/A_c)/(2 + B_c/A_c) - B_c/A_c][1 - (3 B_c/A_c)/(2 + B_c/A_c)]^2\} \quad (46)$$

$$G_c = F_c B_c / A_c \quad (47)$$

where

$A_c$  is the empirical constant in historical API collapse equation;

$B_c$  is the empirical constant in historical API collapse equation;

$C_c$  is the empirical constant in historical API collapse equation;

$F_c$  is the empirical constant in historical API collapse equation;

$f_{ymn}$  is the specified minimum yield strength;

$G_c$  is the empirical constant in historical API collapse equation.

USC units:

$$A_c = 2.8762 + 0.10679 \times 10^{-5} f_{ymn} + 0.21301 \times 10^{-10} f_{ymn}^2 - 0.53132 \times 10^{-16} f_{ymn}^3 \quad (48)$$

$$B_c = 0.026233 + 0.50609 \times 10^{-6} f_{ymn} \quad (49)$$

$$C_c = -465.93 + 0.030867 f_{ymn} - 0.10483 \times 10^{-7} f_{ymn}^2 + 0.36989 \times 10^{-13} f_{ymn}^3 \quad (50)$$

$$F_c = 46.95 \times 10^6 [(3 B_c / A_c) / (2 + B_c / A_c)]^3 / \{ f_{ymn} [(3 B_c / A_c) / (2 + B_c / A_c) - B_c / A_c] [1 - (3 B_c / A_c) / (2 + B_c / A_c)]^2 \} \quad (51)$$

$$G_c = F_c B_c / A_c \quad (52)$$

where

$A_c$  is the empirical constant in historical API collapse equation;

$B_c$  is the empirical constant in historical API collapse equation;

$C_c$  is the empirical constant in historical API collapse equation;

$F_c$  is the empirical constant in historical API collapse equation;

$f_{ymn}$  is the specified minimum yield strength;

$G_c$  is the empirical constant in historical API collapse equation.

## 6.6 Application of Collapse Pressure Equations to Line Pipe

**6.6.1** The collapse pressure equations presented in this section are empirical relations derived from tests on pipe representative of the casing and tubing inventories listed in API 5CT. Application of these relations outside the range of yield strengths and D/t ratios contained in API 5CT is not recommended. These equations do not apply to cold expanded pipe because Bauschinger effects significantly reduce collapse resistance. Some line pipe grades listed in API 5L have a rough casing equivalent in API 5CT. However, the API 5L pipe inventory contains D/t ratios that often exceed casing D/t ratios.

**6.6.2** Cold-expanded line pipe—for example, UOE or JCOE line pipe—can exhibit collapse performance lower than the design equations in this clause. Pipeline design guidelines API 1111 and DNVGL-ST-F101 provide correlations that have been shown to yield more accurate collapse design rating predictions when compared to collapse test results. Both pipeline guidelines provide factors to add a design margin to the predicted collapse pressure, resulting in a design rating approach commensurate with this clause.

**6.6.3** For line pipe that is not cold-expanded and has a yield strength and  $D/t$  falling within the limits of the sizes and thicknesses listed in **API 5CT**, application of the equations in this section should yield reasonable estimates of minimum collapse pressure. Nevertheless, as with the application of any of the equations in this document, sound engineering judgment should prevail.

## 6.7 Example Calculations

Calculate the collapse resistance  $p_c$  of size Label 1: 7, mass Label 2: 26, Grade P110 casing with axial stress of 11,000 psi and internal pressure 1000 psi. Wall thickness is 0.362 in.

Table 9 presents the results of the calculation in both SI and USC units.

**Table 9—Example Calculation: Collapse Resistance with Internal Pressure and Tension**

Term	SI		USC	
	Value	Units	Value	Units
Load				
$\sigma_a$	75.842	MPa	11,000	psi
$p_i$	6.895	MPa	1000	psi
Geometry				
$D$	177.80	mm	7.000	in.
$t$	9.19	mm	0.362	in.
$D/t$	19.35			
Material				
$f_{ymn}$	758	MPa	110,000	psi
Calculations				
$f_{ycom}$	713.7	MPa	103,508	psi
$A_c$	3.156			
$B_c$	0.0786			
$C_c$	18.33	MPa	2658	psi
$F_c$	2.050			
$G_c$	0.0511			
Equation	Plastic			
$p_c - p_i$	42.1	MPa	6100	psi

## 7 Joint Strength

### 7.1 General

**7.1.1** Joint strength is a measure of the structural integrity of a threaded connection and does not include consideration of leak resistance. For casing applications, where installation of the tubular string is considered permanent, the limit load can be based on either yield or fracture/pull-out of the connector. For tubing applications, where the tubular string can be repeatedly recovered from and re-installed in the wellbore, the limit load is usually based on yield of the connector.

**7.1.2** In this and other sections, the following abbreviations may be used:

- BC, buttress thread and coupling;
- BC SC, buttress thread with special clearance coupling;
- EU, external upset end;
- EU SC, external upset end with special clearance coupling;
- LC, long thread and coupling (round thread);
- NU, nonupset end;
- STC, short thread and coupling (round thread).

### 7.2 API Casing Connection Tensile Joint Strength

#### 7.2.1 General

The following tensile joint strength performance properties apply to casing connections manufactured in accordance with API 5B and API 5CT.

#### 7.2.2 Round Thread Casing Joint Strength

##### 7.2.2.1 General

Round thread casing joint strength is calculated as the minimum of fracture of the pipe under the last perfect thread, of a joint failing by thread jump-out or pull-out, or of fracture in the coupling. With certain coupling dimensions, the strength of the coupling can be less than that of the pipe body. The coupling fracture strength is calculated at the root of the coupling thread coincident with the end of the pipe in the power-tight position.

##### 7.2.2.2 Assumptions and Limitations

The round thread casing joint strength equation ignores the possible presence of internal and external pressure. The effect of casing curvature on joint strength is also ignored. The coefficients in some of the equations were originally developed in USC units. It is suggested that users perform these calculations in USC units and then convert the result to SI units if needed.

##### 7.2.2.3 Data Requirements

**7.2.2.3.1** The following input data are necessary to complete the calculation for round thread casing joint strength:



- $D$  specified pipe outside diameter, in inches;
- $f_{umnp}$  specified minimum tensile strength of the pipe body, in pounds per square inch;
- $f_{ymnp}$  specified minimum yield strength of the pipe body, in pounds per square inch;
- $L_{et}$  engaged thread length,  $[= L_4 - M]$  for nominal make-up, in accordance with API 5B, in inches;
- $t$  specified pipe wall thickness, in inches.

**7.2.2.3.2** The following input data are necessary to complete the calculation for round thread coupling fracture strength:

- $A$  hand-tight standoff, in turns;
- $E_1$  pitch diameter at the hand-tight plane, in accordance with API 5B, in inches;
- $f_{umc}$  actual tensile strength of a representative tensile specimen from the coupling, in pounds per square inch;
- $H$  is the thread height of a round thread equivalent Vee thread, as follows:  
2.1996 mm (0.08660 in.) for 10 TPI  
2.7496 mm (0.10825 in.) for 8 TPI;
- $L_1$  length from the end of the pipe to the hand-tight plane, in accordance with API 5B, in inches;
- $p$  is the thread pitch, in inches;
- $s_{rn}$  root truncation of the pipe thread of round threads, as follows:  
0.36 mm (0.014 in.) for 10 TPI  
0.43 mm (0.017 in.) for 8 TPI;
- $T_d$  taper (on diameter), 0.0625 mm/mm (0.0625 in./in.);
- $W$  specified coupling outside diameter, in accordance with API 5CT, in inches.

#### 7.2.2.4 Design Equations

Round thread casing joint strength is calculated by taking the minimum of the fracture strength of the pipe threads, the pull-out strength, and the fracture strength of the coupling:

$$P_j = 0.95 A_{jp} f_{umnp} \text{ (fracture strength)} \quad (53)$$

or

$$P_j = 0.95 A_{jp} L_{et} [(0.74D^{-0.59} f_{umnp}) / (0.5L_{et} + 0.14D) + f_{ymnp} / (L_{et} + 0.14D)] \text{ (pull-out strength)} \quad (54)$$

or

$$P_j = 0.95 A_{jc} f_{umc} \text{ (coupling fracture strength)} \quad (55)$$

where

$$A_{jp} = \pi/4 [(D - 0.1425)^2 - d^2] \quad (56)$$

$$A_{jc} = \pi/4 (W^2 - d_1^2) \quad (57)$$

$$d_1 = E_1 - (L_1 + pA)T_d + H - 2s_{rn} \quad (58)$$

and

- A* is the hand-tight standoff, in turns;
- A<sub>jc</sub>* is the area of the coupling cross section, in square inches;
- A<sub>jp</sub>* is the area of the pipe cross section under the last perfect thread, in square inches;
- D* is the specified pipe outside diameter, in inches;
- d* is the pipe inside diameter,  $d = D - 2t$ , in inches;
- d<sub>1</sub>* is the diameter at the root of the coupling thread at the end of the pipe in the power-tight position, in inches;
- E<sub>1</sub>* is the pitch diameter at the hand-tight plane, in accordance with API 5B, in inches;
- f<sub>umnp</sub>* is the specified minimum tensile strength of the pipe body, in pounds per square inch;
- f<sub>umc</sub>* is the specified minimum tensile strength of the coupling, in pounds per square inch;
- f<sub>ymnp</sub>* is the specified minimum yield strength of the pipe body, in pounds per square inch;
- H* is the thread height of a round thread equivalent Vee thread, as follows:  
2.1996 mm (0.08660 in.) for 10 TPI  
2.7496 mm (0.10825 in.) for 8 TPI;
- L<sub>et</sub>* is the engaged thread length,  $[= L_4 - M]$  for nominal make-up, in accordance with API 5B, in inches;
- L<sub>1</sub>* is the length from the end of the pipe to the hand-tight plane, in accordance with API 5B, in inches;
- p* is the thread pitch, in inches;
- P<sub>j</sub>* is the joint strength, in pounds;
- s<sub>rn</sub>* is the root truncation of the pipe thread of round threads, as follows:  
0.36 mm (0.014 in.) for 10 TPI  
0.43 mm (0.017 in.) for 8 TPI;
- t* is the specified pipe wall thickness, in inches;
- T<sub>d</sub>* is the taper (on diameter), 0.0625 mm/mm (0.0625 in./in.);
- W* is the specified coupling outside diameter, in accordance with **API 5CT**, in inches.

## 7.2.3 Buttress Thread Casing Joint Strength

### 7.2.3.1 General

Buttress thread casing joint strength is calculated as the minimum of the pipe strength and of the coupling fracture strength calculated at the root of the coupling thread coincident with the end of the pipe in the power-tight position.

### 7.2.3.2 Assumptions and Limitations

The buttress thread casing joint strength equations are based on the following assumptions.

- a) The buttress thread does not fail by pull-out.

NOTE This assumption is contradicted by some test data for larger  $D/t$  ratios.

- b) The effect of internal and external pressure is ignored.
- c) The effect of casing curvature is ignored.
- d) The coefficients in some of the equations were originally developed in USC units. It is suggested that users perform these calculations in USC units and then convert the result to SI units if needed.

### 7.2.3.3 Data Requirements

The following input data are **necessary** to complete the calculation for buttress thread casing joint strength:

- $D$  specified pipe outside diameter, in inches;
- $E_7$  pitch diameter, in accordance with API 5B, in inches;
- $f_{umnc}$  specified minimum tensile strength of the coupling, in pounds per square inch;
- $f_{umnp}$  specified minimum tensile strength of the pipe body, in pounds per square inch;
- $f_{ymnp}$  specified minimum yield strength of the pipe body, in pounds per square inch;
- $h_B$  buttress thread height, equal to 1.575 mm (0.062 in.);
- $I_B$  length from the face of the buttress thread coupling to the base of the triangle in the hand-tight position, as follows:
  - 10.16 mm (0.400 in.) for Label 1: 4-1/2
  - 12.70 mm (0.500 in.) for sizes between Label 1: 5 and Label 1: 13-3/8, inclusive
  - 9.52 mm (0.375 in.) for sizes greater than Label 1: 13-3/8;
- $L_7$  length of perfect threads, in accordance with API 5B, in inches;
- $t$  specified pipe wall thickness, in inches;
- $T_d$  taper (on diameter), equal to:
  - 0.0625 mm/mm (0.0625 in./in.) for sizes less than or equal to Label 1: 13-3/8
  - 0.0833 mm/mm (0.0833 in./in.) for sizes greater than Label 1: 13-3/8;
- $W$  specified coupling outside diameter, in accordance with **API 5CT**, in inches.

### 7.2.3.4 Design Equation

Buttress thread casing joint strength is calculated by taking the minimum of the pipe thread strength and the coupling thread strength:

$$P_j = 0.95A_p f_{umnp} [1.008 - 0.0396(1.083 - f_{ymnp}/f_{umnp})D] \text{ (pipe thread strength)} \quad (59)$$

or

$$P_j = 0.95A_{jc} f_{umnc} \text{ (coupling thread strength)} \quad (60)$$

where

- $A_{jc}$  is the area of the coupling cross section,  $A_{jc} = \pi/4 (W^2 - d_1^2)$ , in square inches;
- $A_p$  is the area of the pipe cross section,  $A_p = \pi/4 (D^2 - d^2)$ , in square inches;
- $D$  is the specified pipe outside diameter, in inches;
- $d$  is the pipe inside diameter,  $d = D - 2t$ , in inches;
- $d_1$  is the diameter at the root of the coupling thread at the end of the pipe in the power-tight position, in inches;
- $f_{umnc}$  is the specified minimum tensile strength of the coupling, in pounds per square inch;
- $f_{umnp}$  is the specified minimum tensile strength of the pipe body, in pounds per square inch;
- $f_{ymnp}$  is the specified minimum yield strength of the pipe body, in pounds per square inch;
- $P_j$  is the joint strength, in pounds;
- $t$  is the specified pipe wall thickness, in inches;
- $W$  is the specified coupling outside diameter, in accordance with **API 5CT**, in inches;

and

$$d_1 = E_7 - (L_7 + I_B) T_d + h_B \quad (61)$$

where

- $E_7$  is the pitch diameter, in accordance with API 5B, in inches;
- $h_B$  is the buttress thread height, equal to 1.575 mm (0.062 in.);
- $I_B$  is the length from the face of the buttress thread coupling to the base of the triangle in the hand-tight position, as follows:
  - 10.16 mm (0.400 in.) for Label 1: 4-1/2
  - 12.70 mm (0.500 in.) for sizes between Label 1: 5 and Label 1: 13-3/8, inclusive
  - 9.52 mm (0.375 in.) for sizes greater than Label 1: 13-3/8;
- $L_7$  is the length of perfect threads, in accordance with API 5B;

$T_d$  is the taper (on diameter), as follows:  
 0.0625 mm/mm (0.0625 in./in.) for sizes less than or equal to Label 1: 13-3/8  
 0.0833 mm/mm (0.0833 in./in.) for sizes greater than Label 1: 13-3/8.

## 7.3 API Tubing Connection Tensile Joint Strength

### 7.3.1 General

The following tensile joint strength performance properties apply to tubing connections manufactured in accordance with API 5B and API 5CT.

### 7.3.2 Nonupset Tubing Joint Strength

#### 7.3.2.1 Introduction

Nonupset tubing joint strength is calculated as the product of the yield strength and the area of the pipe cross section under the last perfect thread. The areas of the critical sections of regular tubing couplings and special-clearance couplings are, in all instances, greater than the governing critical areas of the pipe part of the joint and do not affect the strength of the joint.

#### 7.3.2.2 Assumptions and Limitations

The nonupset tubing joint strength equation ignores the possible presence of internal and external pressure. The effect of tubing curvature on joint strength is also ignored. The coefficients in some of the equations were originally developed in USC units. It is suggested that users perform these calculations in USC units and then convert the result to SI units if needed.

#### 7.3.2.3 Data Requirements

The following input data are necessary to complete the calculation for nonupset tubing joint strength:

- $D$  specified pipe outside diameter, in inches;
- $D_4$  major diameter, in accordance with API 5B, in inches;
- $f_{ymn}$  specified minimum yield strength, in psi;
- $h_s$  round thread height, as follows:  
 1.3122 mm (0.05560 in.) for 10 TPI  
 1.8098 mm (0.07125 in.) for 8 TPI;
- $t$  specified pipe wall thickness, in inches.

#### 7.3.2.4 Design Equation

The joint strength in tension of nonupset tubing is defined by the following expression:

$$P_j = f_{ymn} \left\{ \pi/4 [(D_4 - 2h_s)^2 - d^2] \right\} \quad (62)$$

Pull-out strength of non-upset tubing is used for calculating make-up torque and defined by the following expression:

$$P_{jpo} = 0.95 A_{jp} L_{et} [(0.74 D_4^{0.59} f_{umnp}) / (0.5 L_{et} + 0.14 D_4) + f_{ymnp} / (L_{et} + 0.14 D_4)] \quad (63)$$

where

$A_{jp}$  is equal to  $\pi/4 [(D_4 - 2h_s)^2 - d^2]$ , in square inches;

$D$  is the specified pipe outside diameter, in inches;

$d$  is the pipe inside diameter,  $d = D - 2t$ , in inches;

$D_4$  is the major diameter, in accordance with API 5B, in inches;

$f_{umnp}$  is the specified minimum tensile strength of the pipe body, in psi;

$f_{ymnp}$  is the specified minimum yield strength of the pipe body, in psi;

$f_{ymn}$  is the specified minimum yield strength, in psi;

$h_s$  is the round thread height, as follows:

1.3122 mm (0.05560 in.) for 10 TPI

1.8098 mm (0.07125 in.) for 8 TPI;

$L_{et}$  is the engaged thread length for nominal make-up, in accordance with API 5B ( $= L_4 - M$ ), in inches ;

$P_j$  is the joint strength, in pounds;

~~$P_{jpo}$  is the joint pull-out strength, in pounds;~~

$t$  is the specified pipe wall thickness, in inches.

### 7.3.3 Upset Tubing Joint Strength

#### 7.3.3.1 General

Upset tubing joint strength is calculated as the product of the yield strength and the area of the body of the pipe. The area of the section under the last perfect thread of API upset tubing is greater than the area of the body of the pipe. The areas of the critical sections of regular tubing couplings, special-clearance couplings, and the box of integral-joint tubing are, in all instances, greater than the governing critical areas of the pipe part of the joint and do not affect the strength of the joint.

#### 7.3.3.2 Assumptions and Limitations

The upset tubing joint strength equation ignores the possible presence of internal and external pressure. The effect of tubing curvature on joint strength is also ignored. The coefficients in some of the equations were originally developed in USC units. It is suggested that users perform these calculations in USC units and then convert the result to SI units if needed.

#### 7.3.3.3 Data Requirements

The following input data are **necessary** to complete the calculation for upset tubing joint strength:

$D$  specified pipe outside diameter, in inches;

$f_{ymn}$  specified minimum yield strength, in psi;

$t$  specified pipe wall thickness, in inches.

### 7.3.3.4 Design Equation

**7.3.3.4.1** The joint strength in tension of upset tubing is defined by the following expression:

$$P_j = f_{ymn} [\pi/4 (D^2 - d^2)], \text{ in pounds.} \quad (64)$$

**7.3.3.4.2** Pull-out strength of upset tubing is used for calculating make-up torque and defined by the following expression:

$$P_{jpo} = 0.95 A_{jp} L_{et} [(0.74 D_4^{0.59} f_{umnp}) / (0.5 L_{et} + 0.14 D_4) + f_{ymnp} / (L_{et} + 0.14 D_4)] \quad (65)$$

where

$A_{jp}$  is the area under the last thread,  $\pi/4 [(D_4 - 2h_s)^2 - d^2]$ , in square inches;

$d$  is the pipe inside diameter,  $d = D - 2t$ , in inches;

$D$  is the specified pipe outside diameter, in inches;

$D_4$  is the major diameter, in accordance with API 5B, in inches;

$f_{umnp}$  is the specified minimum tensile strength of the pipe body, in psi;

$f_{ymnp}$  is the specified minimum yield strength of the pipe body, in psi;

$f_{ymn}$  is the specified minimum yield strength, in psi;

$h_s$  is the round thread height, as follows:

1.3122 mm (0.05560 in.) for 10 TPI

1.8098 mm (0.07125 in.) for 8 TPI;

$L_{et}$  is the engaged thread length for nominal make-up, in accordance with API 5B ( $= L_4 - M$ ), in inches;

$P_j$  is the joint strength, in pounds;

~~$P_{jpo}$  is the joint pull-out strength, in pounds;~~

$t$  is the specified pipe wall thickness, in inches.

## 8 Pressure Performance for Couplings

### 8.1 General

Internal pressure capacity for threaded and coupled pipe is the same as for plain-end pipe, except where a lower pressure is necessary to avoid yielding the coupling or leakage due to insufficient interface pressure leak resistance at the  $E_1$  or  $E_7$  plane as calculated below. For integral joint tubing, the box is considered the coupling.

## 8.2 Internal Yield Pressure of Round Thread and Buttress Couplings

**8.1.1** The internal yield pressure for the coupling is calculated from:

$$p_{iYc} = f_{ymnc} (W - d_1) / W \quad (66)$$

where

$f_{ymnc}$  is the specified minimum yield strength of the coupling, in psi;

$d_1$  is the diameter at the root of the coupling thread at the end of the pipe in the power-tight position, in inches;

$p_{iYc}$  is the internal pressure at yield for coupling, in psi;

$W$  is the specified coupling outside diameter, in accordance with **API 5CT**, in inches.

**8.1.2** For round thread casing and tubing:

$$d_1 = E_1 - (L_1 + pA)T_d + H - 2s_m, \text{ in inches;} \quad (67)$$

where

$A$  is the hand-tight standoff, in turns;

$E_1$  is the pitch diameter at the hand-tight plane, in accordance with API 5B, in inches;

$H$  is the thread height of a round thread equivalent Vee thread, as follows:  
2.1996 mm (0.08660 in.) for 10 TPI  
2.7496 mm (0.10825 in.) for 8 TPI;

$L_1$  is the length from the end of the pipe to the hand-tight plane, in accordance with API 5B, in inches;

$p$  is the thread pitch, in inches;

$s_m$  is the root truncation of the pipe thread of round threads, as follows:  
0.36 mm (0.014 in.) for 10 TPI  
0.43 mm (0.017 in.) for 8 TPI;

$T_d$  is the taper (on diameter), 0.0625 mm/mm (0.0625 in./in.).

**8.1.3** For buttress thread casing:

$$d_1 = E_7 - (L_7 + I_B)T_d + h_B, \text{ in inches} \quad (68)$$

where

$E_7$  is the pitch diameter, in accordance with API 5B, in inches;

$h_B$  is the buttress thread height, equal to 1.575 mm (0.062 in.);

$I_B$  is the length from the face of the buttress thread coupling to the base of the triangle in the hand-tight position, as follows:



10.16 mm (0.400 in.) for Label 1: 4-1/2  
 12.70 mm (0.500 in.) for sizes between Label 1: 5 and Label 1: 13-3/8, inclusive  
 9.52 mm (0.375 in.) for sizes greater than Label 1: 13-3/8;

$L_7$  is the length of perfect threads, in accordance with API 5B, in inches;

$T_d$  is the taper (on diameter), as follows:  
 0.0625 mm/mm (0.0625 in./in.) for sizes less than or equal to Label 1: 13-3/8  
 0.0833 mm/mm (0.0833 in./in.) for sizes greater than Label 1: 13-3/8.

### 8.3 Interface Pressure of Round Thread or Buttress Couplings

**8.3.1** The interface pressure at the  $E_1$  or  $E_7$  plane is calculated from Equation (69). Equation (69) is based on the seal being at the  $E_1$  plane for round threads and the  $E_7$  plane for buttress threads where the coupling is the weakest and the interface pressure the lowest. Also, Equation (69) is based on the interference pressure between the pipe and coupling threads resulting from make-up and the internal pressure itself, with stresses in the elastic range.

$$p_{iS} = ET_d N p (W^2 - E_s^2) / 2 E_s W^2 \quad (69)$$

where

$E$  is Young's modulus, in psi;

$E_s$  is the pitch diameter at plane of seal, as follows:

$E_1$  for round thread, in inches

$E_7$  for buttress thread casing, in inches;

$N$  is the number of thread turns make-up, as follows:

$A$  for round thread casing and tubing (in accordance with API 5B)

$A + 1.5$  for buttress thread casing smaller than 16

$A + 1$  for buttress thread casing 16 and larger;

$p$  is the thread pitch, as follows:

3.175 mm (0.125 in.) for 8-round thread casing and tubing

2.540 mm (0.100 in.) for 10-round thread tubing

5.080 mm (0.200 in.) for buttress thread casing;

~~$p_{iS}$  is the internal pressure at thread separation, in psi;~~

$T_d$  is the taper (on diameter), as follows:

0.0625 mm (0.0625 in./in.) for round thread casing and tubing

0.0625 mm (0.0625 in./in.) for buttress casing smaller than 16

0.0833 mm (0.0833 in./in.) for buttress thread casing 16 and larger;

$W$  is the specified coupling outside diameter, in accordance with **API 5CT**, in inches;

where

$A$  is the hand-tight standoff, in turns;

$E_1$  is the pitch diameter at the hand-tight plane, in accordance with API 5B, in inches;

$E_7$  is the pitch diameter, in accordance with API 5B, in inches.

**8.3.2** The interface pressure between the pin and box as a result of make-up is as follows:

$$p_1 = ET_d Np(W^2 - E_s^2)(E_s^2 - d^2)/E_s^2(W^2 - d^2), \text{ in psi} \quad (70)$$

where

$E$  is Young's modulus, in psi;

$E_s$  is the pitch diameter at plane of seal, as follows:

$E_1$  for round thread, in inches

$E_7$  for buttress thread casing, in inches;

$d$  is the pipe inside diameter,  $d = D - 2t$ , in inches;

$N$  is the number of thread turns make-up, as follows:

$A$  for round thread casing and tubing (in accordance with API 5B)

$A + 1.5$  for buttress thread casing smaller than 16

$A + 1$  for buttress thread casing 16 and larger;

$p$  is the thread pitch, as follows:

3.175 mm (0.125 in.) for 8-round thread casing and tubing

2.540 mm (0.100 in.) for 10-round thread tubing

5.080 mm (0.200 in.) for buttress thread casing;

$T_d$  is the taper (on diameter), as follows:

0.0625 mm (0.0625 in./in.) for round thread casing and tubing

0.0625 mm (0.0625 in./in.) for buttress casing smaller than 16

0.0833 mm (0.0833 in./in.) for buttress thread casing 16 and larger;

$W$  is the specified coupling outside diameter, in accordance with **API 5CT**, in inches;

where

$A$  is the hand-tight standoff, in turns;

$E_1$  is the pitch diameter at the hand-tight plane, in accordance with API 5B, in inches;

$E_7$  is the pitch diameter, in accordance with API 5B, in inches;

$D$  is the specified pipe outside diameter, in inches;

$t$  is the specified pipe wall thickness, in inches.

**8.3.3** Subsequent to make-up, internal pressure,  $p_i$ , causes a change in the interface pressure by an amount  $p_2$ :

$$p_2 = p_i d^2(W^2 - E_s^2)/E_s^2(W^2 - d^2), \text{ in psi} \quad (71)$$

where

$E_s$  is the pitch diameter at plane of seal, as follows:

$E_1$  for round thread, in inches

$E_7$  for buttress thread casing, in inches;

$d$  is the pipe inside diameter,  $d = D - 2t$ , in inches

$p_i$  is the internal pressure, in psi;

$W$  is the specified coupling outside diameter, in accordance with **API 5CT**, in inches;

where

$E_1$  is the pitch diameter at the hand-tight plane, in accordance with API 5B, in inches;

$E_7$  is the pitch diameter, in accordance with API 5B, in inches;

$D$  is the specified pipe outside diameter, in inches;

$t$  is the specified pipe wall thickness, in inches.

**8.3.4** Since the external box diameter is always greater than the contact diameter, which in turn is always greater than the internal pipe diameter,  $p_2$  will always be less than  $p_1$ . Therefore, when the total interface pressure  $p_1 + p_2$  equals the internal pressure  $p_i$ , the connection has reached the interface threshold  $p_{is}$ . In other words, if  $p_i$  were greater than  $p_1 + p_2$ , leakage would occur:

$$p_1 + p_2 = p_i = p_{is}, \text{ in psi.} \quad (72)$$

Substituting the appropriate values for  $p_1$  and  $p_2$  into Equation (72) and simplifying produces Equation (69).

## 8.4 Design Equation for Collapse of Pipe Body

### 8.4.1 General

The minimum collapse strength for pipe with no axial force or internal pressure is given by a series of equations, depending on the specified minimum yield strength and cross-sectional dimensions of the tube body.

### 8.4.2 Yield Strength Collapse Pressure Equation

**8.4.2.1** The yield strength collapse pressure is not a true collapse pressure, but rather the external pressure,  $p_{Yp}$ , that generates minimum yield stress,  $f_{ymn}$ , on the inside wall of a tube as calculated by:

$$p_{Yp} = 2f_{ymn} [(D/t) - 1] / [(D/t)^2] \quad (73)$$

where

$D$  is the specified pipe outside diameter;

$f_{ymn}$  is the specified minimum yield strength;

~~$p_{Yp}$  is the pressure for yield strength collapse;~~

$t$  is the specified pipe wall thickness.

**8.4.2.2** Equation (73) for yield strength collapse pressure is applicable for  $D/t$  values up to the value of  $D/t$  corresponding to the intersection with the plastic collapse [Equation (37)]. This intersection is calculated by:

$$(D/t)_{Yp} = \{[(A_c - 2)^2 + 8(B_c + C_c/f_{ymn})]^{1/2} + (A_c - 2)\} / [2(B_c + C_c/f_{ymn})] \quad (74)$$

where

$A_c$  is the empirical constant in historical API collapse equation;

$B_c$  is the empirical constant in historical API collapse equation;

$C_c$  is the empirical constant in historical API collapse equation;

$f_{ymn}$  is the specified minimum yield strength.

The parameters used to calculate collapse pressures depend on the pipe yield strength and on the axial load, as explained in later subsections.

The applicable  $D/t$  ratios for yield strength collapse are shown in Table 5.

**Table 5—Yield Collapse Pressure Equation Range**

<b>Grade <sup>a</sup></b>	<b><i>D/t</i> Range <sup>b</sup></b>
H40	16.40 and less
-50	15.24 and less
J55, K55	14.81 and less
-60	14.44 and less
-70	13.85 and less
C75, E75	13.60 and less
L-N-80	13.38 and less
C90	13.01 and less
R95, T95, X95	12.85 and less
-100	12.70 and less
P105, G105	12.57 and less
P110	12.44 and less
-120	12.21 and less
Q125	12.11 and less
-130	12.02 and less
S135	11.92 and less
-140	11.84 and less
-150	11.67 and less
-155	11.59 and less
-160	11.52 and less
-170	11.37 and less
-180	11.23 and less
<sup>a</sup> Grades indicated without letter designation are not API grades but are grades in use or grades being considered for use and are shown for information purposes. <sup>b</sup> The <i>D/t</i> range values were calculated from Equations (36), (43) or (48), (44) or (49), and (45) or (50).	

### 8.4.3 Plastic Collapse Pressure Equation

**8.4.3.1** The minimum collapse pressure for the plastic range of collapse is calculated by:

$$p_P = f_{ymn} [A_c / (D/t) - B_c] - C_c \quad (75)$$

where

$A_c$  is the empirical constant in historical API collapse equation;

$B_c$  is the empirical constant in historical API collapse equation;

$C_c$  is the empirical constant in historical API collapse equation;

$D$  is the specified pipe outside diameter;

$f_{ymn}$  is the specified minimum yield strength;

~~$p_p$  is the pressure for plastic collapse;~~

$t$  is the specified pipe wall thickness.

**8.4.3.2** The equation for minimum plastic collapse pressure is applicable for  $D/t$  values ranging from  $(D/t)_{yp}$ , Equation (36) for yield strength collapse pressure, to the intersection with Equation (39) for transition collapse pressure  $(D/t)_{pt}$ . Values for  $(D/t)_{pt}$  are calculated by means of:

$$(D/t)_{pt} = [f_{ymn} (A_c - F_c)] / [C_c + f_{ymn} (B_c - G_c)] \quad (76)$$

where

$A_c$  is the empirical constant in historical API collapse equation;

$B_c$  is the empirical constant in historical API collapse equation;

$C_c$  is the empirical constant in historical API collapse equation;

$F_c$  is the empirical constant in historical API collapse equation;

$f_{ymn}$  is the specified minimum yield strength;

$G_c$  is the empirical constant in historical API collapse equation.

The factors and applicable  $D/t$  range for the plastic collapse equation are shown in Table 6.

**Table 6—Equation Factors and  $D/t$  Range for Plastic Collapse**

Grade <sup>a</sup>	$A_c$	$B_c$	$C_c$ psi	$D/t$ Range <sup>b</sup>
H40	2.950	0.0465	754	16.40 to 27.01
–50	2.976	0.0515	1056	15.24 to 25.63
J55, K55	2.991	0.0541	1206	14.81 to 25.01
–60	3.005	0.0566	1356	14.44 to 24.42
–70	3.037	0.0617	1656	13.85 to 23.38
C75, E75	3.054	0.0642	1806	13.60 to 22.91
L-N-80	3.071	0.0667	1955	13.38 to 22.47
C90	3.106	0.0718	2254	13.01 to 21.69
R95, T95, X95	3.124	0.0743	2404	12.85 to 21.33
–100	3.143	0.0768	2553	12.70 to 21.00
P105, G105	3.162	0.0794	2702	12.57 to 20.70
P110	3.181	0.0819	2852	12.44 to 20.41
–120	3.219	0.0870	3151	12.21 to 19.88
Q125	3.239	0.0895	3301	12.11 to 19.63
–130	3.258	0.0920	3451	12.02 to 19.40
S135	3.278	0.0946	3601	11.92 to 19.18
–140	3.297	0.0971	3751	11.84 to 18.97
–150	3.336	0.1021	4053	11.67 to 18.57
–155	3.356	0.1047	4204	11.59 to 18.37
–160	3.375	0.1072	4356	11.52 to 18.19
–170	3.412	0.1123	4660	11.37 to 17.82
–180	3.449	0.1173	4966	11.21 to 17.47
<sup>a</sup> Grades indicated without letter designation are not API grades but are grades in use or grades being considered for use and are shown for information purposes. <sup>b</sup> The $D/t$ range values and equation factors were calculated from Equations (36), (38), (43) or (48), (44) or (49), (45) or (50), (46) or (51), and (47) or (52).				

#### 8.4.4 Transition Collapse Pressure Equation

**8.4.4.1** The minimum collapse pressure for the plastic to elastic transition zone  $p_T$  is calculated by:

$$p_T = f_{ymn} [F_c/(D/t) - G_c] \quad (77)$$

where

$D$  is the specified pipe outside diameter;

$F_c$  is the empirical constant in historical API collapse equation;

$f_{ymn}$  is the specified minimum yield strength;

$G_c$  is the empirical constant in historical API collapse equation;

~~$p_T$  is the pressure for transition collapse;~~

$t$  is the specified pipe wall thickness.

**8.4.4.2** ~~The equation for  $p_T$~~  Equation (77) is applicable for  $D/t$  values from  $(D/t)_{pt}$ , Equation (38) for plastic collapse pressure, to the intersection  $(D/t)_{te}$  with Equation (41) for elastic collapse. Values for  $(D/t)_{te}$  are calculated by

$$(D/t)_{te} = [2 + B_c/A_c]/[3(B_c/A_c)] \quad (78)$$

where

$A_c$  is the empirical constant in historical API collapse equation;

$B_c$  is the empirical constant in historical API collapse equation.

The factors and applicable  $D/t$  range for the transition collapse pressure equation are shown in Table 7.



**Table 7 — Equation Factors and  $D/t$  Range for Transition Collapse**

Grade <sup>a</sup>	$F_c$	$G_c$	$D/t$ Range <sup>b</sup>
H40	2.063	0.0325	27.01 to 42.64
–50	2.003	0.0347	25.63 to 38.83
J55, K55	1.989	0.0360	25.01 to 37.21
–60	1.983	0.0373	24.42 to 35.73
–70	1.984	0.0403	23.38 to 33.17
C75, E75	1.990	0.0418	22.91 to 32.05
L-N-80	1.998	0.0434	22.47 to 31.02
C90	2.017	0.0466	21.69 to 29.18
R95, T95, X95	2.029	0.0482	21.33 to 28.36
–100	2.040	0.0499	21.00 to 27.60
P105, G105	2.053	0.0515	20.70 to 26.89
P110	2.066	0.0532	20.41 to 26.22
–120	2.092	0.0565	19.88 to 25.01
Q125	2.106	0.0582	19.63 to 24.46
–130	2.119	0.0599	19.40 to 23.94
S135	2.133	0.0615	19.18 to 23.44
–140	2.146	0.0632	18.97 to 22.98
–150	2.174	0.0666	18.57 to 22.11
–155	2.188	0.0683	18.37 to 21.70
–160	2.202	0.0700	18.19 to 21.32
–170	2.231	0.0734	17.82 to 20.60
–180	2.261	0.0769	17.47 to 19.93

<sup>a</sup> Grades indicated without letter designation are not API grades but are grades in use or grades being considered for use and are shown for information purposes.

<sup>b</sup> The  $D/t$  range values and equation factors were calculated from Equations (36), (38), (43) or (48), (44) or (49), (45) or (50), (46) or (51), and (47) or (52).

#### 8.4.5 Elastic Collapse Pressure Equation

The minimum collapse pressure for the elastic range of collapse is calculated by:

$$p_E = H C \times 10^6 / [(D/t) (D/t - 1)^2] \quad (79)$$

where

$D$  is the specified pipe outside diameter;

$H C$  is the stress conservation factor,  $3.237 \times 10^5$  MPa ( $46.95 \times 10^6$  psi);

$p_E$ —is the pressure for elastic collapse;

$t$  is the specified pipe wall thickness.

The applicable  $D/t$  range for elastic collapse is shown in Table 8.

**Table 8 —  $D/t$  Range for Elastic Collapse**

Grade <sup>a</sup>	$D/t$ Range <sup>b</sup>
H40	42.64 and greater
–50	38.83 and greater
J55, K55	37.21 and greater
–60	35.73 and greater
–70	33.17 and greater
C75, E75	32.05 and greater
L-N-80	31.02 and greater
C90	29.18 and greater
R95, T95, X95	28.36 and greater
–100	27.60 and greater
P105, G105	26.89 and greater
P110	26.22 and greater
–120	25.01 and greater
Q125	24.46 and greater
–130	23.94 and greater
S135	23.44 and greater
–140	22.98 and greater
–150	22.11 and greater
–155	21.70 and greater
–160	21.32 and greater
–170	20.60 and greater
–180	19.93 and greater

<sup>a</sup> Grades indicated without letter designation are not API grades but are grades in use or grades being considered for use and are shown for information purposes.

<sup>b</sup> The  $D/t$  range values were calculated from Equations (40), (43) or (48), and (44) or (49).

#### 8.4.6 Collapse Pressure under Axial Tension Stress and Internal Pressure

**8.4.6.1** The above formulas for collapse resistance  $p_c$  are valid for zero axial stress and zero internal pressure. The external pressure at collapse under combined axial stress and internal pressure can be

calculated by replacing the collapse resistance  $p_c$  by the collapse pressure differential  $p_c - p_i$  and at the same time modifying the yield stress to a combined loading equivalent grade according to Equation (42) for  $\sigma_a + p_i \geq 0$

$$f_{ycom} = \{[1 - 0.75((\sigma_a + p_i) / f_{ymn})^2]^{1/2} - 0.5 (\sigma_a + p_i) / f_{ymn}\} f_{ymn} \quad (80)$$

**NOTE** Equation (80) is based on the Hencky-von Mises maximum strain energy of distortion theory of yielding.

where

~~$f_{ycom}$  is the combined loading equivalent grade, the equivalent yield strength in the presence of axial stress and internal pressure;~~

$f_{ymn}$  is the specified minimum yield strength;

$\sigma_a$  is the component of axial stress not due to bending;

$p_i$  is the internal pressure.

**8.4.6.2** Collapse resistance equation factors and  $D/t$  ranges for the combined loading equivalent grade are then calculated by means of Equations (36), (38), (40), (43) or (48), (44) or (49), (45) or (50), (46) or (51), and (47) or (52). Using equation factors for the combined loading equivalent grade, collapse resistance under axial stress and internal pressure is calculated by means of Equations (35), (37), (41) and (39).

**8.4.6.3** ~~API collapse resistance equations are not valid for the yield strength of combined loading equivalent grade ( $f_{ycom}$ ) less than 24,000 psi.~~ If the yield strength of combined loading equivalent grade ( $f_{ycom}$ ) is less than 24,000 psi, use the collapse equation determined for  $f_{ycom}$  equal to 24,000 psi, but limited by the value obtained from Equation (73).

## 8.5 Equations for Empirical Constants

The following equations may be used to calculate the empirical constants in the historical API collapse equations. There are two versions of each equation, one each for SI and USC units.

### SI units

$$A_c = 2.8762 + 0.15489 \times 10^{-3} f_{ymn} + 0.44809 \times 10^{-6} f_{ymn}^2 - 0.16211 \times 10^{-9} f_{ymn}^3 \quad (4381)$$

$$B_c = 0.026233 + 0.73402 \times 10^{-4} f_{ymn} \quad (4482)$$

$$C_c = -3.2125 + 0.030867 f_{ymn} - 0.15204 \times 10^{-5} f_{ymn}^2 + 0.778 \times 10^{-9} f_{ymn}^3 \quad (4583)$$

$$F_c = 3.237 \times 10^5 [(3 B_c/A_c)/(2 + B_c/A_c)]^3 / \{f_{ymn} [(3 B_c/A_c)/(2 + B_c/A_c) - B_c/A_c][1 - (3 B_c/A_c)/(2 + B_c/A_c)]^2\} \quad (4684)$$

$$G_c = F_c B_c/A_c \quad (4785)$$

where

$A_c$  is the empirical constant in historical API collapse equation;

$B_c$  is the empirical constant in historical API collapse equation;

$C_c$  is the empirical constant in historical API collapse equation;

$F_c$  is the empirical constant in historical API collapse equation;

$f_{ymn}$  is the specified minimum yield strength;

$G_c$  is the empirical constant in historical API collapse equation.

## USC units

$$A_c = 2.8762 + 0.10679 \times 10^{-5} f_{ymn} + 0.21301 \times 10^{-10} f_{ymn}^2 - 0.53132 \times 10^{-16} f_{ymn}^3 \quad (4886)$$

$$B_c = 0.026233 + 0.50609 \times 10^{-6} f_{ymn} \quad (4987)$$

$$C_c = -465.93 + 0.030867 f_{ymn} - 0.10483 \times 10^{-7} f_{ymn}^2 + 0.36989 \times 10^{-13} f_{ymn}^3 \quad (5088)$$

$$F_c = 46.95 \times 10^6 [(3 B_c/A_c)/(2 + B_c/A_c)]^3 / \{ f_{ymn} [(3 B_c/A_c)/(2 + B_c/A_c) - B_c/A_c] [1 - (3 B_c/A_c)/(2 + B_c/A_c)]^2 \} \quad (5489)$$

$$G_c = F_c B_c/A_c \quad (5290)$$

where

$A_c$  is the empirical constant in historical API collapse equation;

$B_c$  is the empirical constant in historical API collapse equation;

$C_c$  is the empirical constant in historical API collapse equation;

$F_c$  is the empirical constant in historical API collapse equation;

$f_{ymn}$  is the specified minimum yield strength;

$G_c$  is the empirical constant in historical API collapse equation.

## 8.6 Application of Collapse Pressure Equations to Line Pipe

**8.6.1** The collapse pressure equations presented in this clause are empirical relations derived from tests on pipe representative of the casing and tubing inventories listed in API 5CT or ISO 11960. Application of these relations outside the range of yield strengths and  $D/t$  ratios contained in API 5CT or ISO 11960 is not recommended. These equations do not apply to cold expanded pipe because Bauschinger effects significantly reduce collapse resistance.

**8.6.2** Some line pipe grades listed in API 5L have a rough casing equivalent in API 5CT or ISO 11960. However, the API 5L inventory of line pipe contains  $D/t$  ratios that often exceed casing  $D/t$  ratios significantly.

**8.6.3** For line pipe having a yield strength and  $D/t$  falling within the limits of the sizes and thicknesses listed in API 5CT or ISO 11960, application of the equations in this clause should yield reasonable estimates of minimum collapse pressure. Nevertheless, as with the application of any of the equations in this document, sound engineering judgment should prevail.

## 8.7 Example Calculations

Calculate the collapse resistance  $p_c$  of size Label 1: 7, mass Label 2: 26, grade P110 casing with axial stress of 11,000 psi and internal pressure 1000 psi. Wall thickness is 0.362 in.

The following table presents the results of the calculation in both SI and USC units.

**Table 9 — Example Calculation — Collapse Resistance with Internal Pressure and Tension**

Term	SI		USC	
	Value	Units	Value	Units
Load				
$\sigma_a$	75.842	MPa	11,000	psi
$p_i$	6.895	MPa	1000	psi
Geometry				
$D$	177.80	mm	7.000	in.
$t$	9.19	mm	0.362	in.
$D/t$	19,35			
Material				
$f_{ymn}$	758	MPa	110,000	psi
Calculations				
$f_{ycom}$	713.7	MPa	103,508	psi
$A_c$	3.156			
$B_c$	0.0786			
$C_c$	18.33	MPa	2658	psi
$F_c$	2.050			
$G_c$	0.0511			
Equation	Plastic			
$p_c - p_i$	42.1	MPa	6100	psi

## 9 Calculated Masses

### 9.1 General

The density of martensitic chromium steel (L80 Type 13Cr) is different than carbon steel. A mass correction factor of 0.989 may be used.

NOTE The dimensional symbols and corresponding numerical values used in the equations for calculation of masses in Section 11 are given in API 5B and [API 5CT](#).

## 9.2 Nominal Linear Masses

**9.2.1** Nominal linear mass is used in connection with pipe having end finish such as threads and couplings, upset and threaded ends, upset ends, and so forth, primarily for the purpose of identification in ordering. It is also used generally in the design of casing and tubing strings as the basis for determining joint safety factors in tension.

**9.2.2** Nominal linear mass is approximately equal to the calculated theoretical mass per foot of a 6.10 m (20 ft) length of threaded and coupled tubing and a 9.14 m (30 ft) length of threaded and coupled casing, based on the dimensions of the connection in use for the class of product when the particular diameter and wall thickness were introduced. Some nominal linear masses are based on sharp thread connections that were in use before this specification was adopted. The same nominal linear masses are used for short thread connections, long thread connections, buttress thread connections, and the various proprietary connections offered to the oil industry. For calculation of nominal linear masses not listed in the casing or tubing list tables of API 5CT, the length of a piece of pipe is taken as 9.14 m (30 ft) for casing and 6.10 m (20 ft) for tubing. Nominal linear masses for upset drill pipe for weld-on tool joints are based on the calculated mass-per-foot values of the original threaded and coupled drill pipe.

**9.2.3** In determining the nominal linear mass from calculated linear masses, it would appear that historical rounding was implemented with no definite procedure. Rounding increments of 0.01, 0.05, 0.1, and 0.5 should be used for adding isolated new nominal linear masses not listed in the casing or tubing list tables of API 5CT, selecting the increment most compatible with adjacent nominal linear masses.

## 9.3 Calculated Plain-end Mass

Plain-end mass per unit length for API 5CT is calculated by:

$$w_{pe} = k_m k_{wpe} (D - t) t \quad (7491)$$

where

$D$  is the specified pipe outside diameter, in millimeters or inches;

$k_m$  is the mass correction factor, as follows:  
1.000 for carbon steel  
0.989 for martensitic chromium steel;

$k_{wpe}$  is the mass per unit length conversion factor, equal to:  
0.0246615 for SI units  
10.69 for USC units;

$t$  is the specified pipe wall thickness, in millimeters or inches;

$w_{pe}$  is the plain-end mass per unit length, in kilograms per meter or pounds per foot.

## 9.4 Calculated Threaded and Coupled Mass

### 9.4.1 General

The calculated threaded and coupled mass per unit length is based on a length measured from the outer face of the coupling to the end of the pipe, as shown in Figure 2, and is calculated from Equation (7492). For calculation of casing nominal linear masses not listed in the casing or tubing list tables of API 5CT, the buttress connection shall be used.

$$w_{tc} = \{[L_j - k_{isl} (N_L + 2J)/2] w_{pe} + \text{mass of coupling} - \text{mass removed in threading two pipe ends}\} / L_j \quad (7492)$$

NOTE The mill end of the coupling is assumed to be installed to the power-tight axial position.

where

$k_{isl}$  is the length conversion factor, equal to:

0.001 for SI units

1/12 for USC units;

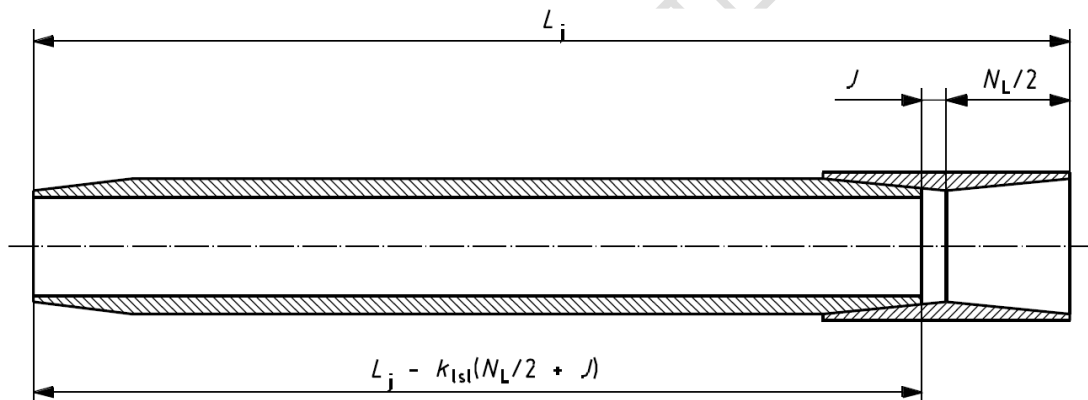
$J$  is the distance from end of pipe to center of coupling in power-tight position, in accordance with API 5B, in millimeters or inches;

$L_j$  is the length of a standard piece of pipe, 9.14 m (30 ft) for casing and 6.10 m (20 ft) for tubing;

$N_L$  is the coupling length, in accordance with API 5CT, in millimeters or inches;

~~$w_{tc}$  is the threaded and coupled mass per unit length;~~

$w_{pe}$  is the plain-end mass per unit length, in kilograms per meter or pounds per foot.



#### Key

$L_j$  length of standard piece of pipe, in meters or feet

$N_L$  coupling length, in accordance with API 5CT, in millimeters or inches

$J$  distance from end of pipe to center of coupling in power-tight position, in accordance with API 5B

$k_{isl}$  length conversion factor, equal to 0.001 for SI units and 1/12 for USC units

**Figure 2—Threaded and Coupled Pipe**

#### 9.4.2 Direct Calculation of $e_m$ for Threaded and Coupled Pipe

The end finishing mass gained in kilograms (pounds) is calculated by:

$$e_m = -k_{isl} (N_L/2 + J) w_{pe} + \text{mass of coupling} - \text{mass removed in threading two pipe ends} \quad (7593)$$

where

~~$e_m$  is the mass gain due to end finishing, in kilograms or pounds;~~

$k_{\text{SI}}$  is the length conversion factor, equal to:  
0.001 for SI units  
1/12 for USC units;

$J$  is the distance from end of pipe to center of coupling in power-tight position, in accordance with API 5B, in millimeters or inches;

$N_L$  is the coupling length, in accordance with API 5CT, in millimeters or inches;

$w_{\text{pe}}$  is the plain-end mass per unit length, in kilograms per meter or pounds per foot;

and

mass of coupling is determined according to the appropriate part of 11.9.7 below;

mass removed in threading is determined according to the appropriate part of 11.9.8 below.

## 9.5 Calculated Upset and Threaded Mass for Integral Joint Tubing

### 9.5.1 General

The calculated upset and threaded mass is based on a standard length as shown in Figure 3 and is calculated by Equation (94).



#### Key

$L_j$  length of standard piece of pipe, in meters or feet

Figure 3—Upset Pipe

The upset and threaded mass per unit length is calculated by Equation (94), in kilograms per meter (pounds per foot).

$$w_{\text{ij}} = w_{\text{pe}} + (\text{mass of upsets} - \text{mass removed in threading two ends})/L_j \quad (94)$$

where

$L_j$  length of a standard piece of pipe, in meters or feet;

$w_{\text{ij}}$  upset and threaded mass per unit length, in kilograms per meter or pounds per foot;

$w_{\text{pe}}$  plain-end mass per unit length, in kilograms per meter or pounds per foot.



## 9.5.2 Direct Calculation of $e_m$ for Upset and Threaded Pipe

The end finishing mass gained is calculated by [Equation \(95\)](#):

$$e_m = \text{mass of upsets} - \text{mass removed threading two pipe ends} \quad (7795)$$

where

mass of upsets is determined according to the appropriate part of [449.9 below](#);

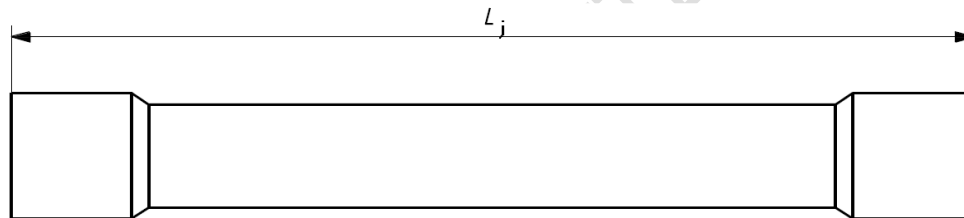
mass removed in threading is determined according to the appropriate part of [449.8 below](#).

## 9.6 Calculated Upset Mass

### 9.6.1 General

Calculated upset mass of upset drill pipe for weld-on tool joints is necessary for determination of  $e_m$ , the mass gain due to end finishing by upsetting, and is calculated by Equation (78).

The calculated upset mass,  $w_u$ , is based on a 6.10 m (20 ft) length measured end to end, including the upsets as shown in Figure 4.



#### Key

$L_j$  length of standard piece of pipe, in meters or feet

**Figure 4—Upset Pipe—Both Ends**

[The upset mass per unit length is calculated by Equation \(96\), in kilograms per meter \(pounds per foot\).](#)

$$w_u = w_{pe} + (\text{mass of upsets})/L_j \quad (7896)$$

where

$L_j$  is the length of a standard piece of pipe, in meters or feet;

$w_{pe}$  is the plain-end mass per unit length, in kilograms per meter or pounds per foot;

~~$w_u$  is the upset mass per unit length, in kilograms per meter or pounds per foot.~~

## 9.6.2 Direct Calculation of $e_m$ for Upset Pipe

The end finishing mass gained is calculated by:

$$e_m = \text{mass of upsets} \quad (7997)$$

where

mass of upsets is determined according to the appropriate part of 449.9 below.

## 9.7 Calculated Coupling Mass

### 9.7.1 General

Coupling masses are calculated as shown in 449.7.2 for line pipe and round thread casing and tubing, and in 11.8.3 for buttress thread casing.

### 9.7.2 Calculated Coupling Mass for Line Pipe and Round Thread Casing and Tubing

#### 9.7.2.1 General

**9.7.2.1.1** Coupling masses for line pipe are calculated on the basis of the dimensions shown in the 1942 edition of API 5L (8<sup>th</sup> Edition), which are identical with those shown in the 1971 edition of API 5L (26<sup>th</sup> Edition).

**9.7.2.1.2** Coupling masses for round thread casing are calculated on the basis of the dimensions shown in the 1942 standards except for Label 1: 18-5/8 short and Label 1: 20 long round thread casing, which are based on hand-tight dimensions identical with the 1971 standard values.

**9.7.2.1.3** Coupling masses shown for Label 1: 18 5/8 long round threads and for Label 1: 16 round threads are based on the old sharp thread form and dimensions. The hand-tight standoff values in the 1971 standards were made one thread turn larger than those in the 1942 standards. Recalculation on the basis of the 1971 hand-tight dimensions would result in slightly different coupling masses.

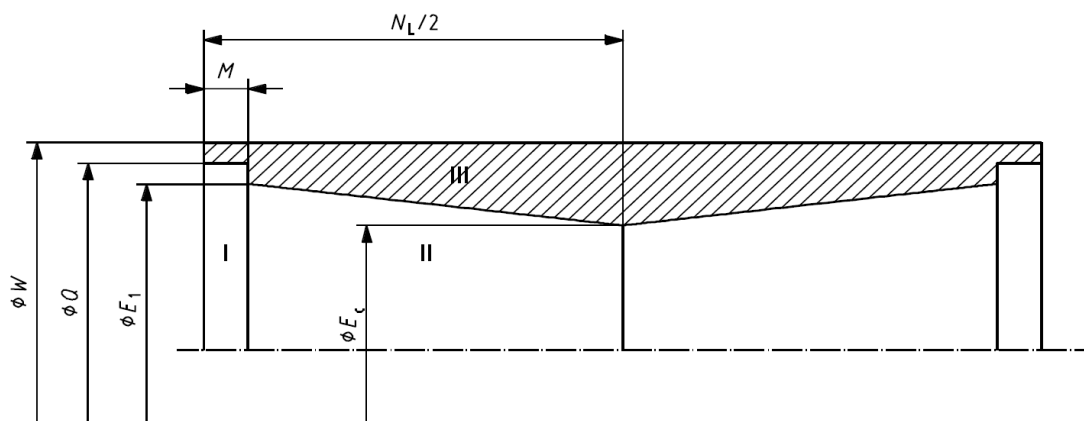
**9.7.2.1.4** Nonupset tubing coupling masses are based on 1942 coupling dimensions, except for the Label 1: 1.050, Label 1: 1.315, and Label 1: 1.660 sizes, which were based on coupling dimensions added in 1962 (19<sup>th</sup> Edition of API 5L). The 1971 dimensions are identical with those from which the present coupling masses were calculated.

**9.7.2.1.5** External upset tubing coupling masses are based on 1942 coupling dimensions except for the Label 1: 1.050 and Label 1: 1.315 sizes, which were based on coupling dimensions added in 1954 (13<sup>th</sup> Edition of API 5L). For regular diameter couplings, the dimensions used in calculating masses are identical with those in the 1971 standards. The special clearance coupling masses are based on the diameters introduced in the 1958 (17<sup>th</sup> Edition of API 5L) standards, which are identical with those in the 1971 standards. In calculating the masses of the special clearance couplings, an allowance is made for the mass removed by the special bevel. However, the masses were calculated several years before special clearance couplings were introduced into the standards in 1958 on the basis of a 12° degree bevel rather than the 20° bevel introduced in the 1962 standard. The masses were not recalculated for the change in bevel dimensions adopted for the special bevel in 1962. The special bevel is also available on regular diameter couplings, but separate listings of masses for these couplings are shown in the standards.

~~Masses for line pipe couplings and round thread casing and tubing couplings are calculated from Equations (81) to (89), with reference to Figures 5 and 6.~~

### 9.7.2.2 Couplings without Special Bevel Mass Allowance

Masses for line pipe couplings and round thread casing and tubing couplings are calculated from Equations (99) to (107), with reference to Figures 5 and 6.



#### Key

- $N_L$  coupling length, in accordance with API 5CT, in millimeters or inches  
 $M$  length from the face of the coupling to the hand-tight plane for line pipe and for round thread casing and tubing, in accordance with API 5B  
 $W$  specified coupling outside diameter, in accordance with API 5CT  
 $Q$  diameter of coupling recess, in accordance with API 5B  
 $E_1$  pitch diameter at the hand-tight plane, in accordance with API 5B  
 $E_c$  pitch diameter, at center of coupling  
 I, II, III represent Volumes I, II, III, respectively [see Equations (82100), (83101), and (85103)]

**Figure 5—Pipe Coupling**

$$m_c = 0.5672 k_m (\text{Vol III}) \quad (8098)$$

$$E_c = E_1 - (N_L/2 - M) T_d \quad (8499)$$

$$\text{Vol I} = 0.7854 M Q^2 \quad (82100)$$

$$\text{Vol II} = 0.2618 (N_L/2 - M) (E_1^2 + E_1 E_c + E_c^2) \quad (83101)$$

$$\text{Vol (I + II + III)} = 0.7854 N_L W^2/2 \quad (84102)$$

$$\text{Vol III} = \text{Vol (I + II + III)} - \text{Vol I} - \text{Vol II} \quad (85103)$$

where

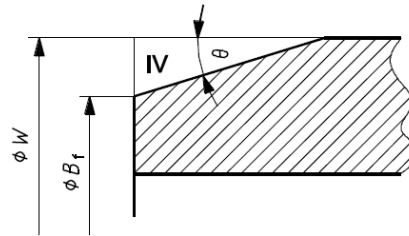
$k_m$  is the mass correction factor, as follows:  
 1.000 for carbon steel  
 0.989 for martensitic chromium steel;

$m_c$  is the coupling mass;

$T_d$  is the taper, 0.0625.

Calculations for coupling masses are expressed in pounds. The final calculated mass is rounded to two decimals with no intermediate rounding in the calculations.

#### 9.7.2.3.1.1.1 Coupling Mass Removed by Special Bevel



#### Key

$W$  specified coupling outside diameter, in accordance with API 5CT

$B_f$  maximum bearing face diameter, in accordance with dimension  $b$  in API 5CT

$\theta$  angle of bevel

IV represents Volume IV [see Equation (87)]

Figure 6—Coupling with Special Bevel

#### 9.7.2.3 Coupling Mass Removed by Special Bevel

Equation (86104), which is used to calculate the mass allowance for the special bevel on special clearance couplings for external upset tubing, is approximate. The exact equation for Vol IV is shown as Equation (87105).

$$\text{Vol IV} = 0.7854 (W - B_f) (W^2 - B_f^2) / 2 \tan \theta \quad (86104)$$

$$\text{Vol IV} = (W - B_f) [0.785W^2 - 0.2618 (B_f^2 + B_f W + W^2)] / \tan \theta \quad (87105)$$

The coupling mass removed by special bevel,  $m_{\text{crsb}}$ , is calculated as follows:

$$m_{\text{crsb}} = 0.5672 k_m (\text{Vol IV}) \quad (88106)$$

where

$k_m$  is the mass correction factor, as follows:

1.000 for carbon steel

0.989 for martensitic chromium steel.

#### 9.7.2.4 Coupling Mass with Special Bevel

The mass of a coupling with special bevel is calculated by subtracting the coupling mass removed by the special bevel, Equation (88106)—above, from the mass of the coupling without a special bevel, Equation (8998). Calculations for coupling masses are in pounds. The final calculated mass is rounded to two decimals with no intermediate rounding in the calculations.

$$m_{\text{csb}} = m_c - m_{\text{crsb}} \quad (89107)$$

where

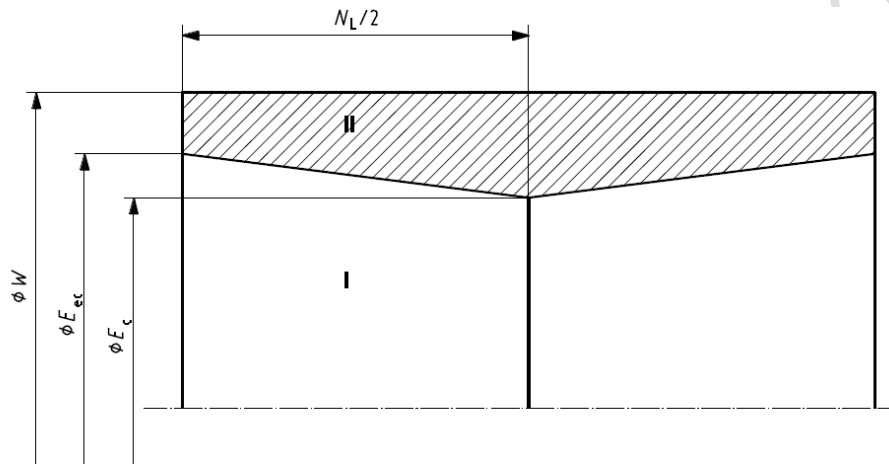
$m_c$  is the coupling mass;

$m_{crsb}$  is the coupling mass removed by special bevel;

$m_{csb}$  is the coupling mass with special bevel.

### 9.7.3 Calculated Coupling Mass for Buttress Thread Casing

Coupling masses for buttress thread casing are calculated by Equations (90108) to (95), with reference to Figure 7.



#### Key

$N_L$  coupling length, in accordance with API 5CT, in millimeters or inches

$W$  specified coupling outside diameter, in accordance with API 5CT

$E_c$  pitch diameter, at center of coupling

$E_{ec}$  pitch diameter, at end of coupling

I, II represent Volumes I, II, respectively [see Equations (92110) and (94112)]

**Figure 7—Mass Calculations for Buttress Thread Couplings**

The pitch diameter at the center and end of the coupling are calculated by Equations (108) and (109) respectively.

$$E_c = E_7 - (L_7 + J) T_d \quad (90108)$$

$$E_{ec} = E_7 + (g + X) T_d \quad (94109)$$

where

$E_7$  is the pitch diameter, in accordance with API 5B;

$g$  is the length of imperfect threads, in accordance with API 5B;

$J$  is the distance from end of pipe to center of coupling in power-tight position, in accordance with API 5B;

$L_7$  is the length of perfect threads, in accordance with API 5B;

$X$  is 0.300 for sizes less than Label 1: 16  
is 0.200 for sizes Label 1: 16 and larger;

$T_d$  is the taper, as follows:  
0.0625 for sizes less than Label 1: 16  
0.0833 for sizes Label 1: 16 and larger.

$$\text{Vol I} = 0.2618 (N_L/2)(E_{ec}^2 + E_{ec}E_c + E_c^2) \quad (92110)$$

$$\text{Vol (I + II)} = 0.7854 (N_L/2)W^2 \quad (93111)$$

$$\text{Vol II} = \text{Vol (I + II)} - \text{Vol I} \quad (94112)$$

The coupling mass of buttress thread casing,  $m_{cB}$ , is calculated as follows:

$$m_{cB} = 0.5672 k_m (\text{Vol II}) \quad (95113)$$

where

$k_m$  is the mass correction factor, as follows:  
1.000 for carbon steel  
0.989 for martensitic chromium steel.

Calculations for coupling masses are in pounds. The final calculated mass is rounded to two decimals with no intermediate rounding in the calculations.

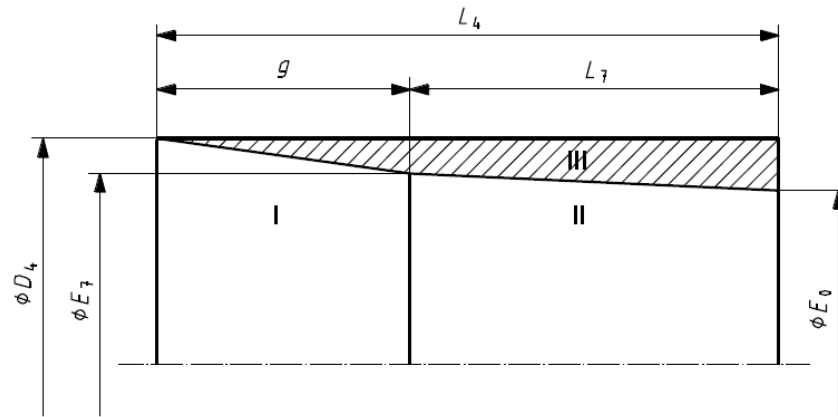
## 9.8 Calculated Mass Removed during Threading

### 9.8.1 General

The mass removed in threading pipe or pin ends is calculated in accordance with 419.8.2.

### 9.8.2 Calculated Mass Removed during Threading Pipe or Pin Ends

The mass removed by threading pipe or pin ends is calculated from Equations (97115) to (106) with reference to Figures 8 and 9.



### Key

$L_4$  pin thread length, in accordance with API 5B, in millimeters or inches

$D_4$  upset outside diameter of upset pipe and pipe outside diameter of nonupset pipe and buttress thread casing, in accordance with API 5B

$g$  length of imperfect threads, in accordance with API 5B

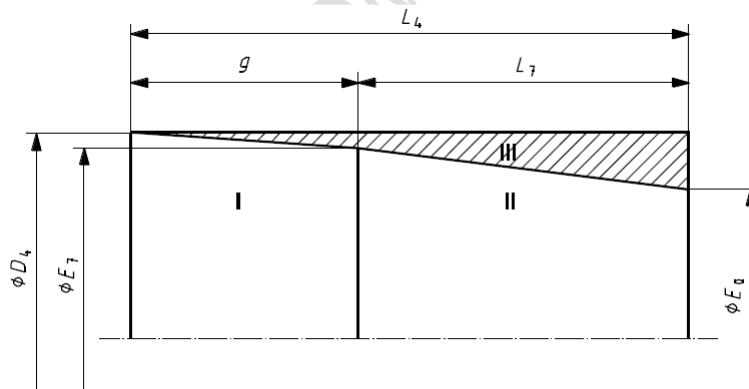
$L_7$  length of perfect threads, in accordance with API 5B

$E_7$  pitch diameter, in accordance with API 5B

$E_0$  pitch diameter, at end of pipe

I, II, III represent Volumes I, II, III, respectively [see Equations (97), (98), and (100)]

**Figure 8—Round Threads and Line Pipe Threads**



### Key

$L_4$  pin thread length, in accordance with API 5B, in millimeters or inches

$D_4$  upset outside diameter of upset pipe and pipe outside diameter of nonupset pipe and buttress thread casing, in accordance with API 5B

$g$  length of imperfect threads, in accordance with API 5B

$L_7$  length of perfect threads, in accordance with API 5B

$E_7$  pitch diameter, in accordance with API 5B

$E_0$  pitch diameter, at end of pipe

I, II, III represent Volumes I, II, III, respectively [see Equations (97), (98), and (100)]

**Figure 9—Buttress Threads**

The pitch diameter at the end of the pipe is calculated by Equation (114).

$$E_0 = E_7 - L_7 T_d \quad (96114)$$

where

$T_d$  is 0.0625 for all round threads and for buttress threads in size less than Label 1: 16  
is 0.0833 for buttress threads in sizes Label 1: 16 and larger.

For Figures 8 and 9:

$$\text{Vol I} = 0.2618g (D_4^2 + D_4E_7 + E_7^2) \quad (97115)$$

$$\text{Vol II} = 0.2618 (L_4 - g) (E_7^2 + E_7E_0 + E_0^2) \quad (98116)$$

$$\text{Vol (I + II + III)} = 0.7854L_4D_4^2 \quad (99117)$$

$$\text{Vol III} = \text{Vol (I + II + III)} - \text{Vol I} - \text{Vol II} \quad (100118)$$

The mass removed by threading,  $m_{rt}$ , is calculated as follows:

$$m_{rt} = 0.2836k_m (\text{Vol III}) \quad (101119)$$

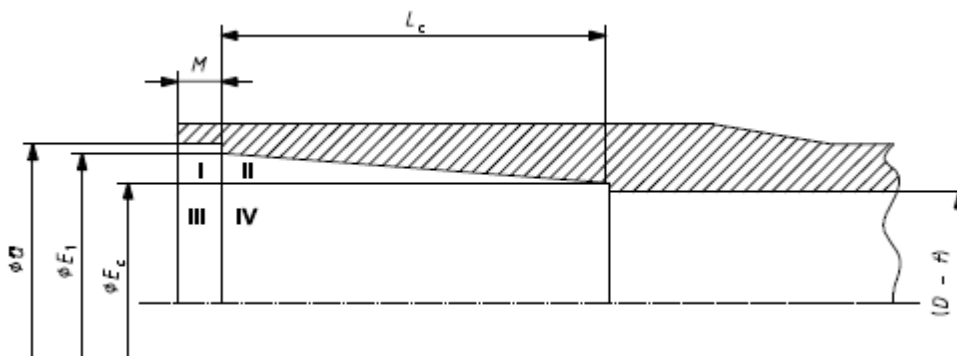
where

$k_m$  is the mass correction factor, as follows:  
1.000 for carbon steel  
0.989 for martensitic chromium steel.



### 9.8.3 Calculated Mass Removed during Threading Integral Joint Tubing Box Ends

The mass removed by threading and recessing the box ends of integral joint tubing is calculated from Equations (402120) to (408126), with reference to Figure 10.



#### Key

- $M$  length from the face of the coupling to the hand-tight plane for line pipe and for round thread casing and tubing, in accordance with API 5B
- $L_c$  minimum length of full crest threads from end of pipe, in accordance with API 5B
- $Q$  diameter of coupling recess, in accordance with API 5B
- $E_1$  pitch diameter at the hand-tight plane, in accordance with API 5B
- $E_c$  pitch diameter, at center of coupling
- $D$  specified pipe outside diameter, in accordance with API 5CT
- $t$  specified pipe wall thickness, in accordance with API 5CT
- I, II, III, IV represent Volumes I, II, III, IV, respectively [see Equations (404122) to (407125)]

Figure 10—Integral Joint Tubing

$$L_c = L_1 + J + A \quad (402120)$$

$$E_c = E_1 - L_c T_d \quad (403121)$$

where

$J$  is the end of pipe to thread run-out in box power-tight

$$\text{Vol (I + III)} = 0.7854MQ^2 \quad (404122)$$

$$\text{Vol (II + IV)} = 0.2618L_c (E_1^2 + E_1E_c + E_1^2) \quad (405123)$$

$$\text{Vol (III + IV)} = 0.7854 (M + L_c)(D - t)^2 \quad (406124)$$

$$\text{Vol (I + II)} = \text{Vol (I + III)} + \text{Vol (II + IV)} - \text{Vol (III + IV)} \quad (407125)$$

The integral joint mass removed by threading and recessing,  $m_{irt}$ , is calculated as follows:

$$m_{irt} = 0.2836k_m [\text{Vol (I + II)}] \quad (408126)$$

where

$k_m$  is the mass correction factor, as follows:

1.000 for carbon steel

0.989 for martensitic chromium steel.

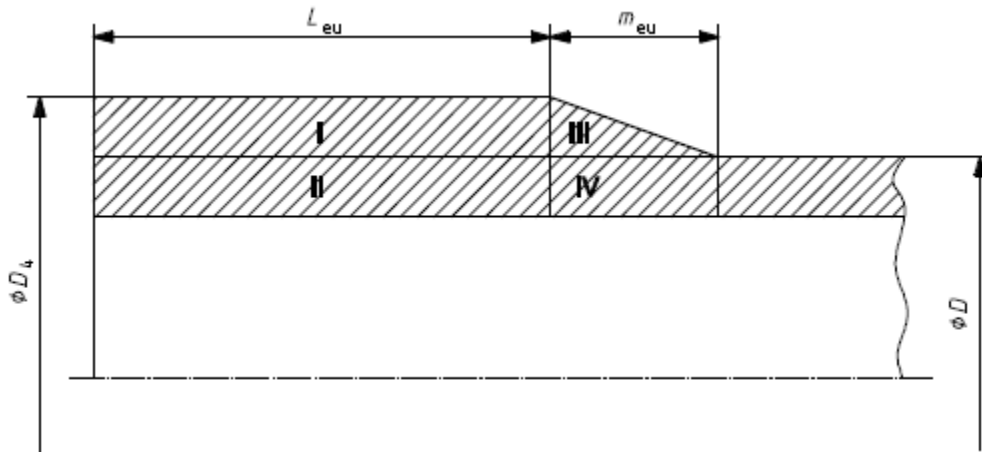
## 9.9 Calculated Mass of Upsets

### 9.9.1 General

The mass added when upsetting pipe or pin ends is calculated in accordance with 11.9.2 to 11.9.4.

### 9.9.2 Calculated Mass of External Upsets

The mass added by an external upset is calculated by Equations (409127) to (443131), with reference to Figure 11.



#### Key

$D_4$  major diameter, in accordance with API 5B

$D$  specified pipe outside diameter

$L_{eu}$  length from end of pipe to start of taper, in accordance with API 5CT

$m_{eu}$  length of box upset taper, in accordance with API 5CT

I, II, III, IV represent Volumes I, II, III, IV, respectively [see Equations (409127) to (442130)]

Figure 11—External Upset

$$\text{Vol (I + II)} = 0.7854 L_{eu} D_4^2 \quad (409127)$$

$$\text{Vol (III + IV)} = 0.2618 m_{eu} (D_4^2 + D_4 D + D^2) \quad (440128)$$

$$\text{Vol (II + IV)} = 0.7854 (L_{eu} + m_{eu}) D^2 \quad (444129)$$

$$\text{Vol (I + III)} = \text{Vol (I + II)} + \text{Vol (III + IV)} - \text{Vol (II + IV)} \quad (442130)$$

The external upset mass,  $m_{exu}$ , is calculated as follows:

$$m_{exu} = 0.2836 k_m [\text{Vol (I + III)}] \quad (443131)$$

where

$k_m$  is the mass correction factor, as follows:

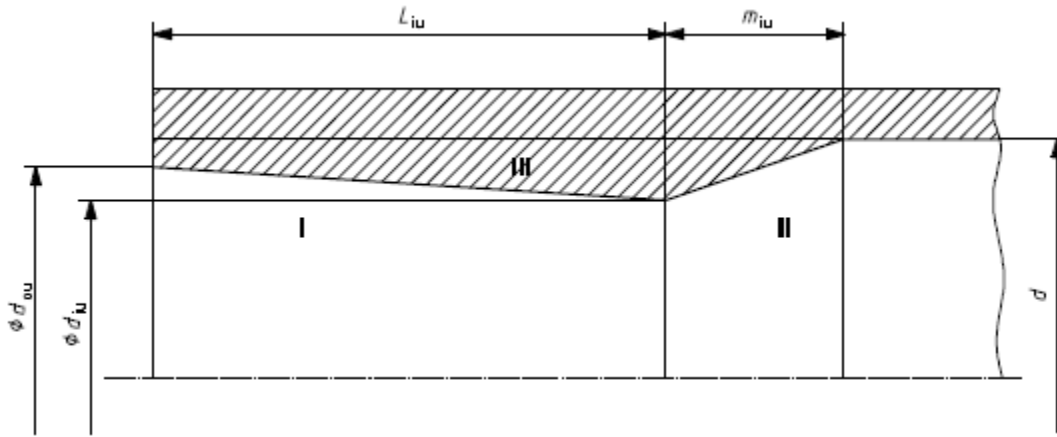
1.000 for carbon steel

0.989 for martensitic chromium steel.

NOTE Calculations for the mass of an external upset are expressed in pounds and carried to four decimals.

### 9.9.3 Calculated Mass of Internal Upsets

The mass added by an internal upset is calculated by Equations (414.132) to (418.136), with reference to Figure 12.



#### Key

$d$  pipe inside diameter

$d_{ou}$  inside diameter at end of upset pipe

$L_{iu}$  length of pin upset, in accordance with API 5CT

$d_{iu}$  inside diameter of pin upset, in accordance with API 5CT

$m_{iu}$  length of pin upset taper, in accordance with API 5CT

I, II, III represent Volumes I, II, III, respectively [see Equations (414.132) to (417.135)]

Figure 12—Internal Upset

$$\text{Vol I} = 0.2618L_{iu}(d_{ou}^2 + d_{ou}d_{iu} + d_{iu}^2) \quad (414.132)$$

$$\text{Vol II} = 0.2618m_{iu}(d^2 + dd_{iu} + d_{iu}^2) \quad (415.133)$$

$$\text{Vol (I + II + III)} = 0.7854d^2(L_{iu} + m_{iu}) \quad (416.134)$$

$$\text{Vol III} = \text{Vol (I + II + III)} - \text{Vol I} - \text{Vol II} \quad (417.135)$$

The internal upset mass,  $m_{inu}$ , is calculated as follows:

$$m_{inu} = 0.2836k_m (\text{Vol III}) \quad (418.136)$$

where

$k_m$  is the mass correction factor, as follows:

1.000 for carbon steel

0.989 for martensitic chromium steel.

NOTE Calculations for the mass of an internal upset are expressed in pounds and carried to four decimals.

#### 9.9.4 Calculated Mass of External–Internal Upsets

The mass added by an external–internal upset is calculated as the sum of the mass of an external upset calculated from Equation (443131), and the mass of an internal upset calculated from Equation (448136).

The external–internal upset mass,  $m_{eiu}$ , is calculated as follows:

$$m_{eiu} = m_{inu} + m_{exu} \quad (449137)$$

where

$m_{exu}$  is the external upset mass;

$m_{inu}$  is the internal upset mass.

NOTE Calculations for the mass of an external–internal upset are expressed in pounds and carried to four decimals.

## 10 Elongation

The minimum elongation over 50.8 mm (2 in.) is calculated from:

$$\varepsilon_{el} = k_{el} A_s^{0.2} / f_{umn}^{0.9} \quad (429138)$$

where

$A_s$  is the cross-sectional area of the tensile test specimen, in square millimeters (square inches), based on specified outside diameter or nominal specimen width and specified wall thickness, rounded to the nearest 10 mm<sup>2</sup> (0.01 in.<sup>2</sup>) or 490 mm<sup>2</sup> (0.75 in.<sup>2</sup>), whichever is smaller;

$\varepsilon_{el}$  is the minimum gauge length extension in 50.8 mm (2.0 in.), expressed in percent rounded to the nearest 0.5 % below 10 % and to the nearest unit percent for 10 % and larger;

$f_{umn}$  is the specified minimum tensile strength, in MPa (psi);

$k_{el}$  is the elongation constant, equal to:

1942.57 for SI units

625,000 for USC units.

## 11 Flattening Tests

### 11.1 Flattening Tests for Casing and Tubing

The distance between plates for flattening tests for welded casing and tubing is calculated from the equations shown in Table 10.

**Table 10—Casing and Tubing Flattening Tests: Distance between Plates**

<sup>(1)</sup> Grade	<sup>(2)</sup> $D/t$ Ratio	<sup>(3)</sup> Maximum Distance between Plates in.
H40	16 and over	$0.5D$
	Less than 16	$D(0.830 - 0.0206 D/t)$
J55 and K55	16 and over	$0.65D$
	3.93 to 16	$D(0.980 - 0.0206 D/t)$
	Less than 3.93	$D(1.104 - 0.0518 D/t)$
N80 <sup>a</sup>	9 to 28	$D(1.074 - 0.0174 D/t)$
L80	9 to 28	$D(1.074 - 0.0194 D/t)$
R95 <sup>a</sup>	9 to 28	$D(1.080 - 0.0178 D/t)$
Q125 <sup>b</sup>	All	$D(1.092 - 0.0140 D/t)$
$D$ is the specified pipe outside diameter, in millimeters or inches; $t$ is the specified pipe wall thickness, in millimeters or inches.		
<sup>a</sup> If the flattening test of R95 or N80 fails at 12 or 6 o'clock, the flattening should continue until the remaining portion of the specimen fails at the 3 or 9 o'clock position; premature failure at the 12 or 6 o'clock position should not be considered basis for rejection. <sup>b</sup> See API 5CT; flattening should be a minimum of $0.85D$ .		

In Table 10:

$D$  is the specified pipe outside diameter, in millimeters or inches;

$t$  is the specified pipe wall thickness, in millimeters or inches.

### 11.2 Flattening Tests for Line Pipe

The maximum distance between plates for flattening tests for line pipe is calculated from the equations shown in Table 11.

**Table 11—Line Pipe Flattening Tests: Distance between Plates**

<sup>(1)</sup> Grade	<sup>(3)</sup> Maximum Distance between Plates in.
Less than X52	$3.07t/(0.07 + 3t/D)$
X52 and higher	$3.05t/(0.05 + 3t/D)$
<del><i>D is the specified pipe outside diameter, in millimeters or inches;</i></del> <del><i>t is the specified pipe wall thickness, in millimeters or inches.</i></del>	

In Table 11:

~~*D is the specified pipe outside diameter, in millimeters or inches;*~~

~~*t is the specified pipe wall thickness, in millimeters or inches.*~~

Flattening tests for line pipe are addressed in API 5L. **Hydrostatic Test Pressures**

## 12 **Hydrostatic Test Pressures**

### 12.1 Hydrostatic Test Pressures for Plain-end Pipe and Integral Joint Tubing

**12.1.13.1** The hydrostatic test pressures for plain-end pipe and integral-joint tubing are calculated according to the following equation, except for Grade A25 line pipe, Grade A and Grade B line pipe in sizes less than Label 1: 2-3/8, and threaded and coupled line pipe in sizes Label 1: 6-5/8 and less, which were determined arbitrarily.

**12.1.13.2** The hydrostatic test pressure,  $p_{ht}$ , expressed in kilopascals or pounds per square inch, is calculated as follows:

$$p_{ht} = 2\sigma_f t/D \quad (12.1.13.2)$$

where

$D$  is the specified pipe outside diameter, in millimeters or inches;

$t$  is the specified pipe wall thickness, in millimeters or inches;

$\sigma_f$  is the fiber stress corresponding to the percent of specified yield strength as given in Table 12, in kilopascals or pounds per square inch.

**12.1.13.3** Alternative test pressures should be used when specified on the purchase agreement and when agreed by the purchaser and manufacturer in accordance with **API 5CT**.

### 12.2 Hydrostatic Test Pressure for Threaded and Coupled Pipe

**12.2.24.1** The hydrostatic test pressure for threaded and coupled pipe is the same as for plain-end pipe, except where a lower pressure is **necessary** to avoid leakage due to insufficient internal yield pressure of the coupling or insufficient interface pressure at the  $E_1$  or  $E_7$  plane as calculated in Section 10.

**124.24.2** The test pressure should be based on the lowest of the test pressure determined for plain-end pipe in 14.1, or 80 % of the internal coupling yield pressure result from Equation (66) in 10.2, or the interface pressure result from Equation (69) in 10.3.

**Table 12—Factors for Test Pressure Equations**

(1) Grade	(2) Label 1	Fiber Stress as Percent of Specified Minimum Yield Strength		Maximum Test Pressure <sup>a</sup> kPa (psi)	
		(3) Standard Test Pressures	(4) Alternative Test Pressures	(5) Standard	(6) Alternative
A and B	2-3/8 – 3-1/2	60	75	17,240 (2500)	17,240 (2500)
A and B	Over 3-1/2	60	75	19,300 (2800)	19,300 (2800)
X Grades	4-1/2 and under	60	75	20,680 (3000)	20,680 (3000)
X Grades	5-9/16	75	b	20,680 (3000)	b
X Grades	6-5/8 and 8-5/8	75	b	20,680 (3000)	b
X Grades	10-3/4 – 18	85	b	20,680 (3000)	b
X Grades	20 and larger	90	b	20,680 (3000)	b
H40, J55, and K55	9-5/8 and under	80	80	20,680 (3000)	68,950 (10,000)
H40, J55, and K55	10-3/4 and larger	60	80	20,680 (3000)	68,950 (10,000)
M65	All sizes	60	80	20,680 (3000)	68,950 (10,000)
L80 and N80	All sizes	80	b	68,950 <sup>c</sup> (10,000)	b
C90	All sizes	80	b	68,950 <sup>c</sup> (10,000)	b
R95	All sizes	80	b	68,950 <sup>c</sup> (10,000)	b
T95	All sizes	80	b	68,950 <sup>c</sup> (10,000)	b
P110	All sizes	80	80	68,950 <sup>c</sup> (10,000)	d
Q125	All sizes	80	80	68,950 <sup>c</sup> (10,000)	d

<sup>a</sup> Higher test pressures are permissible by agreement between purchaser and manufacturer.  
<sup>b</sup> No alternative test pressure.  
<sup>c</sup> Plain-end pipe is tested to 20,680 kPa (3000 psi) maximum unless a higher pressure is agreed upon by the purchaser and manufacturer.  
<sup>d</sup> No maximum test pressure, except that plain-end pipe is tested to 20,680 kPa (3000 psi) unless a higher pressure is agreed upon by the purchaser and manufacturer.

### 13 Make-up Torque for Round Thread Casing and Tubing

**132.1** The values of optimum make-up torque listed in **API 5C1** (in ft-lb) were taken as 1 % of the calculated joint pull-out strength for round thread casing, non-upset tubing, and upset tubing as determined from Equations (54), (63), and (65), respectively.

**132.2** The API round thread joint pull-out strength equation contains several variables believed to affect make-up torque. Using a modification of the joint strength equation for establishing torque values, the torque values obtained by dividing the calculated pull-out value by 100 to be generally comparable to those values obtained by field make-up tests where the API modified thread compound was used.

**132.3** Subsequently, optimum and maximum torques were dropped for large diameter (sizes Label 1: 16 to Label 1: 20) casing, and minimum torque was changed to 1 % of pull-out strength.

**132.4** Action was taken to eliminate minimum and maximum torque values (formerly 75 % and 125 % of the optimum make-up torque, respectively) and emphasize position on make-up.

### 14 Guided Bend Tests for Submerged Arc Welded Line Pipe

#### 14.1 General

**143.1.1** Dimensions for the jig for guided bend tests for submerged arc welded line pipe are calculated from Equation (422140) with reference to Figure 13.

**143.1.2** The critical dimension on guided bend test jig,  $A_{gbtj}$ , denoted as dimension A in **API 5L**, is calculated as follows:

$$A_{gbtj} = [1.15 (D - 2t)] / [\varepsilon_{eng} D / t - 2\varepsilon_{eng} - 1] - t \quad (422140)$$

where

$D$  is the specified pipe outside diameter, in millimeters or inches;

$t$  is the specified pipe wall thickness, in millimeters or inches;

$\varepsilon_{eng}$  is the engineering strain;

where the value of  $\varepsilon_{eng}$  depends on grade and additional dimensions (in units of inches), defined by:

$$R_A = A_{gbtj} / 2;$$

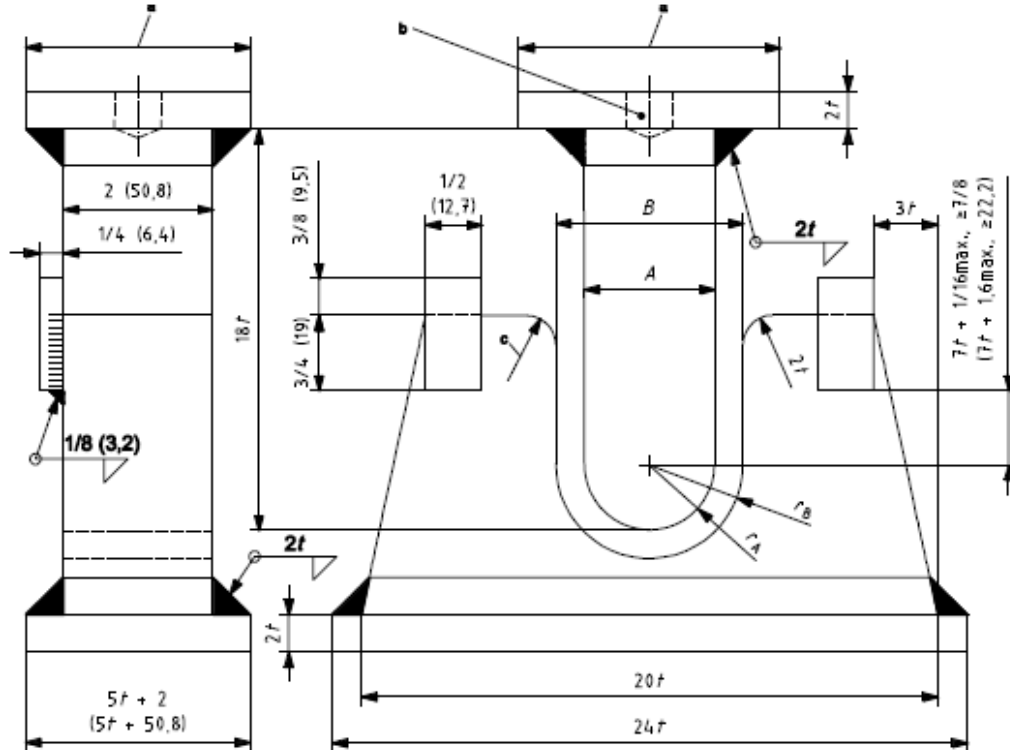
$$B = A_{gbtj} + 2t + 0.125 \text{ in.};$$

$$R_B = B / 2.$$

NOTE See Table 13.



Dimensions in inches (millimeters)



**Key**

$A = A_{gbtj}$  as defined in Equation (122)

$B = A_{gbtj} + 2t + 0.0125$  in.

$t$  specified pipe wall thickness ( $t = 1$  in.)

$a$  as required

$b$  tapped mounting hole

$c$  shoulders hardened and greased hardened rollers may be substituted

**Figure 13—Guided Bend Test Jig**

**Table 13—Values of Strain for Guided Bend Test**

Grade	Strain $\epsilon_{eng}$
A	0.1675
B	0.1375
X42	0.1375
X46	0.1325
X52	0.1275
X56	0.1200
X60	0.1125
X65	0.1100
X70	0.1025
X80	0.0900

## 14.2 Background

### 14.2.1 Values of $\epsilon_{\text{eng}}$

The engineering strain,  $\epsilon_{\text{eng}}$ , is calculated as follows:

$$\epsilon_{\text{eng}} = 3000 (0.64)^{0.2} / f_{\text{umnp}}^{0.9} \quad (1423/141)$$

where

$f_{\text{umnp}}$  is the specified minimum tensile strength of the pipe body, expressed in pounds per square inch.

### 14.2.2 Values of $A_{\text{gbtj}}$

The values of dimension  $A_{\text{gbtj}}$  in Appendix G of API 5L:2004 (43<sup>rd</sup> Edition) are calculated from Equation (1422/140) and rounded as shown in Table 14.

**Table 14—Standard Values for Dimension  $A_{\text{gbtj}}$  in Guided Bend Test**

Unit	Dimension A										
mm	25.4	30.5	35.6	40.6	48.3	55.9	66.0	78.7	94.0	111.8	132.1
	157.5	188.0	223.5	266.7	320.0	383.5	459.7	551.2	660.4	792.5	—
in.	1.0	1.2	1.4	1.6	1.9	2.2	2.6	3.1	3.7	4.4	5.2
	6.2	7.4	8.8	10.5	12.6	15.1	18.1	21.7	26.0	31.2	—

## 15 Determination of Minimum Impact Specimen Size for API Couplings and Pipe

### 15.1 Critical Thickness

The Charpy V-notch (CVN) absorbed energy requirements for API couplings are based on the coupling thickness at the critical thickness. The critical thickness for API couplings is defined as the thickness at the root of the thread at the middle of the coupling, based on the specified coupling diameter and the specified thread dimensions. The critical thickness for all API couplings is provided in Table 15 for various pipe; the critical thickness is the specified wall thickness.

**Table 15—Critical Thickness of Various API Couplings**

Pipe OD mm (in.)	NU mm (in.)	EU mm (in.)	EU SC mm (in.)	BC SC mm (in.)	BC mm (in.)	LC mm (in.)	STC mm (in.)
26.67 (1.050)	4.29 (0.169)	5.36 (0.211)	—	—	—	—	—
33.40 (1.315)	5.36 (0.211)	6.55 (0.258)	—	—	—	—	—
42.16 (1.660)	6.07 (0.239)	6.10 (0.240)	—	—	—	—	—
48.26 (1.900)	4.98 (0.196)	6.38 (0.251)	—	—	—	—	—
60.32 (2.375)	7.72 (0.304)	7.62 (0.300)	5.69 (0.224)	—	—	—	—
73.02 (2.875)	9.65 (0.380)	9.09 (0.358)	6.45 (0.254)	—	—	—	—
88.90 (3.500)	11.46 (0.451)	11.53 (0.454)	7.47 (0.294)	—	—	—	—
101.60 (4.000)	11.53 (0.454)	11.63 (0.458)	—	—	—	—	—
114.30 (4.500)	11.05 (0.435)	12.52 (0.493)	—	6.58 (0.259)	11.35 (0.447)	12.04 (0.474)	11.74 (0.462)
127.00 (5.000)	—	—	—	6.76 (0.266)	12.17 (0.479)	12.98 (0.511)	12.47 (0.491)
139.70 (5.500)	—	—	—	6.81 (0.268)	12.28 (0.481)	13.06 (0.514)	12.57 (0.495)
168.28 (6.625)	—	—	—	6.96 (0.274)	11.91 (0.469)	12.90 (0.508)	12.32 (0.485)
177.80 (7.000)	—	—	—	7.11 (0.280)	13.46 (0.530)	14.43 (0.568)	13.72 (0.540)
193.68 (7.625)	—	—	—	8.71 (0.343)	13.61 (0.536)	12.01 (0.473)	13.87 (0.546)
219.08 (8.625)	—	—	—	8.94 (0.352)	15.29 (0.602)	16.43 (0.647)	15.54 (0.612)
244.48 (9.625)	—	—	—	8.94 (0.352)	15.29 (0.602)	16.69 (0.657)	15.60 (0.614)
273.05 (10.750)	—	—	—	8.94 (0.352)	15.29 (0.602)	—	15.70 (0.618)
298.45 (11.750)	—	—	—	—	15.29 (0.602)	—	15.70 (0.618)
339.73 (13.375)	—	—	—	—	15.29 (0.602)	—	15.70 (0.618)
406.40 (16.000)	—	—	—	—	16.94 (0.667)	—	16.05 (0.632)
473.10 (18.625)	—	—	—	—	21.69 (0.854)	—	20.80 (0.819)
508.00 (20.000)	—	—	—	—	16.94 (0.667)	17.09 (0.673)	16.10 (0.634)
NOTE The coupling blank thickness is greater than indicated above, due to the thread height and manufacturing allowance to avoid black crested threads.							

## 15.2 Calculated Coupling Blank Thickness

The appropriate thread height is added to the critical thickness provided in Table 15, and the result is divided by 0.875 to determine the calculated thickness of the coupling blank. The coupling blank thicknesses calculated in this manner are provided in Table 16.

**Table 16—Calculated Couplings Blank Thickness for API Couplings**

Pipe OD mm (in.)	NU mm (in.)	EU mm (in.)	EU SC mm (in.)	BC SC mm (in.)	BC mm (in.)	LC mm (in.)	STC mm (in.)
26.67 (1.050)	6.53 (0.257)	7.72 (0.304)	—	—	—	—	—
33.40 (1.315)	7.72 (0.304)	9.09 (0.358)	—	—	—	—	—
42.16 (1.660)	8.56 (0.337)	8.59 (0.338)	—	—	—	—	—
48.26 (1.900)	7.32 (0.288)	8.92 (0.351)	—	—	—	—	—
60.32 (2.375)	10.44 (0.411)	10.77 (0.424)	8.56 (0.337)	—	—	—	—
73.02 (2.875)	12.65 (0.498)	12.47 (0.491)	9.45 (0.372)	—	—	—	—
88.90 (3.500)	14.68 (0.578)	15.24 (0.600)	10.59 (0.417)	—	—	—	—
101.60 (4.000)	15.24 (0.600)	15.37 (0.605)	—	—	—	—	—
114.30 (4.500)	14.68 (0.578)	16.38 (0.645)	—	9.32 (0.367)	14.78 (0.582)	15.82 (0.623)	15.47 (0.609)
127.00 (5.000)	—	—	—	9.52 (0.375)	15.70 (0.618)	16.89 (0.665)	16.31 (0.642)
139.70 (5.500)	—	—	—	9.58 (0.377)	15.77 (0.621)	16.99 (0.669)	16.43 (0.647)
168.28 (6.625)	—	—	—	9.75 (0.384)	15.42 (0.607)	16.81 (0.662)	16.15 (0.636)
177.80 (7.000)	—	—	—	9.93 (0.391)	17.20 (0.677)	18.54 (0.730)	17.73 (0.698)
193.68 (7.625)	—	—	—	11.91 (0.469)	17.35 (0.683)	18.69 (0.736)	17.91 (0.705)
219.08 (8.625)	—	—	—	12.01 (0.473)	19.28 (0.759)	20.85 (0.821)	19.84 (0.781)
244.48 (9.625)	—	—	—	12.01 (0.473)	19.28 (0.759)	21.13 (0.832)	19.89 (0.783)
273.05 (10.750)	—	—	—	12.01 (0.473)	19.28 (0.759)	—	20.02 (0.788)
298.45 (11.750)	—	—	—	—	19.28 (0.759)	—	20.02 (0.788)
339.73 (13.375)	—	—	—	—	19.28 (0.759)	—	20.02 (0.788)
406.40 (16.000)	—	—	—	—	21.16 (0.833)	—	20.40 (0.803)
473.10 (18.625)	—	—	—	—	26.59 (1.047)	—	25.86 (1.018)
508.00 (20.000)	—	—	—	—	21.16 (0.833)	21.59 (0.850)	20.47 (0.806)

### 15.3 Calculated Wall Thickness for Transverse Specimens

The calculated wall thickness necessary for full-size, three-quarter-size, and one-half-size transverse impact test specimens for API couplings, including a 0.020 in. OD and a 0.020 in. ID machining allowance, is determined according to Equation (124) and provided in Table 17.

**Table 17—Transverse Impact Specimen Size Required for API Couplings**

(1)	(2)	(3)	(4)	(5)	(6)
Label 1	Connection	Coupling Outside Diameter mm (in.)	Calculated Wall Thickness Required to Machine Transverse Charpy Impact Specimens <sup>a</sup> mm (in.)		
			Full Size	3/4 Size	1/2 Size
3-1/2	NU	107.95 (4.250)	18.54 (0.730)	16.05 (0.632)	13.54 (0.533)
	EU	114.30 (4.500)	18.06 (0.711)	15.57 (0.613)	13.06 (0.514)
4	NU	120.65 (4.750)	17.65 (0.695)	15.14 (0.596)	12.65 (0.498)
	EU	127.00 (5.000)	17.27 (0.680)	14.78 (0.582)	12.27 (0.483)
4-1/2	NU	132.08 (5.200)	17.02 (0.670)	14.50 (0.571)	12.01 (0.473)
	EU	141.30 (5.563)	16.59 (0.653)	14.10 (0.555)	11.58 (0.456)
	STC/LC/BC	133.35 (5.250)	16.94 (0.667)	14.45 (0.569)	11.96 (0.471)
5	STC/LC/BC	147.32 (5.800)	16.33 (0.643)	13.84 (0.545)	11.35 (0.447)
5-1/2	STC/LC/BC	160.02 (6.300)	15.90 (0.626)	13.39 (0.527)	10.90 (0.429)
6-5/8	STC/LC/BC	187.71 (7.390)	15.14 (0.596)	12.62 (0.497)	10.13 (0.399)
7	STC/LC/BC	200.03 (7.875)	14.86 (0.585)	12.37 (0.487)	9.88 (0.389)
7-5/8	STC/LC/BC	215.90 (8.500)	14.58 (0.574)	12.07 (0.475)	9.58 (0.377)
8-5/8	STC/LC/BC	244.48 (9.625)	14.15 (0.557)	11.66 (0.459)	9.14 (0.360)
9-5/8	STC/LC/BC	269.88 (10.625)	13.84 (0.545)	11.35 (0.447)	8.84 (0.348)
10-3/4	STC/BC	298.45 (11.750)	13.56 (0.534)	11.07 (0.436)	8.56 (0.337)
11-3/4	STC/BC	323.85 (12.750)	13.36 (0.526)	10.87 (0.428)	8.36 (0.329)
13-3/8	STC/BC	365.13 (14.375)	13.11 (0.516)	10.59 (0.417)	8.10 (0.319)
16	STC/BC	431.80 (17.000)	12.78 (0.503)	10.29 (0.405)	7.77 (0.306)
18-5/8	STC/BC	508.00 (20.000)	12.50 (0.492)	10.01 (0.394)	7.52 (0.296)
20	STC/LC/BC	533.40 (21.000)	12.45 (0.490)	9.93 (0.391)	7.44 (0.293)

<sup>a</sup> Wall thicknesses provide a 0.51 mm (0.020 in.) OD and 0.51 mm (0.020 in.) ID machining allowance.

NOTE Orientation and specimen size is calculated on coupling material/stock wall and not coupling critical thickness.

$$\text{Minimum wall thickness (inches)} = (W/2) - [(W/2)^2 - 1.1722]^{0.5} + 0.040 \text{ in.} + k_i(0.3937) \quad (\text{124142})$$

where

$k_i$  is the factor for determining minimum wall thickness for transverse impact specimens, as follows:  
 1.00 for full-size specimens  
 0.75 for three-quarter-size specimens  
 0.50 for one-half-size specimens;

$W$  is the specified coupling outside diameter, in accordance with **API 5CT**, in inches.

## 15.4 Calculated Wall Thickness for Longitudinal Specimens

The calculated wall thickness necessary for full-size, three-quarter-size, and one-half-size longitudinal impact test specimens for API couplings, including a 0.020 in. OD and a 0.020 in. ID machining allowance, is determined according to Equation (125143) and provided in Table 18.

$$\text{Minimum wall thickness (inches)} = (W/2) - [(W/2)^2 - 0.3875]^{0.5} + 0.040 \text{ in.} + k_i(0.3937) \quad (\text{125143})$$

where

$k_i$  is the factor for determining minimum wall thickness for transverse impact specimens, as follows:  
 1.00 for full-size specimens  
 0.75 for three-quarter-size specimens  
 0.50 for one-half-size specimens;

$W$  is the specified coupling outside diameter, in accordance with **API 5CT**, in inches.

**Table 18—Longitudinal Impact Specimen Size Required for API Couplings**

(1)	(2)	(3)	(4)	(5)	(6)
Label 1	Connection	Coupling Outside Diameter mm (in.)	Calculated Wall Thickness Required to Machine Longitudinal Charpy Impact Specimens <sup>a</sup> mm (in.)		
			Full Size	3/4 Size	1/2 Size
1.050	NU	33.35 (1.313)	11.79 (0.464)	9.27 (0.365)	6.78 (0.267)
	EU	42.16 (1.660)	11.61 (0.457)	9.12 (0.359)	6.63 (0.261)
1.315	NU	42.16 (1.660)	11.61 (0.457)	9.12 (0.359)	6.63 (0.261)
1.660	NU	52.17 (2.054)	11.51 (0.453)	8.99 (0.354)	6.50 (0.256)
	EU	55.88 (2.200)	11.46 (0.451)	8.97 (0.353)	6.48 (0.255)
1.900	NU	55.88 (2.200)	11.46 (0.451)	8.97 (0.353)	6.48 (0.255)
	EU	63.50 (2.500)	11.40 (0.449)	8.92 (0.351)	6.40 (0.252)
2-3/8	NU	73.02 (2.875)	11.35 (0.447)	8.86 (0.349)	6.35 (0.250)
	EU	77.80 (3.063)	11.33 (0.446)	8.84 (0.348)	6.35 (0.250)
2-7/8	NU	88.90 (3.500)	11.30 (0.445)	8.78 (0.346)	6.30 (0.248)
	EU	93.17 (3.668)	11.28 (0.444)	8.78 (0.346)	6.27 (0.247)
3-1/2	NU	107.95 (4.250)	11.25 (0.443)	8.74 (0.344)	6.25 (0.246)
	EU	114.30 (4.500)	11.23 (0.442)	8.74 (0.344)	6.22 (0.245)
4	NU	120.65 (4.750)	11.23 (0.442)	8.71 (0.343)	6.22 (0.245)
	EU	127.00 (5.000)	11.20 (0.441)	8.71 (0.343)	6.22 (0.245)
4-1/2	NU	132.08 (5.200)	11.20 (0.441)	8.71 (0.343)	6.20 (0.244)
	EU	141.30 (5.563)	11.20 (0.441)	8.69 (0.342)	6.20 (0.244)
	STC/LC/BC	133.00 (5.250)	12.93 (0.509)	10.41 (0.410)	7.92 (0.312)
<sup>a</sup> Wall thicknesses provide a 0.51 mm (0.020 in.) OD and 0.51 mm (0.020 in.) ID machining allowance.					
NOTE Orientation and specimen size is calculated on coupling material/stock wall and not coupling critical thickness.					

## 15.5 Minimum Specimen Size for API Couplings

The calculated wall thickness of the coupling blank (see 17.2) is compared to the calculated wall thickness **necessary** for an impact test specimen (see Table 16 and Table 17). The minimum size impact test specimen that should be selected from Table 16 or Table 17 is the largest impact test specimen having a calculated wall thickness that is less than the calculated wall thickness of the coupling blank for the connection of interest. See Table 19 for the minimum acceptable size transverse specimens and Table 20 for the minimum acceptable size longitudinal specimens. Table 19 and Table 20 are used to determine the impact specimen orientation and size as **necessary** in **API 5CT**.

**Table 19—Minimum Size Transverse Charpy Impact Test Specimens for Various API Couplings**

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)
Label 1	Minimum Permissible Size Transverse Charpy Impact Test Specimens <sup>a</sup>						
	NU	EU	EU	Special Clearance <sup>c</sup> BC	BC	LC	STC
3-1/2	1/2	1/2	1/2	—	—	—	—
4	3/4	3/4	—	—	—	—	—
4-1/2	3/4	3/4	—	b	3/4	3/4	3/4
5	—	—	—	1/2	3/4	Full	3/4
5-1/2	—	—	—	1/2	3/4	Full	Full
6-5/8	—	—	—	Full	Full	Full	Full
7	—	—	—	3/4	Full	Full	Full
7-5/8	—	—	—	Full	Full	Full	Full
8-5/8	—	—	—	Full	Full	Full	Full
9-5/8	—	—	—	Full	Full	Full	Full
10-3/4	—	—	—	Full	Full	—	Full
11-3/4	—	—	—	—	Full	—	Full
13-3/8	—	—	—	—	Full	—	Full
16	—	—	—	—	Full	—	Full
18-5/8	—	—	—	—	Full	—	Full
20	—	—	—	—	Full	Full	Full
NOTE 1 Transverse specimens are not possible for couplings for pipe sizes smaller than 3.50 in.							
NOTE 2 Orientation and specimen size is calculated on coupling material/stock wall and not coupling critical thickness.							
<sup>a</sup> The size of the specimen is relative to a full-size specimen that is 10 mm (0.39 in.) × 10 mm (0.39 in.).							
<sup>b</sup> Should use longitudinal specimen.							
<sup>c</sup> The Charpy impact specimen size assumes that special clearance couplings are machined from standard couplings.							



**Table 20—Minimum Size Longitudinal Charpy Impact Test Specimens for API Couplings for All Pipe Less Than Label 1: 3-1/2 Outside Diameter and for Larger Sizes Where Transverse Test Specimens One-Half Size or Larger Are Not Possible**

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)
Label 1	Minimum Permissible Size Longitudinal Charpy Impact Test Specimens <sup>a</sup>						
	NU	EU	EU	Special Clearance <sup>c</sup> BC	BC	LC	STC
1.050	b	1/2	—	—	—	—	—
1.315	1/2	3/4	—	—	—	—	—
1.660	1/2	1/2	—	—	—	—	—
1.900	1/2	3/4	—	—	—	—	—
2-3/8	3/4	3/4	3/4	—	—	—	—
2-7/8	Full	Full	Full	—	—	—	—
3-1/2	N/A	N/A	N/A	—	—	—	—
4	N/A	N/A	—	—	—	—	—
4-1/2	N/A	N/A	—	3/4	—	—	—
NOTE 1 N/A = Transverse impact test specimens should be used for all tubing connections for pipe Label 1: 3-1/2 OD and larger and for all casing Label 1: 5 OD and larger.							
NOTE 2 Orientation and specimen size is calculated on coupling material/stock wall and not coupling critical thickness.							
<sup>a</sup> The size of the specimen is relative to a full-size specimen that is 10 mm (0.39 in.) × 10 mm (0.39 in.).							
<sup>b</sup> Pipe not thick enough to test based on calculations; however, if the coupling material is slightly thicker than calculated, it will be possible to machine a half-size longitudinal test specimen.							
<sup>c</sup> The Charpy impact specimen size assumes that special clearance couplings are machined from standard couplings.							

## 15.6 Impact Specimen Size for Pipe

Procedures specified in 17.5.3 and 17.5.4 are used to determine the wall thickness necessary for impact test specimens for pipe except that the OD term is the specified pipe OD. Tables specifying the calculated wall thickness necessary to machine full-size, three-quarter-size, and one-half-size transverse and longitudinal impact test specimens are provided in API 5CT.

## 15.7 Larger Size Specimens

In some cases, it can be possible to machine larger impact test specimens if either:

- the coupling blank is thicker than that calculated in 17.2;
- the full 0.020 in. OD and 0.020 in. ID machining allowances are not utilized;
- the impact test specimens are partially rounded due to the OD curvature of the original tubular product (see API 5CT).

## 15.8 Reference Information

See Reference [5] for the correlation of  $K_c$  to CVN for high-strength steel. The longitudinal requirements are based on a longitudinal-to-transverse ratio of 1.33 for grades J55/K55 and 2.0 for higher-strength grades.

The requirement for pipe in SR 16 of **API 5CT** is based on the minimum specified yield strength rather than the maximum specified yield strength used for couplings. This choice is made since the stress level of the pipe is typically expected to be less than the stress level of the couplings.

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## Annex A (informative)

### Discussion of Equations for Triaxial Yield of Pipe Body

#### A.1 Triaxial Yield of Pipe Body

##### A.1.1 General

The criterion for triaxial pipe body yield is that proposed by von Mises. The elastic state leading to incipient yield consists of the superposition of:

- radial and circumferential stress as determined by the Lamé equations for a thick cylinder,
- uniform axial stress due to all sources except bending,
- axial bending stress for a Timoshenko beam,
- torsional shear stress due to a moment aligned with the axis of the pipe.

##### A.1.2 Equations for Elastic Pipe Stresses

###### A.1.2.1 General

Yield, as defined in this subsection, assumes a material for which the elastic limit, the proportional limit, and the yield stress coincide. Further, yield of the material marks the boundary between elastic and inelastic behavior. This boundary has no relation to standardized definitions of minimum yield strength. Standard definitions, such as the minimum yield strength as specified in [API 5CT](#), are more appropriately discussed in conjunction with the design equation.

The limit state for pipe body yield addresses the stress state at which yield is about to occur. That is, the pipe body is still entirely elastic, with one or more locations just reaching yield. Therefore, the stresses defining the yield limit state in the pipe body can be defined with equations based on linear elastic behavior.

###### A.1.2.2 Lamé Equations

Given a tube exposed to internal and external pressure, the radial stress,  $\sigma_r$ , and the circumferential (or hoop) normal stress,  $\sigma_h$ , in the tube are given by:

$$\sigma_r = [(p_i d^2 - p_o D^2) - (p_i - p_o) d^2 D^2 / (4r^2)] / (D^2 - d^2) \quad (\text{A.1})$$

$$\sigma_h = [(p_i d^2 - p_o D^2) + (p_i - p_o) d^2 D^2 / (4r^2)] / (D^2 - d^2) \quad (\text{A.2})$$

where

$D$  is the specified pipe outside diameter;

$d$  is the pipe inside diameter,  $d = D - 2t$ ;

$p_i$  is the internal pressure;

$p_o$  is the external pressure;

$r$  is the radial coordinate,  $(d/2) \leq r \leq (D/2)$ ;

$t$  is the specified pipe wall thickness.

The elastic radial and hoop stresses do not depend on the axial load.

#### A.1.2.3 Uniform Axial Stress

The gravitational force field, along with other environmental loads (e.g. hydrostatic pressure on shoulders, changes in temperature and pressure, landing practice) give rise to an axial force,  $F_a$ . The resulting axial stress,  $\sigma_a$ , i.e. the component of axial stress not due to bending, assumed uniform across any cross section, is as follows:

$$\sigma_a = F_a / A_p \quad (A.3)$$

where

$A_p$  is the area of the pipe cross section,  $A_p = \pi/4 (D^2 - d^2)$ ;

$D$  is the specified pipe outside diameter;

$d$  is the pipe inside diameter,  $d = D - 2t$ ;

$t$  is the specified pipe wall thickness;

$F_a$  is the axial force.

In some cases,  $F_a$  is known and the axial stress is determined from Equation (A.3). In other instances, the axial stress is known, and  $F_a$  is determined from the axial stress. For example, if the pipe is cemented in a well, then stretching and contracting in the axial direction are not allowed. The axial stress, and hence the axial force, is then partially a function of changes in pressure and temperature. That is, the axial stress and axial force are secondary rather than primary variables. The relation in Equation (A.3) still applies.

#### A.1.2.4 Bending Stress

The axial stress component due solely to bending is given by:

$$\sigma_b = \pm M_b r / I = \pm E c r \quad (A.4)$$

where

$c$  is the tube curvature, the inverse of the radius of curvature to the centerline of the pipe;

$D$  is the specified pipe outside diameter;

$d$  is the pipe inside diameter,  $d = D - 2t$ ;

$E$  is Young's modulus;

$I$  is the moment of inertia of the pipe cross section,  $I = \pi/64 (D^4 - d^4)$ ;

$M_b$  is the bending moment;

$r$  is the radial coordinate,  $(d/2) \leq r \leq (D/2)$ ;

$t$  is the specified pipe wall thickness.

The  $\pm$  sign indicates that the component of axial stress due to bending can be positive (tension) or negative (compression), depending on the location of the point in the cross section. Points in the pipe cross section closer to the center of tube curvature than the centerline of the pipe experience compressive bending stress. Points in the pipe cross section farther from the center of tube curvature than the centerline of the pipe experience tensile bending stress.

The variable  $c$  has units radian/length, which are not the norm for the petroleum industry. The more common measure of  $c$  in the industry is  $^{\circ}/30$  m or  $^{\circ}/100$  ft. If, therefore, the units for  $c$  are radians/meter, and  $c$  is expressed in  $^{\circ}/30$  m, the right-hand side of Equation (A.4) should be multiplied by the constant  $\pi/(180 \times 30) = 5.8178 \times 10^{-4}$ . Similarly, if the units for  $c$  are radians/in., and  $c$  is expressed in  $^{\circ}/100$  ft, the right-hand side of Equation (A.4) should be multiplied by the constant  $\pi/(180 \times 100 \times 12) = 1.4544 \times 10^{-5}$ . If, therefore, the appropriate units for  $c$  are radians per meter, and  $c$  is expressed in  $^{\circ}/30$  m, the right hand side of Equation (A.4) should be multiplied by the constant  $\pi/(180 \times 30) = 5.8178 \times 10^{-4}$ .

### A.1.2.5 Torsional Stress

The torsional shear stress,  $\tau_{ha}$ , acting in the circumferential direction on the pipe cross section is as follows:

$$\tau_{ha} = Tr/J_p \quad (A.5)$$

where

$D$  is the specified pipe outside diameter;

$d$  is the pipe inside diameter,  $d = D - 2t$ ;

$J_p$  is the polar moment of inertia of the pipe cross section,  $J_p = \pi/32 (D^4 - d^4)$ ;

$r$  is the radial coordinate,  $(d/2) \leq r \leq (D/2)$ ;

$T$  is the applied torque;

$t$  is the specified pipe wall thickness.

## A.1.3 Triaxial Yield Limit State Equation

### A.1.3.1 General

Given the internal and external pressures, axial force, and bending and torsional moments, the equivalent stress,  $\sigma_e$ , is defined as follows:

$$\sigma_e = [\sigma_r^2 + \sigma_h^2 + (\sigma_a + \sigma_b)^2 - \sigma_r \sigma_h - \sigma_r(\sigma_a + \sigma_b) - \sigma_h(\sigma_a + \sigma_b) + 3\tau_{ha}^2]^{1/2} \quad (A.6)$$

where

$\sigma_r$  is the radial stress, and  $\sigma_h$ , the circumferential or hoop stress, given by Equations (A.1) and (A.2), respectively;

$\sigma_a$  is the component of axial stress not due to bending, given by Equation (A.3);

$\sigma_b$  is the component of axial stress due to bending, given by Equation (A.4);

$\tau_{ha}$  is the torsional (shear) stress, given by Equation (A.5).

The onset of yield is defined as follows:

$$\sigma_e = f_y \quad (A.7)$$

where  $\sigma_e < f_y$  corresponds to elastic behavior, and

$\sigma_e$  is the equivalent stress;

$f_y$  is the yield strength of a representative tensile specimen.

In the absence of bending and torsion, the highest value of equivalent stress always occurs at the inner radius of the pipe body. In the presence of bending ( $\sigma_b \neq 0$ ), Equation (A.7) should be checked four times, i.e. once at the inner diameter and once at the outer diameter, for each of the possible positive and negative values of  $\sigma_b$ .

### A.1.3.2 Special Cases of the Yield Criterion

#### A.1.3.2.1 Pipe Exposed to Axial Stress Alone

In the absence of internal and external pressure, bending, and torsion, Equation (A.6) reduces to:

$$\sigma_e^2 = \sigma_a^2 \quad (A.8)$$

where

$\sigma_a$  is the component of axial stress not due to bending;

$\sigma_e$  is the equivalent stress.

Yield of the pipe occurs when the axial stress equals  $\pm f_y$ , where  $f_y$  is the yield strength of a representative tensile specimen.

#### A.1.3.2.2 Pipe Exposed to Internal and External Pressure and Axial Stress

In the absence of bending and torsion, Equation (A.6) reduces to:

$$\sigma_e = [\sigma_r^2 + \sigma_h^2 + \sigma_a^2 - \sigma_r\sigma_h - \sigma_r\sigma_a - \sigma_h\sigma_a]^{1/2} \quad (A.9)$$

where

$\sigma_e$  is the equivalent stress;

$\sigma_a$  is the component of axial stress not due to bending;

$\sigma_h$  is the circumferential or hoop stress;

$\sigma_r$  is the radial stress.

Substituting Equations (A.1) and (A.2) into (A.9) and using the yield criterion in Equation (A.7), at the inner radius of the pipe body:

$$f_y^2 = [(p_i d^2 - p_o D^2)/(D^2 - d^2)]^2 + 3(p_i - p_o)^2 D^4/(D^2 - d^2)^2 + \sigma_a^2 - 2[(p_i d^2 - p_o D^2)/(D^2 - d^2)]\sigma_a \quad (\text{A.10})$$

or

$$f_y^2 = [\sigma_a - (p_i d^2 - p_o D^2)/(D^2 - d^2)]^2 + 3[(p_i - p_o) D^2/(D^2 - d^2)]^2 \quad (\text{A.11})$$

where

$D$  is the specified pipe outside diameter;

$d$  is the pipe inside diameter,  $d = D - 2t$ ;

$f_y$  is the yield strength of a representative tensile specimen;

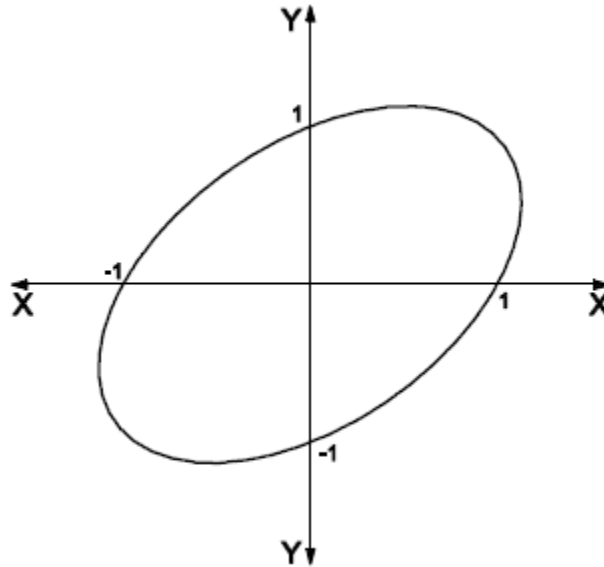
$p_i$  is the internal pressure;

$p_o$  is the external pressure;

$t$  is the specified pipe wall thickness;

$\sigma_a$  is the component of axial stress not due to bending;

which is the equation of an ellipse rotated at 45° about the origin, with semi-major to semi-minor axes in the ratio  $a/b = \sqrt{3}$  (Figure A.1).



#### Key

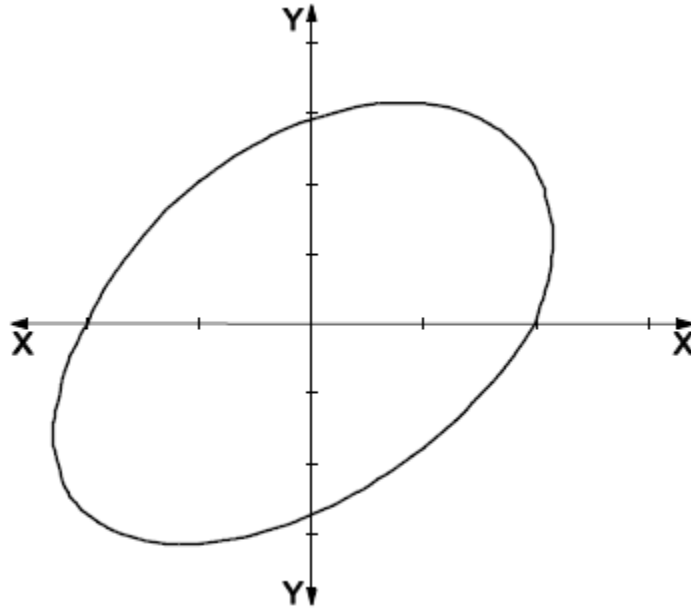
X  $(\sigma_a + p_i)/f_y$

Y  $1/2 [(D/t)^2/(D/t - 1)][(p_i - p_o)/f_y]$

**Figure A.1—von Mises Yield Criterion for a Tube Loaded by Internal and External Pressures and Axial Stress**

### A.1.3.2.3 Alternative Representation of Yield Surface

A consequence of the expression of the yield criterion in terms of internal and external pressures and axial stress is that the pressure and axial stress terms cannot be explicitly separated. One solution, used in [API 5C5](#), is to divide the expression of yield into two special cases representing only external pressure or only internal pressure in combination with axial load. Further, the geometric factor appearing on the abscissa in Figure A.1 is incorporated in the yield surface by simplifying the abscissa to  $p_i$  (upper two quadrants) and  $p_o$  (lower two quadrants). The resulting pictorial representation (see Figure A.2) of yield is similar to that of Figure A.1, with the exception that the yield criterion is not smooth at  $p_i = p_o = 0$ .



#### Key

X axial load, kN  
Y pressure, MPa

**Figure A.2—von Mises Yield Criterion for a Tube Loaded by Internal and External Pressure and Axial Stress, [API 5C5](#) Representation**

The equation for [internal pressure in](#) the upper two quadrants ( $p_o = 0$ ) is as follows:

$$p_i = (-k_B \pm [k_B^2 - 4k_A k_C]^{1/2}) / (2k_A) \quad (\text{A.12})$$

where

$$k_A = k_{pi}^2 + k_{pi} + 1 \quad (\text{A.13})$$

$$k_B = (1 - k_{pi}) \sigma_a \quad (\text{A.14})$$

$$k_C = \sigma_a^2 - f_y^2 \quad (\text{A.15})$$

$$k_{pi} = (D^2 + d^2) / (D^2 - d^2) \quad (\text{A.16})$$



The equation for external pressure in the lower two quadrants ( $p_i = 0$ ) is as follows:

$$p_o = (-k_B \pm [k_B^2 - 4k_A k_C]^{1/2}) / (2k_A) \quad (A.17)$$

where

$$k_A = k_{po}^2 \quad (A.18)$$

$$k_B = k_{po} \sigma_a \quad (A.19)$$

$$k_C = \sigma_a^2 - f_y^2 \quad (A.20)$$

$$k_{po} = 2D^2 / (D^2 - d^2) \quad (A.21)$$

In both instances:

$D$  is the specified pipe outside diameter;

$d$  is the pipe inside diameter,  $d = D - 2t$ ;

$f_y$  is the yield strength of a representative tensile specimen;

~~$p_i$  is the internal pressure;~~

~~$p_o$  is the external pressure;~~

$t$  is the specified pipe wall thickness;

$\sigma_a$  is the component of axial stress not due to bending.

#### A.1.3.2.4 Effective Stress Representation of Yield Surface

It is sometimes convenient to express pipe equations in terms of the effective stress,  $\sigma_{eff}$ , which is defined as:

$$\sigma_{eff} = F_{eff} / A_p = (F_a - p_i A_i - p_o A_o) / A_p \quad (A.22)$$

where

$A_i$  is the area to pipe inside diameter;  $A_i = \pi d^2 / 4$ ;

$A_o$  is the area to pipe outside diameter;  $A_o = \pi D^2 / 4$ ;

$A_p$  is the cross-sectional area of the pipe;  $A_p = A_o - A_i$ ;

$F_a$  is the material axial force;

$F_{eff}$  is the effective axial force;

$D$  is the specified pipe outside diameter;

$d$  is the pipe inside diameter,  $d = D - 2t$ ;

$p_i$  is the internal pressure;

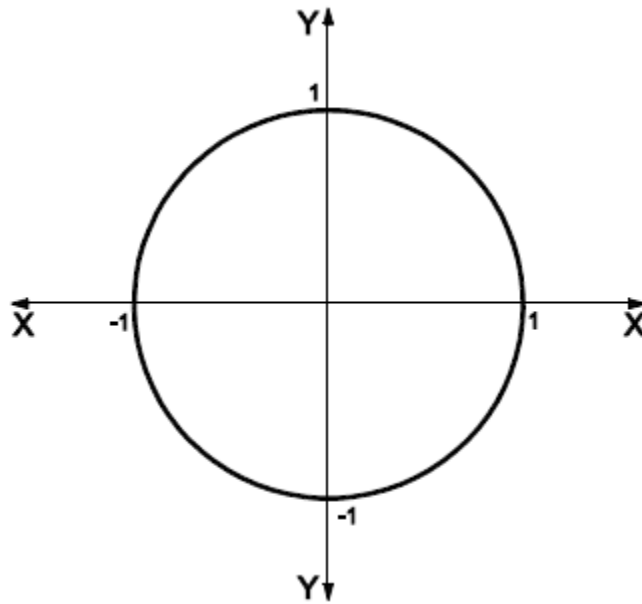
$p_o$  is the external pressure.

In this case, the equivalent of Equation (A.11) is as follows:

$$f_y^2 = \sigma_{\text{eff}}^2 + 3(p_i - p_o)^2 D^4 / (D^2 - d^2)^2 \quad (\text{A.23})$$

where

$f_y$  is the yield strength of a representative tensile specimen, for which the graphical representation is a circle (see Figure A.3).



**Key**

X  $\sigma_{\text{eff}}/f_y$

Y  $\sqrt{3} [D^2/(D^2 - d^2)] [(p_i - p_o)/f_y]$

**Figure A.3—von Mises Yield Criterion Expressed in Terms of Internal and External Pressure and Effective Stress**

### A.1.3.3 Assumptions and Limitations

#### A.1.3.3.1 General

Equations (A.6) and (A.7) are based on the assumptions given in A.1.3.3.1.1 to A.1.3.3.1.4.

##### A.1.3.3.1.1 Concentric, Circular Cross-sectional Geometry

The equations for radial stress, circumferential or hoop stress, bending, and torsion presume the pipe cross section to consist of inner and outer surfaces that are circular and concentric.

### A.1.3.3.1.2 Isotropic Yield

The yield strength of the material of which the pipe is composed is assumed to be independent of direction. An axial sample and a circumferential sample are assumed to possess identical elastic moduli and yield stresses in both tension and compression.

### A.1.3.3.1.3 No Residual Stress

For determination of the onset of yield, residual stresses due to manufacturing processes are assumed to be negligible and are ignored.

### A.1.3.3.1.4 Cross-sectional Instability (Collapse) and Axial Instability (Column Buckling)

Particularly in instances where  $p_o > p_i$ , it is possible for the pipe cross section to collapse due to instability prior to yield. For external pressure greater than internal pressure, see Section 8 on collapse. Similarly, if  $\sigma_{eff} < 0$ , it is possible for the pipe to buckle as a column prior to yield, and the bending stress due to buckling should be included in the yield check.

### A.1.3.3.2 Elongation under Load at Which the Yield Strength Is Determined

The values for the elongation under load at which the yield strength is determined in **API 5CT, API 5DP, and API 5L** for pipe with specified minimum yield strengths of 655 MPa (95,000 psi) or less have been arbitrarily established at 0.5 %.

The values for the elongation under load at which the yield strength is determined in **API 5CT, API 5DP, and API 5L** for pipe with specified minimum yield strengths greater than 655 MPa (95,000 psi) are determined with Equation (A.24), where  $\epsilon_{ymn}$  is the strain at which specified minimum yield strength is determined.

$$\epsilon_{ymn} = (f_{ymn}/E) + 0.002 \quad (A.24)$$

where

$f_{ymn}$  is the specified minimum yield strength;

$E$  is Young's modulus, taken to be  $193 \times 10^3$  MPa ( $28 \times 10^6$  psi).

The calculated values of  $\epsilon_{ymn}$  are rounded to the nearest 0.005.

## A.1.4 Triaxial Yield Design Equation

In all of the general and simplified forms of Equation (A.7), a design equation is formulated with the following substitutions:

replace  $t$  with  $k_{wall} t$  in Equations (A.1) and (A.2) for the radial and circumferential or hoop stresses, respectively, but not in Equations (A.3) to (A.5) for the axial and torsional stresses;

replace  $f_y$  with  $f_{ymn}$ .

The purpose of the design equation is to determine the stress state that results in the onset of pipe yield when the properties of the pipe are at their worst-case, minimum allowable values. The wall thickness of the pipe at all times accounts for the extreme allowable thin-wall eccentricity that comes about naturally as part of the pipe manufacturing process.

## A.2 Initial Yield of Pipe Body, Lamé Equation for Pipe When External Pressure, Bending, and Torsion are Zero

### A.2.1 General

The Lamé equations for the radial and hoop stresses of the pipe are based on the three-dimensional equations of equilibrium for a linear elastic cross section. As such, these equations are triaxial equations and provide the most accurate calculation of pipe stresses. Two equations are provided: open-end with zero axial stress and closed-end with axial stress due to internal pressure acting on the end cap.

### A.2.2 Yield Limit State Equation, Special Case for Capped-end Conditions

Initial yield of a capped-end thick tube is a special case of Equations (A.6) and (A.7) when external pressure, bending, and torsion are zero. The axial stress is generated solely by the action of internal pressure on the ends of the sample (e.g. the capped-end condition). In this case, the effective stress is zero [see Equation (A.22)].

The internal pressure at yield for a capped-end thick tube,  $p_{iYLC}$ , is calculated as follows:

$$p_{iYLC} = f_y (D^2 - d^2) / (\sqrt{3} D^2) \quad (A.25)$$

where

$D$  is the specified pipe outside diameter;

$d$  is the pipe inside diameter,  $d = D - 2t$ ;

$f_y$  is the yield strength of a representative tensile specimen;

$t$  is the specified pipe wall thickness.

This equation is subject to the same assumptions and limitations as the more general expressions from which it is derived (see A.1.3.3).

There is no adjustment to Equation (A.25) for axial tension, as all axial tension is generated by the action of internal pressure on the (closed) ends of the pipe. The more general case, where axial stress is generated by other than the action of internal pressure on the ends of the pipe, is addressed by the triaxial yield criterion, Equations (A.6) and (A.7).

### A.2.3 Yield Design Equation, Special Case for Capped-end Conditions

A design equation for initial yield of the pipe body with capped-end conditions and using the Lamé equations for the radial and hoop stresses should be formulated from Equation (A.9) with the following substitutions:

replace  $t$  with  $k_{wall} t$  in Equations (A.1) and (A.2) for the radial and circumferential or hoop stresses, respectively, but not in Equation (A.3) for the axial stress;

replace  $f_y$  with  $f_{ymn}$ .

The resulting design equation for  $p_{iYLC}$ , internal pressure at yield for a capped-end thick tube, is as follows:

$$p_{iYLC} = f_{ymn} / \{ (3 D^4 + d_{wall}^4) / (D^2 - d_{wall}^2)^2 + d^4 / (D^2 - d^2)^2 - 2 d^2 d_{wall}^2 / [(D^2 - d^2) (D^2 - d_{wall}^2)] \}^{1/2} \quad (A.26)$$

where

$D$  is the specified pipe outside diameter;

$d$  is the pipe inside diameter,  $d = D - 2t$ ;

$d_{\text{wall}}$  is the inside diameter based on  $k_{\text{wall}} t$ ,  $d_{\text{wall}} = D - 2k_{\text{wall}} t$ ;

$f_{\text{ymn}}$  is the specified minimum yield strength;

$k_{\text{wall}}$  is the factor to account for the specified manufacturing tolerance of the pipe wall; e.g. for a tolerance of  $-12.5\%$ ,  $k_{\text{wall}} = 0.875$ ;

$t$  is the specified pipe wall thickness.

Note that the use of different wall thicknesses in the radial/circumferential and axial stresses precludes deriving this design equation directly from Equation (A.25).

#### A.2.4 Yield Limit State Equation, Special Case for Internal Pressure Only with Zero Axial Load

Initial yield of an open-ended thick tube is a special case of Equations (A.6) and (A.7) when the uniform axial stress, external pressure, bending, and torsion are zero. In this case, the internal pressure at yield for an open-ended thick tube,  $p_{iYLo}$ , is as follows:

$$p_{iYLo} = f_y (D^2 - d^2) / (3 D^4 + d^4)^{1/2} \quad (\text{A.27})$$

where

$D$  is the specified pipe outside diameter;

$d$  is the pipe inside diameter,  $d = D - 2t$ ;

$f_y$  is the yield strength of a representative tensile specimen;

$t$  is the specified pipe wall thickness.

This equation is subject to the same assumptions and limitations as the more general expressions from which it is derived (see A.1.3.3).

The more general case where axial stress is non-zero is addressed by the triaxial yield criterion, Equations (A.6) and (A.7).

#### A.2.5 Yield Design Equation, Special Case for Internal Pressure Only with Zero Axial Load

A design equation for initial yield of the pipe body with open-end conditions and using the Lamé equations for the radial and hoop stresses should be formulated from Equation (A.9) with the following substitutions:

replace  $t$  with  $k_{\text{wall}} t$  in Equations (A.1) and (A.2) for the radial and circumferential or hoop stresses, respectively;

replace  $f_y$  with  $f_{\text{ymn}}$ .

The resulting design equation for internal pressure at yield for an open-end thick tube,  $p_{iYLo}$ , is as follows:

$$p_{iYLo} = f_{ymn}(D^2 - d_{wall}^2)/(3D^4 + d_{wall}^4)^{1/2} \quad (A.28)$$

where

$D$  is the specified pipe outside diameter;

$d$  is the pipe inside diameter,  $d = D - 2t$ ;

$d_{wall}$  is the inside diameter based on  $k_{wall} t$ ,  $d_{wall} = D - 2k_{wall} t$ ;

$f_{ymn}$  is the specified minimum yield strength;

$k_{wall}$  is the factor to account for the specified manufacturing tolerance of the pipe wall; e.g. for a tolerance of  $-12.5\%$ ,  $k_{wall} = 0.875$ ;

$t$  is the specified pipe wall thickness.

Since axial stress is absent in this expression, the design equation can be derived directly from Equation (A.27).

### A.2.6 Yield Limit State Equation, Special Case for Thin Wall Pipe with Internal Pressure Only and Zero Axial Load

The Barlow equation for pipe yield, which is the historical API equation, is based on a one-dimensional (not triaxial), approximate formulation of the von Mises yield condition, combined with an approximate expression for the hoop stress in the pipe. In essence, the Barlow equation approximates the hoop stress and then equates this approximation to the yield strength. This approximation is less accurate than the Lamé equation of yield discussed in A.1. Because the Barlow equation neglects axial stress, there is no distinction between pipe with capped ends, pipe with open ends, or pipe with tension end load.

Expanding Equation (A.27) in  $x = t/D$  gives:

$$p_{iYLo} = 2f_y x (1 - 5x^2/2 - x^3 + 63x^4/8 + 11x^5/2 + \dots) \quad (A.29)$$

where

$f_y$  is the yield strength of a representative tensile specimen;

$x$  is the ratio of specified wall thickness to specified outside diameter;  $x = t/D$ .

Making the thin wall assumption (that is, terms in  $x^2$  and above can be neglected) results in:

$$p_{iYLo} = 2f_y x = 2f_y t/D \quad (A.29)$$

where

$D$  is the specified outside diameter;

$f_y$  is the yield strength of a representative tensile specimen;

$t$  is the specified pipe wall thickness;

which is Barlow's equation. It is subject to the same assumptions and limitations as the more general expressions from which it is derived (see A.1.3.3).

### A.2.7 Yield Design Equation, Special Case for Thin Wall Pipe with Internal Pressure Only and Zero Axial Load (Historical API Equation)

Initial yield of a thin tube is defined by the following expression, where  $p_{iYAPI}$  is the internal pressure at yield for a thin tube:

$$p_{iYAPI} = 2f_{ymn}k_{wall} t/D \quad (A.29)$$

where

$D$  is the specified pipe outside diameter;

$f_{ymn}$  is the specified minimum yield strength;

$k_{wall}$  is the factor to account for the specified manufacturing tolerance of the pipe wall; e.g. for a tolerance of -12.5 %,  $k_{wall} = 0.875$ ;

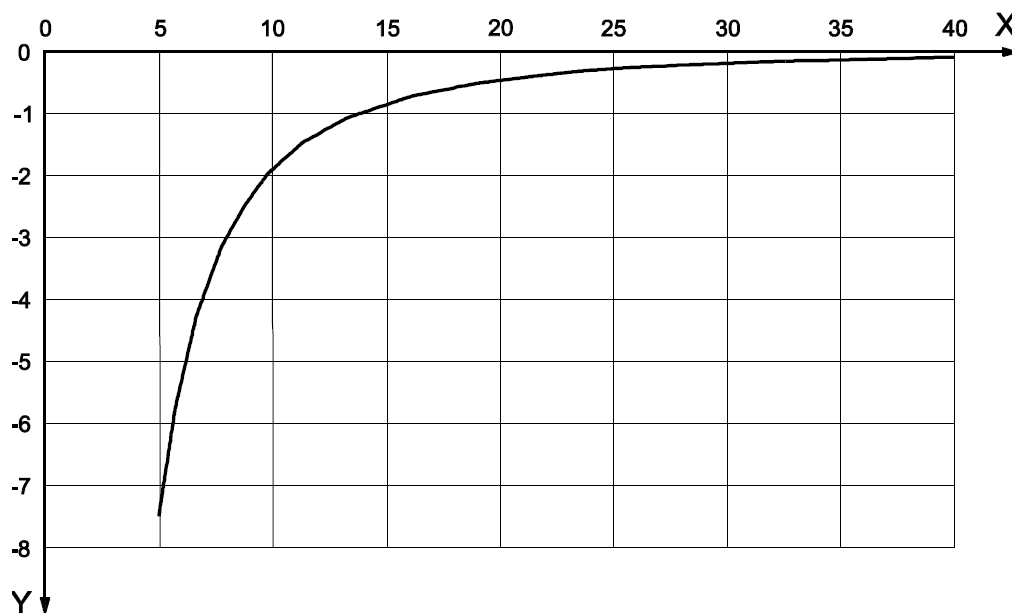
$t$  is the specified pipe wall thickness.

This equation is subject to the same assumptions and limitations as the more general expressions from which it is derived (see A.1.3.3).

### A.2.8 Comparison of Thick and Thin Wall Equations for Internal Pressure Only and Zero Axial Load

Figure A.4 compares Equations (A.28) and (A.29) for the range of diameter: thickness ratio values typical of oil field tubulars. Two significant conclusions are as follows:

- for stated yield stress and cross-sectional dimensions, the thin wall equation (Barlow) predicts a higher internal pressure resistance than the thick wall equation;
- the difference between the limit pressures predicted by the two equations is less than 8 % for the range of diameter:thickness ratios typical of oil field tubulars ( $D/t > 4.9$ ).



**Key**

X  $D/t$

Y difference in  $p/f_y$ , (thick wall/thin wall - 1)  $\times$  100 %

**Figure A.4—Comparison of Thick and Thin Wall Predictions of Yield Pressure for Internal Pressure Only and Zero Axial Load,  $k_{wall} = 0.875$**



## Annex B (informative)

### Discussion of Equations for Ductile Rupture

#### B.1 Introduction

Internal pressure resistance equations differentiate between yield (Annex A) and rupture of the pipe body, and for rupture, between ductile and brittle material response. Table B.1 outlines the treatment of internal pressure resistance in the subsections to follow.

**Table B.1—Equations for Rating a Pipe Body for Internal Pressure Resistance**

Limit State	Definition	Pertinent Annex
Ductile rupture	Failure of a tube in the plastic deformation range, which is characteristic of pipe with adequate and lasting toughness for the environment in which it is used.	Annex B
Fracture	Failure of a tube due to propagation of a crack.	Annex D

#### B.2 Ductile Rupture of Pipe Body

##### B.2.1 General

The equations for ductile rupture pertain to the actual failure of the pipe body due to internal pressure. Whereas the yield equations of Annex A are intended to describe the onset of permanent plastic deformation and not loss of pressure integrity, the rupture equations are intended to describe the ultimate pressure capacity of the pipe at a pressure that fails the pipe body with loss of internal pressure integrity.

These equations are applicable only when the pipe material in its environment has sufficient toughness to meet a minimum criterion such that the deformation of the pipe in its environment, through to rupture, is ductile and not brittle, even in the presence of small imperfections.

The equations for ductile rupture depend on the minimum physical wall thickness and the pipe outer diameter, the maximum depth of imperfections that have a reasonable probability of passing through the inspection process undetected, the fracture toughness of the material, the work hardening of the material, and the ultimate tensile strength of the pipe. Yield strength has no direct impact on the ductile rupture pressure, except through the correlation of the work-hardening parameter  $n$ .

The ductile rupture equations can be derived from the mechanics of pipe equilibrium combined with a model of pipe plasticity and a model of the effect of imperfections. These derivations are outside the scope of this document (see SPE 48329 and SPE 48330).

The ductile rupture limit state and design equations consist of three interlinked concepts:

- an equation for equilibrium-plasticity-based rupture of a pipe with known physical wall thickness and diameter;

- subtraction of a penalty for wall loss in proportion to depths of imperfections which may not be detected by the manufacturing and inspection system;

a criterion for minimum toughness at which ductile rupture applies.

These equations are applicable to direct pressure and axial loading, but do not describe the capacity of the pipe under fatigue loading. The subtraction to the pipe wall for the presence of imperfections and the interrelated role of pipe toughness are based on a fracture mechanics approach relating  $J_{IC}$  toughness measurements of sample pipes to numerically calculated crack-tip intensities (J-integrals) as a function of imperfection depth. This is explained in detail later in this annex.

Further information can be found in [SPE 102585](#) and [SPE 48320](#).

## B.2.2 Capped-end Ductile Rupture Limit State Equation

### B.2.2.1 General

Ductile rupture of a capped-end tube under internal pressure alone is given by the following expression, in which  $p_{iR}$  is the internal pressure at rupture:

$$p_{iR} = 2k_{dr}t_{dr}f_u/(D - t_{dr}) \quad (B.1)$$

where

$$t_{dr} = t_{min} - k_a a \quad (B.2)$$

and

- $a$  is, for a limit state equation, the actual maximum depth of a crack-like imperfection; for a design equation, the maximum depth of a crack-like imperfection that could likely pass the manufacturer's inspection system;
- $D$  is the specified pipe outside diameter;
- $f_u$  is the tensile strength of a representative tensile specimen;
- $k_a$  is the burst strength factor, having the numerical value 1.0 for quenched and tempered (martensitic structure) or 13Cr products and 2.0 for as-rolled and normalized products based on available test data, and the default value set to 2.0 where the value has not been measured; the value of  $k_a$  can be established for a specific pipe material based on testing;
- $k_{dr}$  is the correction factor based on pipe deformation and material strain hardening, having the numerical value  $[(1/2)^{n+1} + (1/\sqrt{3})^{n+1}]$ ;
- $n$  is the dimensionless hardening index used to obtain a curve fit (see B.2.3.3) of the true stress-strain curve derived from the uniaxial tensile test;
- $t_{min}$  is the actual minimum pipe wall thickness disregarding crack-like imperfections.

The value selected for  $k_{dr}$  renders  $p_{iR}$  the average of rupture pressures predicted using Tresca yield condition and von Mises yield condition for the case of an end-capped pipe. It accounts for the material hardening and the pipe deformation up to rupture.

### B.2.2.2 Origin of the Limit State Equation

The limit state Equation (B.1) is based on the mechanics of equilibrium for capped-end pipe subjected to internal pressure, combined with hardening plasticity. This limit state equation was selected from a review

of six candidate equations. The limit state equations were compared with full-scale pipe rupture data for a wide assortment of pipe grades and pipe diameter-to-wall ( $D/t$ ) ratios. The candidate equations and the data used to evaluate the equations are listed in B.3. For each combination of model and source set of data, the results of the comparisons are expressed in terms of the mean and the standard deviation (coefficient of variation) of the ratio of actual and predicted test pressures. The limit state Equation (B.1) provided the best accuracy across the different datasets. When all the data are combined, Equation (B.1) has a mean of 1.004 and a coefficient of variation of 4.7 % for the ratio of actual to predicted rupture pressure.

So far as can be determined, the test data used to calibrate the rupture equation did not have naturally occurring, sharp-bottomed imperfections present. This is understandable considering the frequency of occurrence of sharp-bottomed imperfections for pipe which have been inspected. Likewise, the fundamental mechanics equation for the limit state starts without a penalty for imperfections. However, the equation for the limit state was then generalized to account for external pressure and axial compression or tension different from capped-end conditions, and furthermore to account for the presence of sharp-bottomed imperfections that can just pass undetected through the inspection system. The basis for the imperfection penalty is explained in B.7.

The limit state Equation (B.1) includes the penalty for the maximum actual imperfection in the pipe. Furthermore, pipe can be and occasionally actually is manufactured with imperfections coincident with minimum wall geometry. If the limit state equation is used to make a deterministic calculation of the rupture pressure, the limit state equation should assume that a sharp-bottomed imperfection is present 100 % of the time and that the depth of the imperfection is equal to the inspection threshold.

However, if Equation (B.1) is used to make a probabilistic calculation of the rupture pressure, then the calculation can account for the frequency at which the sharp imperfection occurs and the distribution of wall thickness. Analysis in B.7 makes the case that the low-frequency imperfection with depth equal to the inspection threshold has more impact on the rupture pressure than the high-frequency imperfection of more shallow depth. Because of this, the probabilistic rupture calculation is dominated by the large imperfection (with depth equal to the inspection threshold) that rarely occurs.

Hence, the limit state Equation (B.1) for rupture should always include the correction for imperfection depth, and the probabilistic limit state equation should account for the frequency at which the sharp-bottomed imperfection is likely to occur. For the deterministic calculation of the rupture pressure, the limit state analysis should assume that the imperfection frequency is 100 %, and that the imperfection depth will equal the inspection threshold. For the probabilistic calculation of the rupture pressure, the imperfection frequency should be based on inspection data for pipe that already has been subjected to the inspection system.

## **B.2.3 Assumptions and Limitations**

### **B.2.3.1 General**

Equation (B.1) is based on the assumptions explained in B.2.3.2 and B.2.3.3.

### **B.2.3.2 Material with Adequate Toughness**

For Equation (B.1) to be valid, the pipe material should have minimum toughness equal to or exceeding that embodied in **API 5CT** and its supplement SR 16.

Although the fundamental derivation leading to Equation (B.1) does not depend on the shape of the stress-strain curve, the final form of the equation assumes the true stress-strain curve can be adequately fitted for larger strains of, say, between 2 % and the strain at maximum load by the relation for true (Cauchy) stress,  $\sigma_c$ :

$$\sigma_c = C \varepsilon_{ln}^n \quad (B.3)$$

where

$$C = (2.718/n)^n f_u \quad (B.4)$$

and

$f_u$  is the tensile strength of a representative tensile specimen;

$n$  is the dimensionless hardening index used to obtain a curve fit (see B.2.3.3) of the true stress-strain curve derived from the uniaxial tensile test;

$\varepsilon_{ln}$  is the logarithmic strain.

### B.2.3.3 Determination of the Hardening Index

A specimen from material with true stress-strain relation of the form given in Equation (B.3) would reach a maximum load in a uniaxial tensile test at a logarithmic strain of  $n$ . The best method is to derive  $n$  from fitting the actual true stress-strain curve with Equation (B.3) in the relevant strain range of 2 % to the strain at maximum load. Although less accurate, one could alternatively approximate  $n$  equal to the actual logarithmic strain at the maximum load point of the tensile test. If the engineering strain at the maximum load is given, the logarithmic strain,  $\varepsilon_{ln}$ , is as follows:

$$\varepsilon_{ln} = \ln(1 + \varepsilon_{eng}) \quad (B.5)$$

where

$\varepsilon_{eng}$  is the engineering strain.

The relatively flat nature of the stress-strain curve in the plastic region of most OCTG grades makes determination of  $n$  through this last method often rather difficult. As a third alternative, in the absence of stress-strain information, the following values of  $n$  should be used.

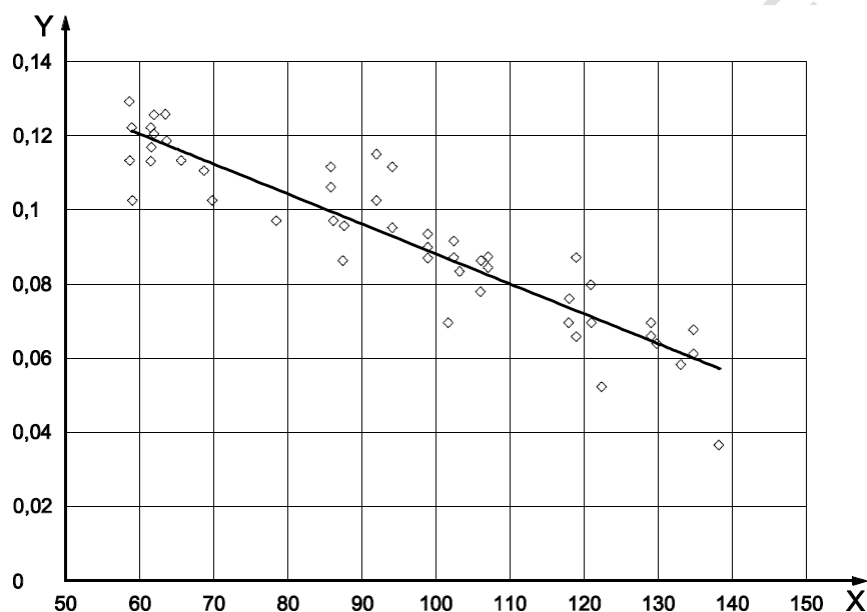
**Table B.2—Suggested Values for Hardening Index in Ductile Rupture Equation**

Grade	$n$
H40	0.14
J55	0.12
K55	0.12
M65	0.12
N80	0.10
L80 Type 1	0.10
L80 Chrome	0.10
C90	0.10
R95	0.09
T95	0.09
P110	0.08

Q125	0.07
------	------

If the grade of the material is unknown but is not high hardening, the hardening index can alternatively be determined from the correlation given in Figure B.1. The effort expended to determine  $n$  should be weighed against the fact that the equation for ductile rupture is relatively insensitive to this quantity for commonly used OCTG. However, if a high-hardening material such as duplex steel is used, it is important to determine  $n$  to avoid a nonconservative rupture strength prediction. Values of  $n$  for these materials can be as high as 0.30.

The parameter  $n$  has relatively small effect on the value of  $p_{iR}$ , but should not be ignored. At  $n = 0.12$  a 14 % change in  $n$  is necessary to produce a 1 % change in predicted rupture pressure. Two materials may have the same tensile strength, but if one material has  $n = 0.12$  it will have a 4 % lower rupture strength than a material with  $n = 0.06$ . When selecting a value of  $n$  for OCTG grades in the absence of experimental data, estimating  $n$  using specified minimum yield strength,  $f_{ymn}$ , is conservative.



#### Key

X measured yield strength, ksi

Y  $n$

NOTE Least squares fit of data results in  $n = 0.1693 - 0.000812 \times \text{measured yield strength}$ ; coefficient of variance = 0.10.

**Figure B.1—Correlations for Hardening Index from Typical Experimental Data for Steel Grades Listed in Table B.2**

## B.2.4 Adjustment for the Effect of Axial Force and External Pressure

### B.2.4.1 General

The ductile rupture strength Equation (B.1) was developed for the situation of an end-capped pipe, where the axial tension is determined by the internal pressure acting on the closed inner pipe surface area. This is a special case of a more general situation where a pipe can reach a maximum internal pressure load, that is a rupture load, under the simultaneous action of arbitrary external pressure and arbitrary axial force. The combined loads together determine when the pipe is going to yield and how it will plastically deform towards the point of rupture. A fundamental criterion when this rupture load is attained can still be

expressed, but this will now be a more involved equation governed by the equation of the von Mises or Tresca yield surface in terms of axial stress, radial stress, and hoop stress.

Moreover, rupture is only the prevailing failure mechanism if the axial force is not too large. For large axial force and smaller internal overpressure, a maximum axial load (a precursor to necking and axial splitting of the pipe) is reached before the maximum pressure phenomenon occurs.

Below, equations for both rupture and necking under combined loads are described, together with a criterion to identify which phenomenon occurs first. The equation is given in terms of “effective axial tension” associated with “effective axial stress” defined in A.1.3.2.4. For effective axial tension, these approximate equations are very accurate when compared to the exact theoretical Equation (24); performance against combined loading test data is given in B.6.2.

For negative values of effective axial force, i.e. effective axial compression, the pipe may buckle as a column, depending on the quality of lateral support. If buckling is adequately suppressed, the equation for rupture under combined loads is valid also for effective axial compression. However, for higher values of effective axial compression, it is the phenomenon of local buckling of the pipe wall (“wrinkling”) that presents the governing failure mechanism. Therefore, there exists a value of effective axial compression that limits the validity of the exact combined loading rupture equation.

#### B.2.4.2 Ductile Rupture under Combined Loads

In the presence of external pressure and axial force different from capped-end conditions, the general equation for ductile rupture is as follows:

$$p_{iRa} = p_o + \min[1/2(p_M + p_{ref T}), p_M] \quad (B.6)$$

with

$$p_M = p_{ref M} [1 - k_R (F_{eff}/F_{uts})^2]^{1/2} \quad (B.7)$$

where

$$F_a = \pi t (D - t) \sigma_a \quad (B.8)$$

$$F_{eff} = F_a + p_o \pi t (D - t) - p_M t (D - t) / [t_{dr} (D - t_{dr})] \pi / 4 (D - 2 t_{dr})^2 \quad (B.9)$$

$$F_{uts} = \pi t (D - t) f_u \quad (B.10)$$

$$p_{uts} = 2 t_{dr} f_u / (D - t_{dr}) \quad (B.11)$$

$$p_{ref} = 1/2 (p_{ref M} + p_{ref T}), \text{ used in Figure B.2} \quad (B.12)$$

$$p_{ref M} = (2/\sqrt{3})^{1+n} (1/2)^n p_{uts} \quad (B.13)$$

$$p_{ref T} = (1/2)^n p_{uts} \quad (B.14)$$

$$k_R = (4^{1-n} - 1)/3^{1-n} \quad (B.15)$$

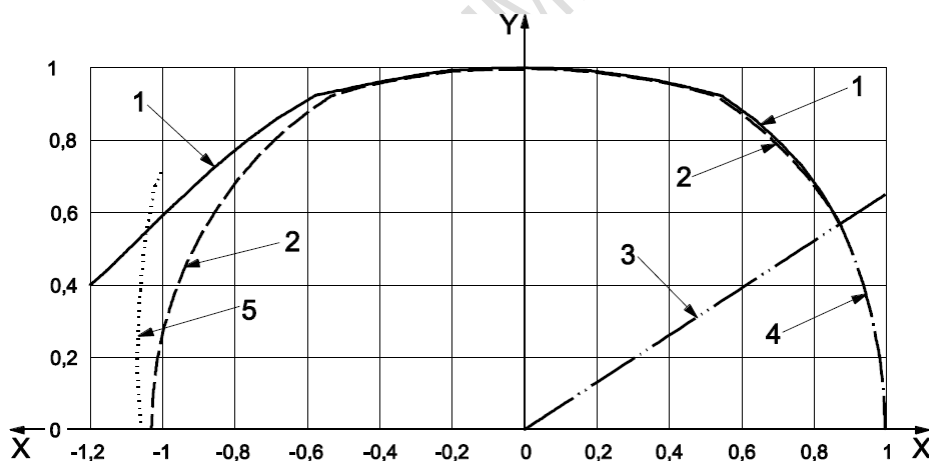
$$t_{dr} = t_{min} - k_a a \quad (B.16)$$

and

$a$  is, for a limit state equation, the actual maximum depth of a crack-like imperfection; for a design equation, the maximum depth of a crack-like imperfection that could likely pass the manufacturer's inspection system;

- $D$  is the specified pipe outside diameter;
- $F_a$  is the axial force;
- $F_{\text{eff}}$  is the effective axial load, i.e. for a perfect pipe the axial load additional to the end-cap forces induced by internal and external pressures;
- $f_u$  is the tensile strength of a representative tensile specimen;
- $k_a$  is the burst strength factor, having the numerical value 1.0 for quenched and tempered (martensitic structure) or 13Cr products and 2.0 for as-rolled and normalized products based on available test data, and the default value set to 2.0 where the value has not been measured; the value of  $k_a$  can be established for a specific pipe material based on testing;
- $n$  is the dimensionless hardening index used to obtain a curve fit (see B.2.3.3) of the true stress-strain curve derived from the uniaxial tensile test;
- $p_{iR}$  is the internal pressure at ductile rupture of an end-capped pipe;
- $p_{iRa}$  is the  $p_{iR}$  adjusted for axial load and external pressure;
- $p_o$  is the external pressure;
- $t$  is the average pipe wall thickness;
- $t_{\text{min}}$  is the minimum pipe wall thickness disregarding crack-like imperfections.

Equation (B.6) is illustrated in Figure B.2, together with the exact equation.



#### Key

- X effective axial force ( $F_{\text{eff}}/F_{\text{uts}}$ )
- Y pressure differential ( $(p_i - p_o)/p_{\text{ref}}$ )
- 1 rupture (exact)
- 2 rupture [Equation (B.6)]
- 3 transition
- 4 necking [Equation (B.18)]
- 5 wrinkling

**Figure B.2—Illustration of the Effect of Effective Axial Force and External Pressure on Ductile Rupture**

Under capped-end conditions, the effective axial load is zero and Equation (B.6) reduces to Equation (B.1).

The rupture equation is valid; i.e. rupture occurs before necking, when:

$$F_{\text{eff}}/F_{\text{uts}} \leq (\sqrt{3}/2)^{1-n} \quad (\text{B.17})$$

#### B.2.4.3 Ductile Necking under Combined Loads

In the presence of internal and external pressure the general equation for ductile necking is as follows:

$$F_{\text{eff}} = F_{\text{uts}} [1 - k_N [(p_i - p_o)/p_{\text{ref M}}]^2]^{1/2} \quad (\text{B.18})$$

where

$$F_a = \pi t (D - t) \sigma_a \quad (\text{B.19})$$

$$F_{\text{eff}} = F_a + p_o \pi t (D - t) - p_M t (D - t)/[t_{\text{dr}} (D - t_{\text{dr}})] \pi/4 (D - 2 t_{\text{dr}})^2 \quad (\text{B.20})$$

$$F_{\text{uts}} = \pi t (D - t) f_u \quad (\text{B.21})$$

$$p_{\text{uts}} = 2 t_{\text{dr}} f_u / (D - t_{\text{dr}}) \quad (\text{B.22})$$

$$p_{\text{ref M}} = (2/\sqrt{3})^{1+n} (1/2)^n p_{\text{uts}} \quad (\text{B.23})$$

$$k_N = 4^{(1-n)} - 3^{(1-n)} \quad (\text{B.24})$$

Under zero pressure conditions, the effective axial load equals the true axial load, and Equation (B.18) for the maximum axial load reverts to the ultimate tensile strength.

The necking equation is valid; i.e. necking occurs before rupture, when:

$$(p_i - p_o)/p_{\text{ref M}} \leq (1/2)^{1-n} \quad (\text{B.25})$$

#### B.2.4.4 Boundary between Rupture and Necking

Comparing Equations (B.6) and (B.18) reveals that the necking criterion is reached earlier than the rupture criterion when:

$$F_{\text{eff}}/F_{\text{uts}} \geq (3/2) (p_i - p_o)/p_{\text{uts}} \quad (\text{B.26})$$

and this criterion (also shown in Figure B.2) describes the boundary between rupture and necking.

#### B.2.4.5 Axisymmetric Wrinkling under Combined Loads

Figure B.2 shows that, in the axial compression range, i.e. for negative values of the effective axial load, Equation (B.6) is conservative when compared to both the exact rupture equation and the local pipe wall buckling limit, called wrinkling. Although it would be easy to construct an equation such as Equation (B.7) with a different factor  $k_R$  that would better fit the exact rupture curve in the effective axial compression range, it is perceived such a separate equation would not have great practical impact.



## B.2.5 Ductile Rupture and Necking Design Equations

Minimum ductile rupture and necking of a tube is defined by replacing  $t_{dr}$  with  $k_{wall} t - k_a a_N$  and  $f_u$  with  $f_{umn}$  in the limit state Equations (B.1), (B.9) to (B.11), and (B.20) to (B.22):

$$p_{iR} = 2k_{dr}f_{umn}(k_{wall}t - k_a a_N)/[D - (k_{wall}t - k_a a_N)] \quad (B.27)$$

where

$a_N$  is the imperfection depth associated with a specified inspection threshold, i.e. the maximum depth of a crack-like imperfection that could reasonably be missed by the pipe inspection system [e.g. for a 5 % imperfection threshold inspection in a 12.7 mm (0.500 in.) wall thickness pipe,  $a_N = 0.635$  mm (0.025 in.)];

$D$  is the specified pipe outside diameter;

$f_{umn}$  is the specified minimum tensile strength;

$k_a$  is the burst strength factor, having the numerical value 1.0 for quenched and tempered (martensitic structure) or 13Cr products and 2.0 for as-rolled and normalized products based on available test data, and the default value set to 2.0 where the value has not been measured; the value of  $k_a$  can be established for a specific pipe material based on testing;

$k_{wall}$  is the factor to account for the specified manufacturing tolerance of the pipe wall; e.g. for a tolerance of -12.5 %,  $k_{wall} = 0.875$ ;

$p_{iR}$  is the internal pressure at ductile rupture of an end-capped pipe;

$t$  is the specified pipe wall thickness.

The factor  $k_{wall}$  addresses minimum pipe body wall thickness without considering imperfections. This value may be adjusted if minimum wall is guaranteed by a particular process or purchasing agreement.

Crack-like imperfections are accounted for by  $a_N$ . The term  $k_a a_N$  represents a further reduction in minimum wall thickness associated with crack-like imperfections that are outside the sensitivity setting of the inspection equipment and assumed coincident with the location of minimum wall thickness. This stacking of minimum wall thickness and a crack-like imperfection depends on the frequency of occurrence of thin wall and the frequency of occurrence of sharp-bottomed imperfections approaching the depth of the inspection threshold.

For the deterministic calculation of rupture pressure, it is necessary to calculate a conservative ductile rupture design pressure. In this case, the frequency of occurrence of the imperfection is set to 100 % and the imperfection depth equals the inspection threshold.

For the probabilistic calculation of rupture pressure, the depth of the imperfection still equals the depth of the inspection threshold, but the calculation takes account of the actual frequency of occurrence of thin wall and the actual frequency of occurrence of sharp-bottomed imperfections with depth comparable to the inspection threshold.

## B.3 Selection of a Ductile Rupture Model

### B.3.1 General

Six closed-form, analytical models were evaluated as candidates for the ductile rupture model. In the absence of a sharp-bottomed imperfection, the closed form candidates were as follows:

- ad-hoc Barlow equation

$$p_{iR} = 2f_u t / D \quad (\text{B.28})$$

- ad-hoc von Mises equation

$$p_{iR} = f_u (D^2 - d^2) / (\sqrt{3} D^2) \quad (\text{B.29})$$

- Klever-Stewart equation (see SPE 48329 and SPE 48330)

$$p_{iR} = 2k_{dr} f_u t_{\min} / (D - t_{\min}) \quad (\text{B.30})$$

- ad-hoc Paslay equation (see SPE 48327)

$$p_{iR} = 2f_u t_{\min} / (D - t_{\min}) \quad (\text{B.31})$$

- ad-hoc Moore equation

$$p_{iR} = f_u (D^2 - d^2) / (D^2 + d^2) \quad (\text{B.32})$$

Nadai equation

$$p_{iR} = 2f_u \ln[D / (D - 2t)] / \sqrt{3} \quad (\text{B.33})$$

where, in the above equations,

$D$  is the specified pipe outside diameter;

$d$  is the pipe inside diameter,  $d = D - 2t$ ;

$f_u$  is the tensile strength of a representative tensile specimen;

$k_{dr}$  is the correction factor based on pipe deformation and material strain hardening, having the numerical value  $[(1/2)^{n+1} + (1/\sqrt{3})^{n+1}]$ ;

$n$  is the dimensionless hardening index used to obtain a curve fit (see B.2.3.3) of the true stress-strain curve derived from the uniaxial tensile test;

$p_{iR}$  is the internal pressure at ductile rupture of end-capped pipe;

$t$  is the specified pipe wall thickness;

$t_{\min}$  is the actual minimum pipe wall thickness, disregarding crack-like imperfections.

For the Klever-Stewart model, the value selected for  $k_{dr}$  renders  $p_{IR}$  the average of rupture pressures predicted using Tresca yield condition and von Mises yield condition for end-capped pipes. It accounts for the material hardening and the pipe deformation up to rupture.

Of the many alternative rupture models existing in the published literature, the above list provides sufficient diversity to ensure an accurate final selection. The names corresponding to the models indicate either the developer of the model or the advocate for considering the model. The term “ad-hoc” is used for the first two models because they are generalizations of the Barlow and Mises yield equations, where yield strength has been replaced by tensile strength. There is no fundamental mechanistic justification or derivation for the ad-hoc models; only an appeal to their generalization from the yield equations. Likewise, the fourth and fifth candidates are ad-hoc models. Alternatively, the Klever-Stewart and Nadai equations can be derived from fundamental physical principles, specifically different levels of approximation of the equations of equilibrium. Through their fundamental derivations, the Nadai and Klever-Stewart equations depend on the pipe tensile strength. The Paslay equation can be shown to be a special case of the Klever-Stewart model.

In the forms shown here, these equations address internal pressure without external pressure. All equations are for capped-end pipe and therefore include axial tension acting on end caps of the pipe model, e.g. tension equal to the internal pressure times the bore area of the pipe.

The candidate models were compared with capped-end burst test data from full-sized casing and tubing. On the basis of that accuracy, a single rupture equation was recommended. The recommended model was then generalized to account for external pressure and axial load other than that from pressure on end caps, and to account for the influence of a sharp-bottomed imperfection (ductile rupture).

### B.3.2 Values of $n$ Used to Evaluate the Klever-Stewart Model

The Klever-Stewart model requires a stress-strain parameter,  $n$ , obtained by fitting the true stress-strain curve in the range of 2 % to the uniform strain (the strain at maximum load in a tensile test). Alternatively,  $n$  may be approximated as the true (logarithmic) strain at which a tensile stress-strain coupon experiences maximum load (maximum engineering stress). Whenever available, actually measured values for  $n$  were used in evaluating the model against rupture test data. Otherwise, the value of  $n$  used in the model is based on a regression fit of both round and strip stress-strain coupons. See Table B.3.

**Table B.3—Values of  $n$  Used to Evaluate the Klever-Stewart Model**

API Grade	Yield Strength psi	$n$
H40	40,000	0.137
J55	55,000	0.125
K55	55,000	0.125
M65	65,000	0.117
N80	80,000	0.104
L80	80,000	0.104
C90	90,000	0.096
R95	95,000	0.092
T95	95,000	0.092
P110	110,000	0.080
Q125	125,000	0.068

## B.4 Pipe Rupture Datasets Used to Validate the Rupture Models

One hundred six data points from full-scale pipe rupture tests under capped-end conditions were donated from three industry sources in seven different datasets as listed below. Five of the datasets contain pipe failures with measured wall thickness, whereas two of the datasets contain pipe failures with specified wall thickness. In general, the evaluations of model accuracy were not sensitive to the manner in which wall thickness was reported. All tests reported the specified pipe outside diameter and the measured tensile strength. The lists below indicate the source company and dataset.

### Capped-end rupture datasets with measured pipe wall thickness:

- Shell Btest1,
- Hydril Measured Wall,
- Grant Prideco,
- Shell Super Duplex,
- Shell Not-Worn.

### Capped-end rupture datasets with specified pipe wall thickness:

- Shell Pipeline,
- Hydril Nominal.

All tubes are believed to have been tested in ductile condition and without the presence of sharp, crack-like imperfections. The absence of sharp imperfections is understandable, as the pipes were well inspected, and the frequency of occurrence of small (below inspection threshold) sharp-bottomed imperfections is very low.

## B.5 Comparison between the Different Rupture Models and Pipe Rupture Data under Capped-end Conditions

Table B.4 compares the rupture models with the donated pipe rupture data. For each data point used in the comparison, the ratio of the actual rupture test pressure to the prediction of the model evaluated at the thinnest reported section of pipe wall thickness is reported. All calculations use the specified pipe diameters because only the specified diameters were reported. All calculations use the measured tensile strengths of the samples. For each set of comparisons, the mean and the standard deviation are also reported.

**Table B.4—Comparison of Rupture Predictions for Candidate Models**

Data Source			(1)	(2)	(3)	(4)	(5)	(6)
			Ad-hoc Barlow	Ad-hoc Mises	Klever-Stewart	Paslay	Moore	Nadai
			Actual/ Predicted	Actual/ Predicted	Actual/ Predicted	Actual/ Predicted	Actual/ Predicted	Actual/ Predicted
Shell-Pipeline Nominal Wall	28 data	Mean	1.071	1.021	0.991	0.973	0.984	1.172
		Stdev	0.059	0.085	0.036	0.035	0.038	0.117
		COV, %	5.5	8.4	3.6	3.6	3.9	10.0
Hydril Data Nominal Wall	11 data	Mean	1.092	1.023	0.997	1.010	1.017	0.978
		Stdev	0.052	0.052	0.060	0.061	0.058	0.079
		COV, %	4.8	5.0	6.0	6.0	5.7	8.0
Shell Btest1 Measured Wall	18 data	Mean	1.125	1.052	1.014	1.042	1.050	1.043
		Stdev	0.051	0.080	0.029	0.030	0.032	0.102
		COV, %	4.5	7.6	2.9	2.9	3.0	9.8
Hydril Data Measured Wall	5 data	Mean	1.150	1.052	1.046	1.086	1.090	1.086
		Stdev	0.013	0.012	0.013	0.013	0.013	0.067
		COV, %	1.1	1.1	1.2	1.2	1.2	6.2
Not-Worn Pipe Dataset	2 data	Mean	1.075	Model eliminated	0.979	1.021	Model eliminated	1.003
		Stdev	0.025		0.010	0.007		0.006
		COV, %	2.3		1.0	0.7		0.6
Super Duplex Dataset	4 data	Mean	0.990		0.982	0.940		1.044
		Stdev	0.008		0.009	0.008		0.049
		COV, %	0.8		0.9	0.9		4.7
Grant Prideco Measured Wall	38 data	Mean			1.026	1.062		
		Stdev			0.041	0.043		
		COV, %			4.0	4.0		
Average mean			1.09	1.03	1.01	1.03	1.02	1.09
Average COV, %			4.4	7.0	3.6	3.6	3.7	8.8

Table B.4 also reports the average of all the means and the average of all the coefficients of variance (COVs). In some cases, the comparison between equation and test data was stopped when progress was obtained with a more accurate and fundamental model.

Table B.5 shows the average of the means from the above comparisons (weighted according to the amount of data in each population) and the corresponding standard deviation between means. This was calculated to provide a measure of the scatter in the predictive ability of each equation to represent the mean rupture pressure of the population.

**Table B.5—Illustration of Equation Performance over Population**

Parameter	(1)	(2)	(3)	(4)	(5)	(6)
	Ad-hoc Barlow	Ad-hoc Mises	Klever- Stewart	Paslay	Moore	Nadai
	Actual/ Predicted	Actual/ Predicted	Actual/ Predicted	Actual/ Predicted	Actual/ Predicted	Actual/ Predicted
Mean of means	1.08	1.04	1.00	1.02	1.04	1.05
Stdev of means	0.050	0.015	0.023	0.047	0.039	0.063
COV of means	4.6 %	1.5 %	2.3 %	4.6 %	3.8 %	5.9 %

Based on the results in Tables B.4 and B.5, the Klever-Stewart equation was selected to predict ductile rupture.

## B.6 Comparison between the Recommended Rupture Model and Pipe Rupture Data under Capped-end Conditions

### B.6.1 General

Table B.6 compares the measured and predicted rupture pressure from the Klever-Stewart model when all the capped-end test data from the sets listed in B.4 are combined into a single dataset. The mean ratio of actual to predicted is 1.004, the standard deviation is 0.047, and the coefficient of variation is 4.7 %.

Figure B.3 summarizes the same data in graphical form.

**Table B.6—Statistical Evaluation of Klever-Stewart Rupture Model for All Data**

Test Number	Measured Yield Strength psi	Measured Ultimate Tensile Strength psi	<i>n</i> Value	Measured Outer Diameter in.	Measured Minimum Wall Thickness in.	<i>D/t</i> Ratio	Measured Rupture Pressure psi	Calculated Rupture Pressure psi	Actual Burst/ Predicted Burst	Dataset
1	97,150	120,350	0.18 <sup>a</sup>	10.709	0.535	20.00	12,050	12,217	0.986	Super Duplex
2	97,150	121,945	0.21 <sup>a</sup>	10.709	0.539	19.85	12,108	12,246	0.989	Super Duplex
3	79,750	109,765	0.20 <sup>a</sup>	10.709	0.539	19.85	10,919	11,091	0.984	Super Duplex
4	97,150	122,670	0.18 <sup>a</sup>	10.709	0.539	19.85	12,166	12,549	0.969	Super Duplex
5	121,800	142,100	0.05 <sup>a</sup>	13.465	0.524	25.71	11,687	12,021	0.972	Not-Worn Set
6	134,850	155,150	0.06 <sup>a</sup>	9.922	0.606	16.36	20,735	20,996	0.988	Not-Worn Set

7	67,600	107,000	0.11	7.064	0.352	20.07	11,270	11,268	1.000	Shell Btest1
8	100,900	109,400	0.09	3.530	0.254	13.90	17,300	17,319	0.999	Shell Btest1
9	131,400	144,200	0.06	7.805	0.575	13.57	25,210	23,776	1.060	Shell Btest1
10	120,700	131,000	0.07	7.068	0.524	13.49	22,090	21,631	1.021	Shell Btest1
11	88,600	105,000	0.10	7.076	0.408	17.34	13,370	13,038	1.025	Shell Btest1
12	122,600	133,200	0.07	7.076	0.407	17.39	17,190	16,779	1.024	Shell Btest1
13	92,900	104,000	0.09	9.724	0.388	25.06	8860	8790	1.008	Shell Btest1
14	88,000	102,000	0.10	9.929	0.531	18.70	11,830	11,692	1.012	Shell Btest1
15	87,500	106,200	0.10	3.502	0.567	6.18	42,740	41,611	1.027	Shell Btest1
16	100,300	121,200	0.09	2.649	0.154	17.20	16,440	15,270	1.077	Shell Btest1
17	112,600	130,000	0.08	7.070	0.473	14.95	19,740	19,143	1.031	Shell Btest1
18	131,400	144,200	0.06	7.805	0.577	13.53	24,900	23,865	1.043	Shell Btest1
19	131,400	144,200	0.06	7.089	0.585	12.12	25,900	26,889	0.963	Shell Btest1
20	84,900	102,000	0.10	11.134	0.560	19.88	11,140	10,942	1.018	Shell Btest1

**Table B.6—Statistical Evaluation of Klever-Stewart Rupture Model for All Data** *(continued)*

Test Number	Measured Yield Strength psi	Measured Ultimate Tensile Strength psi	<i>n</i> Value	Measured Outer Diameter in.	Measured Minimum Wall Thickness in.	<i>D/t</i> Ratio	Measured Rupture Pressure psi	Calculated Rupture Pressure psi	Actual Burst/ Predicted Burst	Dataset
21	97,600	116,000	0.09	2.776	0.333	8.34	33,540	32,232	1.041	Shell Btest1
22	72,300	80,700	0.11	2.382	0.124	19.21	9140	8920	1.025	Shell Btest1
23	127,000	147,000	0.07	3.524	0.282	12.50	25,980	26,451	0.982	Shell Btest1
24	145,000	149,000	0.05	3.514	0.287	12.24	26,800	27,661	0.969	Shell Btest1
25	128,700	140,000	0.06	9.625	0.538	17.89	17,770	17,161	1.035	Hydril M-Wall
26	128,700	140,000	0.06	9.625	0.531	18.13	17,720	16,925	1.047	Hydril M-Wall

27	119,000	143,000	0.07	16.000	0.857	18.67	17,700	16,675	1.061	Hydril M-Wall
28	119,000	143,000	0.07	16.000	0.870	18.39	17,700	16,942	1.045	Hydril M-Wall
29	119,000	143,000	0.07	16.000	0.870	18.39	18,160	16,942	1.072	Hydril M-Wall
30	120,670	134,130	0.07	7.056	0.511	13.81	23,080	21,595	1.069	Grant Prideco
31	120,670	134,130	0.07	7.063	0.533	13.25	22,600	22,576	1.001	Grant Prideco
32	120,670	134,130	0.07	7.054	0.528	13.36	21,870	22,378	0.977	Grant Prideco
33	120,670	134,130	0.07	7.049	0.527	13.38	21,960	22,349	0.983	Grant Prideco
34	120,670	134,130	0.07	7.060	0.543	13.00	21,030	23,046	0.913	Grant Prideco
35	120,670	134,130	0.07	7.048	0.524	13.45	22,080	22,215	0.994	Grant Prideco
36	129,800	142,700	0.06	7.695	0.490	15.70	20,190	20,104	1.004	Grant Prideco
37	129,800	142,700	0.06	7.691	0.491	15.66	20,060	20,159	0.995	Grant Prideco
38	129,800	142,700	0.06	7.700	0.496	15.52	20,195	20,353	0.992	Grant Prideco
39	129,800	142,700	0.06	7.694	0.468	16.44	19,900	19,145	1.039	Grant Prideco
40	129,800	142,700	0.06	7.694	0.476	16.16	20,055	19,494	1.029	Grant Prideco
41	129,800	142,700	0.06	7.698	0.502	15.33	20,230	20,622	0.981	Grant Prideco

**Table B.6—Statistical Evaluation of Klever-Stewart Rupture Model for All Data** *(continued)*

Test Number	Measured Yield Strength psi	Measured Ultimate Tensile Strength psi	<i>n</i> Value	Measured Outer Diameter in.	Measured Minimum Wall Thickness in.	<i>D/t</i> Ratio	Measured Rupture Pressure psi	Calculated Rupture Pressure psi	Actual Burst/ Predicted Burst	Dataset
42	124,000	132,700	0.07	7.663	0.472	16.24	19,208	17,991	1.068	Grant Prideco
43	124,000	132,700	0.07	7.679	0.474	16.20	19,338	18,032	1.072	Grant Prideco
44	128,700	144,800	0.06	9.746	0.539	18.08	17,985	17,551	1.025	Grant Prideco
45	128,700	144,800	0.06	9.729	0.533	18.25	17,975	17,376	1.034	Grant Prideco
46	128,700	144,800	0.06	9.730	0.535	18.19	18,025	17,443	1.033	Grant Prideco



47	128,700	144,800	0.06	9.724	0.536	18.14	18,080	17,489	1.034	Grant Prideco
48	128,700	144,800	0.06	9.722	0.525	18.52	17,835	17,113	1.042	Grant Prideco
49	128,700	144,800	0.06	9.738	0.542	17.97	18,005	17,669	1.019	Grant Prideco
50	105,900	125,350	0.08	9.660	0.542	17.82	16,500	15,252	1.082	Grant Prideco
51	133,900	144,810	0.06	9.953	0.620	16.05	20,853	19,969	1.044	Grant Prideco
52	133,900	144,810	0.06	9.931	0.620	16.02	20,631	20,016	1.031	Grant Prideco
53	124,355	138,976	0.07	11.831	0.509	23.24	13,300	12,908	1.030	Grant Prideco
54	124,355	138,976	0.07	11.837	0.497	23.82	12,900	12,583	1.025	Grant Prideco
55	124,355	138,976	0.07	11.840	0.482	24.56	12,950	12,184	1.063	Grant Prideco
56	124,355	138,976	0.07	11.823	0.527	22.43	13,550	13,395	1.012	Grant Prideco
57	127,150	136,000	0.07	13.456	0.497	27.07	11,523	10,791	1.068	Grant Prideco
58	127,150	136,000	0.07	13.473	0.509	26.47	11,740	11,047	1.063	Grant Prideco
59	127,150	136,000	0.07	13.443	0.495	27.16	11,447	10,756	1.064	Grant Prideco
60	127,150	136,000	0.07	13.465	0.489	27.54	11,840	10,603	1.117	Grant Prideco
61	127,150	136,000	0.07	13.451	0.505	26.64	11,414	10,975	1.040	Grant Prideco
62	127,150	136,000	0.07	13.459	0.504	26.70	11,914	10,946	1.088	Grant Prideco

**Table B.6—Statistical Evaluation of Klever-Stewart Rupture Model for All Data** *(continued)*

Test Number	Measured Yield Strength psi	Measured Ultimate Tensile Strength psi	<i>n</i> Value	Measured Outer Diameter in.	Measured Minimum Wall Thickness in.	<i>D/t</i> Ratio	Measured Rupture Pressure psi	Calculated Rupture Pressure psi	Actual Burst/ Predicted Burst	Dataset
63	93,450	107,400	0.09	13.682	0.626	21.86	11,226	10,475	1.072	Grant Prideco
64	94,450	107,400	0.09	13.673	0.617	22.16	9744	10,330	0.943	Grant Prideco
65	95,450	107,400	0.09	13.669	0.612	22.33	10,312	10,251	1.006	Grant Prideco
66	96,450	170,400	0.09	13.670	0.621	22.01	10,750	10,413	1.032	Grant Prideco

67	96,258	113,072	0.09	16.080	0.474	33.92	7330	6996	1.048	Grant Prideco
68	62,700	86,400	0.12	5.562	0.750	7.42	25,460	26,976	0.944	Shell Pipeline
69	62,700	86,400	0.12	5.562	0.750	7.42	25,800	26,976	0.956	Shell Pipeline
70	62,700	86,400	0.12	5.562	0.750	7.42	25,980	26,976	0.963	Shell Pipeline
71	62,700	86,400	0.12	5.562	0.750	7.42	26,320	26,976	0.976	Shell Pipeline
72	62,700	86,400	0.12	5.562	0.750	7.42	26,090	26,976	0.967	Shell Pipeline
73	61,600	76,100	0.12	5.562	0.750	7.42	24,930	23,747	1.050	Shell Pipeline
74	65,700	86,200	0.12	6.625	0.562	11.79	15,900	16,030	0.992	Shell Pipeline
75	65,700	86,200	0.12	6.625	0.562	11.79	15,870	16,030	0.990	Shell Pipeline
76	47,800	72,800	0.13	6.625	0.562	11.79	13,170	13,418	0.982	Shell Pipeline
77	47,800	72,800	0.13	6.625	0.562	11.79	13,270	13,418	0.989	Shell Pipeline
78	47,700	73,100	0.13	6.625	0.562	11.79	12,900	13,472	0.958	Shell Pipeline
79	47,700	73,100	0.13	6.625	0.562	11.79	12,940	13,472	0.960	Shell Pipeline
80	51,160	77,540	0.13	4.500	0.345	13.04	12,810	12,823	0.999	Shell Pipeline
81	51,160	77,540	0.13	4.500	0.345	13.04	12,800	12,823	0.998	Shell Pipeline
82	51,160	77,540	0.13	4.500	0.345	13.04	12,800	12,823	0.998	Shell Pipeline
83	74,500	90,500	0.11	12.750	0.750	17.00	11,370	11,398	0.998	Shell Pipeline

**Table B.6—Statistical Evaluation of Klever-Stewart Rupture Model for All Data** *(continued)*

Test Number	Measured Yield Strength psi	Measured Ultimate Tensile Strength psi	<i>n</i> Value	Measured Outer Diameter in.	Measured Minimum Wall Thickness in.	<i>D/t</i> Ratio	Measured Rupture Pressure psi	Calculated Rupture Pressure psi	Actual Burst/ Predicted Burst	Dataset
84	74,500	90,500	0.11	12.750	0.750	17.00	11,140	11,398	0.977	Shell Pipeline
85	74,500	90,500	0.11	12.750	0.750	17.00	11,250	11,398	0.987	Shell Pipeline
86	57,660	78,680	0.12	8.625	0.495	17.42	8710	9572	0.910	Shell Pipeline

87	57,660	78,680	0.12	8.625	0.495	17.42	8550	9572	0.893	Shell Pipeline
88	57,660	78,680	0.12	8.625	0.495	17.42	8640	9572	0.903	Shell Pipeline
89	43,990	73,740	0.13	8.625	0.507	17.01	8760	9140	0.958	Shell Pipeline
90	43,990	73,740	0.13	8.625	0.507	17.01	8770	9140	0.960	Shell Pipeline
91	43,990	73,740	0.13	8.625	0.507	17.01	8830	9140	0.966	Shell Pipeline
92	78,920	95,010	0.11	8.625	0.875	9.86	21,720	21,663	1.003	Shell Pipeline
93	78,920	95,010	0.11	8.625	0.875	9.86	21,930	21,663	1.012	Shell Pipeline
94	71,540	91,560	0.11	8.625	1.000	8.63	23,560	24,161	0.975	Shell Pipeline
95	71,540	91,560	0.11	8.625	1.000	8.63	23,690	24,161	0.981	Shell Pipeline
96	121,000	132,800	0.07	2.875	0.308	9.33	30,410	32,863	0.925	Hydril N-Wall
97	120,700	131,500	0.07	2.875	0.308	9.33	29,710	32,536	0.913	Hydril N-Wall
98	116,000	126,200	0.08	2.875	0.308	9.33	29,800	31,152	0.957	Hydril N-Wall
99	115,400	124,200	0.08	2.875	0.308	9.33	28,750	30,649	0.938	Hydril N-Wall
100	116,400	131,300	0.07	13.625	0.625	21.80	12,460	12,989	0.959	Hydril N-Wall
101	100,700	111,000	0.09	3.500	0.254	13.78	17,960	17,733	1.013	Hydril N-Wall
102	145,700	166,000	0.05	5.500	0.304	18.09	21,600	20,280	1.065	Hydril N-Wall
103	142,000	162,400	0.05	5.500	0.304	18.09	20,830	19,803	1.052	Hydril N-Wall
104	145,400	170,200	0.05	5.500	0.304	18.09	20,700	20,790	0.996	Hydril N-Wall

**Table B.6—Statistical Evaluation of Klever-Stewart Rupture Model for All Data** (continued)

Test Number	Measured Yield Strength psi	Measured Ultimate Tensile Strength psi	<i>n</i> Value	Measured Outer Diameter in.	Measured Minimum Wall Thickness in.	<i>D/t</i> Ratio	Measured Rupture Pressure psi	Calculated Rupture Pressure psi	Actual Burst/ Predicted Burst	Dataset
105	90,500	130,700	0.10	7.625	0.430	17.73	11,680	12,589	0.928	Hydril N-Wall
106	85,100	97,400	0.10	9.625	0.545	17.66	12,320	11,843	1.040	Hydril N-Wall
								Mean	1.006	

	Stdev	0.045	
	COV	0.045	
<sup>a</sup> These <i>n</i> -values denote the measure for the pipe tested to rupture; all other <i>n</i> -values are based on regression fit of OCTG stress-strain data; <i>n</i> -OCTG = 0.1693-0.000812 × yield (ksi) with yield being actual measured yield, ksi.			

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**Key**

X *D/t* ratio

Y ratio of actual to predicted

NOTE Mean = 1.006; COV = 4.5 %.

**Figure B.3—Ratio of Actual to Predicted Rupture Pressures for Klever-Stewart Model**

## B.6.2 Additional Pipe Rupture Data under Combined Pressure and Axial Tension

Only a small amount of data was available reporting measured pipe rupture pressure at axial load different from the capped-end load. These data are summarized in Table B.7 and compared with the predictions of the Klever-Stewart equation generalized to include the influence of axial force. Overall, the equation becomes more conservative at increasing axial tension. No data were available to evaluate the model at compressive axial load.

**Table B.7—Comparison of Rupture Equation (B.6) and Necking Equation (B.18) with Shell Mini-pipe<sup>[12]</sup> Test Data at Axial Tension Exceeding the Capped-end Load**

Test Number	Outside Diameter <i>in.</i>	Wall Thickness <i>in.</i>	Test		Check Equation (B.22)	Model		
			$F_{eff}/F_{uts}$	$p/p_{uts}$		$F_{eff}/F_{uts}$	$p/p_{uts}$ <sup>a</sup>	Actual/ Predicted
1	0.90	0.1065	1.029	0.000	N	1.000	0.000	1.029
2	0.90	0.1065	0.922	0.500	N	0.910	0.500	1.012
3	0.90	0.1065	0.000	0.922	R	0.000	1.027	0.966
4	0.90	0.1065	0.395	0.983	R	0.395	0.985	0.998
5	0.90	0.1065	0.468	0.965	R	0.468	0.967	0.998
6	0.90	0.1065	0.774	0.765	R	0.774	0.731	1.046
Mean								1.008
COV								0.025
<sup>a</sup> Italic numbers are considered given.								

In Table B.7, the effective axial tension is the amount of tension in excess of the load from pressure acting on the end caps of the pipe.

Table B.8 offers additional validation of the Klever-Stewart equation with rupture data for the special case of zero axial tension. The tests in this instance were conducted on ring specimens rather than tubes.

**Table B.8—Comparison of Rupture Equation and Test Data at Zero Axial Tension, Ring Tests**

Number	$f_u$ MPa	$n$	$D$ mm	$t_{\min}$ mm	$D/t_{\min}$	$P_{IR}$ MPa		Actual/ Predicted
						Test	Model	
1	563	0.13	914.2	22.20	41.18	26.30	25.99	1.012
2	563	0.13	914.2	22.20	41.18	26.40	25.99	1.016
3	563	0.13	914.2	15.54	58.83	18.70	19.27	0.971
4	563	0.13	914.2	15.76	58.00	19.50	19.55	0.997
5	563	0.13	914.2	11.54	79.19	14.80	14.24	1.040
6	563	0.13	914.2	10.43	87.62	13.00	12.85	1.012
7	563	0.13	914.2	6.88	132.84	8.56	8.43	1.015
8	563	0.13	914.2	6.66	137.27	8.08	8.16	0.990
9	563	0.13	914.2	6.88	132.84	8.22	8.43	0.975
Mean								1.003
COV								0.022

## B.7 The Role of Imperfections in the Ductile Rupture Equation

### B.7.1 General

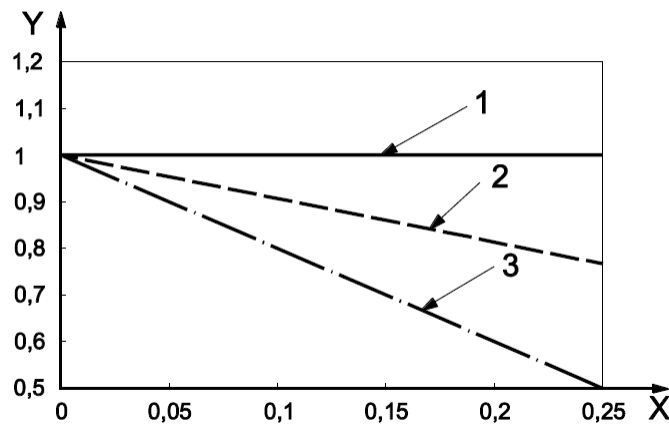
The ductile rupture equation is recommended only with pipe small wall eccentricity and inspection-threshold sized imperfection depth included in the equation. The equation should not be used without accounting for these imperfections. In the deterministic equation of rupture, thin-wall eccentricity is accounted for through the minimum possible physical wall thickness. In the probabilistic equation of rupture, thin-wall eccentricity is accounted for through the mean and standard deviation of the minimum wall thickness on a per length basis.

In a deterministic formulation of rupture, sharp-bottomed imperfections are accounted for through the maximum size imperfection that **can** pass undetected through the inspection threshold; that is, an imperfection with depth equal to the inspection threshold. It is assumed that each length of pipe has an imperfection of this depth. This conservative equation uses the worst-case possible frequency for the imperfection.

In a probabilistic formulation of rupture, the depth of the sharp-bottom imperfection is still set equal to the inspection threshold, but the occurrence of the imperfection is based on statistical observations. In this case, the penalty for the presence of the imperfection accounts for the mean and the standard deviation of the frequency of occurrence of sharp-bottomed imperfections in pipe that has been inspected. A typical frequency of occurrence of imperfections of all kinds during secondary inspection is 2 % to 5 %. But for sharp-bottomed imperfections, the frequency can be much less, for example 0.5 % to 0.05 %. This frequency can have significant impact on the probability of rupture at a particular pressure.

Figure B.4 is a conceptual illustration of the role that the sharp-bottomed imperfection plays in decreasing the rupture strength of the pipe.

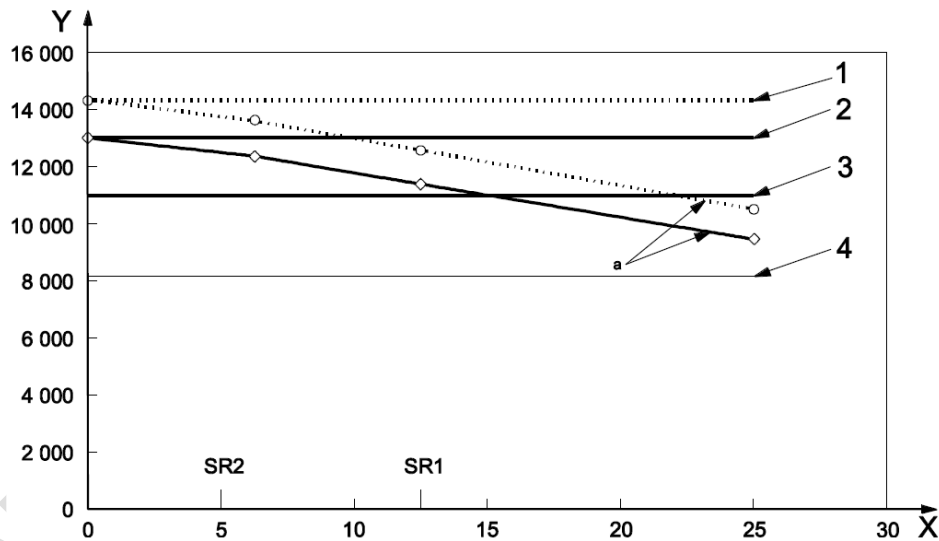
Figure B.5 is a more in-depth illustration of the role of the imperfection penalty for 9-5/8 inch OD, 53.5 lb/ft, P110 casing.



#### Key

- X crack depth,  $a/t$   
Y  $p_{crit}/p_{ref}$   
1 no cracks  
2 crack, high toughness  
3 crack, low toughness

**Figure B.4—Rupture De-rating for Cracks (Crack Assumed on Minimum Wall)**



#### Key

- X crack depth as a fraction of wall thickness, %  
Y internal pressure, psi  
SR1, SR2 Supplemental Requirements as defined in [API 5CT](#)  
1 mean rupture without a flaw  
2 minimum rupture without a flaw  
3 rating based on minimum yield  
4 historical design on yield divided by a design factor  
a influence of crack-like imperfections in pipe with good toughness

**Figure B.5—Influence of the Imperfection Penalty for 9-5/8 in., 53.5 lb/ft, P110 Casing**

To have a conservative deterministic calculation of the rupture pressure, the penalty for sharp imperfections should be applied on top of the physical thin-wall geometry. This is because this case does occur in pipe manufacturing, and the deterministic model should capture the minimum pressure that is possible for pipe that is produced in compliance with manufacturing specifications. If one seeks to address how rare this combination of imperfections is, then a probabilistic rupture equation should be applied.

## B.7.2 Fracture Mechanics Basis for Burst Strength Factor in Ductile Rupture Equation

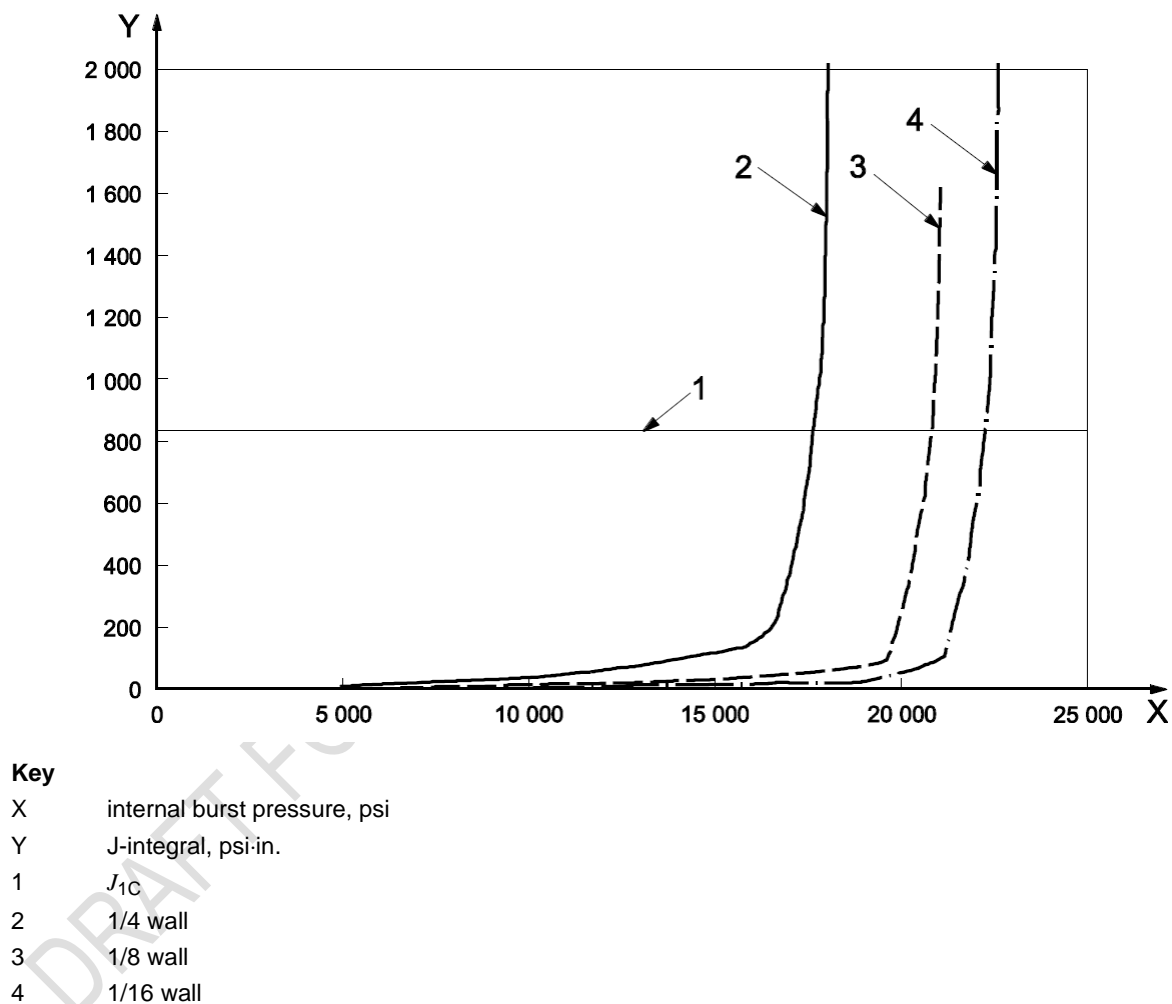
The burst strength factor,  $k_a$ , in the ductile rupture equation is an ad-hoc generalization of the Klever-Stewart ductile rupture equation. The burst strength factor is based on the fracture mechanics concept of crack growth as measured by the material property  $J_{1C}$  and the J-integral. To determine  $k_a$ , first  $J_{1C}$  values were measured for representative samples of pipe and coupling material. Table B.9 is an example of such data.

Table B.9—Typical  $J_{1C}$  Values

Grade	Test	Grade	Test
J55	900	13Cr95	1200
J55	925	13Cr95	1225
K55	501	C100	1100
K55	612	C100	1200
K55	640	CYP110	836
K55	875	P110	340
K55	1000	P110	360
L80	710	P110	360
L80	720	P110	402
L80	743	P110	418
L80	750	P110	450
L80	780	P110	455
L80	780	P110	490
L80	810	P110	520
L80	863	P110	550
L80	925	P110	580
L80	925	P110	585
L80	940	P110	585
L80	950	P110	640
L80	1020	P110	660
L80	1025	P110	660
N80	850	P110	675
N80	1025	P110	700
C90	610	P110	752

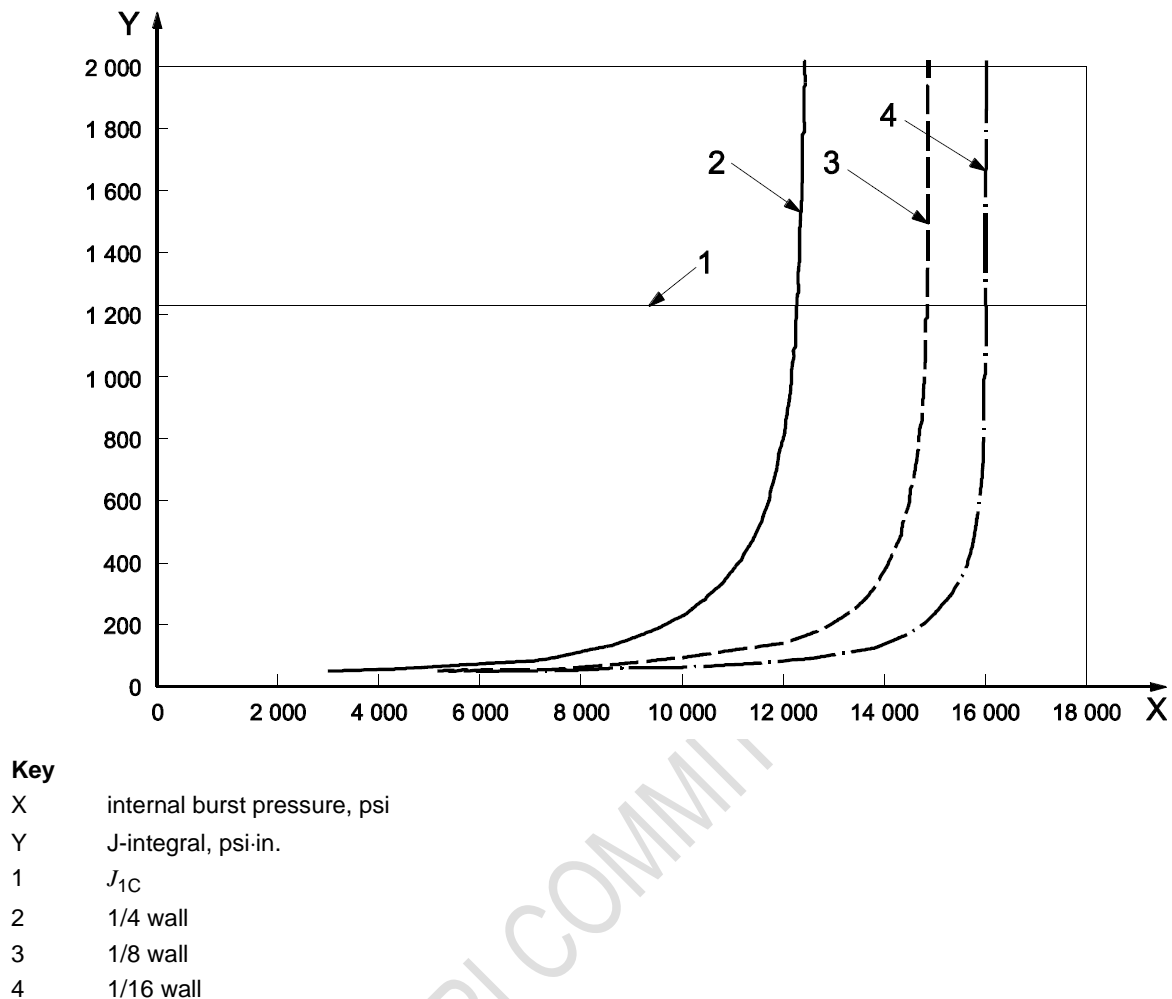
C90	682		P110	800
R95	472		P110	848
R95	485		P110	921

Next, for several pipes, the value of the J-Integral was calculated using the finite element method. The calculation of the J-integral is based on the crack depth, the internal pressure acting on the pipe, and the measured stress-strain curve of the same pipe material that supplied the sample for  $J_{1C}$  measurement. Figures B.6 and B.7 show typical curves of the J-integral. The figures also show an overlay of the measured  $J_{1C}$  value. Theoretically, the point where the J-integral equals  $J_{1C}$  represents the failure pressure of the pipe, and the decrease of the failure pressure with depth of the crack indicates the increasing influence of the crack depth.



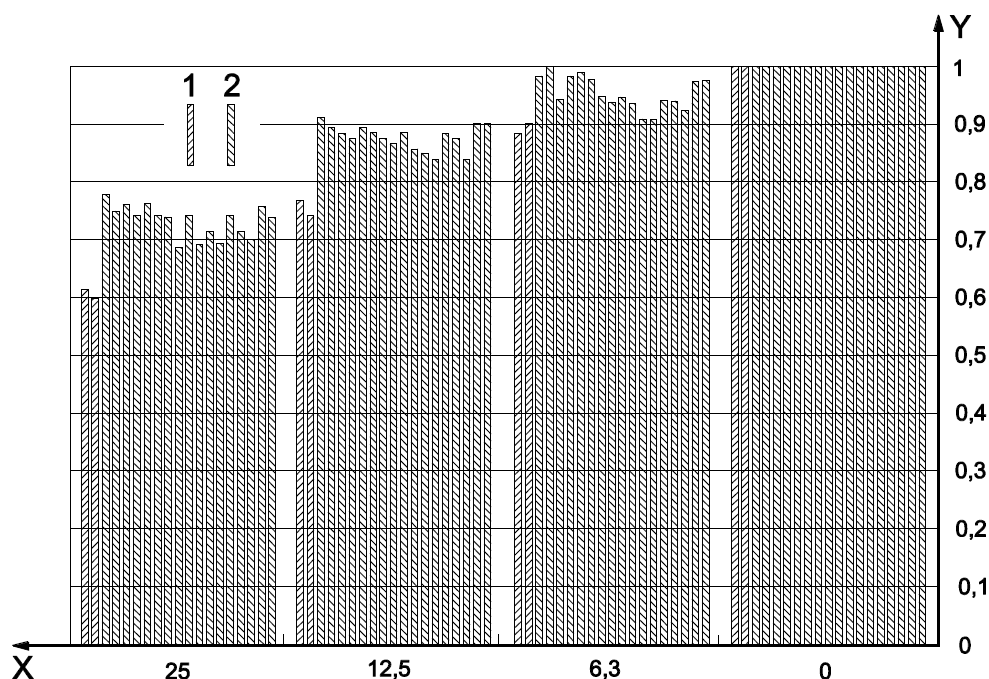
**Figure B.6—J-integral and  $J_{1C}$  for a P110 Pipe**





**Figure B.7—J-integral and  $J_{1C}$  for a 13-Cr-95 Pipe**

Based on the change of the intersection between  $J_{1C}$  and the J-integral with internal pressure for several calculations and measurements, Figure B.8 shows the predicted ratio of rupture pressure with an imperfection divided by rupture pressure without an imperfection, for different imperfection depths and grades of pipe. Figure B.8 shows that in general for the quenched and tempered pipes, the influence of the sharp imperfection is about equal to the imperfection depth. This is the origin of the burst strength parameter  $k_a = 1$  for quenched and tempered pipes in the rupture equation. However, Figure B.8 also shows that for some pipes, such as J55 and K55, the influence of the sharp imperfection is approximately equal to twice the imperfection depth. For this reason, when the toughness behavior of the pipe is unknown, the burst strength parameter  $k_a = 2$  is used in the **Kleiver-Stewart ductile rupture equation**.



#### Key

- X flaw depth/nominal wall thickness, %
- Y burst pressure with flaw/burst pressure without flaw
- 1 non-Q&T (Grade K)

#### 2 Q&T (various grades)

Figure B.8—Influence of Crack Depth on Ductile Rupture

## B.8 Template for Probabilistic Calculation of Ductile Rupture Strength

### B.8.1 General

The actual ductile rupture pressure for a given pipe is uncertain because many of the factors affecting it are random. A deterministic calculation of the rupture pressure assumes that all of the factors are known with absolute certainty and the equations used to calculate the strength are exact. The calculation assumes that if the pressure exceeds the calculated strength, it ruptures. In this case, a given set of input parameters corresponds to a single, deterministic predicted pressure. To ensure that the prediction is conservative and errs on the safe side, the deterministic calculation should use the worst-case allowable set of input parameters for a pipe in compliance with its product specifications. Of course, this is not reality. The actual ductile rupture pressure could be above or below the predicted value. In a design situation, a set of worst-case input parameters is used to calculate a lower bound pressure.

In contrast to the deterministic equation, a probabilistic estimate of the strength accounts for the uncertainties in the input parameters and results in a relation between the applied pressure and the rupture probability. In this manner, the calculation provides the likely ductile rupture pressure instead of the worst possible ductile rupture pressure.

Why use a probabilistic model for pipe ductile rupture strength? A simple universal design factor can result in inconsistent failure probabilities or inconsistent risks. A probabilistic strength calculation can be used to

justify a low economical design when the failure economics is low and a higher failure probability can be tolerated. In contrast, it can be used to produce a highly reliable design when the economics and safety consequences of failure are large. The goal of a probabilistic model of ductile rupture can be to determine the failure probability, either with deterministic loads or in conjunction with a randomized load model.

The following paragraphs briefly summarize the steps needed to develop a simple probabilistic model for ductile rupture. These steps follow the generalized procedure:

determine the failure mode (in this case ductile rupture);

determine the physics of failure and express failure mathematically as a limit state function,  $g(\bar{x})$ , where the component fails whenever  $g(\bar{x}) < 0$ ;

determine uncertainty models for the variables,  $\bar{x}$  in the limit state, where possible basing these models on statistical analysis of these parameters;

finally, estimate the failure probability as the probability that  $g(\bar{x}) < 0$ ; in the case summarized, this can be estimated using Monte Carlo simulation, or first- and second-order reliability methods (FORM/SORM)—in the steps below, FORM is demonstrated.

In the pipe ductile rupture case, a possible limit state could be  $g(\bar{x}) = R(\bar{x}) - p_a(\bar{x})$ , where  $R(\bar{x})$  is the equation used to calculate the rupture pressure from size and strength parameters, and  $p_a(\bar{x})$  is the applied pressure. The limit state function,  $g(\bar{x})$ , depends on a vector consisting of all the significant random parameters affecting failure. These random variables could include material properties such as yield and tensile strength, geometric properties such as metal thickness and loads such as the lifetime maximum applied pressure, or the uncertainty in model idealization. The limit state divides the space of all possibilities into two sets: the safe set, where failure does not occur, and the set where failure occurs.

The limit state input parameters,  $\bar{x}$ , are modelled as random variables. In the case where all the parameters are mutually independent, each parameter is assigned an appropriate probability distribution function. These distribution functions should be based on a statistical analysis of the measurements of the parameters. Input parameters used to model the idealization uncertainties should be based where possible on measurements of actual and predicted loads or strengths.

With the limit state and the probabilistic models for each of the input parameters, the probability of failure,  $\phi_f$ , is determined as follows:

$$\phi_f = \int_{g(\bar{x}) < 0} f(\bar{x}) d\bar{x} \quad (B.34)$$

where

$f(\bar{x})$  is the joint probability density function of the variables in  $\bar{x}$ ;

$g(\bar{x})$  is the limit state function;

$\bar{x}$  is a vector of random variables.

To calculate the probability of ductile rupture at a particular pressure, a closed-form solution of the probability integral usually is not possible. In practice, methods other than the direct integration are used to estimate the failure probability. The most familiar method is Monte Carlo simulation, which is not usually recommended for rare events ( $P < 10^{-3}$ ) such as pipe ductile rupture because of the large number of individual calculations that are necessary to estimate a small failure probability. In cases where the probability is small, there are other methods such as FORM/SORM. The Gaussian central moment method, summarized below, demonstrates one such method to estimate rupture probability. This method is not too accurate; however, it can be used to approximate failure probabilities and can be used to investigate the sensitivity of a given design to various input parameters. Furthermore, unlike FORM/SORM or simulation, the Gaussian central moment method depends only on the mean and the standard deviation of the various input variables, and as a result, it is easy to apply.

The probabilistic rupture pressure is based on the deterministic model for rupture pressure,  $p_{iR}$ , including the effect of sharp imperfections, where  $p_{iR}$  is the internal pressure at ductile rupture of an end-capped pipe:

$$p_{iR} = 2k_{dr}f_u(t_{min} - k_a a_N) / [D - (t_{min} - k_a a_N)] \quad (B.35)$$

where

- $a_N$  is the imperfection depth associated with a specified inspection threshold, i.e. the maximum depth of a crack-like imperfection that could reasonably be missed by the pipe inspection system;  
e.g. for a 5 % imperfection threshold inspection in a 12.7 mm (0.500 in.) wall thickness pipe,  $a_N = 0.635$  mm (0.025 in.);
- $D$  is the specified pipe outside diameter;
- $f_u$  is the tensile strength of a representative tensile specimen;
- $k_a$  is the burst strength factor, having the numerical value 1.0 for quenched and tempered (martensitic structure) or 13Cr products and 2.0 for as-rolled and normalized products based on available test data, and the default value set to 2.0 where the value has not been measured; the value of  $k_a$  can be established for a specific pipe material based on testing;
- $k_{dr}$  is the correction factor based on pipe deformation and material strain hardening, having the numerical value  $[(1/2)^{n+1} + (1/\sqrt{3})^{n+1}]$ ;
- $n$  is the dimensionless hardening index used to obtain a curve fit (see B.2.3.3) of the true stress-strain curve derived from the uniaxial tensile test;
- $t_{min}$  is the actual minimum pipe wall thickness disregarding crack-like imperfections.

The rupture pressure,  $p_{iR}$  is used with the applied pressure,  $p_i$ , to form the limit state function:

$$g = C_{iR}p_{iR}(f_u, n, t, D) - p_i \quad (B.36)$$

where

$C_{iR}$  is the random variable that represents model uncertainty.

This function is less than zero when the applied pressure,  $p_i$ , exceeds the ductile rupture resistance,  $C_{iR}p_{iR}(f_u, n, t, D)$ .

Using this limit state, the ductile rupture probability can be estimated using closed-form equations from mean value FORM:

$$\bar{g} = \bar{p}_i - \bar{C}_{iR} \bar{p}_{iR}(\bar{f}_u, \bar{n}, \bar{t}, \bar{D}) \quad (\text{B.37})$$

$$s_g^2 = s_p^2 \left( \frac{dg}{dp_i} \right)_{\bar{x}=\bar{X}}^2 + s_u^2 \left( \frac{dg}{df_u} \right)_{\bar{x}=\bar{X}}^2 + \dots + s_D^2 \left( \frac{dg}{dD} \right)_{\bar{x}=\bar{X}}^2 \quad (\text{B.38})$$

$$\beta = \frac{\bar{g}}{s_g} \quad (\text{B.39})$$

$$p_{\text{crude}} = \Phi(-\beta) \quad (\text{B.40})$$

where

the “barred” variables are the means;

the  $s_x^2$  are the estimated variances (squared standard deviations) of the random variables;

the derivatives  $dg/dx$  are evaluated at the mean values for  $f_u, n, t, \dots$ , and so forth;

$\beta$  is the crude, first-order reliability index;

$\Phi(-\beta)$  is the cumulative probability function of a standard unit normal random variable (mean = 0 and standard deviation = 1) evaluated at  $-\beta$ .

This estimated probability corresponds to a length of pipe that has a significant defect. The ductile rupture equation includes the effect of a defect with a depth equal to the inspection notch depth,  $a_N$ . This is the largest defect that should be expected in an inspected length of pipe, depending on the quality of the inspection. Larger defects are possible because inspection could miss a significant defect. However, in general not every length has a defect as deep or as sharp-bottomed as assumed by this equation. Consequentially, when using this formulation, it is important to consider the effect of imperfection frequency; for example, a low imperfection frequency expected in a run of pipe where a high-quality inspection was used. In this case, almost all lengths of pipe with defects greater than the threshold are culled, and only a few lengths will have any significant defect.

## B.8.2 Approach to Random Variables in the Probabilistic Rupture Equation

In the probabilistic formulation of rupture pressure, there are four variables that are treated as random, such that the mean and the standard deviation (or coefficient of variation) of these variables impact the probability of rupture at a particular pressure:

the pipe ultimate tensile strength;

the pipe diameter;

the pipe physical (no imperfection) minimum wall thickness;

the frequency at which sharp-bottomed imperfections occur in pipe that has received primary inspection.

At the same time, there are three variables that are treated as fixed (deterministic) in the probabilistic formulation:

the stress-strain curve parameter  $n$ ;

the material toughness parameter  $m$ ;

the maximum imperfection depth, set equal to the inspection threshold setting  $a_N$ .

It is appropriate that care be used to select the frequency of sharp-bottomed imperfections in the probabilistic treatment of the rupture equation. First, this frequency should be set to the reject rate of sharp-bottomed imperfections and not to the reject rate including round-bottom imperfections. Second, the value of this frequency can depend on the equipment and methods used for the combination of primary and secondary inspection.

If pipe is first inspected (by the mill) to SR 2 (5 %) depth and then re-inspected to SR 1 (12.5 %), the imperfection depth should be set equal to the inspection threshold used for the SR 1 inspection, but the frequency in this case is less than in the case where SR 2 depth is used for the second inspection.

Alternatively, if SR 2 also is used for the second inspection, then the mean and standard deviation of the imperfection frequency should be based on the frequency observed at the second inspection, and the imperfection depth used in the probabilistic rupture equations should be set equal to the inspection threshold used in the second SR 2 inspection.

When the type of inspection equipment is changed for either the first or second inspection, this will usually result in a different mean and standard deviation of imperfection frequency in the probabilistic rupture calculations.

The imperfection frequency is an important parameter in the probabilistic ductile rupture calculation. For the deterministic ductile rupture calculation, the frequency of occurrence of the imperfection is 100 %, and the penalty for the imperfection is severe and conservative. For the probabilistic calculation, the penalty for the imperfection only occurs a small part of the time.

### **B.8.3 Approach to Probable Imperfection Depth and Frequency**

The inspection threshold allows passage of imperfections equal or smaller in size than the setting of the inspection threshold. The threshold setting usually is slightly smaller than the imperfection depth of the inspection, but this depends on the setting and line speed of individual inspection equipment. Even with perfect imperfection detection, it is important to recognize that the threshold setting allows pipe imperfections to pass when the imperfection depths are equal or smaller than the depth of the threshold setting.

There are two potential limits to the way that imperfections smaller than the inspection threshold impact the rupture strength of the pipe. At one extreme, rarely occurring, large imperfections equal in size to the threshold setting impose the largest penalty against rupture strength. At the other extreme, frequently occurring imperfections much smaller than the inspection threshold setting do impact the rupture strength, not due to their depth but due to their frequency.

This section provides a comparison of the likely impact of imperfection depth compared with imperfection frequency. It is found that the rare deep imperfection (equal in depth to the inspection threshold) tends to

have more dominant impact compared with the smaller more frequent imperfections. On the strength of this premise, the probabilistic equation for ductile pipe rupture has based the rupture strength of the pipe on the probable wall thickness combined with a penalty for the probable imperfection with depth set equal to the inspection threshold setting (to be conservative) and with frequency set equal to the frequency at which sharp crack-like or seam lap imperfections are detected and proven during secondary inspection.

What effect does the shape of the imperfection frequency distribution below the inspection limit have on the probability of failure? As a first stage of the answer to this question, the distribution of the maximum defect size was investigated to determine how the shape of the imperfection size distribution below the inspection threshold affects the largest expected imperfection size in a string.

The following assumptions were made.

After the inspection, the distribution of defects is truncated at the threshold. In this case, the numerical study assumes this is 5 %.

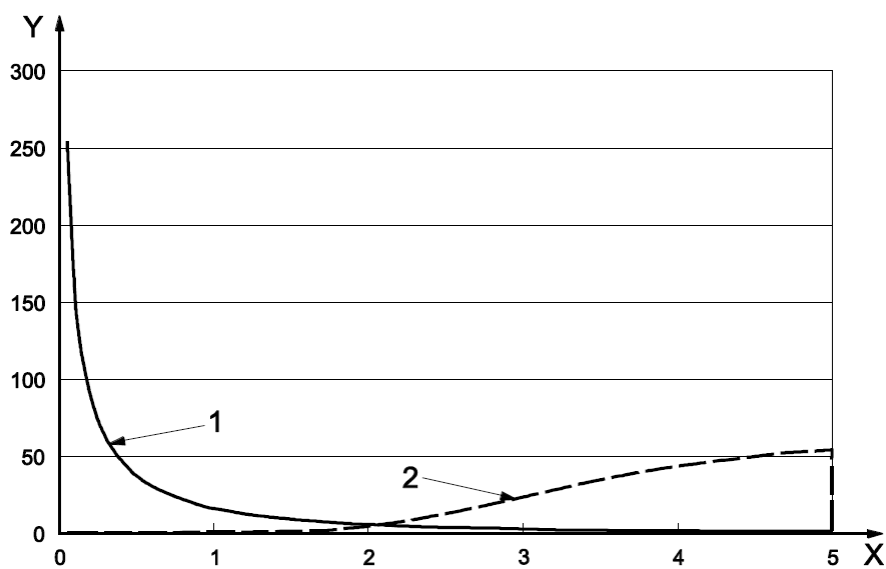
The imperfection frequency of defects greater than 5 % is 3 %; there is a 3 % probability that the deepest imperfection in an uninspected length of casing is deeper than 5 % of the specified wall.

The distribution of imperfections is modelled by a Weibull distribution. This distribution is chosen only for convenience, because its shape can be easily modified using the distribution's shape parameter.

The following charts compare the shape of the distribution of imperfection size with the distribution of the deepest extreme imperfection in 50 lengths. These four charts parameterize the shape of the distribution of imperfection size in an arbitrary length by the  $b$ -parameter of the Weibull distribution used to model each. This distribution is truncated at a depth of 5 % specified. As  $b$  increases from 0.5 to 4, the coefficient of variation of the distribution decreases from 224 % to 28 %. The probability density functions for these are shown by the solid line in Figures B.9 through B.12.

The dashed lines in the charts are the probability densities for the deepest defect in 50 lengths. It is related to the probability density and cumulative probability functions of the underlying distribution of the deepest imperfection in an arbitrary length. This distribution is also truncated at a depth of 5 % of specified. These extreme-value probability density functions show that the majority of the probability weight is toward the upper limit of 5 %. Because the underlying distribution of imperfection size is truncated at the imperfection threshold, the most likely depth imperfection depth is equal to the imperfection threshold (5 %). This effect becomes more pronounced as the  $b$  parameter in the underlying distribution increases.

Even when the probability distribution is biased toward 0 % ( $b = 0.5$ ), the most likely deepest defect is 5 % of specified. In this case, the distribution is evenly spread over the interval between 3.5 % and 5 %.

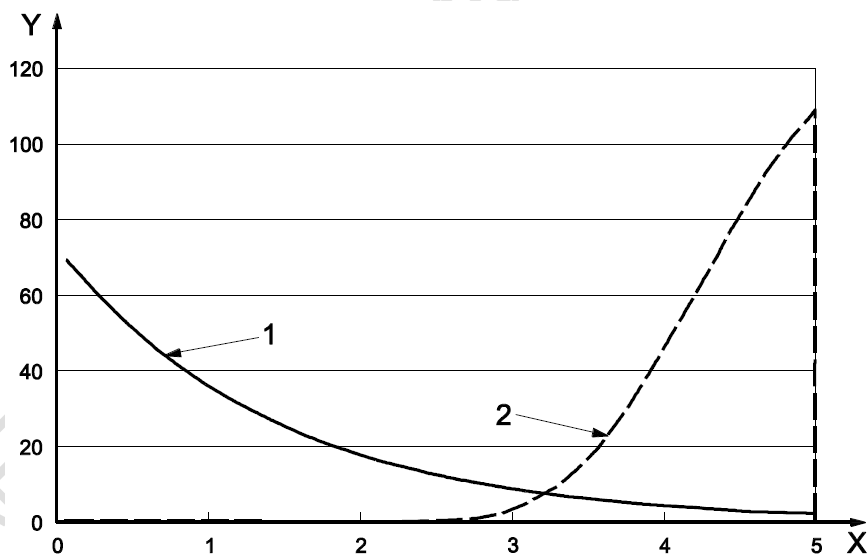


**Key**

X defect, % of nominal      1 single joint  
Y PDF (probability density function)      2 maximum of 50 joints

NOTE  $b = 0.5$ ; mean = 0.81 %; standard deviation = 1.82 %.

**Figure B.9—Distribution Bias to Very Small Defects—Very High COV**



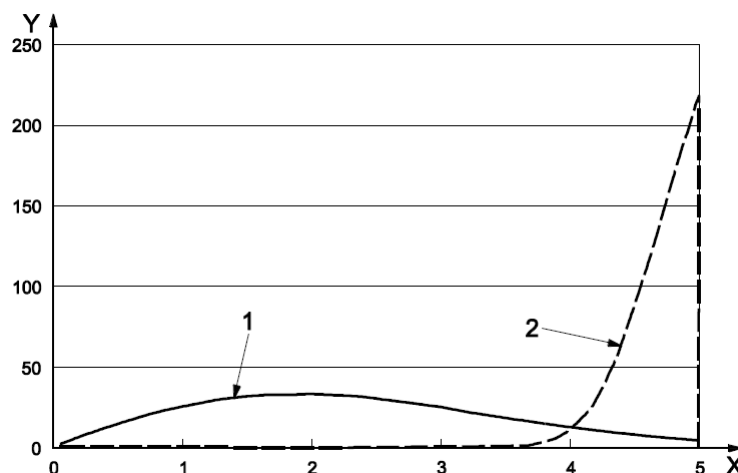
**Key**

X defect, % of nominal      1 single joint  
Y PDF (probability density function)      2 maximum of 50 joints

NOTE  $b = 1.0$ ; mean = 1.43 %; standard deviation = 1.43 %.

**Figure B.10—Exponential Distribution of Defect Sizes—Large COV**





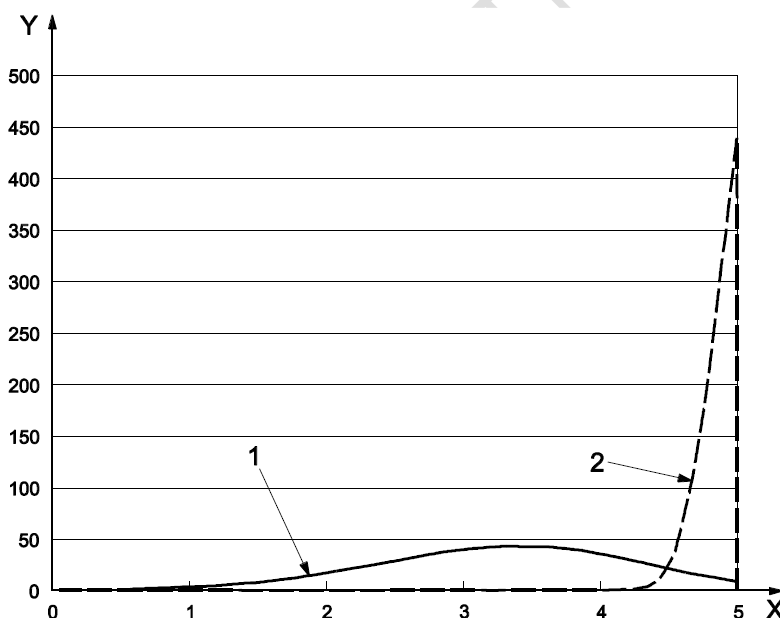
**Key**

X defect, % of nominal  
Y PDF (probability density function)

1 single joint  
2 maximum of 50 joints

NOTE  $b = 2.0$ ; mean = 2.37 %; standard deviation = 1.24 %.

**Figure B.11—Distribution of Defects More Evenly Dispersed in Interval—COV ~ 50 %**



**Key**

X defect, % of nominal  
Y PDF (probability density function)

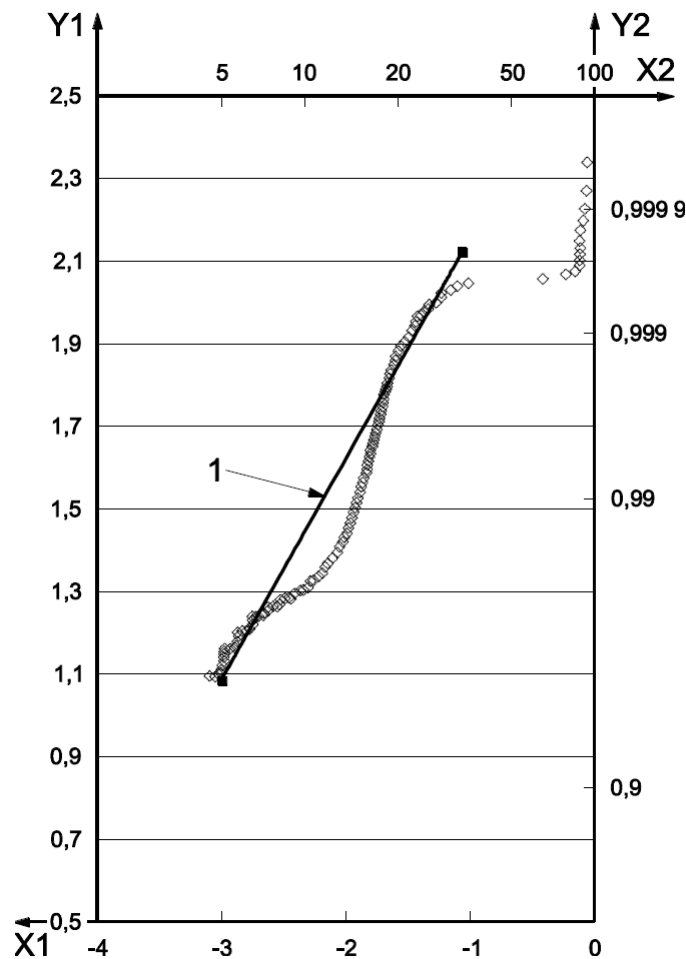
1 single joint  
2 maximum of 50 joints

NOTE  $b = 4.0$ ; mean = 3.31 %; standard deviation = 0.93 %.

**Figure B.12—Distribution Bias to Larger but Undetected Sizes—COV ~ 30 %**

This  $b = 0.5$  distribution is **generally** a good match for the actual defect distribution, when only defect depth is analysed exclusive of the possibility of stacking. A rough analysis of defect sizes deeper than 5 % of specified wall thickness was made. Here, it was assumed that the probability of exceeding 5 % depth is

about 5 % per length of pipe. This analysis finds that the upper tail of defect depths corresponds to a  $b$ -parameter value of about 0.55 (see Figure B.13).



#### Key

- $X_1$  ln (% of depth)
- $X_2$  % of nominal
- $Y_1$  ln  $[-\ln(1-\text{probability})]$
- $Y_2$  CDF (cumulative distribution function)
- 1 Weibull fit with  $b$ -parameter = 0.55

**Figure B.13—Upper Tail of Seam Lap Defect Depths Plotted on a Weibull Probability Scale**

The four charts in Figure B.14 show how the probability distribution for pipe ductile rupture varies based on the different below-the-threshold distribution shapes. These are based on the following assumptions.

The ratio of the pressure capacity to the applied pressure,  $p_{\text{cap}}/p_{\text{app}}$ , is normal.

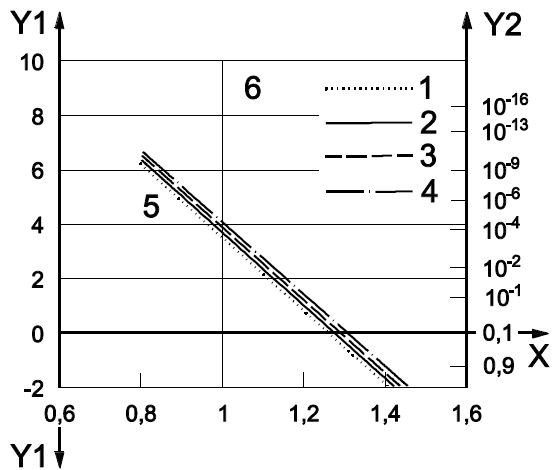
- The mean factor of safety is 1.3 and 1.5; the mean factor of safety is the ratio of the mean ductile rupture pressure to the applied pressure; e.g. if a simple hoop stress equals tensile strength defines

failure, then this factor of safety SF is equal to the mean of the ratio between the pressure capacity and the applied pressure.

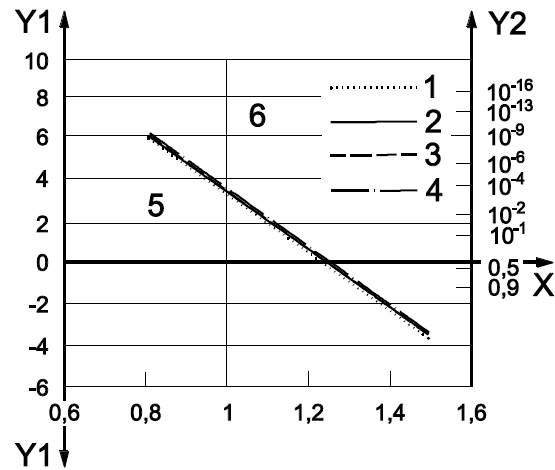
- The coefficient of variability of this safety factor is 5 %.
- The normalized strength  $p_{cap}/p_{app}$  of an imperfect length is simply  $(1 - d_i)p_{cap}/p_{app}$ , where  $d_i$  is the imperfection depth as a percent or fraction of specified wall.
- The distribution for defects is a truncated Weibull distribution, where the defect depth is truncated at the threshold depth (5 %); the shape of this distribution is varied using the b-parameter, where  $b = 0.5$  is most variable, but the majority of defects are near 0 % in depth, and  $b = 4$  indicates a low variability but deeper mean depth.

These charts show that the shape of the distribution below the threshold does not significantly change the distribution of the ratio  $p_{cap}/p_{app}$ . The failure probabilities can be read off these charts by noting the probabilities associated with  $p_{cap}/p_{app} = 1$ . The bar charts in Figures B.15 and B.16 compare the probability of failure for the length with the deepest imperfection in a string of 50 lengths with the probability of failure for a single arbitrary imperfect length.

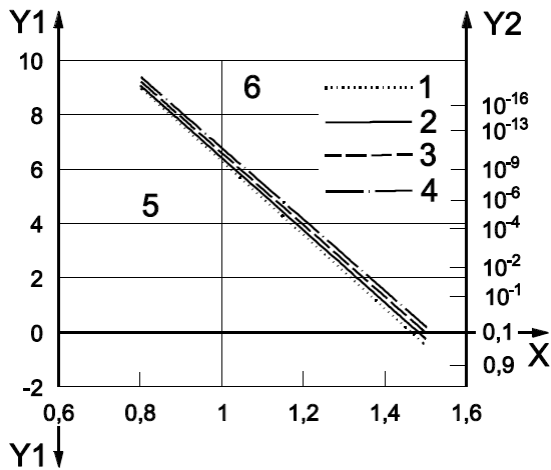
This case study suggests that the rare large imperfection has more impact on the failure probability than the cumulative effect of frequently occurring “below-the-threshold” small imperfections. This has been used to develop part of the framework for the probabilistic calculation of pipe ductile rupture strength. The rupture probability for a length is a function of the probability distributions for material ultimate strength, wall thickness, and the uncertainty of idealization, but the presence of an imperfection is modelled deterministically. In the probabilistic ductile rupture equation, the depth of the imperfection is set equal to the maximum allowed, i.e. the depth of the inspection threshold. The effect of the imperfection on the probability of failure is accounted for by using the frequency of occurrence for threshold-deep imperfections. In the recommended model, this is set equal to the frequency of occurrence of sharp imperfections as measured during secondary inspection.



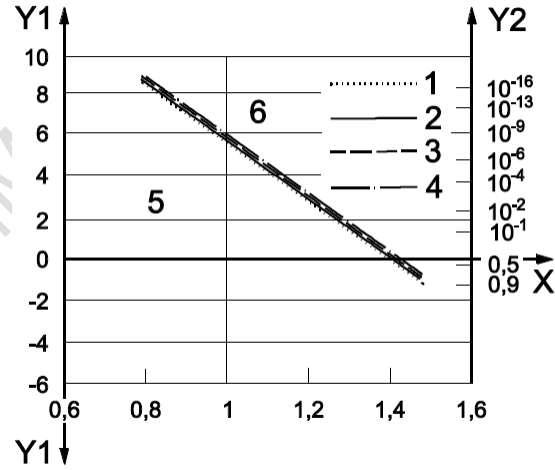
a) Single joint — mean factor of safety of 1.3



b) Deepest defect in 50 joints — mean factor of safety of 1.3



c) Single joint — mean factor of safety of 1.5

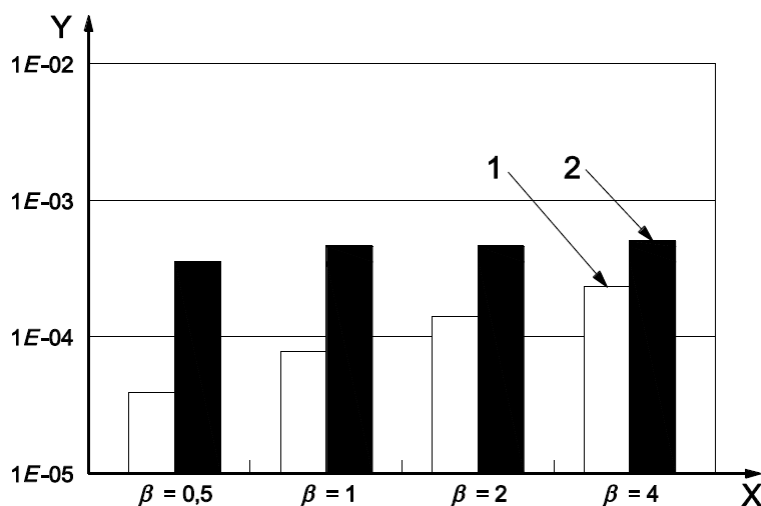


d) Deepest defect in 50 joints — mean factor of safety of 1.5

#### Key

- X  $p_{cap}/p_{app}$
- Y reliability index
- 1  $b = 4.0$
- 2  $b = 2.0$
- 3  $b = 1.0$
- 4  $b = 0.5$
- 5 fail
- 6 safe

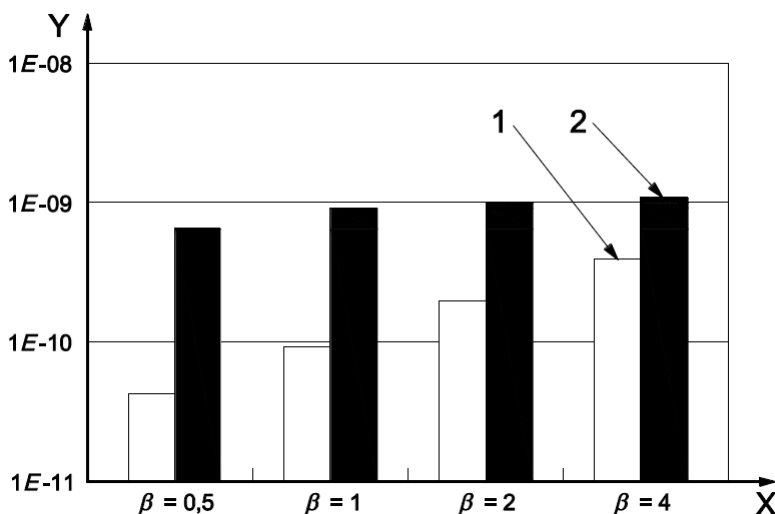
Figure B.14—Effect of “Below-the-Threshold” Imperfection Distribution on the Probability Distribution for Ductile Rupture Strength



**Key**

- X Weibull shape factor,  $b$
- Y probability of failure
- 1 single joint
- 2 string of 50 joints

**Figure B.15—Comparison of Shape Factor Effect on the Probability of Failure of a Single Length and a String (Mean Safety Factor of 1.3)**



**Key**

- X Weibull shape factor,  $b$
- Y probability of failure
- 1 single joint
- 2 string of 50 joints

**Figure B.16—Comparison of Shape Factor Effect on the Probability of Failure of a Single Length and a String (Mean Safety Factor of 1.5)**

## Annex C (informative)

### Rupture Test Procedure

#### C.1 Specimen Ends

Pipe rupture tests should be performed with the pipe ends capped by either welded, slip-in plugs (most common) or threaded end caps (API or premium connections). Although other test setups are possible, such as end plugs that are self-restraining via a center bar, they are rarely used and produce a different stress state and test results. Plug ends that are self-restraining via a center bar are not acceptable. The capped-end test condition produces an axial stress on the pipe via the internal fluid pressure acting on the end caps. The axial stress is about equal to one-half the average hoop membrane stress and is the load condition that, according to the von Mises theory of yield, produces the maximum possible internal pressure.

#### C.2 Minimum Specimen Length

##### C.2.1 Background

Analysis from Timoshenko's Theory of Plates and Shells was considered, in addition to results from a simple FEA model. End-effect calculations were made for several pipe sizes and both light and heavy pipe walls. This resulted in a recommendation of a minimum pipe (pup) length of at least  $8.4(R/t)^{1/2}$ , which was later changed to  $L_p = D + 6(Dt)^{1/2}$ , which is given in Figure 1 of API 5C5 (Second Edition).

This length provides the following:

distance of  $3(Dt)^{1/2}$  from the specimen end cap necessary to remove end effects from the end cap;

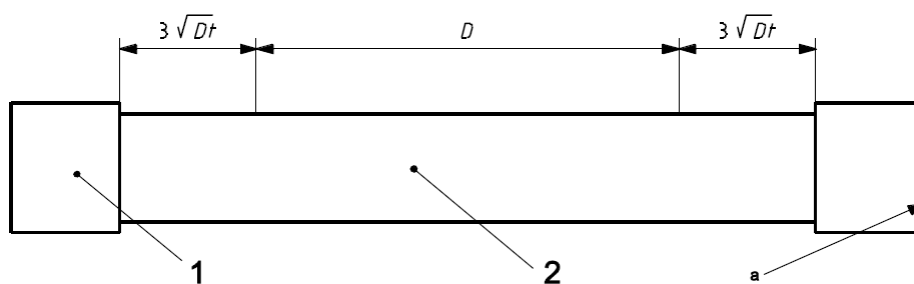
distance of  $D$  (one pipe diameter) of pipe that is removed from any end effects and behaves as an infinitely long cylinder (full length of pipe);

distance of  $3(Dt)^{1/2}$  from the specimen coupling or connection necessary to remove end effects from the coupling/connection.

API 5C5 have also adopted this minimum length for specimen pup length.

##### C.2.2 Specification of Minimum Length

To follow this rupture test procedure, adopt the minimum length from API 5C5 as shown in Figure C.1.



#### Key

- $D$  specified pipe outside diameter
- $t$  specified pipe wall thickness
- 1 end cap
- 2 pipe (pup)
- a coupling centerline

**Figure C.1—Illustration of Minimum Specimen Length for a Pipe Rupture Test**

### C.3 Application of Pressure

Pipe rupture tests are to be performed using water and an internal filler bar. The filler bar is used to reduce the volume of water and therefore minimize the stored energy for safety purposes. This also reduces the size of the rupture in the pipe, which has not been of any consequence. In [API 5C5](#), the pressure application rate is limited to 34.474 MPa/min (5000 psi/min). When pressuring to rupture, ductile pipe will begin to yield and, for typical lab-type pumps, the pressure rate becomes much less as the pipe swells. When a pressure above the yield pressure is reached and shut-in (valve closed), the pipe briefly continues to swell and the pressure drops accordingly. For this reason, the pressure should be continuously applied at a low rate of 6.895 MPa/min (1000 psi/min) or less.

A pressure transducer should be used to measure/record the fluid pressure. A transducer located at the opposite end of the specimen from the pressure line removes spikes in the observed pressure that are related to stroking of the pump. A digital data acquisition system is preferred, with a recording rate of at least every 5 s. This rate will capture the maximum pressure reached and the final rupture pressure that is typically a few percent less than the maximum pressure obtained.

## Annex D (informative)

### Discussion of Equations for Fracture

#### D.1 Material-induced Fracture of the Pipe Body

This annex provides equations that may be used to calculate the pressure at which a pipe will fail due to propagation of a pre-existing sharp crack or due to initiation and growth of a new sharp crack. These elastic-plastic equations apply to pipe that is ductile, brittle, or anywhere between the two extremes. These equations represent an extension of other established fracture mechanics standards to the full range of oil and gas environments. Use of the equations and their related material test data can require substantial expertise, and it is not always straightforward to generate the needed property data. The equations can be used provided the fracture toughness ( $K_{I\text{mat}}$ ) of the material can be determined experimentally for the particular environment of interest. Currently, there is no proposed framework for the case for which  $K_{I\text{mat}}$  cannot be measured. It may be possible to determine  $K_{I\text{mat}}$  from empirically based calculations using other  $K_{I\text{mat}}$  data.

There are two types of fracture failure phenomena: failure due to unstable propagation of a pre-existing crack, and failure due to initiation and stable growth of a crack where there previously was no detectable crack. The first failure phenomenon (addressed in D.2) occurs due to stress-intensity overloading of the crack tip, and failure is determined as a function of the applied stress, the crack size, and the fracture toughness of the material in a particular environment. This failure phenomenon addresses a crack of given size and the conditions under which the crack either propagates or arrests.

The second failure phenomenon (addressed in D.3) is environmental cracking, which occurs due to a combination of stress, material, and environment without requiring any pre-existing imperfection. This failure phenomenon addresses conditions that produce stable growth of a crack that may not have initially existed. Once created, the crack grows in a stable way until it becomes large enough to satisfy the fracture mechanics condition for unstable propagation to failure. Environmental cracking can occur independently of fracture propagation, so both the equations in D.2 and the threshold stress criterion in D.3 need to be satisfied to prevent failure through fracture. This means that there are two limit states that need to be satisfied to prevent fracture, and both limit states depend on the stress and the material fracture toughness in its environment.

In a previous API work group study (API PRAC II), it was found that residual stresses from pipe manufacturing processes can dramatically affect fracture in sour environments. Although residual stresses in OCTG pipes are generally low, in some cases these residual stresses are tensile in nature and can induce a significant increase in the crack stress intensity factor; this increase could be critical when pipes are in sour environments that deteriorate material fracture resistance (fracture toughness). API 579 (ASME FFS) and BS 7910 have considered residual stresses as a secondary stress in the FAD analysis; more general descriptions about residual stress effects on fracture can be found in those standards. Specific residual stress cases for OCTG pipes, however, were not included in those standards and are discussed in this annex.

#### D.2 The Crack Propagation Model

##### D.2.1 General

The approach to fracture here is similar to that used in determining performance of pressure vessels and is extensively utilized to predict the fitness-for-service of these structures. Integrity of cracked structures



has been successfully safeguarded using fracture mechanics standards such as the British PD 6493, now revised and re-issued as BS 7910, and API 579.

A pipe performance calculation based on tensile strength, yield strength, and other material and dimensional properties does not address failure due to propagation of crack-like imperfections where failure is determined by the stress intensity around the crack. When the applied stress intensity factor,  $K$ , reaches a critical value referred to as  $K_{I\text{mat}}$ , the crack propagates and pipe rupture is imminent. The value of  $K_{I\text{mat}}$  is a function of both the environment and the material. The units of  $K_{I\text{mat}}$  are  $\text{MPa}\cdot\text{m}^{1/2}$  ( $\text{ksi}\cdot\text{in}^{1/2}$ ).

Under  $\text{H}_2\text{S}$  environments, the value of  $K_{I\text{mat}}$  will be smaller than in non- $\text{H}_2\text{S}$  environments, and the fracture mode can control the failure pressure of the pipe. The value of  $K_{I\text{mat}}$  changes based on environmental exposure ( $\text{H}_2\text{S}$ , temperature, pH, and so forth). Once this value is determined in laboratory testing for a specific material in a unique environment, it can be used to evaluate the integrity of a pipe with imperfections for that environment.  $K_{I\text{mat}}$  can be considered to be the amount of fracture toughness necessary to prevent or stop further crack propagation in the environment. To preserve the integrity of the pipe,  $K_{I\text{mat}}$  needs to be sufficiently high to prevent the propagation of a crack in the service environment.

A failure assessment diagram (FAD) can be utilized to assess the integrity of the pipe over the whole range of brittle and ductile fractures. The FAD is a plot of the stress intensity ratio ( $K_r$  or  $J_r$ ) on the ordinate versus the load ratio ( $L_r$ ) on the abscissa, where  $K_r$  is the ratio of applied  $K$  to  $K_{I\text{mat}}$ , and  $L_r$  is a ratio of applied load to limit load. The limit load in this case commonly represents an approximation of the load where the cracked pipe yields without crack growth. For further understanding of FAD concepts, see the fracture mechanics standards in [SPE 97577](#). The FAD corresponding to the fracture propagation Equations (D.1) and (D.3) covers both the elastic and combined elastic-plastic behaviors of the material. For materials that are ductile in a particular environment, the value of  $K_{I\text{mat}}$  will be high and the fracture pressure will correspond to the elastic-plastic part of the FAD curve. For materials that are brittle in a particular environment, the value of  $K_{I\text{mat}}$  will be low and the fracture pressure will correspond to the elastic part of the FAD curve.

A “fit-for-purpose” assessment of pipe performance can be made by using  $K_{I\text{mat}}$  corresponding to a specific environment of interest. The evaluation of fracture pressure of casing and tubing in a specific environment requires the measurement of  $K_{I\text{mat}}$  in that environment and comparison to the maximum applied  $K$  within the component. The applied  $K$  depends on the pipe geometry, the imperfection geometry, and the applied load. Within any one chemistry, increasing the yield strength generally results in a lower value of  $K_{I\text{mat}}$ . However,  $K_{I\text{mat}}$  can increase or decrease with increasing yield strength due to changes of chemistry and heat treatment and manufacturing process. Temperature, pH and concentration of sulfide ions all affect the environmental fracture toughness. As temperature and pH increase, the environmental fracture toughness of the material also increases. Microstructure also can cause variation in the environmental fracture toughness. Materials with higher transformation products, such as bainite and pearlite, have lower environmental fracture toughness when compared to martensitic materials. Increasing the partial pressure of  $\text{H}_2\text{S}$  decreases the environmental fracture toughness. Partial pressure of  $\text{H}_2\text{S}$  is calculated by multiplying the absolute pressure by the mole fraction of  $\text{H}_2\text{S}$  in the gas.

## D.2.2 Assumptions and Limitations

The following are assumptions relative to the FAD approach.

Only Mode I failure is considered; this is crack propagation perpendicular to the applied load, i.e. a deepening of the crack, as opposed to Mode II, sliding, and Mode III, tearing.

Elastic-plastic fracture mechanics starting from the “J-integral” is used as the general basis for the FAD curve; the applied  $K$  is the linear elastic solution for a crack in the pipe wall; the intersection of  $K_I$  and the FAD curve determines the fracture pressure; the depth of the crack-like imperfection should be set equal to the depth of the inspection gate setting.

The pipe is infinitely long, with an infinitely long longitudinal crack-like imperfection.

Pressure acts on the faces of the crack.

The ductile rupture limit state with influence of axial load ( $p_{IRa}$ , see 7.5) represents the ultimate ductile failure mode.

### D.2.3 Experimental determination of $K_{I\text{mat}}$

A variety of fracture mechanics tests have been used to develop  $K_{I\text{mat}}$  data. The determination of  $K_{I\text{mat}}$ , which is the threshold stress intensity factor for crack growth in Mode I, plane-strain loading, is a complex matter as API 5CT grades in a sulphide stress corrosion (sour) environment exhibit little or no evidence of ductility before crack propagation. Results are valid only for the materials used to develop the concept

The  $K_{I\text{SSC}}$ , determined from double cantilever beam (DCB) specimen has been extensively used to evaluate  $K_{I\text{mat}}$  for oil field materials. This specimen is described in NACE TM0177-2016 as Method D. The DCB is notched or pre-cracked and then its arms are held open at constant displacement by loading with a wedge. The DCB is loaded such that the applied  $K$  level is above the  $K_{I\text{SSC}}$  for the material and exposed to the test environment (e.g.  $\text{H}_2\text{S}$  aqueous). With initiation, the crack grows, the load drops (under approximately constant displacement), and the applied  $K$  drops until it reaches  $K_{I\text{SSC}}$ , thereby halting crack propagation. After an appropriate period has elapsed after the halting of crack growth, the measured force necessary to remove the wedge and the measured crack length are used to calculate the applied  $K$  at the end of the test. At this point,  $K_{I\text{SSC}}$  is equal to the applied  $K$ .

API conducted programs<sup>[1]</sup> that attempted to evaluate  $K_{I\text{mat}}$  through extensive experimental work on a high-strength, sour service material aligned with Grade API 5CT C110. This work evaluated the variance of  $K_{I\text{SSC}}$  with sour environment (particularly a mild sour condition of a 0.07 bar  $\text{H}_2\text{S}$  and pH of 4.0), mechanical properties (including hardness), manufacturing route, metallurgies, and temperature.

The following properties were obtained on five casing samples having average hardness values ranging from 25.6 to 28.5 HRC.

Average  $K_{I\text{SSC}}$  values at room temperature for Grade C110 were reported as 20.8 to 27.5 ksi-in<sup>1/2</sup> (arm displacement 0.51 mm, +0.03/-0.05 mm) for 1 bar  $\text{H}_2\text{S}$  and 2.7 pH environment, and 35.8 to 50.5 ksi-in<sup>1/2</sup> (arm displacement 0.89 mm, +0.03/-0.05 mm) for 0.07 bar  $\text{H}_2\text{S}$  and 4.0 pH environment, respectively. Tests at 40°F gave reductions between 25 and 35 percent below corresponding room temperature values, with a higher reduction for 1 bar  $\text{H}_2\text{S}$  and 2.7 pH environment. (NACE TM0177 Solution A).

The results demonstrated that  $K_{I\text{SSC}}$  strongly depends on the initial arm displacement used in the test. To isolate this effect and obtain results more aligned with  $K_{I\text{mat}}$ , the concept of  $K_{\text{LIMIT limit}}$  was proposed.  $K_{\text{LIMIT limit}}$  is obtained by testing at different arm displacements and evaluating the results against a  $K_{I\text{SSC}}/K_{\text{app}}$  line.

$K_{LIMIT}$  values at room temperature for Grade C110 were reported between 17.8 and 23.6 ksi-in<sup>1/2</sup> for 1 bar H<sub>2</sub>S and 2.7 pH environment, and 31.7 to 48.1 ksi-in<sup>1/2</sup> for 0.07 bar H<sub>2</sub>S and 4.0 pH environment, respectively.

For a pipe having 28.5 HRC hardness,  $K_{LIMIT}$  values at 40°F were reported as 9.4 ksi-in<sup>1/2</sup> for 1 bar H<sub>2</sub>S and 2.7 pH environment, and 30.2 ksi-in<sup>1/2</sup> for 0.07 bar H<sub>2</sub>S and 4.0 pH environment, respectively.

The  $K$  applied in service to a pipe depends on its geometry, the imperfection geometry, and the applied load. It was proposed that an environmental tests of a pipe specimen (mini-pipe) could be used to study the accuracy of the fracture equations. These tests were conducted on high-strength, sour service material aligned with Grade API 5CT-C110 [2].

Tests with stress intensities of 25.0 ksi-in<sup>1/2</sup> on both ID and OD notches experienced crack growth in mild conditions. This suggests some consistency with ID or OD equations from API 579.

The experimental results did not match the FAD equation in all cases. The analysis of the test results using failure assessment diagrams indicated that the measured  $K_{LIMIT}$  was 50 percent higher than the fracture toughness that would predict tube failures based on NACE TM0177 method D tests. This discrepancy could be a result of a combination of several factors:

- Significant differences may exist between ID or OD environments and the environment of the DCB specimens used for obtaining  $K_{LIMIT}$ . H<sub>2</sub>S concentration and solution pH during mini-pipe tests are not identical to those to which the crack tip is normally exposed during DCB testing.
- The method used for determining  $K_{LIMIT}$  as the intersection of the  $K_{ISSC}/K_{applied}$  line (45°) might not be the correct approach.
- The FAD equation is sensitive to the variability of notch dimensions and pipe dimensions (particularly wall thickness) and the effect of residual stress.
- Both ID hydrogen charging for the mini-pipe tests compared with all-aspect hydrogen charging for the DCB tests and pressure fluctuation during the mini-pipe tests contributed to test uncertainty. For example, tests at constant pressure did not show crack growth at pressures where failures were obtained before.

Application of FAD models to casing performance in sour environments is not common. While there have been numerous tests to validate crack propagation models, only a fraction of these utilized OCTG materials or were carried out in sour H<sub>2</sub>S environments. An even smaller fraction of these tests were carried out on full-size samples of as-produced pipe containing the inherent characteristics of pipe intended for downhole application.

In addition to testing mini-pipe specimens, API conducted test programs [3] on full-scale joints of tubulars with a grade aligned with Grade API 5CT-C110.

A full-length artificial notch was machined on the outside surface of an-as manufactured Grade C110 casing. Small pipe specimens (“tiny pipe”) were machined from the wall of an adjacent pipe section.  $K_{LIMIT}$  was also evaluated on the same casing.

For small pipe specimens, the performed burst tests at 0.07 bar and pH 4.0 conditions with OD and ID exposure showed pressures between 68% to 75% of the FAD prediction, whereas the single test performed only with ID exposure matched the FAD prediction within 1%. The test performed at 1 bar and pH 2.7 with OD and ID exposure failed at approximately 69% of the FAD prediction.

The full size pipes did not show failure when tested at significantly higher pressures than the FAD prediction. Pressures between 2 to 6 times higher than the FAD prediction applied in both 0.07 bar H<sub>2</sub>S, pH 4.0 or 1 bar H<sub>2</sub>S, pH 2.7 did not create either crack initiation or crack propagation from the existing notch. One possible cause could be the blunt shape at the notch bottom, as compared to the sharp crack tip for which the FAD equation is derived.

Structural integrity procedures for the assessment of crack-like flaws usually recommend the use of fracture toughness values that are either the lower bound or have been obtained from testing conditions (geometry, loading mode, etc.) as close as possible to the in-service structure.

$K_{LIMIT}$   $K_{limit}$  is obtained from crack-arrest tests under constant displacement conditions. This condition is different than the one faced by a crack in a pipe. Other fracture mechanic tests expose the crack to loading conditions more similar to those experienced by a pipe in service. Besides the applied  $K_I$ , other factors can affect the crack propagation in aggressive environments, such as time of exposure (subcritical crack growth) or triaxial stress constraints at crack-tip.

SENT specimens under constant load conditions provide an increasing  $K_I$  with crack length. FEM analysis shows that SENT tests better represent the crack-tip stress field existing in a pipe when loaded with internal pressure. Further investigation of SENT specimens and other fracture mechanic tests may be warranted.

Determination of fracture toughness in sour service continues to be a challenge for the industry. In the context of the limited amount of testing being performed, this work has highlighted the limitation of  $K_{LIMIT}$   $K_{limit}$ ; however,  $K_{LIMIT}$   $K_{limit}$  is still an improvement for the estimation of  $K_{Imat}$  when compared with the  $K_{ISSC}$ . Efforts to generate enough data to establish the uncertainty or precision of the fracture toughness parameters ( $K_{Imat}$ ) are also an important consideration for future work.

## D.2.4 Fracture limit state equation

A recent study <sup>[AAA]</sup> found that the rotary straightening process, which generates most of the residual stresses in pipe, normally produces compressive residual stress at pipe ID and tensile residual stress at pipe OD. The distribution of residual stress, however, is not uniform. Thus, for design purposes, it is recommended to include tensile residual stress into the assessment but not compressive residual stress. Special cases such as OD H<sub>2</sub>S exposure, non-straightened pipes, or CRA pipes require more study and are not included in the scope of this annex.

A general fracture mechanics based FAD curve can be described as:

$$K_r = (1 - 0.14L_r^2) \left( 0.3 + 0.7 \exp[-0.65L_r^6] \right) \quad (D.1)$$

where

$K_r$  is the stress intensity ratio (see Equation (D.3));

$L_r$  is the load ratio and, assuming internal pressure is the only load considered:

$$L_r = \sqrt{3}/2 \left( p_{iF} / f_y \right) \left[ (R_i + a) / (t - a) \right] \quad (D.2)$$

$p_{iF}$  is the internal pressure;  
 $f_y$  is the yield strength of a representative tensile specimen;  
 $R_i$  is the pipe inside radius:  $R_i = R_o - t$ ;  
 $R_o$  is the pipe outside radius:  $R_o = D/2$ ;  
 $D$  is the pipe specific outside diameter;  
 $t$  is the pipe actual wall thickness where flaw exists;  
 $a$  is, for a limit state equation, the maximum actual depth of a crack-like imperfection; for a design equation, the maximum depth of a crack-like imperfection that could likely pass the manufacture's inspection system.

The right hand side of Equation (D.1) is the FAD curve.

The left hand side of the Equation (D.1) is the stress intensity ratio, which includes the effects of residual stresses on pipe fracture assessment. According to API 579 (ASME FFS) 2016 version, Annex 9G – Stress analysis for crack-like flaw:

$$K_r = \frac{K_I^P + \Phi K_I^{SR}}{K_{Imat}} \quad (D.3)$$

where

$K_r$  is the stress intensity ratio for FAD curve;  
 $K_I^P$  is the stress intensity factor associated with primary stress (internal pressure in present case);  
 $\Phi$  is the plasticity interaction factor;  
 $K_I^{SR}$  is the stress intensity factor associated with secondary stress (residual stress in present case);  
 $K_{Imat}$  is the fracture toughness of the material in a particular environment.

If internal pressure is the only load considered in the analysis,  $K_I^P$  can be calculated by following equations:

$$K_{I-ID}^P = \frac{p_{iF} R_o^2}{R_o^2 - R_i^2} \left[ 2G_0 - 2G_1 \left( \frac{a}{R_i} \right) + 3G_2 \left( \frac{a}{R_i} \right)^2 - 4G_3 \left( \frac{a}{R_i} \right)^3 + 5G_4 \left( \frac{a}{R_i} \right)^4 \right] \sqrt{\pi a} \quad (D.4)$$

$$K_{I-OD}^P = \frac{p_{iF} R_i^2}{R_o^2 - R_i^2} \left[ 2G_0 + 2G_1 \left( \frac{a}{R_o} \right) + 3G_2 \left( \frac{a}{R_o} \right)^2 + 4G_3 \left( \frac{a}{R_o} \right)^3 + 5G_4 \left( \frac{a}{R_o} \right)^4 \right] \sqrt{\pi a} \quad (D.5)$$

where

$K_{I-ID}^P$  is the  $K_I^P$  for flaws at pipe ID;  
 $K_{I-OD}^P$  is the  $K_I^P$  for flaws at pipe OD;  
 $G_n$  are the influence coefficients, which are listed in Table D.1.

**Table D. 1 – Values of  $G_0$  to  $G_4$  for FAD curve to change first column**

D/t	t/Ri	a/t	Inside Surface					Outside Surface				
			G <sub>0</sub>	G <sub>1</sub>	G <sub>2</sub>	G <sub>3</sub>	G <sub>4</sub>	G <sub>0</sub>	G <sub>1</sub>	G <sub>2</sub>	G <sub>3</sub>	G <sub>4</sub>
61.98800	0.01667	0	1.120000	0.682000	0.524500	0.440400	0.379075	1.120000	0.682000	0.524500	0.440400	0.379075
		0.2	1.353978	0.771359	0.574252	0.473441	0.403767	1.354559	0.771574	0.574391	0.473496	0.403822
		0.4	2.076759	1.047310	0.727245	0.572880	0.479971	2.075495	1.046684	0.726942	0.572615	0.479723
		0.6	3.857250	1.698243	1.078613	0.797885	0.644619	3.858610	1.698758	1.079099	0.798289	0.644759
		0.8	9.818255	3.796811	2.179989	1.488012	1.131273	9.814990	3.795084	2.177882	1.486756	1.130544
42.00000	0.02500	0	1.120000	0.682000	0.524500	0.440400	0.379075	1.120000	0.682000	0.524500	0.440400	0.379075
		0.2	1.351845	0.770679	0.573795	0.473108	0.403649	1.352815	0.770943	0.573978	0.473214	0.403667
		0.4	2.064088	1.042414	0.724534	0.571046	0.478588	2.058880	1.039137	0.721589	0.568690	0.477008
		0.6	3.780308	1.670205	1.063426	0.788181	0.637578	3.783314	1.671252	1.064248	0.788820	0.637814
		0.8	9.046439	3.526527	2.038831	1.400363	1.069409	9.072502	3.534701	2.042477	1.402231	1.070735
22.00000	0.05000	0	1.120000	0.682000	0.524500	0.440400	0.379075	1.120000	0.682000	0.524500	0.440400	0.379075
		0.2	1.345621	0.768292	0.572560	0.472331	0.402984	1.348153	0.769051	0.572972	0.472583	0.403085
		0.4	2.028188	1.028989	0.717256	0.566433	0.475028	2.028188	1.028734	0.717129	0.566281	0.474824
		0.6	3.573882	1.594673	1.023108	0.762465	0.618437	3.584289	1.598763	1.025243	0.763840	0.619628
		0.8	7.388754	2.946567	1.736182	1.211533	0.936978	7.522466	2.992945	1.760192	1.226597	0.947337
12.00000	0.10000	0	1.120000	0.682000	0.524500	0.440400	0.379075	1.120000	0.682000	0.524500	0.440400	0.379075
		0.2	1.332691	0.763153	0.569758	0.470495	0.401459	1.338976	0.765213	0.570770	0.471069	0.401850
		0.4	1.957764	1.002123	0.702473	0.556857	0.467621	1.964321	1.004607	0.703748	0.557628	0.468296
		0.6	3.223438	1.466106	0.953655	0.718048	0.585672	3.270363	1.483681	0.963144	0.724069	0.590347
		0.8	5.543784	2.300604	1.398958	1.000682	0.789201	5.839919	2.403771	1.452694	1.034485	0.812508
7.00000	0.20000	0	1.120000	0.682000	0.524500	0.440400	0.379075	1.120000	0.682000	0.524500	0.440400	0.379075
		0.2	1.307452	0.753466	0.564298	0.466913	0.398757	1.324199	0.759716	0.567636	0.468987	0.400407
		0.4	1.833200	0.954938	0.676408	0.539874	0.454785	1.861734	0.964913	0.681357	0.542909	0.457058
		0.6	2.734052	1.287570	0.857474	0.656596	0.540720	2.864663	1.335823	0.883295	0.673156	0.553201
		0.8	3.940906	1.739955	1.106210	0.818230	0.661258	4.412961	1.903704	1.191550	0.871484	0.697846

Note: Interpolation of the influence coefficients,  $G_i$ , may be used for intermediate values of  $t/R_i$  and  $a/t$ . This table is from API 579 (ASME FFS), 2016.

$K_I^{SR}$  for surface flaw in OCTG pipes can be estimated by the following equation:

$$K_I^{SR} = Q\sigma\sqrt{\pi a} \quad (D.6)$$

where

$\sigma$  is the residual stresses, mainly hoop residual stress, measured by split ring method (see Annex I.4.4) or other method;

$Q$  is the geometry factor.

Regarding the geometry factor  $Q$ , it is recommended to use the following equation<sup>1</sup>:

$$Q = [G_0 - 2G_1(a/t)] \quad (D.7)$$

The plasticity interaction factor  $\phi$  in Equation (D.3) is a crack-tip plasticity adjustment factor. It can be affected by residual stress magnitude and primary load. For example, as  $L_r$  increases, the crack-tip plasticity magnifies the total driving force caused by the combination of primary load and residual stress. After  $\phi$  reaches a peak as  $L_r$  continues increasing, it starts decreasing due to mechanical stress relaxation. For OCTG pipes in sour environment, however, the primary load is relatively low, and the residual stress level is also generally low compared to pipe yield strength. So  $\phi$  can be specialized and simplified for OCTG cases. For getting an accurate value of  $\phi$ , please refer to API 579 (ASME FFS) Part 9 "Assessment of crack-like flaws".



When Equations (D.1) to (D.7) are combined,  $p_{iF}$  can be obtained. It should be noted that  $p_{iF}$  cannot be explicitly solved from above equations. An iterative approach, either numerical or graphical, can be used to solve  $p_{iF}$ . It should also be noted that if the  $p_{iF}$  calculated from above equations is less than  $p_{iRa}$  (adjusted ductile rupture  $p_{iR}$  for axial load and external pressure),  $p_{iF} = p_{iRa}$ .

## D.2.45 Design equation for fracture of the pipe body due to the presence of crack-like imperfections

A recent study <sup>[AAA]</sup> concluded that the rotary straightening process, which generates most of the residuals stresses in pipe, normally produces compressive residual stress at pipe ID and tensile residual stress at pipe OD. The distribution of residual stress, however, is not uniform. Thus, for design purposes, it is recommended to include tensile residual stress into the assessment of OD defects but compressive residual stress. Special cases such as OD H<sub>2</sub>S exposure, non-straightened pipes, or CRA pipes require further study and are not included in the scope of this annex.

The fracture design equation for a normal case (i.e., H<sub>2</sub>S presents at pipe ID and no tensile residual stress at ID) is:

$$(1 - 0.14L_r^2)(0.3 + 0.7 \exp[-0.65L_r^6]) = [P_{iF} R_o^2 (\pi a)^{1/2}] / [(R_o^2 - R_{iwall}^2) K_{Imat}] \times \{2G_0 - 2G_1[a/R_{iwall}] + 3G_2[a/R_{iwall}]^2 - 4G_3[a/R_{iwall}]^3 + 5G_4[a/R_{iwall}]^4\} \quad (D.38)$$

or  $P_{iF} = P_{iRa}$ , if  $P_{iRa}$  is less than the solution from Equation (D.38),

where

$$L_r = \sqrt{3/2} (p_{iF} / f_y) [(R_{iwall} + a) / (k_{wall} t - a)] \quad (D.49)$$

and

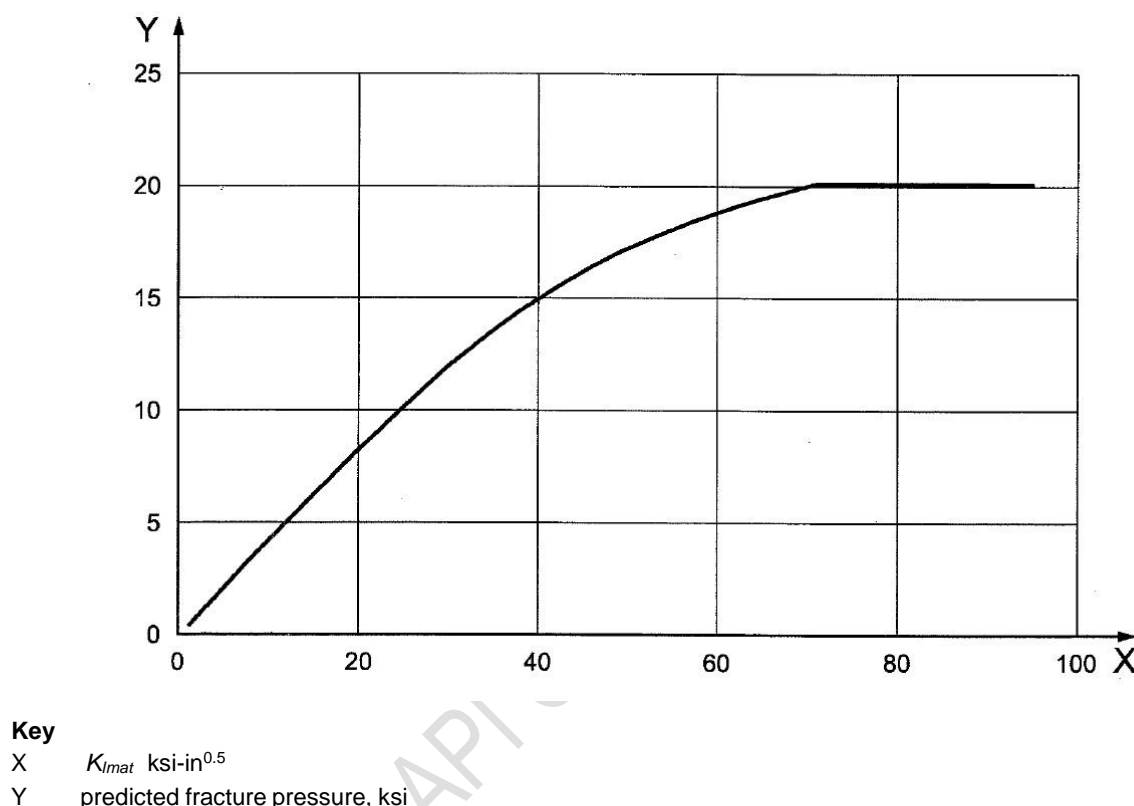
- $a$  is, for a limit state equation, the maximum actual depth of a crack-like imperfection; for a design equation, the maximum depth of a crack-like imperfection that could likely pass the manufacturer's inspection system;
- $R_{iwall}$  is the inside radius based on  $K_{wall}$ ;  $R_{iwall} = D/2 - k_{wall}t$ ;
- $R_o$  is the specified pipe outside radius;  $R_o = D/2$ ;
- $f_{ymn}$  is the specified minimum yield strength;
- $K_{mat}$  is the fracture toughness of the material in a particular environment;
- $K_{wall}$  is the factor to account for the specified manufacturing tolerance of the pipe wall (for example, for a tolerance of -12.5%,  $k_{wall} = 0.875$ );
- $L_r$  is the load ratio;
- $P_{iF}$  is the internal pressure at fracture;
- $P_{iR}$  is the internal pressure at ductile rupture of an end-capped pipe;
- $P_{iRa}$  is  $P_{iR}$  adjusted for axial load and external pressure;
- $t$  is the specified pipe wall thickness;
- $G_n$  are the influence coefficients for inside surface, which are listed in Table D.1.

In Equation (D.38), the G-influence coefficients are the same as listed in Table D.1 (inside surface). For calculation of the fracture design pressure,  $d_{wall}$  in Table D.1 is the specified inner diameter of the pipe.

The left side of Equation (D.38) is the FAD curve. The right side of Equation (D.38) is the stress intensity ratio  $K_r$ . Equation (D.38) is an equation for failure due to propagation of a pre-existing crack. Equation

(D.38) is not an equation for environmentally induced failure of a material which does not have a large crack.

Figure D.1 shows an example of the predicted fracture pressure based on Equations (D.38) and (D.49) for a 7.0 in. diameter, 0.730 in. wall, C90 casing as a function of the fracture toughness  $K_{I\text{mat}}$ , assuming 5% imperfection (inspection gate) combined with  $k_{\text{wall}} = 0.875$ .



**Figure D.1 — Influence of  $K_{I\text{mat}}$  on fracture propagation pressure (7 in. diameter, 0.730 in. wall, C90 with 5% imperfection, 0.875 wall factor)**

## D.3 Fracture due to Environmental Crack Initiation

### D.3.1 General

For a material in a particular environment, environmental crack initiation can occur and result in fracture (failure) when there was no pre-existing crack in the material. This is due to a combination of crack initiation and stable environmental crack growth to failure. For this reason, both the fracture propagation equations in D.2 and the environmental crack initiation criterion in D.3 need to be satisfied to prevent failure through fracture. Environmental crack initiation is said to occur when the von Mises equivalent stress exceeds what is called the threshold ~~stress~~  $\sigma_{th}$  ( $\sigma_{th}$ ) of the material. The fracture pressure for environmental crack initiation is the pressure that causes the von Mises equivalent stress to equal the threshold stress, i.e.:



$$\sigma_e = \sigma_{th} \quad (D.10)$$

where

$\sigma_e$  is the equivalent stress;

$\sigma_{th}$  is the threshold stress.

The threshold stress can vary with the particular material and environment. See NACE TM0177-96 for an explanation of threshold stress. The threshold stress usually is determined by a series of NACE Method-A tensile tests in a specific environment. Above the threshold stress, the tensile specimen fails during the NACE Method-A tensile test; below the threshold stress, the tensile specimen passes the NACE Method-A tensile test. In an H<sub>2</sub>S environment, the threshold stress usually is less than the yield strength of the material.

Environmental crack initiation can typically start at the bottom of a corrosion pit and depends on the combination of the environment (CO<sub>2</sub>, pH, H<sub>2</sub>S), temperature, material microstructure, and mechanical stress. Below a “threshold” combination of these factors, crack initiation does not occur, and beyond the threshold, stress crack initiation does occur. For most applications, the mechanical loads on the pipe are constant and the crack, once initiated, will grow to failure. The situation thus deteriorates until the crack reaches the size where unstable crack propagation leads to final fracture. The time between crack initiation and unstable fracture is uncertain, and therefore it is prudent to avoid crack initiation altogether by keeping the von Mises equivalent stress sufficiently low compared with the threshold stress.

The fracture initiation limit pressure is the pressure for which the von Mises equivalent stress equals the threshold stress [Equation (D.10)], where the von Mises equivalent stress is calculated using measured pipe dimensions. The fracture initiation design pressure is the pressure for which the von Mises equivalent stress equals the threshold stress, where the von Mises equivalent stress is calculated using specified pipe dimensions and the pipe wall thickness tolerance factor  $k_{wall}$ . A margin between applied (service) equivalent stress and the apparent threshold stress should be maintained to account for scatter on the estimation of the threshold stress.

Just as with the yield equation, the von Mises equivalent stress is used to combine the individual, three-dimensional stresses into a single parameter to compare with the threshold stress. The equivalent stress is used because, based on data in [SPE 97577](#), it appears to provide the most accurate combination of stresses that brings about crack initiation failure of pipe in an H<sub>2</sub>S environment. Equation (D.5) is applicable only when the internal pressure exceeds the external pressure. Test data in axial compression suggest that the equation can cease to apply, i.e. that crack initiation failure might not occur, when the mean hydrostatic stress becomes compressive. That is, in the absence of torsion, the equation can cease to apply when the sum of the axial stress, radial stress, and hoop stress added together becomes negative.

### D.3.2 Example Calculation

Calculate the design crack initiation failure pressure for a 244.48 mm (9-5/8 in.), 13.84 mm (0.545 in.) wall thickness, C90 grade [ $f_{ymn} = 621$  Mpa (90,000 psi)] casing subjected to internal pressure with capped-end conditions, if the threshold stress is 90 % of the yield strength of the pipe material and the  $k_{wall} = 0.875$ .

First, the von Mises equivalent stress is set equal to the threshold stress, which is 90 % of the yield stress [Equation (D.10)]. For the loading combination where the pipe has capped-end conditions and there are no torsional and bending stresses, then Equation (D.10) reduces to the following, similar in form to the equation for  $p_{iYLC}$ , Equation (8) of 6.6.1.1:

$$p_{iF} = 0.9 f_{ymn} / \{ (3 D^4 + d_{wall}^4) / (D^2 - d_{wall}^2)^2 + d^4 / (D^2 - d^2)^2 - 2 d^2 d_{wall}^2 / [(D^2 - d^2) (D^2 - d_{wall}^2)] \}^{1/2} \quad (D.11)$$

where

$D$  is the specified pipe outside diameter;

$d$  is the pipe inside diameter,  $d = D - 2t$ ;

$d_{\text{wall}}$  is the inside diameter based on  $k_{\text{wall}}$   $t$ ,  $d_{\text{wall}} = D - 2k_{\text{wall}} t$ ;

$f_{\text{ymn}}$  is the specified minimum yield strength;

$k_{\text{wall}}$  is the factor to account for the specified manufacturing tolerance of the pipe wall; e.g. for a tolerance of  $-12.5\%$ ,  $k_{\text{wall}} = 0.875$ ;

$p_{\text{iYLC}}$  is the internal pressure at yield for a capped-end thick tube;

$t$  is the specified pipe wall thickness.

Then the crack initiation fracture pressure for the example pipe is 60.6 Mpa (8788 psi).

## Annex E (informative)

### Commentary—Historical API Collapse Equations

#### E.1 Collapse Pressure Equations

##### E.1.1 General

The minimum collapse pressures given in Annex K are calculated by means of Equations (E.1), (E.3), (E.5), and (E.7).

Equations (E.2), (E.4), and (E.6) for the intersections between the four collapse pressure equations have been determined algebraically, and used for calculating the applicable  $D/t$  range for each collapse pressure equation. Factors  $A_c$ ,  $B_c$ ,  $C_c$ ,  $F_c$ , and  $G_c$  have been calculated using Equations (E.20), (E.21), (E.22), (E.25), and (E.26). When determining the appropriate equation to be used for calculating collapse resistance for a particular  $D/t$  ratio and minimum yield strength, the  $D/t$  ranges determined by Equations (E.2), (E.4), and (E.6) govern, rather than the collapse equation that gives the lowest collapse pressure. The  $D/t$  ranges are given in Tables E.1, E.2, E.3, and E.4.

Theoretical studies of the effect of ovality on tubular collapse resistance consistently indicate that an ovality of 1 % to 2 % can affect a reduction in collapse resistance on the order of 25 %. However, experimental/empirical investigations indicate a much smaller effect. Test data indicate that ovality is only one of many pipe parameters that influence collapse (including residual stress, isotropy, shape of stress-strain curve/microstructure, and yield strength). Thorough review of industry collapse data indicates that the influence of ovality does not warrant singling out the ovality as a dominant parameter. A workgroup on collapse resistance concluded that the effect of ovality on tubular collapse has been handled during the adjustment of average collapse predictions to minimum performance values and that ovality should not be awarded the status of an independent variable in an API equation for collapse performance.

The collapse equations presented here were originally developed in USC units and should only be used in these units.

##### E.1.2 Yield Strength Collapse Pressure Equation

The yield strength collapse pressure is not a true collapse pressure, but rather the external pressure,  $p_{yp}$ , that generates minimum yield stress,  $f_{ymn}$ , on the inside wall of a tube as calculated by Equation (E.1):

$$p_{yp} = 2 f_{ymn} [(D/t) - 1]/[(D/t)^2] \quad (E.1)$$

where

$D$  is the specified pipe outside diameter;

$f_{ymn}$  is the specified minimum yield strength;

$t$  is the specified pipe wall thickness.

Equation (E.1) for yield strength collapse pressure is applicable for  $D/t$  values up to the value of  $D/t$  corresponding to the intersection with the plastic collapse [Equation (E.3)]. This intersection,  $(D/t)_{yp}$ , is calculated by Equation (E.2) as follows:

$$(D/t)_{yp} = \{[(A_c - 2)^2 + 8(B_c + C_c/f_{ymn})]^{1/2} + (A_c - 2)\} / [2(B_c + C_c/f_{ymn})] \quad (E.2)$$

where

$A_c$  is the empirical constant in historical API collapse equation;

$B_c$  is the empirical constant in historical API collapse equation;

$C_c$  is the empirical constant in historical API collapse equation;

$f_{ymn}$  is the specified minimum yield strength.

The parameters used to calculate collapse pressures depend on the pipe yield strength and on the axial load, as explained in later subsections.

The applicable  $D/t$  ratios for yield strength collapse are shown in Table E.1.

**Table E.1—Yield Collapse Pressure Equation Range**

Grade <sup>a</sup>	$D/t$ Range <sup>b</sup>
H40	16.40 and less
-50	15.24 and less
J55, K55	14.81 and less
-60	14.44 and less
-70	13.85 and less
C75, E75	13.60 and less
L-N-80	13.38 and less
C90	13.01 and less
R95, T95, X95	12.85 and less
-100	12.70 and less
P105, G105	12.57 and less
P110	12.44 and less
-120	12.21 and less
Q125	12.11 and less
-130	12.02 and less
S135	11.92 and less
-140	11.84 and less
-150	11.67 and less
-155	11.59 and less
-160	11.52 and less
-170	11.37 and less
-180	11.23 and less

<sup>a</sup> Grades indicated without letter designation are not API grades but are grades in use or grades being considered for use and are shown for information purposes.

<sup>B</sup> The  $D/t$  range values were calculated from Equations (E.2), (E.20), (E.21), and (E.22).

### E.1.3 Plastic Collapse Pressure Equation

The minimum collapse pressure for the plastic range of collapse,  $p_P$ , is calculated by Equation (E.3):

$$p_P = f_{ymn} [A_c/(D/t) - B_c] - C_c \quad (E.3)$$

where

$A_c$  is the empirical constant in historical API collapse equation;

$B_c$  is the empirical constant in historical API collapse equation;

$C_c$  is the empirical constant in historical API collapse equation;

$D$  is the specified pipe outside diameter;

$f_{ymn}$  is the specified minimum yield strength;

$t$  is the specified pipe wall thickness.

The equation for minimum plastic collapse pressure is applicable for  $D/t$  values ranging from  $(D/t)_{yp}$ , Equation (E.2) for yield strength collapse pressure, to the intersection with Equation (E.5) for transition collapse pressure  $(D/t)_{pt}$ . Values for  $(D/t)_{pt}$  are calculated by means of Equation (E.4):

$$(D/t)_{pt} = [f_{ymn} (A_c - F_c)] / [C_c + f_{ymn} (B_c - G_c)] \quad (E.4)$$

where

$A_c$  is the empirical constant in historical API collapse equation;

$B_c$  is the empirical constant in historical API collapse equation;

$C_c$  is the empirical constant in historical API collapse equation;

$F_c$  is the empirical constant in historical API collapse equation;

$f_{ymn}$  is the specified minimum yield strength;

$G_c$  is the empirical constant in historical API collapse equation.

The factors and applicable  $D/t$  ranges for the plastic collapse equation are shown in Table E.2.

**Table E.2—Equation Factors and  $D/t$  Range for Plastic Collapse**

Grade <sup>a</sup>	$A_c$	$B_c$	$C_c$ psi	$D/t$ Range <sup>b</sup>
H40	2.950	0.0465	754	16.40 to 27.01
–50	2.976	0.0515	1056	15.24 to 25.63
J55, K55	2.991	0.0541	1206	14.81 to 25.01
–60	3.005	0.0566	1356	14.44 to 24.42
–70	3.037	0.0617	1656	13.85 to 23.38
C75, E75	3.054	0.0642	1806	13.60 to 22.91
L-N-80	3.071	0.0667	1955	13.38 to 22.47
C90	3.106	0.0718	2254	13.01 to 21.69
R95, T95, X95	3.124	0.0743	2404	12.85 to 21.33
–100	3.143	0.0768	2553	12.70 to 21.00
P105, G105	3.162	0.0794	2702	12.57 to 20.70
P110	3.181	0.0819	2852	12.44 to 20.41
–120	3.219	0.0870	3151	12.21 to 19.88
Q125	3.239	0.0895	3301	12.11 to 19.63
–130	3.258	0.0920	3451	12.02 to 19.40
S135	3.278	0.0946	3601	11.92 to 19.18
–140	3.297	0.0971	3751	11.84 to 18.97
–150	3.336	0.1021	4053	11.67 to 18.57
–155	3.356	0.1047	4204	11.59 to 18.37
–160	3.375	0.1072	4356	11.52 to 18.19
–170	3.412	0.1123	4660	11.37 to 17.82
–180	3.449	0.1173	4966	11.21 to 17.47
<sup>a</sup> Grades indicated without letter designation are not API grades but are grades in use or grades being considered for use and are shown for information purposes. <sup>B</sup> The $D/t$ range values and equation factors were calculated from Equations (E.2), (E.4), (E.20), (E.21), (E.22), (E.25), and (E.26).				

### E.1.4 Transition Collapse Pressure Equation

The minimum collapse pressure for the plastic to elastic transition zone,  $p_T$ , is calculated by Equation (E.5):

$$p_T = f_{ymn} [F_c / (D/t) - G_c] \quad (E.5)$$

where

$D$  is the specified pipe outside diameter;

$F_c$  is the empirical constant in historical API collapse equation;

$f_{ymn}$  is the specified minimum yield strength;

$G_c$  is the empirical constant in historical API collapse equation;

$p_T$  is the pressure for transition collapse;

$t$  is the specified pipe wall thickness.

The equation for  $p_T$  is applicable for  $D/t$  values from  $(D/t)_{pt}$ , Equation (E.4) for plastic collapse pressure, to the intersection  $(D/t)_{te}$  with Equation (E.7) for elastic collapse. Values for  $(D/t)_{te}$  are calculated by Equation (E.6):

$$(D/t)_{te} = [2 + B_c / A_c] / [3(B_c / A_c)] \quad (E.6)$$

where

$A_c$  is the empirical constant in historical API collapse equation;

$B_c$  is the empirical constant in historical API collapse equation.

The factors and applicable  $D/t$  ranges for the transition collapse pressure equation are shown in Table E.3.

**Table E.3—Equation Factors and  $D/t$  Range for Transition Collapse**

Grade <sup>a</sup>	$F_C$	$G_C$	$D/t$ Range <sup>b</sup>
H40	2.063	0.0325	27.01 to 42.64
–50	2.003	0.0347	25.63 to 38.83
J55, K55	1.989	0.0360	25.01 to 37.21
–60	1.983	0.0373	24.42 to 35.73
–70	1.984	0.0403	23.38 to 33.17
C75, E75	1.990	0.0418	22.91 to 32.05
L-N-80	1.998	0.0434	22.47 to 31.02
C90	2.017	0.0466	21.69 to 29.18
R95, T95, X95	2.029	0.0482	21.33 to 28.36
–100	2.040	0.0499	21.00 to 27.60
P105, G105	2.053	0.0515	20.70 to 26.89
P110	2.066	0.0532	20.41 to 26.22
–120	2.092	0.0565	19.88 to 25.01
Q125	2.106	0.0582	19.63 to 24.46
–130	2.119	0.0599	19.40 to 23.94
S135	2.133	0.0615	19.18 to 23.44
–140	2.146	0.0632	18.97 to 22.98
–150	2.174	0.0666	18.57 to 22.11
–155	2.188	0.0683	18.37 to 21.70
–160	2.202	0.0700	18.19 to 21.32
–170	2.231	0.0734	17.82 to 20.60
–180	2.261	0.0769	17.47 to 19.93

<sup>a</sup> Grades indicated without letter designation are not API grades but are grades in use or grades being considered for use and are shown for information purposes.

<sup>B</sup> The  $D/t$  range values and equation factors were calculated from Equations (E.2), (E.4), (E.20), (E.21), (E.22), (E.25), and (E.26).

### E.1.5 Elastic Collapse Pressure Equation

The minimum collapse pressure for the elastic range of collapse,  $p_E$ , is calculated by Equation (E.7):

$$p_E = \frac{H C}{[(D/t) (D/t - 1)^2]} \quad (E.7)$$

where

$D$  is the specified pipe outside diameter;

$H C$  is the stress conservation factor,  $3.237 \times 10^5$  Mpa ( $46.95 \times 10^6$  psi);

$t$  is the specified pipe wall thickness.



The applicable  $D/t$  range for elastic collapse is shown in Table E.4.

**Table E.4— $D/t$  Range for Elastic Collapse**

Grade <sup>a</sup>	$D/t$ Range <sup>b</sup>
H40	42.64 and greater
–50	38.83 and greater
J55, K55	37.21 and greater
–60	35.73 and greater
–70	33.17 and greater
C75, E75	32.05 and greater
L-N-80	31.02 and greater
C90	29.18 and greater
R95, T95, X95	28.36 and greater
–100	27.60 and greater
P105, G105	26.89 and greater
P110	26.22 and greater
–120	25.01 and greater
Q125	24.46 and greater
–130	23.94 and greater
S135	23.44 and greater
–140	22.98 and greater
–150	22.11 and greater
–155	21.70 and greater
–160	21.32 and greater
–170	20.60 and greater
–180	19.93 and greater
<sup>a</sup> Grades indicated without letter designation are not API grades but are grades in use or grades being considered for use and are shown for information purposes. <sup>B</sup> The $D/t$ range values were calculated from Equations (E.6), (E.20), and (E.21).	

### E.1.6 Collapse Pressure under Axial Tensile Stress and Internal Pressure

The above equations for collapse resistance  $p_c$  are valid for zero axial stress and zero internal pressure. The external pressure at collapse under combined axial stress and internal pressure can be calculated by replacing the collapse resistance  $p_c$  by the collapse pressure differential  $p_c - p_i$  and at the same time modifying the yield stress to a combined loading equivalent grade according to Equation (E.8) for  $\sigma_a + p_i \geq 0$ :

$$f_{ycom} = \{[1 - 0.75((\sigma_a + p_i) / f_{ymn})^2]^{1/2} - 0.5 (\sigma_a + p_i) / f_{ymn}\} f_{ymn} \quad (E.8)$$

where

$f_{ycom}$  is the combined loading equivalent grade, the equivalent yield strength in the presence of axial stress and internal pressure;

$f_{ymn}$  is the specified minimum yield strength;

$\sigma_a$  is the component of axial stress not due to bending;

$p_i$  is the internal pressure.

Collapse resistance equation factors and  $D/t$  ranges for the combined loading equivalent grade are then calculated by means of Equations (E.2), (E.4), (E.6), (E.20), (E.21), (E.22), (E.25), and (E.26). Using equation factors for the combined loading equivalent grade, collapse resistance under axial stress and internal pressure is calculated by means of Equations (E.1), (E.3), (E.5), and (E.7).

If the yield strength of combined loading equivalent grade ( $f_{ycom}$ ) is less than 24,000 psi, use the collapse equation determined for  $f_{ycom}$  equal to 24,000 psi, but limited by the value obtained from Equation (E.1). ~~API collapse resistance equations are not valid for the yield strength of combined loading equivalent grade ( $f_{ycom}$ ) less than 24,000 psi.~~

Equation (E.8) is based on the Hencky-von Mises maximum strain energy of distortion theory of yielding.

## E.2 Derivation of Collapse Equations

### E.2.1 General

Of the four equations used for collapse pressure, those for yield strength collapse and elastic collapse were derived on a theoretical basis, the plastic equation was derived empirically from 2488 collapse tests for grades K55, N80, and P110, whereas the plastic/elastic transition collapse pressure equation was determined on an arbitrary basis.

### E.2.2 Yield Strength Collapse Pressure Equation Derivation

For heavy wall pipe, the use of plastic collapse Equation (E.3) or  $p_P$  could result in compression stresses ~~equalling~~ equaling or exceeding the yield strength. Although there was experimental evidence that the collapse pressure could exceed the external pressure causing yielding, it was thought unsafe to use a collapse pressure value causing yielding. Therefore, the yield strength collapse is based on the pressure that generates minimum yield stress on the inside wall of the tube, calculated by means of the Lamé equation. The derivation of the Lamé equation can be found in books covering theoretical elastic stress analysis.

### E.2.3 Plastic Collapse Pressure Equation Derivation

Equation (E.3) for plastic collapse pressure,  $p_P$ , and factors  $A_C$ ,  $B_C$ , and  $C_C$  were derived by statistical regression analysis from 402 collapse tests on K55, 1440 collapse tests on N80, and 646 collapse tests on P110 seamless casing. The data were gathered to represent the  $D/t$  ranges typically involved in plastic collapse for the particular grades. The regression analysis resulted in the equations of the **Stewart-type** shown in **Table E.5**, which were published in ASME Paper No. 1969-100. These regression equations [Equations (E.9) and (E.10)] for average collapse pressure are substantially the same as those on which the collapse values given in **API 5C2 (11<sup>th</sup> Edition)** were based.

The difference in the new equations from the old arises from the method of determining minimum values from the average values. The new minimum values were determined by subtracting a constant pressure determined for the particular grade from the average, while the old minimum values were determined by reducing the average values by 25 %.

**Table E.5—Average Plastic Collapse Pressure Regression Equations**

Grade	Average Plastic Collapse Regression Equation	Coefficient of Det. $R^2$	Standard Error $S_p$	Equation Number
K55	$p_P = 164,450/(D/t) - 2976$	0.6478	435	(E.9)
N80	$p_P = 245,600/(D/t) - 5336$	0.8627	719	(E.10)
P110	$p_P = 349,800/(D/t) - 9020$	0.7720	1048	(E.11)

Statistical minimum values for the regression equations are based on one-sided tolerance limits developed by the following methods that can be found in **Reference [118]**. Equations (E.12), (E.13), (E.14), and (E.15) for one-sided tolerance limits are developed by such methods. These tolerance limits are subtracted from the average collapse pressure equations to obtain minimum collapse pressure equations.

$$C_C = t_p(\theta_p) Z_p S_p \quad (E.12)$$

$$t_p(\theta_p) \equiv \{u_{1-\theta} + u_p [(1 - u_p^{2/2f})/N_t + u_{1-\theta}^{2/2f}]^{1/2} / (1 - u_p^{2/2f})\} \quad (E.13)$$

$$t_{0.95}(0.005) = \{2.570 + 1.645 [(1 - 1.3530/(N_t - 1))/N_t + 3.30245/(N_t - 1)]^{1/2} / [1 - 1.3530/(N_t - 1)]\} \quad (E.14)$$

$$Z_p = [1 + 1/N_t + (t/D - a_{t/D})^2 / (N_t s_{t/D})]^{1/2} \quad (E.15)$$

where

$a_{t/D}$  is the average value of  $t/D$  ratios used in the regression;

$C_C$  empirical constant in historical API collapse equation;

$D$  specified pipe outside diameter;

$f$  degrees of freedom =  $N_t - 1$ ;

$N_t$  number of tests;

$S_p$  standard error of estimate of the regression equation;

- $s_{t/D}$  standard deviation of  $t/D$  ratios used in the regression;
- $t$  specified pipe wall thickness;
- $t_p$  tolerance interval corresponding to a confidence level of  $p$  that the proportion of the population not included does not exceed;
- $u_p$  fractile corresponding to confidence level  $p$ ;
- $u_{1-\theta}$  fractile, the deviation from the mean of a standardized normal cumulative distribution that includes the fraction  $1 - \theta_p$  of the population;
- $Z_p$  correction factor for variation in  $t/D$  from average;
- $\theta_p$  the proportion of the population not included.

Equation (E.13) was taken directly from Reference [118]. Equation (E.15) provides a correction for variation from average  $t/D$  used in the regression and is based on information taken from Reference [119].

The quantity  $C_c$  is to be regarded as a tolerance limit to be subtracted from the average collapse pressure equation to obtain the minimum collapse pressure equation. Regarding the term  $(t/D - a_{t/D})$ , the maximum absolute value of this quantity occurring in the test data is to be used in Equation (E.15) for calculating  $Z_p$ .

Equation (E.14) was obtained from Equation (E.13) by taking the confidence level to be 0.95 and  $\theta_p = 0.005$  and substituting the corresponding values of  $u_p = u_{0.95} = 1.645$  and  $u_{1-\theta} = u_{0.995} = 2.574$  obtained from a table of probability integrals.

Values for the tolerance limit  $C_c$  were calculated using Equations (E.12) through (E.15) and are shown in Table E.9.

Subtracting the tolerance limit  $C_c$  values from the average collapse pressure Equations (E.9), (E.10), and (E.11), the following Equations (E.16), (E.17), and (E.18) presented in Table E.6 for minimum collapse pressures,  $p_p$ , are obtained.

**Table E.6—Minimum Plastic Collapse Equations for Grades K, N, and P**

Grade	Average Plastic Collapse Regression Equation	Equation Number
K55	$p_p = 164,450/(D/t) - 4181$	(E.16)
N80	$p_p = 245,600/(D/t) - 7291$	(E.17)
P110	$p_p = 349,800/(D/t) - 11,875$	(E.18)

These equations for minimum plastic collapse pressure are based on the conception that there is a 95 % probability or confidence level that the collapse pressure will exceed the minimum stated with no more than 0.5 % failures.

While Equations (E.16), (E.17), and (E.18) could be used in the form shown, they have been converted to the following standard form, primarily to facilitate extrapolation and interpolation to obtain collapse equations for other grades for which adequate collapse test data are not available from which to obtain equations directly:

$$p_P = f_{ymn} [A_c / (D/t) - B_c] - C_c \quad (E.19)$$

where

$A_c$  is the empirical constant in historical API collapse equation;

$B_c$  is the empirical constant in historical API collapse equation;

$C_c$  is the empirical constant in historical API collapse equation;

$D$  is the specified pipe outside diameter;

$f_{ymn}$  is the specified minimum yield strength;

$p_P$  is the pressure for plastic collapse;

$t$  is the specified pipe wall thickness.

The following factors  $A_c$ ,  $B_c$ , and  $C_c$  for grades K55, N80, and P110, given in Table E.7, were curve-fit to provide equations for determining these factors for other grades by extrapolation and interpolation.

**Table E.7—Plastic Collapse Equation Factors for Grades K, N, and P**

Grade	$A_c$	$B_c$	$C_c$
K55	2.990	0.0541	1205
N80	3.070	0.0667	1955
P110	3.180	0.0820	2855

$$A_c = 2.8762 + 0.10679 \times 10^{-5} f_{ymn} + 0.21301 \times 10^{-10} f_{ymn}^2 - 0.53132 \times 10^{-16} f_{ymn}^3 \quad (E.20)$$

$$B_c = 0.026233 + 0.50609 \times 10^{-6} f_{ymn} \quad (E.21)$$

$$C_c = -465.93 + 0.030867 f_{ymn} - 0.10483 \times 10^{-7} f_{ymn}^2 + 0.36989 \times 10^{-13} f_{ymn}^3 \quad (E.22)$$

where

$A_c$  is the empirical constant in historical API collapse equation;

$B_c$  is the empirical constant in historical API collapse equation;

$C_c$  is the empirical constant in historical API collapse equation;

$f_{ymn}$  is the specified minimum yield strength.

Factors for grades K55, N80, and P110 calculated using Equations (E.20), (E.21), and (E.22) are given in Table E.8.

**Table E.8—Plastic Collapse Equation Factors for Grades K, N, and P as Calculated**

Grade	$A_c$	$B_c$	$C_c$
-------	-------	-------	-------

K55	2.991	0.05410	1206
N80	3.071	0.06670	1955
P110	3.181	0.08192	2852

The maximum deviation of the factors determined by the equations from those determined by regression analysis is 0.122 %.

The tolerance limit for grades K55, N80, and P110 to be subtracted from average collapse equations to convert to a minimum base is shown in Table E.9.

**Table E.9—Tolerance Limit  $C_c$  to Be Subtracted from Average Collapse Equations to Convert to a Minimum Base**

Grade	$C_c$
K55	1205
N80	1955
P110	2855

As additional data become available, these equations can be verified or modified where necessary. Analysis of collapse test data should conform to the principles followed in developing the present equations.

## E.2.4 Transition Collapse Pressure Equation Derivation

When the curves of the equations for average plastic collapse pressures are extended to higher  $D/t$  values, they intersect the average elastic collapse pressure curve. However, as the curves for minimum plastic collapse pressures are extended to higher  $D/t$  values, they fall below the minimum elastic collapse pressure curve without intersecting it. To overcome this anomaly, a plastic/elastic transition collapse pressure equation has been developed that intersects the  $D/t$  value where the average plastic collapse pressure equation gives a collapse pressure of zero and is tangent to the minimum elastic collapse pressure Equation (E.7). This equation is used to determine minimum collapse pressures between its tangency to the elastic collapse pressure curve and its intersection with the plastic collapse pressure curve. This is shown in Figure E.1 for grade N80 casing.

The equation for plastic/elastic transition collapse pressure,  $p_T$ , is of the **Stewart-form** as follows:

$$p_T = f_{ymn} [F_c / (D/t) - G_c] \quad (E.23)$$

where

$D$  is the specified pipe outside diameter;

$F_c$  is the empirical constant in historical API collapse equation;

$f_{ymn}$  is the specified minimum yield strength;

$G_c$  is the empirical constant in historical API collapse equation;

$t$  is the specified pipe wall thickness.

The two conditions mentioned:

intersection with the average collapse pressure curve  $p_P$  (average) =  $f_{ymn} [A_c/(D/t) - B_c]$ , where  $p_P$  (average) = 0

tangent to the elastic curve

$$p_E = 46.95 \times 10^6 / [(D/t) (D/t - 1)^2] \quad (E.24)$$

where

$D$  is the specified pipe outside diameter;

$p_E$  is the pressure for elastic collapse;

$t$  is the specified pipe wall thickness;

permit evaluation of  $A_c$  and  $B_c$  according to Equations (E.25) and (E.26) as follows:

$$F_c = 46.95 \times 10^6 [(3 B_c/A_c)/(2 + B_c/A_c)]^3 / \{ f_{ymn} [(3 B_c/A_c)/(2 + B_c/A_c) - B_c/A_c] [1 - (3 B_c/A_c)/(2 + B_c/A_c)]^2 \} \quad (E.25)$$

$$G_c = F_c B_c/A_c \quad (E.26)$$

where

$A_c$  is the empirical constant in historical API collapse equation;

$B_c$  is the empirical constant in historical API collapse equation;

$C_c$  is the empirical constant in historical API collapse equation;

$F_c$  is the empirical constant in historical API collapse equation;

$f_{ymn}$  is the specified minimum yield strength;

$G_c$  is the empirical constant in historical API collapse equation.

## E.2.5 Elastic Collapse Pressure Equation Derivation

The minimum elastic collapse pressure equation was derived from the theoretical elastic collapse pressure equation developed by Clinedinst<sup>[54]</sup>, where  $p_E$  is the pressure for elastic collapse:

$$p_E = 2^E / (1 - \nu^2) / [(D/t) (D/t - 1)^2] \quad (E.27)$$

where

$D$  is the specified pipe outside diameter;

$E$  is Young's modulus;

$t$  is the specified pipe wall thickness;

$\nu$  is Poisson's ratio.

The curve plotted from the equation for theoretical elastic collapse, assuming  $E = 30 \times 10^6$  and  $\nu = 0.3$ , was found to be an adequate upper boundary for collapse pressure as determined by test.

The average collapse resistance equation was taken as 95 % of the theoretical equation for elastic collapse resistance, rounded to two decimals. The minimum elastic collapse resistance equation was taken as 75 % of the average elastic collapse resistance equation, rounded to three decimals:

$$p_E = \frac{H C}{46.95} \times 10^6 / [(D/t) (D/t - 1)^2] \quad (E.28)$$

where

$D$  is the specified pipe outside diameter;

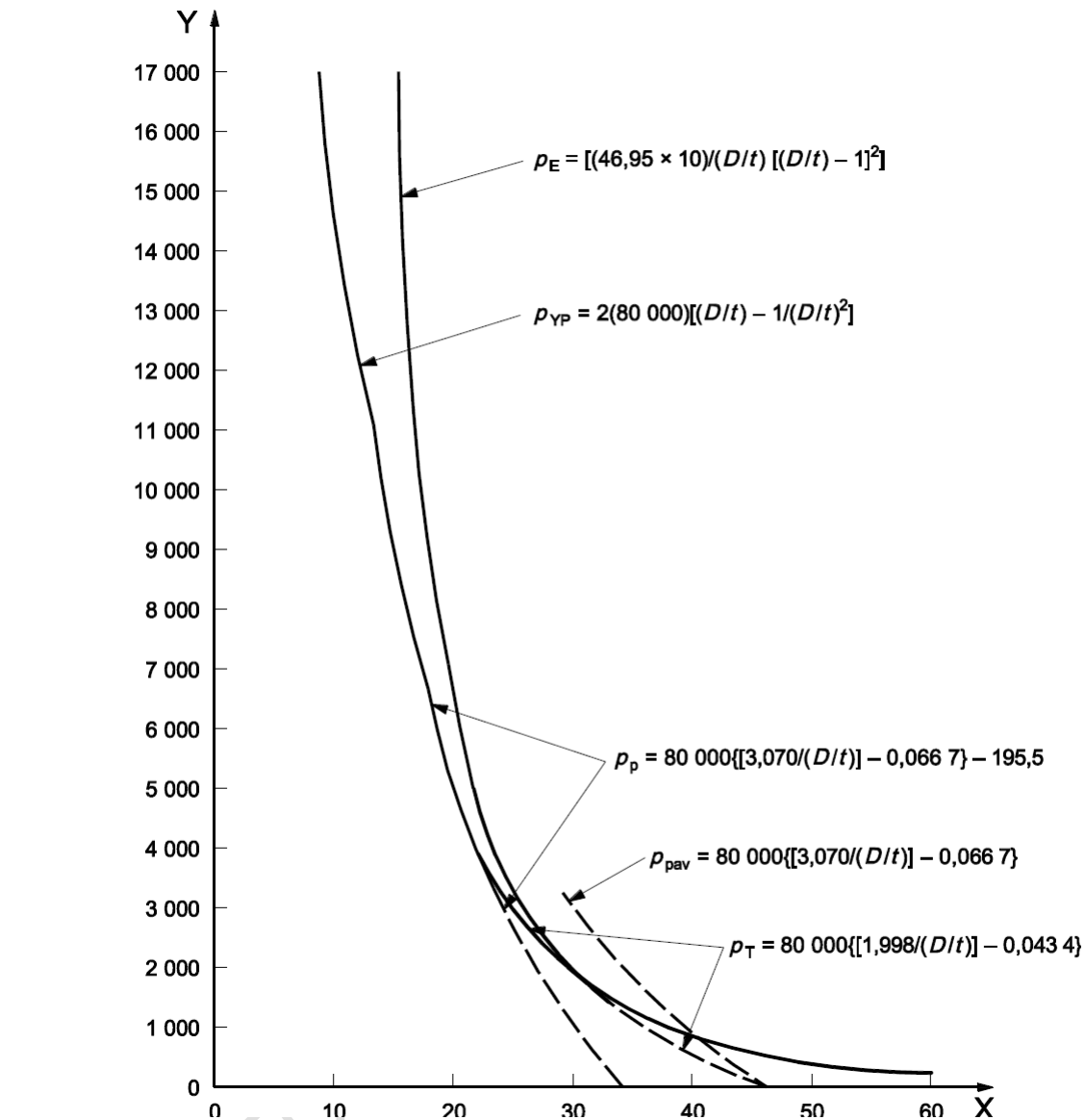
$H C$  is the stress conservation factor,  $3.237 \times 10^5$  Mpa ( $46.95 \times 10^6$  psi);

$p_E$  is the pressure for elastic collapse;

$t$  is the specified pipe wall thickness.

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#### Key

- X  $D/t$
- Y collapse pressure, psi
- $D$  is the specified pipe outside diameter
- $t$  is the specified pipe wall thickness
- $p_E$  pressure for elastic collapse
- $p_{YP}$  pressure for yield strength collapse
- $p_P$  pressure for plastic collapse
- $p_{Pav}$  pressure for average plastic collapse
- $p_T$  pressure for transition collapse

Figure E.1—Grade N80 Transition Collapse Equation Derivation

## Annex F (informative)

### Development of Probabilistic Collapse Performance Properties

#### F.1 Introduction

##### F.1.1 Limitations of Previous Approach

The technical basis of the **API 5C3 (6<sup>th</sup> Edition)** collapse strength equations was developed in the early 1960s. Various limitations have been identified since their first publication, as follows.

Some of the collapse tests were for short specimens ( $L/D = 2$ ), which are now known to overestimate the collapse strength of real pipe **[65], [98], [99]**.

The collapse strength equations resulted in a widely varying margin between the ultimate and design collapse strengths over the  $D/t$  range for well tubulars <sup>[75]</sup>, and thus also in predicted failure probability **(see SPE 48331)**.

The mean value equations were relatively poor predictors of ultimate collapse strength, and modern formulations have been shown to be much more accurate **(Table F.2)**.

The same equations were used for both quenched and tempered (Q&T) and non-Q&T pipe. However, recent work has shown that the two classes have different collapse behavior, and therefore need individual strength equations.

The collapse test specimens were manufactured using a wide range of production methods (e.g. seamless and welded, cold and hot rotary straightening), and no attempt was made to determine the various effects on collapse strength. Subsequent work <sup>[44]</sup> has demonstrated that both straightening and heat treatment have a significant effect on collapse strength, and thus should be included explicitly in any modern treatment.

The formulation for plastic collapse strength was implicitly based on the assumption that collapse strength is proportional to the specified minimum, rather than the actual, yield stress. This is acceptable **if** the ratio of actual to specified minimum yield stress is constant for all grades. However, analysis of production quality data has shown that this ratio varies considerably by grade, and thus that the previous treatment can and should be improved.

Finally, the previous approach could not accommodate non-API grades, such as high-collapse (HC) pipe.

Given the case, it can be asked why the new probabilistic treatment is given here, as an informative annex, with the old (1963) method retained in Section 8. There was a broad consensus that the new collapse strength equations (Sections F.2 and F.4) are more accurate than the old; that the statistical data for a given production case (F.3.1 to F.3.3) can be reliably determined; and that the probabilistic method (Section F.5) gives the correct results in such cases.

However, at present there is no consensus on how pipe data for all **API** production should be characterized for calculation of design ratings. The text of F.3.4 represents the efforts in this regard. There is some belief that the ratings thus calculated (Section F.6) are already an improvement on the old, and therefore should now replace the old Section 8. Others, however, believed that the industry needs more time to consider the new method and, in particular, the question of characterization of probabilistic data for worldwide production.

The new work has therefore been kept as informative at this stage. It is hoped that as the industry becomes more familiar with the method, the question of data characterization can be fully resolved. In this event, the document will be reissued with the new guidance (Section F.6) replacing the existing Section 8.

### F.1.2 Choice of Method

Design collapse strength can be determined either directly, from collapse test results (Annex G), or indirectly using production quality statistics (that is, the statistics of parameters such as yield stress, outside diameter, and wall thickness), and probabilistic analysis with a predictive equation for ultimate collapse strength (Annex H).

The direct method requires collapse test data for every pipe case (outside diameter, weight, and grade), for a representative range of mills. As these data were not available, it was necessary to use the second approach. The advantage of the indirect method is that production quality data are more readily available than collapse data, and in larger quantities. Collapse data are only necessary to choose the ultimate strength equation and determine any empirical adjustment(s).

The accuracy of performance properties obtained using the indirect approach depends on the accuracy of both the calculation method and the probabilistic data. Comparison for individual pipe sizes (Table F.1) shows that, in general, the direct and indirect calculation methods give very similar results for both the large and small dataset analyses.

The probabilistic data were a combination of measurement data contributed by participating mills, and potentially governing design cases chosen by the workgroup, based on production experience. Although any such choices will be arbitrary to some extent, comparison with ensemble collapse test data (Figures F.12 and F.13) shows that the desired target reliability level (TRL) is being achieved.

Moreover, the new performance properties give a near-uniform predicted safety level (Figure F.10), and therefore correct the primary limitation of the old collapse strength equations (Annex E).

The application of the method is described in the following subsections. Limited space precludes more than an overview of analysis details, and the reader is referred to the literature [52], [81], [103] for a fuller treatment.

**Table F.1—Comparison of Design Collapse Strengths**

<sup>a</sup> Collapse strength distribution not Gaussian.												
0.9625	Mean				0.9900	Mean					250	Total
0.9735	8.643	8.658	8.414	0.9906	8.840	8.859	8.757	0.0634	10.47	60	Hot	
0.9184	6.896	7.363	6.333	0.9885	7.139	7.608	7.057	0.0802	8.896	21	Cold	
0.9850	8.323	8.189	8.198	0.9933	8.539	8.403	8.482	0.0709	10.38	99	Hot	
0.9731	7.282	7.153	7.086	0.9877	7.538	7.409	7.445	0.0806	9.397	70	Cold	
0.9843	Mean	0.9946			Mean					656	Total	
0.9822	8.916	8.677	8.757	0.9958	9.060	8.836	9.022	0.0708	11.03	129	None	
0.9884	7.097	6.966	7.015	0.9956	7.209	7.079	7.177	0.0548	8.357	116	None	
0.9766	5.595	5.502	5.464	0.9937	5.751	5.663	5.715	0.0533	6.625	36	None	
0.9789	9.131	8.670	8.938	0.9937	9.377	8.951	9.318	0.0583	10.96	49	None	
0.9883	7.031	6.636	6.949	0.9941	7.178	6.791	7.136	0.0529	8.261	84	None	
0.9766	4.572	4.366	4.465	0.9932	4.709	4.519	4.677	0.0592	5.518	42	None	
0.9981	10.53	9.769	10.51	1.0009	10.73	10.00	10.74	0.0353	11.81	56	None	
0.9795	8.019	7.740	7.855	0.9897	8.178	7.905	8.094	0.0560	9.459	76	None	
0.9903	5.649	5.374	5.594	0.9945	5.787	5.510	5.755	0.0502	6.609	68	None	
Direct/ Correl. Indirec †	Correl- ated	Indep- endent	Direct (Collapse Test)	Direct/ Correl. Indirec †	ksi			COV	Mean ksi			
					Correl- ated	Indep- endent	Direct (Collap se Test)					
Small Dataset Ratings				Large Dataset Ratings				Collapse Test		Sampl	ening Straigh Rotary	

5-1/2 in. 17.0 lb/ft P110
5-1/2 in. 17.0 lb/ft
4-1/2 in. 11.6 lb/ft P110
4-1/2 in. 11.6 lb/ft P110
<b>Tenaris 2004</b>
9-5/8 in. 53.5 lb/ft HC95
9-5/8 in. 47.0 lb/ft HC95
9-5/8 in. 43.5 lb/ft HC95
7-5/8 in. 33.7 lb/ft HC95
7-5/8 in. 29.7 lb/ft HC95
7-5/8 in. 26.4 lb/ft HC95
7 in. 29.0 lb/ft HC95
7 in. 26.0 lb/ft HC95
7 in. 23.0 lb/ft HC95
<b>Nippon 1977-2000</b>

**Table F.2—Predictive Accuracies (Q&T Only)**

Actual Collapse Strength/Predicted Collapse Strength														
Abbas and 1995-1995	API Bulletin 5C3 (Average)		API Bull. 5C3 /Clinedinst 1985		Haagsma and Schaap 1981		Jianzeng and Taihe 2001		Klever-Tamano		Tamano et al. 1983		Tamano Modified 4	
	Mean	COV	Mean	COV	Mean	COV	Mean	COV	Mean	COV	Mean	COV	Mean	COV
0.069			1.035	<b>0.049</b>	0.955	0.067	0.930	0.069	0.956	0.055	0.968	0.064	0.964	0.064
0.046	0.978	0.083	1.016	0.048	0.925	0.082	0.973	0.060	0.989	0.044	1.004	0.051	<b>0.996</b>	<b>0.042</b>
0.072	1.088	0.136	1.054	0.080	0.944	0.100	0.978	0.077	0.992	<b>0.057</b>	<b>1.003</b>	0.069	0.981	0.060
0.104	1.026	0.107	1.060	0.097	0.947	0.115	0.994	0.110	<b>1.004</b>	<b>0.093</b>	1.019	0.104	0.996	0.095
<b>0.077</b>	1.019	0.085	1.055	0.093	0.956	0.107	0.988	0.095	<b>1.000</b>	0.078	1.017	0.087	0.997	0.077
0.066	1.017	0.097	1.065	0.071	0.966	0.083	0.974	0.082	0.993	<b>0.063</b>	<b>1.006</b>	0.073	0.987	0.066
0.108	1.037	0.136	1.062	0.099	0.965	0.106	<b>1.004</b>	<b>0.080</b>	1.030	0.102	1.035	0.085	1.019	0.098
0.098	1.114	0.168	1.052	<b>0.082</b>	1.009	0.084	0.976	0.087	<b>1.003</b>	0.084	1.021	0.084	1.016	0.090
0.080	1.040	0.116	1.050	0.077	0.958	0.093	0.977	0.083	<b>0.996</b>	<b>0.072</b>	1.009	0.077	0.994	0.074
0.080			1.049	0.081	0.952	0.097	0.981	0.083	<b>0.997</b>	<b>0.071</b>	1.010	0.077	0.992	0.072
0.024					0.975	0.044			0.993	<b>0.011</b>	1.021	0.032	<b>1.000</b>	0.015
0.125	0.961	0.111	<b>1.013</b>	0.114	0.949	<b>0.109</b>	0.950	0.118	0.944	0.115	0.979	0.118	0.949	0.122
0.076	1.158	0.066	1.132	<b>0.054</b>	<b>1.054</b>	0.075	1.073	0.085	1.073	0.065	1.101	0.078	1.069	0.070
0.052	1.124	0.072	1.135	0.063	1.030	0.073	1.016	0.073	1.005	<b>0.050</b>	1.039	0.063	<b>1.000</b>	0.052
0.054	1.082	0.050	1.135	0.057	1.027	0.064	1.007	0.079	1.009	<b>0.046</b>	1.037	0.066	<b>1.006</b>	0.052
0.056	1.107	0.057	1.077	0.048	0.988	0.065	0.990	0.079	<b>1.000</b>	<b>0.053</b>	1.017	0.069	0.988	0.055
<b>0.058</b>	1.127	0.076	1.124	0.059	1.030	0.063	<b>1.003</b>	0.079	0.995	0.059	1.028	0.069	0.992	0.060
0.058	1.126	0.069	1.034	0.061	0.901	0.063	0.919	0.055	<b>0.968</b>	<b>0.054</b>	0.952	0.056	0.946	0.057
0.090	1.037	0.087	1.074	<b>0.062</b>	0.985	0.088	1.008	0.088	<b>1.001</b>	0.072	1.029	0.082	0.997	0.082
0.049	1.218	0.148	1.089	0.059	1.014	<b>0.048</b>	0.967	0.062	0.973	0.056	<b>0.997</b>	0.052	0.968	0.051
0.069	1.104	0.082	1.090	0.064	<b>0.998</b>	0.072	0.993	0.080	0.996	<b>0.063</b>	1.020	0.073	0.991	0.067
0.068	1.108	0.088	1.115	0.073	1.017	0.080	1.005	0.084	<b>1.001</b>	<b>0.064</b>	1.031	0.076	0.996	0.067
0.021					1.032	0.046			1.007	<b>0.013</b>	1.039	0.047	<b>1.007</b>	0.023
NOTE 1 Of the datasets above, those used for calibration of empirical coefficients were as follows:														
—Klever-Tamano: all;														
—Tamano modified 4: all except manufacturer FD00, Nippon 1988-2000, and DEA-130;														
—Abbasian and Parfitt 1995-9a: all except Japanese RR 1987, manufacturer FD00, Nippon 1988-2000, and DEA-130;														
—API 5C3/Clinedinst 1985.														
NOTE 2 Boldface type denotes the best fit for each dataset.														

NOTE 1 Of the datasets above, those used for calibration of empirical coefficients were as follows:

—Kleiver-Tamano: all;

—Tamano modified 4: all except manufacturer FD00, Nippon 1988-2000, and DEA-130;

—Abbassian and Parfitt 1995-9a: all except Japanese RR 1987, manufacturer FD00, Nippon 1988-2000, and DEA-130;

—API 5C3/Clinedinst 1985;

NOTE 2 Boldface type denotes the best fit for each dataset.

a. Includes application from Grant-Driscoll 2000.

		Mean			
Dataset	Axial Force	Position(s) for Geometry Measurements	Tests		
API product	No	Multiple	89	0.983	
Mannesmann 1983	No	Not known	106	1.023	
API 1985	No	Pipe center	433	0.996	
Nippon 1977-87	No	Pipe center	95	1.020	
Nippon 1988-2000	No	Pipe end	129	1.019	
Manufacturer FD00	No	Pipe center	52	1.019	
DEA-130	Yes	Collapse point	96	1.040	
API 1981	Yes	Multiple	63	1.030	
Mannesmann 1983					
Dataset average			1138	1.016	
Ensemble average <sup>a</sup>				1.012	
Dataspace average <sup>a</sup>				1.026	
HC product	No	Pipe end	141	0.977	
API 1982	No	Pipe end	107	1.093	
API 1987	No	Pipe center	794	1.016	
Nippon 1977-87	No	Collapse point	304	1.019	
Vallourec 1987-98	No	Not known	54	1.005	
Japanese RR 1987	No	Pipe center	291	1.012	
Nippon 1988-2000	No	Pipe end	75	0.960	
Manufacturer FD00	No	Pipe center	26	1.019	
DEA-130	Yes	Collapse point	56	0.986	
API 1981					
Dataset average			1848	1.010	
Ensemble average				1.014	
Dataspace average				1.026	

## F.2 Selection of Ultimate Limit State Equation

### F.2.1 Collapse Test Data

Table F.2 lists the collapse test datasets [88], [100] made available to the workgroup, and the manufacturer of each sample is identified in all but five datasets. The anonymous datasets were provided in confidence as part of the 1982, 1985, and 1987 API collapse test programs, the 1999-2000 API data survey, and Drilling Engineering Association project DEA-130.

The collapse test ensemble contained 3171 tests, broken down as follows:

Q&T pipe: 2986 tests (1138 for API grades, 1848 for HC);

non-Q&T pipe: 185 tests, all for API grades.

All tests were for long specimens ( $L/D \geq 7$ ), and in each case the relevant strength and geometry properties (yield stress, average outside diameter, average wall thickness, eccentricity, ovality, and residual stress) were accurately measured prior to collapse testing. All datasets were QA checked, duplicate and suspect lines rejected, and approved by unanimous consensus of the workgroup.

### F.2.2 Candidate Ultimate Limit State Equations

Eleven sets of predictive equations were evaluated against the Q&T collapse test results, namely:

Abbassian and Parfitt 1999a;

API 5C3 (6<sup>th</sup> Edition) [for mean strength, i.e. with the down-rating factors for design collapse strength removed];

API 5C3 mean strength (yield, elastic)/Clinedinst 1985 (plastic) [56];

Haagsma and Schaap 1981 [62];

SPE 26317;

Jianzeng and Taihe 2001 [68];

SPE 48332;

SPE 90904;

Taman~~181xceed~~o et al. 1983 [101];

Tamano modified 4 [48];

Tokimasa and Tanaka 1986 [105].

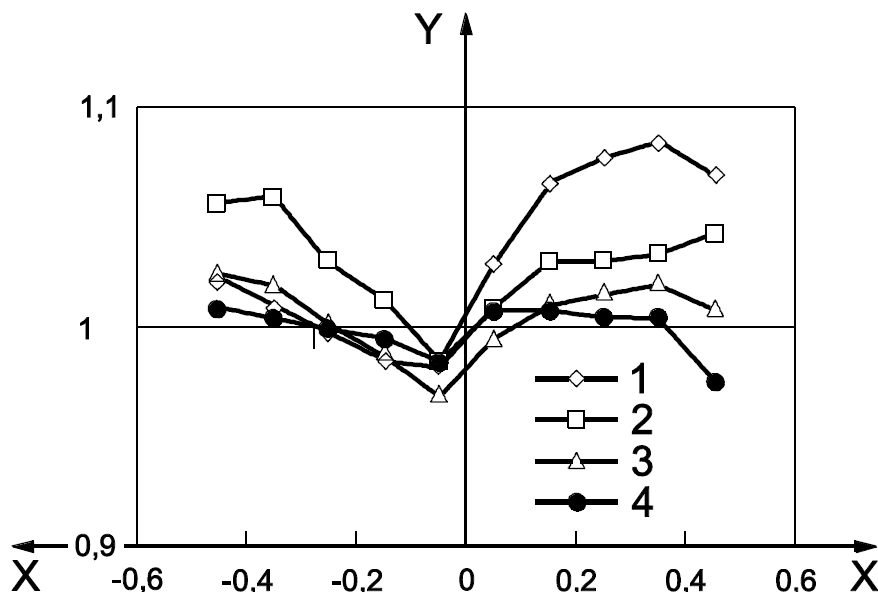
The equations of most of these are discussed in the referenced papers, with three exceptions. **Abbassian and Parfitt 1999a** is an extension of **SPE 29458**; the additions are empirically derived down-rating factors for eccentricity and residual stress, and an empirical adjustment to the elastic collapse strength term to avoid overprediction.

NOTE 1 Elastic collapse occurs at high  $D/t$ , for which failure occurs via buckling; yield collapse occurs at low  $D/t$ , for which failure occurs via yielding.

**Tamano modified 4 is a modification of Tamano et al. 1983.** It involved recalibration of the empirical coefficients on the ovality, eccentricity, and residual stress terms, and addition of an empirical adjustment to the elastic collapse strength term, to give a flat actual/predicted strength response over the input and output dataspace.

NOTE 2 The dataspace is the range of values that physical pipe can assume; the input dataspace is thus the ranges of the strength and geometry variables (yield stress, average outside diameter, average wall thickness, and so forth) (Figure F.2), and the output dataspace is the collapse strength range (Figure F.1).

**Klever-Tamano is a new model inspired by Tamano et al. 1983.** A global adjustment was made to obtain the correct limit behavior, and separate factors applied to the elastic and yield terms, allowing each to be individually calibrated. The yield and elastic equations were rederived, and suggestions were made on improvements to the old API equations. The various empirical coefficients were calibrated to give a flat actual/predicted strength response over the input and output dataspace (Figure F.2).



#### Key

- X log (yield/elastic strength)
- Y mean of actual/predicted collapse strength
- 1 Tamano et al. 1983
- 2 Abbassian and Parfitt 1999a
- 3 Tamano modified 4
- 4 Klever-Tamano

**Figure F.1—Predictive Accuracy (Mean) vs Dataspace Position**

### F.2.3 Predictive Accuracy

#### F.2.3.1 Mean

Table F.2 summarizes predictive accuracy for mean collapse strength by the ultimate limit state (ULS) equation and dataset, as well as for the API and HC ensembles. In each case, the tabular results show the mean and coefficient of variance of actual collapse strength/predicted collapse strength; thus, the most accurate predictive equation is the one with the mean nearest unity and the coefficient of variance nearest zero. Only the eight best-performing equations are shown.

**NOTE** Coefficient of variance is a dimensionless measure of the spread of a random variable, given by standard deviation/mean.

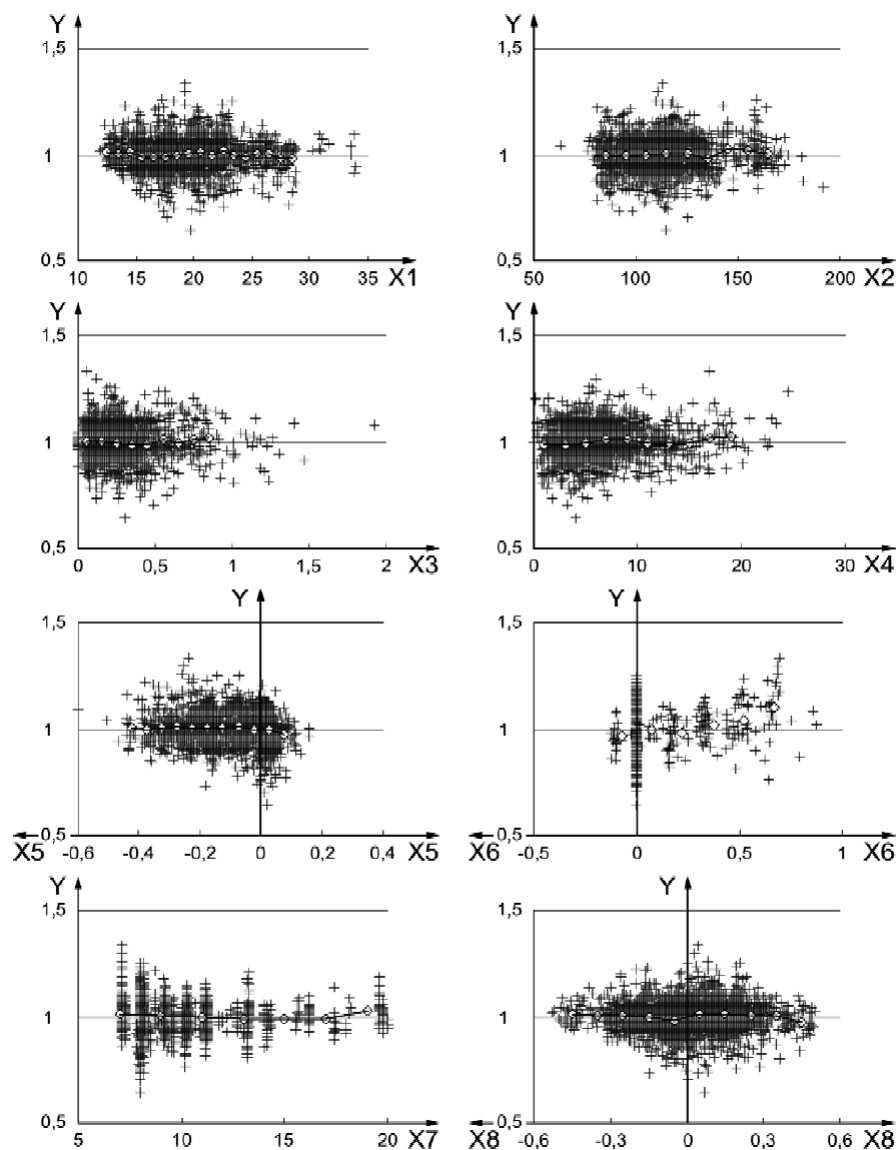
Table F.2 shows that the Klever-Tamano equations have the best combination of a near-unity mean and a low coefficient of variance, for both the API and HC ensembles. Moreover, they give by far the flattest actual/predicted collapse strength response over the dataspace (Figure F.1). Accordingly, the Klever-Tamano equations were taken forward to the next step of the validation process, namely determination of predictive accuracy for collapse strength dispersion.

#### F.2.3.2 Dispersion

Figure F.3 shows predictive accuracy for collapse strength dispersion. The points represent the largest pipe cases (that is, given combinations of OD, weight, and grade) in the combined Nippon 1977-2000 and Tenaris 2004 collapse test datasets (Table F.1). Only pipe cases exceeding 30 tests were used, to ensure



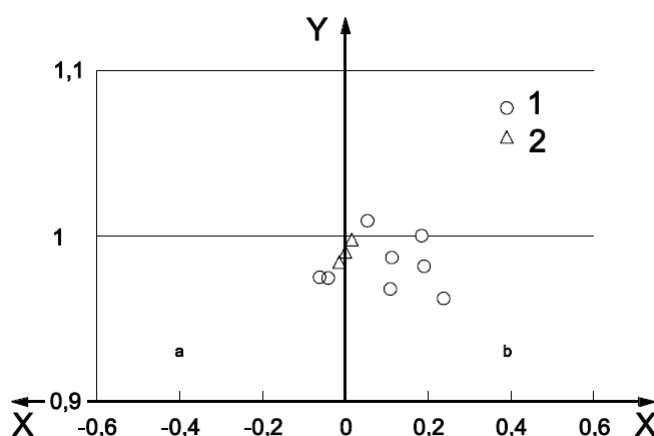
statistical significance, and all cases were checked for homogeneity of production quality variables (see Section F.3). The agreement is very good, with all the points in the range 0.97 to 1.00.



#### Key

- X1 average outside diameter/average wall thickness
- X2 actual yield stress, ksi
- X3 ovality, %
- X4 eccentricity, %
- X5 residual/actual yield stress
- X6 axial/actual yield stress
- X7 sample length/outside diameter
- X8 log (yield/elastic strength)
- Y actual/predicted strength
- + collapse tests
- ◇ mean of collapse tests for each data band

**Figure F.2—Actual/Kleiver-Tamano Predicted Strength vs Dataspace Position**



#### Key

- X log (yield/elastic strength)
- Y actual/predicted coefficient of variance
- 1 Nippon 1977-2000
- 2 Tenaris 2004
- a yield
- b elastic

**Figure F.3—Klever-Tamano Predictive Accuracy (Dispersion) vs Dataspace Position**

NOTE The COVs were calculated including cross-correlation between input variables; this approach improves predictive accuracy, but can be conservatively omitted for calculation of ratings (see F.3.3).

Based on their predictive accuracy for both mean and dispersion, the Klever-Tamano equations were chosen for calculation of collapse ratings.

#### F.2.3.3 Model Uncertainty

The model uncertainty [60] is the statistical variation due to errors and/or limitations in the ULS model; that is, the remaining variability once the effect of variation of the other input parameters has been removed. It is therefore equal to the variation of actual/predicted strength, calculated with the other input variables accurately measured; this is given in the COV columns of Table F.2.

The model uncertainty is included in the probabilistic analysis (see F.5.1.1). This requires knowledge of its probability distribution and PDF parameters. For the Klever-Tamano equations and the Q&T ensemble (2986 collapse tests, Table F.2), mean = 0.999 1, COV = 0.0670, with a Gaussian distribution.

### F.3 Input Variable Data

#### F.3.1 Analysis Methods

##### F.3.1.1 Distribution Type

The production quality data were taken from the collapse test datasets (Table F.2) and the API 1999 data survey. The distribution types for the input variables were determined by plotting the data frequency distributions onto probability scales [52], [103]. The distribution types were as follows:

average OD, average WT, yield stress, residual stress, and model uncertainty: Gaussian;

ovality and eccentricity: two-parameter Weibull.

The PDF parameters were evaluated directly from the moments of the observed datasets [58].

Extreme outliers were removed where necessary [79]. This is because outliers are generally not from the parent process, but are departures from it, typically due to typos in data recording or to production errors. Leaving them in the dataset would therefore lead to unrepresentative PDFs. The proportion of outliers was well below 1 % throughout.

Censoring is best done by the mills, when the data and its correlation to manufacturing process are fresh to hand. If done later, censoring can be performed using the probability scales plot, on which outliers will appear as strong deviations from the best-fit line.

### **F.3.1.2 Ensemble PDFs**

The ensemble PDF for each input variable was obtained by sampling the individual dataset PDFs (Tables F.3 and F.4), using Monte Carlo analysis. This ensures that each dataset has equal weight.

## **F.3.2 Results (Individual Datasets)**

Tables F.3 and F.4 give production quality statistics for each input variable, and Figure F.4 presents the same data in graphical form. Several trends can be identified, as follows:

yield stress bias varies with grade;

mean yield stress varies with batch;

yield stress dispersion varies with mean yield;

yield stress dispersion varies with straightening method (cold/hot);

ovality and eccentricity vary with forming process (seamless/EW);

residual stress varies with straightening method (cold/hot/none).

Table F.3—Production Quality Data (API Survey Only)—Yield Stress

Grade											
H40	J55	K55	L80	L80 9Cr	L80 13Cr	N80	C90	R95	T95	P110	Q125
		1.292 NO 0.0486 CX 191 WE	1.119 NO 0.0387 CX 151 WE			1.243 NO 0.0483 CX 83 WE				1.147 QT 0.0373 CS 133 WE	1.109 QT 0.022.6 CS 121 WE
		1.300 AR 0.0333 CX 220 SS	1.087 QT 0.0239 HR 150 SS			1.214 QT 0.0342 HR 196 SS				1.145 QT 0.0257 HR 220 SS	1.120 QT 0.0272 HR 210 SM
						1.231 QT 0.0331 CX 473 WE				1.117 QT 0.0463 CX 33 WE	
			1.099 QT 0.0366 HR 30 SR			1.205 QT 0.0437 HR 53 SR				1.160 QT 0.0354 HR 46 SR	
		1.091 AR 0.032.6 HR 240 SR	1.089 QT 0.0398 HR 390 SR							1.087 QT 0.0267 HR 300 SR	1.067 QT 0.0133 HR 28 SR
			1.100 QT 0.0344 HR 166 SR			1.233 QT 0.0327 HR 48 SR			1.112 QT 0.0235 HR 36 SR	1.162 QT 0.0291 HR 427 SR	1.117 QT 0.0292 HR 298 SR
	1.198 AR 0.0491 CX 33 SR	1.219 AR 0.0434 CX 49 SR	1.122 QT 0.0401 HR 185 SR			1.216 QT 0.0497 HR 62 SR				1.084 QT 0.0256 HR 335 SR	1.101 QT 0.0322 HR 227 SR
1.364 AR 0.0417 CX 90 WE	1.169 AR 0.0481 CX 684 WE										
	1.199 QT 0.0524 ? 21 SR	1.132 AR 0.0540 ? 39 SR	1.086 QT 0.0472 ? 55 SR			1.187 QT 0.0633 ? 33 SR	1.062 QT 0.0325 HR 42 SS	1.058 QT 0.0176 HR 63 SS	1.066 QT 0.0315 HR 426 SS	1.116 QT 0.0495 ? 26 SR	
1.603 AR 0.0531 CX 98 WE	1.318 AR 0.0655 CX 1094 WE	1.301 NO 0.0521 CX 28 WE	1.098 QT 0.0372 CX 223 WE			1.220 QT 0.0563 CX 594 WE					
					1.126 0.0218 65						
					1.064 0.0262 33						
					1.069 0.0239 250						
1.35 0.0595	1.23 0.0719	1.23 0.0719	1.10 0.0529	1.07 0.0533	1.07 0.0533	1.21 0.0511	1.08 0.0511	1.08 0.0500	1.08 0.0500	1.10 0.0464	1.10 0.0432
			1.10 0.0422	1.07 0.0426	1.07 0.0426	1.21 0.0405	1.08 0.0404	1.08 0.0394	1.08 0.0394	1.10 0.0360	1.10 0.0329

Quantity	
Mean COV Samples	AE04
Mean COV Samples	AP00
Mean COV Samples	CG37
Mean COV Samples	FD00
Mean COV Samples	FF12
Mean COV Samples	HH02
Mean COV Samples	JP01
Mean COV Samples	RH29
Mean COV Samples	SS22
Mean COV Samples	VT01
Mean COV Samples	YF01
Mean COV Samples	1
Mean COV Samples	2
Mean COV Samples	3
Mean <sup>a</sup> COV Samples	CRS ensemble
Mean <sup>a</sup> COV Samples	HRS ensemble

Table F.3—Production Quality Data (API Survey Only)—Yield Stress (*continued*)

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NOTE 1 Type of heat treatment/cold working:			
—AR = as rolled; CR = control rolled; NO = normalized; NT = normalized and tempered; QT = quenched and tempered;			
—CP = cold worked, pilgered; CD = cold worked, drawn.			
NOTE 2 Type of straightening:			
—NS = not straightened; CG = cold gag straightened; CX = cold cross rolled straightened; CS = cold straightened and stress relieved; HR = hot rotary straightened.			
NOTE 3 Process of manufacture:			
—SP = seamless, pilger mill; SM = seamless, plug mill; SR = seamless, retained or floating mandrel mill; SS = seamless, stretch reduced; SE = seamless, hot expanded.			
—WE = welded, electric weld; WL = welded, laser weld; WS = welded, submerged arc weld			
NOTE 4 Mean and COV are of actual yield/nominal yield.			
NOTE 5 The mean mill COVs were calculated from:			
CRS:	COV = -0.0001581 fy + 0.05664		
HRS:	COV = -0.0001517 fy + 0.04743		
where fy is adopted yield stress in ksi. These equations were obtained as the best least squares fit to the mill yield stress vs. COV data (Figure F.4).			
NOTE 6 The ensemble COVs were calculated by factoring the mean mill COVs by the ratio mean (ensemble COV/mean mill COV), where this ratio was developed from the five largest datasets for a given grade and straightening type, as follows:			
Dataset	Ensemble	Mean mill	Ensemble/ COV
K55 CRS	0.0518	0.0455	1.138
L80 HRS	0.0372	0.0341	1.092
N80 HRS	0.0414	0.0327	1.264
P110 HRS	0.0423	0.0291	1.455
Q125 HRS		0.0328	1.235
		Mean	1.237
NOTE 7 Where applicable, datasets were censored by removal of outliers considered not to belong to the parent process, e.g. data entry typos and production errors.			
NOTE 8 The ensemble COV for J/K55 CRS is from Monte Carlo simulation, not the regression line.			
Adopted values for Q&T grades.			

**Table F.4—Production Quality Data (All Datasets)—Other Variables**

Dataset	Quantity	Average OD		Average WT		Ovality		Eccentricity		Residual Stress <sup>a</sup>		
		SL	EW	SL	EW	SL	EW	SL	EW	Cold Str.	Hot Str.	Not Str.
API 1981	Mean	1.0048	—	1.0054	—	0.197	—	5.471	—	−0.209	—	0.0283
	COV	0.00118	—	0.0194	—	0.613	—	0.566	—	0.195	—	0.377
	Samples	140	—	140	—	140	—	140	—	84	—	56
API 1982	Mean	1.0077	1.0098	1.0099	1.0011	0.525	0.492	8.828	5.114	—	—	—
	COV	0.00184	0.00233	0.0307	0.0285	0.558	0.556	0.463	0.605	—	—	—
	Samples	64	77	64	77	64	77	64	77	—	—	—
API 1985	Mean	—	—	—	—	0.298	0.603	7.927	4.534	−0.243	—	—
	COV	—	—	—	—	0.573	0.395	0.482	0.706	0.278	—	—
	Samples	—	—	—	—	321	38	321	38	261	—	—
API 1987	Mean	1.0058	—	1.0058	—	0.345	—	5.769	—	—	−0.123	—
	COV	0.00125	—	0.0264	—	0.588	—	0.479	—	—	0.632	—
	Samples	91	—	91	—	91	—	91	—	—	75	—
Japanese RR 1987	Mean	1.0048	—	1.0176	—	0.239	—	5.204	—	—	—	—
	COV	0.00120	—	0.0225	—	0.433	—	0.291	—	—	—	—
	Samples	54	—	54	—	54	—	54	—	—	—	—
Mannesmann 1983	Mean	1.0017	—	1.0350	—	0.424	—	11.43	—	—	—	—
	COV	0.00276	—	0.0289	—	0.550	—	0.374	—	—	—	—
	Samples	169	—	169	—	169	—	169	—	—	—	—
Nippon 1977-87	Mean	1.0035	—	0.9957	—	0.166	—	4.883	—	−0.269	—	0.0196
	COV	0.00206	—	0.0231	—	0.675	—	0.447	—	0.288	—	0.793
	Samples	1247	—	1247	—	1247	—	1247	—	235	—	710
Vallourec 1987-1998	Mean	1.0055	—	1.0110	—	0.198	—	6.496	—	—	—	—
	COV	0.00149	—	0.0240	—	0.593	—	0.411	—	—	—	—
	Samples	295	—	304	—	303	—	299	—	—	—	—
Nippon 1988-2000	Mean	1.0035	—	0.9997	—	0.184	—	4.928	—	−0.239	—	0.0192
	COV	0.00175	—	0.0244	—	0.727	—	0.428	—	0.440	—	0.705
	Samples	583	—	577	—	575	—	578	—	121	—	426
Manufacturer AE04	Mean	—	1.0058	—	1.0125	—	0.182	—	3.342	—	—	—
	COV	—	0.00195	—	0.0137	—	0.658	—	0.501	—	—	—
	Samples	—	999	—	997	—	997	—	1000	—	—	—
Manufacturer CG37	Mean	—	1.0090	—	1.0276	—	0.534	—	1.857	—	—	—
	COV	—	0.00241	—	0.0133	—	0.425	—	0.485	—	—	—
	Samples	—	62	—	62	—	62	—	62	—	—	—
Manufacturer DA01	Mean	1.0065	—	1.0082	—	0.313	—	1.390	—	—	—	—
	COV	0.00132	—	0.0320	—	0.394	—	0.556	—	—	—	—
	Samples	203	—	208	—	201	—	208	—	—	—	—
	Mean	1.0071	—	1.0068	—	0.241	—	5.170	—	−0.211	−0.142	—



Manufacturer FD00	COV	0.00189	—	0.0217	—	0.338	—	0.317	—	0.383	0.189	—
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**Table F.4—Production Quality Data (All Datasets)—Other Variables (*continued*)**

Dataset	Quantity	Average OD		Average WT		Ovality		Eccentricity		Residual Stress <sup>a</sup>		
		SL	EW	SL	EW	SL	EW	SL	EW	Cold Str.	Hot Str.	Not Str.
	Samples	203	—	132 <sup>b</sup>	—	204	—	194	—	84	54	—
Manufacturer FF12	Mean	1.0056	—	1.0010	—	0.122	—	3.493	—	—	—	—
	COV	0.00102	—	0.0182	—	0.555	—	0.478	—	—	—	—
	Samples	1012	—	1012	—	1009	—	1008	—	—	—	—
Manufacturer HH02	Mean	1.0062	—	1.0114	—	0.269	—	1.276	—	—	—	—
	COV	0.00110	—	0.0357	—	0.389	—	0.592	—	—	—	—
	Samples	957	—	991	—	956	—	987	—	—	—	—
Manufacturer JP01	Mean	1.0064	—	—	—	0.181	—	—	—	—	—	—
	COV	0.00134	—	—	—	0.410	—	—	—	—	—	—
	Samples	655	—	—	—	648	—	—	—	—	—	—
Manufacturer HM03	Mean	—	—	—	—	—	—	—	—	—	−0.149	—
	COV	—	—	—	—	—	—	—	—	—	0.652	—
	Samples	—	—	—	—	—	—	—	—	—	32	—
Tenaris 2002	Mean	—	—	—	—	—	—	—	—	—	−0.185	—
	COV	—	—	—	—	—	—	—	—	—	0.247	—
	Samples	—	—	—	—	—	—	—	—	—	74	—
DEA-130	Mean	1.0064	1.0074	1.0104	0.9903	0.227	0.328	4.699	1.484	−0.206	—	—
	COV	0.00180	0.00192	0.0171	0.0136	0.460	0.539	0.495	0.477	0.297	—	—
	Samples	96	44	94	43	94	44	95	43	65	—	—
Manufacturer FD00	Mean	—	—	—	—	—	—	—	—	−0.282	−0.091	—
	COV	—	—	—	—	—	—	—	—	0.232	0.398	—
	Samples	—	—	—	—	—	—	—	—	93	235	—
Ensemble <sup>c, d</sup>	Mean	1.0059	—	1.0069	—	0.217	—	3.924	—	−0.237	−0.138	0.0224
	COV	0.00181	—	0.0259	—	0.541	—	0.661	—	0.332	0.507	0.628

NOTE 1 SL = seamless, EW = welded.

NOTE 2 The quantities measured were as follows:

Variable	Quantity	Units	Distribution
Yield stress	Actual yield/nominal yield	None	Gaussian
Average OD	Average OD/nominal OD	None	Gaussian
Average WT	Average WT/nominal WT	None	Gaussian
Ovality	(Maximum OD – minimum OD)/average OD	%	Two-parameter Weibull
Eccentricity	(Maximum WT – minimum WT)/average WT	%	Two-parameter Weibull
Residual stress	Residual stress/actual yield stress	None	Gaussian

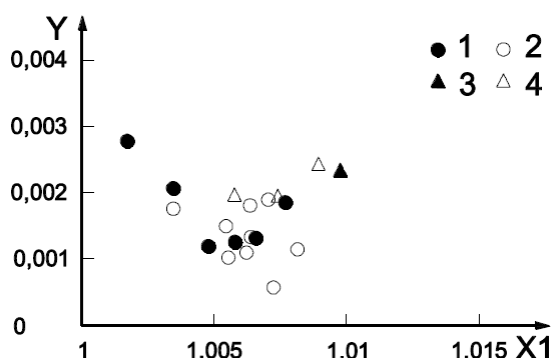
NOTE 3 The measurement basis for the geometry variables was as follows:

—local, single-point: API 1982, API 1987, Nippon 1977-87, Vallourec 1987-1998, Nippon 1988-2000;

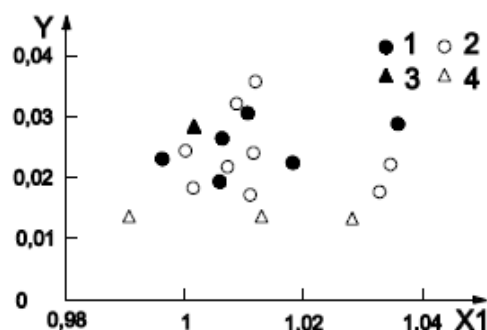
—local, multiple-point average: API 1981, Mannesmann 1983;

—not known: API 1985, Japanese RR 1985-1987.

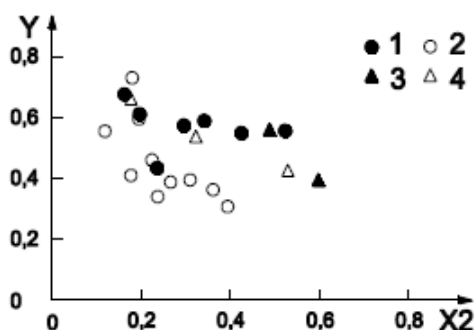
- a Sign convention: stress at ID face, tension positive.
- b The HC dataset (75 lines) had a much higher bias and is therefore not representative of API pipe.
- c With each dataset normalized to the same number of samples.
- d For the geometric properties, the ensemble values were calculated from the post-1987 data only.



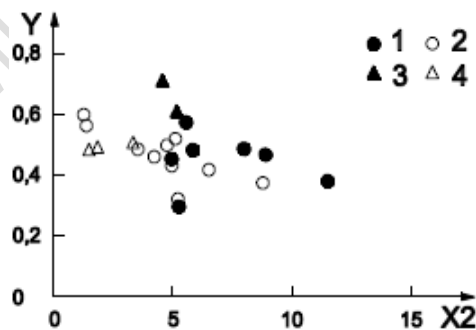
a) Average outside diameter



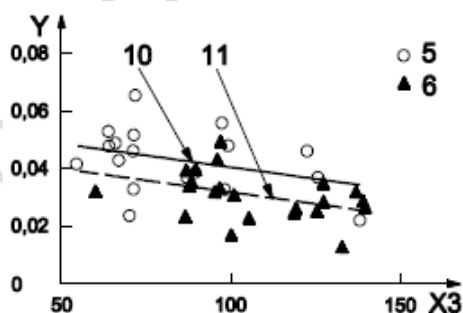
b) Average wall thickness



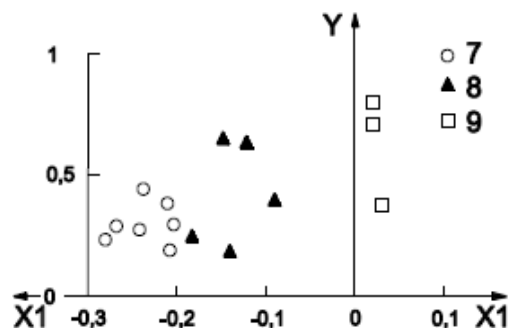
c) Ovality



d) Eccentricity



e) Yield stress (API survey only)



f) Residual stress

**Figure F.4—Pipe Dimension and Stress Data**

**Key**

- X<sub>1</sub> mean
- X<sub>2</sub> mean, %
- X<sub>3</sub> mean, ksi
- Y coefficient of variance
- 1 seamless—pre-1987 data
- 2 seamless—post-1987 data
- 3 EW—pre-1987 data
- 4 EW—post-1987 data
- 5 API, CRS—post-1987 data
- 6 API, HRS—pre-1987 data
- 7 CRS—post-1987 data
- 8 HRS—pre-1987 data
- 9 not rotary straightened—post-1987 data
- 10 trend line for CRS product
- 11 trend line for HRS product

NOTE The quantities measured were as follows:

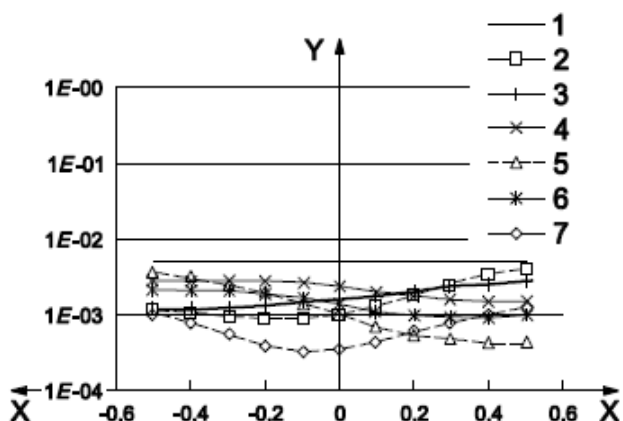
Variable	Quantity	Basis	Units
Average outside diameter	Average outside diameter/nominal outside diameter	Local	
Average wall thickness	Average wall thickness/nominal wall thickness	Local	
Ovality	(Maximum outside diameter – minimum OD)/average OD	Local	%
Eccentricity	(Maximum WT – minimum WT)/average WT	Local	%
Residual stress	Residual stress/actual yield stress	Local	

**Figure F.4—Pipe Dimension and Stress Data** (*continued*)

It is therefore necessary to prepare separate collapse ratings for CRS and hot rotary straightened (HRS) product. In principle, it is also necessary to prepare separate ratings for seamless and EW pipe; however, the EW production data are currently insufficient to permit this. Present results suggest that strengths calculated for seamless pipe are generally conservative for welded pipe from the higher-performing mills [42].

### F.3.3 Input Variable Correlation

Statistical analysis shows that, in general, the input variables have a slight negative cross-correlation. Figure F.5 shows the effect of including cross-correlation. The predicted failure probabilities  $\phi_f$  were calculated using the production quality statistics for each of the larger collapse test datasets, for the down-rating factors obtained from independent variable analysis (F.5.4.1).  $\phi_f$  for correlated variables is lower than for independent variables (compare Figure F.5 with Figure F.10, for which the same down-rating factors give  $\phi_f$  very close to the TRL). It is therefore conservative to omit the effect and treat the input variables as independent.



#### Key

X	log (yield/elastic strength)	4	Nippon 1977 API (433 tests)
Y	predicted failure probability	5	Manufacturer FD00 API (129 tests)
1	TRL	6	Nippon 1977 HC (794 tests)
2	API 1982 (141 tests)	7	API 1987 (107 tests)
3	Nippon 1988 HC (291 tests)		

NOTE L80 seamless CRS,  $k_e = 0.83$ ,  $k_y = 0.855$ .

**Figure F.5—Effect of Input Variable Cross-correlation**

### F.3.4 Choice of Data for Calculation of Ratings

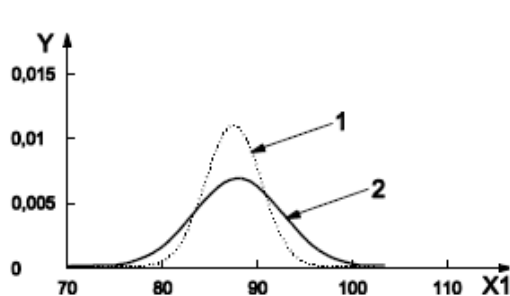
The ensemble data in F.3.2 represents the historical range of aim points for yield stress, wall thickness, and so forth. Ensemble PDFs may govern and shall be considered when developing collapse ratings.

The other potentially governing case is that of a single lot or batch. Production aim points may be placed anywhere within the ranges specified by **API 5CT**, commensurate only with an economic reject rate. The in-lot dispersion is much lower than the ensemble dispersion, and therefore a potentially governing case is the in-lot PDF placed adjacent to the relevant tolerance limit. Figure F.6 illustrates possible PDFs for calculating collapse performance properties, developed as given in Table F.5.

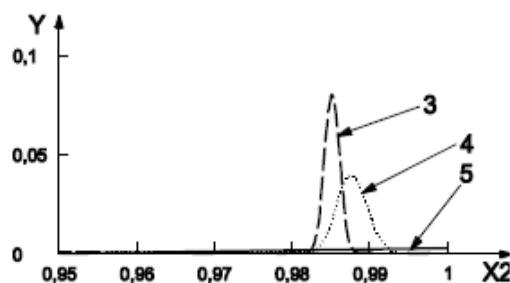
**Table F.5—Development of PDFs for Potentially Governing Cases**

Parameter	0.5 % Exceedence Value Located at	PDF Dispersion Given by
Yield stress	Minimum yield	15,000 psi between 0.5 %- <del>exceedence</del> 194xceedance limits
Average WT	0.9825 specified WT	COV
Ovality	1 %	COV
Eccentricity	20 %	COV
Residual stress (CRS)	Residual/yield = -0.4	COV
Residual stress (HRS)	Residual/yield = -0.3	COV

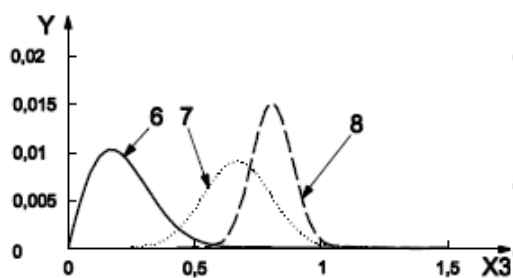
For collapse, either the ensemble or the governing case PDF can govern, and hence both possibilities should be considered.



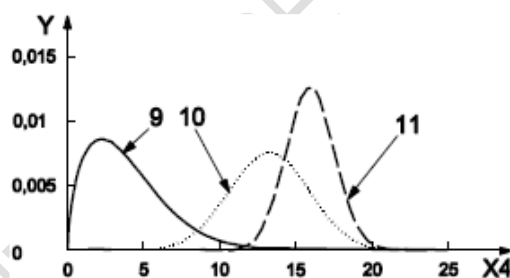
a) Yield stress (L80 CRS)



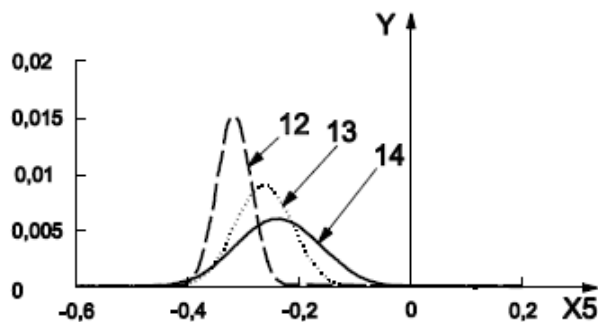
b) Average wall thickness



c) Ovality



d) Eccentricity



e) Residual stress (CRS)

Figure F.6—Possible PDFs for Calculation of Design Collapse Strengths

## Key

- X1 yield stress, ksi
- X2 average/nominal wall thickness
- X3 ovality, %
- X4 eccentricity, %
- X5 residual/yield stress
- Y probability
- 1 governing case, mean = 87.5; coefficient of variance = 0.0333
- 2 dataset ensemble (mean = 88.0; coefficient of variance = 0.0529)
- 3 governing case, mean = 0.9850; coefficient of variance = 0.001
- 4 governing case, mean = 0.9876; coefficient of variance = 0.002
- 5 dataset ensemble (mean = 1.0069; coefficient of variance = 0.0259)
- 6 dataset ensemble (mean = 0.217; coefficient of variance = 0.541)
- 7 governing case, mean = 0.660; coefficient of variance = 0.2
- 8 governing case, mean = 0.795; coefficient of variance = 0.1
- 9 dataset ensemble (mean = 3.924; coefficient of variance = 0.661)
- 10 governing case, mean = 13.2; coefficient of variance = 0.2
- 11 governing case, mean = 15.9; coefficient of variance = 0.1
- 12 dataset ensemble (mean = -0.237; coefficient of variance = 0.332)
- 13 governing case, mean = -0.264; coefficient of variance = 0.2
- 14 governing case, mean = -0.318; coefficient of variance = 0.1

**Figure F.6—Possible PDFs for Calculation of Design Collapse Strengths** *(continued)*

## F.4 Selection of Design Equation

### F.4.1 Ultimate Limit State Equation

SPE 90904 describes the development of the Klever-Tamano equations. A slightly simplified version was used, as below, to calculate the ultimate collapse pressure,  $p_{ult}$ :

$$p_{ult} = \{(p_{e\text{ ult}} + p_{y\text{ ult}}) - [(p_{e\text{ ult}} - p_{y\text{ ult}})^2 + 4p_{e\text{ ult}} p_{y\text{ ult}} H_{t\text{ ult}}]^{1/2}\} / [2(1 - H_{t\text{ ult}})] \quad (\text{F.1})$$

where

$$p_{e\text{ ult}} = k_{e\text{ ult}} 2E / [(1 - \nu^2) (D_{\text{ave}} / t_{e\text{-av}} - 196x_{\text{ceed}}) (D_{\text{ave}} / t_{e\text{-av}} - 196x_{\text{ceed}} - 1)^2] \quad (\text{F.2})$$

$$p_{y\text{ ult}} = k_{y\text{ ul}} 196x_{\text{ceed}} 2f_y (t_{e\text{-av}} / D_{\text{ave}}) [1 + t_{e\text{-av}} / (2D_{\text{ave}})] \quad (\text{F.3})$$

$$H_{t\text{ ult}} = 0.127\text{ ov} + 0.0039\text{ ec} - 0.440(rs/f_y) + h_n, \text{ with the limitation } H_{t\text{ ult}} \geq 0 \quad (\text{F.4})$$

and

$D_{\text{ave}}$  is the average actual outside diameter;

$D_{\text{max}}$  is the maximum actual outside diameter;

$D_{\text{min}}$  is the minimum actual outside diameter;

$E$  is Young's modulus;

$ec$  is the eccentricity, in percent,  $ec = 100 (t_{c \max} - t_{c \min})/t_{c \text{ ave}}$ ;

$f_y$  is the actual yield strength of a representative tensile specimen;

$h_n$  is the stress-strain curve shape factor;

$Ht_{ult}$  is a decrement factor;

$k_{e \text{ uls}}$  is the calibration factor for ultimate elastic collapse: 1.089;

$k_{y \text{ ul}}$  is the calibration factor for ultimate yield collapse: 0.9911;

$ov$  is the ovality, in percent,  $ov = 100 (D_{\max} - D_{\min})/D_{\text{ave}}$ ;

$p_{e \text{ ult}}$  is the ultimate elastic collapse term;

$p_{y \text{ ult}}$  is the ultimate yield collapse term;

$rs$  is the circumferential residual stress at the ID face (negative for compression, positive for tension);

$t_{c \text{ av}}$  is the average actual wall thickness;

$t_{c \max}$  is the maximum actual wall thickness;

$t_{c \min}$  is the minimum actual wall thickness;

$\nu$  is Poisson's ratio.

The quantity  $h_n$  was obtained empirically from collapse test data. The great majority of Q&T product has sharp-kneed stress-strain curves, for which no correction is necessary ( $h_n = 0$ , Figure F.12). However, collapse test results suggest that a small minority of Q&T pipe has rounded-kneed stress-strain curves, which reduce collapse strength, and in this event  $h_n = 0.017$  is necessary to give the desired TRL (Figure F.13; the average for the four grades was taken).

The calibration factors  $k_{e \text{ uls}}$  and  $k_{y \text{ ul}}$  were also obtained empirically, to give the flattest possible actual/predicted collapse strength response in each of the input variables, for the 2986 Q&T collapse tests (Figure F.2). The calibration process did not aim to set mean actual/predicted strength to 1.0 or to minimize its dispersion; nevertheless, it was found that both quantities were very accurately predicted, with mean = 0.9991 and COV = 0.0670.

Collapse test results suggest that Equation (F.1):

— does not apply to very thin wall pipe [ $\log_{10}(p_y/p_e) > 0.4$ ] with very high compressive residual stress ( $rs/f_y < -0.5$ );

— slightly over-predicts collapse strength at low tensile residual stresses (by 2–3 % for  $rs/f_y = 0.075$ , Figure F.2). Preliminary FEA studies indicate that over-prediction continues at higher tensile residual stresses, and caution is advised when using Equation (F.1) for  $rs/f_y > 0.05$ .

## F.4.2 Design Equation (Ensemble PDFs)

### F.4.2.1 External Pressure Only

Equations (F.1) to (F.4) are for ultimate collapse strength; that is, they predict when the casing actually fails. For design, down-rated strengths are used, which contain a safety margin appropriate to the desired TRL. In this case, the margin was obtained by applying multiplicative down-rating factors  $k_{y\text{ des}}$  and  $k_{e\text{ des}}$  to the yield and elastic strengths respectively, as below:

$$p_{\text{des}} = \{(k_{e\text{ des}} p_e + k_{y\text{ des}} p_y) - [(k_{e\text{ des}} p_e - k_{y\text{ des}} p_y)^2 + 4 k_{e\text{ des}} p_e k_{y\text{ des}} p_y H_{t\text{ des}}]^{1/2}\} / [2 (1 - H_{t\text{ des}})] \quad (\text{F.5})$$

where

$$p_e = 2E / [(1 - \nu^2) (D/t) ((D/t) - 1)^2] \quad (\text{F.6})$$

$$p_y = 2f_{y\text{ mn}} (t/D) [1 + t/(2D)] \quad (\text{F.7})$$

and

$D$  is the specified outside diameter;

$E$  is Young's modulus:  $206.9 \times 10^9 \text{ N/m}^2$  ( $30 \times 10^6 \text{ psi}$ );

$f_{y\text{ mn}}$  is specified minimum yield strength;

$H_{t\text{ des}}$  is a decrement factor;

$k_{e\text{ des}}$  is the down-rating factor for design elastic collapse;

$k_{y\text{ des}}$  is the down-rating factor for design yield collapse;

$p_{\text{des}}$  is the design collapse pressure;

$p_e$  is the elastic collapse term;

$p_y$  is the yield collapse term;

$t$  is the specified wall thickness;

$\nu$  is Poisson's ratio: 0.28.

Note that  $p_e$  and  $p_y$  are calculated using the specified dimensions and the specified minimum yield stress, rather than the actual values as for the ultimate strength.

For calculation of case-specific collapse strengths, the decrement factor  $H_{t\text{ des}}$  was obtained from the means of the relevant production variables, to give uniform scaling between the ULS and design strengths. For pipe with sharp-kneed stress-strain curves, and the ensemble means (Table F.4):

$$\text{CRS } H_{t\text{ des}} = (0.127 \times 0.217) + (0.0039 \times 3.924) - [0.440 \times (-0.237)] + 0 = 0.1471 \quad (\text{F.8})$$

$$\text{HRS } H_{t\text{ des}} = (0.127 \times 0.217) + (0.0039 \times 3.924) - [0.440 \times (-0.138)] + 0 = 0.1036 \quad (\text{F.9})$$



Similarly, for pipe with rounded-kneed stress-strain curves:

$$\text{CRS } H_{t_{\text{des}}} = (0.127 \times 0.217) + (0.0039 \times 3.924) - [0.440 \times (-0.237)] + 0.017 = 0.1641 \quad (\text{F.10})$$

$$\text{HRS } H_{t_{\text{des}}} = (0.127 \times 0.217) + (0.0039 \times 3.924) - [0.440 \times (-0.138)] + 0.017 = 0.1206 \quad (\text{F.11})$$

The working in Equations (F.8) to (F.11) is given purely as an example of its later use in Annex H on case-specific collapse strengths. For the probabilistic ratings in Section F.6,  $H_{t_{\text{des}}}$  was calculated from the governing case PDFs, resulting in 0.22 for CRS and 0.20 for HRS (F.5.3.3).

#### F.4.2.2 Combined Loads

Axial tension reduces collapse strength, and internal pressure increases it. This subsection gives a method for calculating collapse strength under one or both combined loads, based on [SPE 90904](#).

Elastic collapse pressure is unaffected by axial tension, so  $\Delta p_e$  is obtained from Equation (F.6) as before:

$$\Delta p_{e \text{ des}} = k_{e \text{ des}}^2 E / [(1 - \nu^2) (D/t) (D/t - 1)^2] \quad (\text{F.12})$$

where  $\Delta p_e$  is interpreted as a pressure difference  $p_o - p_i$ ; the Tresca design yield pressure  $\Delta p_{y \text{ T des}}$  is calculated as follows:

$$\Delta p_{y \text{ T des}} = k_{y \text{ des}} 2 f_{y \text{ mn}} t / (D - t) \quad (\text{F.13})$$

The von Mises design yield pressure  $\Delta p_{y \text{ vme des}}$  is obtained as follows:

$$\Delta p_{y \text{ vme des}} = (4/3^{1/2}) k_{y \text{ des}} f_{y \text{ mn}} [t/(D - t)] [1 - (F_{\text{eff}}/F_{y \text{ des}})^2]^{1/2} = p_o - p_i \quad (\text{F.14})$$

where

$$F_{\text{eff}} = F_a - p_i A_i + p_o A_o \quad (\text{F.15})$$

$$F_{y \text{ des}} = k_{y \text{ des}} f_{y \text{ mn}} A_s \quad (\text{F.16})$$

and

$A_i$  is the area to ID =  $\pi (D - 2t)^2/4$ ;

$A_o$  is the area to OD =  $\pi D^2/4$ ;

$A_s$  is the cross-sectional area =  $A_o - A_i$ ;

$F_a$  is the component of axial force not due to bending, tension positive;

$p_i$  is the internal pressure;

$p_o$  is the external pressure.

Equations (F.13) and (F.14) are approximations that neglect higher order terms.  $F_{\text{eff}}$  is itself a function of  $p_o$  [Equation (F.15)]; thus, Equation (F.14) is solved either by iteration or by using the root-finding function in a mathematical spreadsheet (see example, F.6.3.2). One can also eliminate  $F_{\text{eff}}$  by using (F.15) in (F.14) and solve the resulting quadratic equation for the yield pressure directly.

From the original work, the average of von Mises and Tresca is selected to match FEA results under axial compression [74]. The actual yield collapse pressure is taken as either the von Mises yield pressure or the average of the von Mises (vme) and Tresca (T) yield pressures, depending on the position on the VME ellipse. The von Mises yield pressure calculated from Equation (F.14) is selected if its value is lower than the Tresca yield pressure calculated from Equation (F.13). Thus, for the full range of combined collapse loading: Thus:

$$\Delta p_{y \text{ des}} = (\Delta p_{y \text{ T des}} + \Delta p_{y \text{ vme des}})/2 \quad \text{if } \Delta p_{y \text{ vme des}} > \Delta p_{y \text{ T des}} \quad (\text{F.17})$$

$$\Delta p_{y \text{ des}} = \Delta p_{y \text{ vme des}} \quad \text{if } \Delta p_{y \text{ vme des}} < \Delta p_{y \text{ T des}} \quad (\text{F.18})$$

$\Delta p_{\text{des}}$  is then calculated as in Equation F.5, but with  $\Delta p_e$  and  $\Delta p_y$  in place of  $p_e$  and  $p_y$ :

$$\Delta p_{\text{des}} = \{(\Delta p_{e \text{ des}} + \Delta p_{y \text{ des}}) - [(\Delta p_{e \text{ des}} - \Delta p_{y \text{ des}})^2 + 4\Delta p_{e \text{ des}} \Delta p_{y \text{ des}} H_{t \text{ des}}]^{1/2}\} / [2(1 - H_{t \text{ des}})] \quad (\text{F.19})$$

Finally, the external design pressure  $p_{o \text{ des}}$  is calculated as follows:

$$p_{o \text{ des}} = \Delta p_{\text{des}} + p_i \quad (\text{F.20})$$

## F.5 Risk-calibrated Collapse Ratings

### F.5.1 Analysis Methods

#### F.5.1.1 Model Uncertainty

For probabilistic analysis, Equation (F.1) is multiplied by the model uncertainty [60] giving:

$$p_{\text{ult}} = mu \{ (p_{e \text{ ult}} + p_{y \text{ ult}}) - [(p_{e \text{ ult}} - p_{y \text{ ult}})^2 + 4p_{e \text{ ult}} p_{y \text{ ult}} H_{t \text{ ult}}]^{1/2} \} / [2(1 - H_{t \text{ ult}})] \quad (\text{F.21})$$

where

$mu$  is the model uncertainty;

$p_{e \text{ ult}}$  is the ultimate elastic collapse term, Equation (F.2);

$p_{y \text{ ult}}$  is the ultimate yield collapse term, Equation (F.3);

and  $mu$  is a random variable.

#### F.5.1.2 Calculation of Failure Probabilities

Predicted failure probabilities were calculated using FORM. For independent input variables, a Rackwitz 1976 [92] search routine was employed, with the normal tail approximation [52], [81] for non-Gaussian variables. For correlated variables, the correlated non-Gaussian variable set was mapped to a correlated Gaussian set using the Nataf transformation [83], with transformed correlations as given by Liu and der Kiureghian 1986 [78], and mapped again to an independent nonstandard Gaussian set with the orthogonal transformation [81]. Positions in the independent standard Gaussian space were mapped back into the original variable space via the Jacobian matrix [81]. A Hohenbichler and Rackwitz 1981 [63] search routine was adopted. Both methods were implemented as computer spreadsheets and validated against SORM, Monte Carlo, and textbook examples.

### F.5.1.3 Downrating Factors

The necessary values of  $k_{e\text{ des}}$  and  $k_{y\text{ des}}$  were calculated by iteration to give the flattest possible response over the dataspace for a given TRL.

### F.5.2 Target Reliability Level

In design code calibration, the TRL is often taken as the average predicted failure probability for the previous version of the code. This was obtained by:

calculating design collapse strength, using API 5C3 (6<sup>th</sup> Edition);

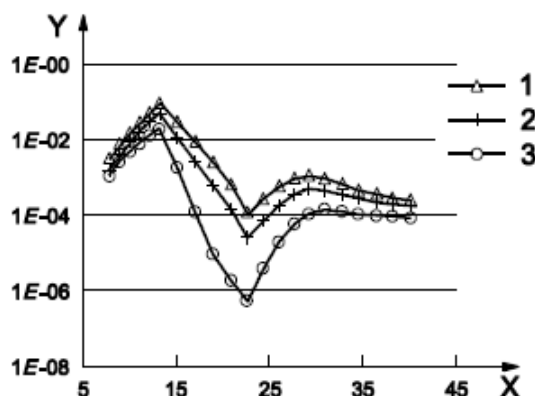
setting the external pressure equal to the design strength;

running the probabilistic analysis with the ensemble input PDFs (Tables F.3 and F.4), and ultimate collapse strength calculated using Equation (F.21).

Figure F.7 shows typical results. Predicted failure probability varies by five orders of magnitude, from 0.0966 at the yield-plastic boundary for CRS pipe, to  $5.11 \times 10^{-7}$  at the plastic-transition boundary for NS pipe. This is far too great a variation to be acceptable; moreover, the average values (Table F.6) are very different and vary with grade. It was therefore concluded that this approach was not suitable, and it was decided to adopt TRL = 0.5 % as specified (but not achieved) by the previous version.

**Table F.6—Average Predicted Failure Probability for API 5C3 (6<sup>th</sup> Edition) (Seamless L80)**

Straightening	TRL
Cold	$1.26 \times 10^{-2}$
Hot	$6.41 \times 10^{-3}$
None	$2.48 \times 10^{-3}$



#### Key

X specified outside diameter/specified wall thickness

Y predicted failure probability

1 CRS

2 HRS

3 not straightened

NOTE L80 seamless.

**Figure F.7—Predicted Failure Probabilities for API 5C3 (6<sup>th</sup> Edition)**

### F.5.3 Selection of Input Data

#### F.5.3.1 Yield Stress

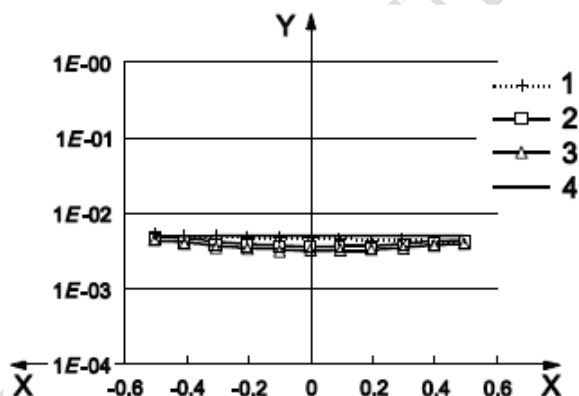
The more onerous of the ensemble and potential governing case PDFs was taken for each grade. In general, the ensemble PDF is more onerous for grades with narrow permissible yield stress ranges (e.g. T95), and the governing case PDF more onerous for grades with wide ranges (e.g. N80).

#### F.5.3.2 Average Wall Thickness

The ensemble condition governs and was used throughout.

#### F.5.3.3 Eccentricity, Ovality, and Residual Stress

These parameters appear in the decrement term  $H_{t_{des}}$  of Equation (F.5) for design collapse strength. For ensemble data, the mean values of each parameter (Table F.4) were used in calculating  $H_{t_{des}}$  [see Equations (F.8) to (F.11)]. This gives a flat reliability response (Figure F.10). For governing case data, using the respective governing case means in Equations (F.8) to (F.11) instead of the ensemble means also gives nearly uniform reliability (Figure F.8).



#### Key

- X log (yield/elastic strength)
- Y predicted failure probability
- 1 ensemble data
- 2 governing case data, coefficients of variance = 0.2
- 3 governing case data, coefficients of variance = 0.1
- 4 TRL

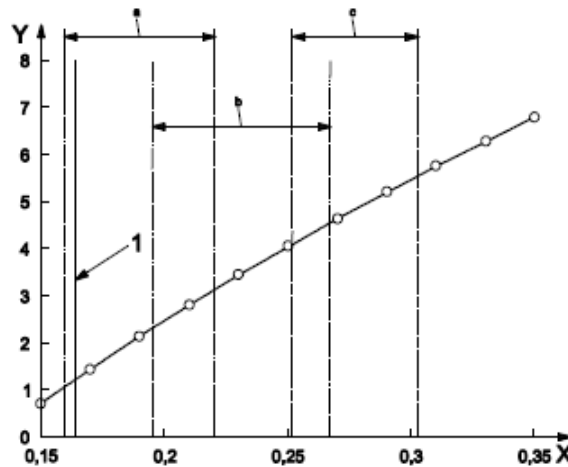
NOTE L80 seamless CRS ( $k_e = 0.825$ ,  $k_y = 0.855$ ).

**Figure F.8—Predicted Failure Probabilities with Potential Governing Case Means in  $H_{t_{des}}$**

This shows that the effect of  $H_{t_{des}}$  is almost independent of the down-rating factors  $k_{e_{des}}$  and  $k_{y_{des}}$ . Therefore, the value of  $H_{t_{des}}$  can be set for some desired governing case combination of eccentricity, ovality, and residual stress, and this value will (to a good approximation) apply for all grades, sizes, and weights.

Figure F.9 shows the effect of  $H_{t_{des}}$  on pipe mass. Using the ensemble PDFs gives an average mass increase of 1.2 % over all pipe sizes, weights, and grades. Using potential governing case PDFs gives mass increases of 1.0 % to 5.6 %, depending on the number of cases taken, and the PDF dispersion assumed for each case (Figure F.6). For calculation of collapse ratings, it was decided to use  $H_{t_{des}} = 0.2$ ; this allows for one moderately severe governing case, or two mild governing cases.

This allowance is additional to that for stress-strain curve shape (F.4.1). The total value of  $H_{t_{des}}$  is therefore 0.22 ( $\cong 0.20 + 0.017$ ) for CRS pipe and 0.20 for HRS.



#### Key

- X  $H_{t_{des}}$
- Y mass change, %, with respect to API 5C3 (6<sup>th</sup> Edition)
- 1 ensemble PDFs
- a one of ovality, eccentricity, residual stress governing case
- b two of ovality, eccentricity, residual stress governing case
- c ovality, eccentricity, residual stress all governing case

**Figure F.9—Effect of  $H_{t_{des}}$  on Pipe Mass**

Table F.7 summarizes the input data used for calculation of design collapse ratings.

**Table F.7—Summary of Probabilistic Data**

Parameter	Type	Distribution	Ensemble Data		Potential Governing Case Data	
			Mean	COV	0.5 % Exceedence Value Located at	PDF Dispersion Given by
Yield stress <sup>a</sup>	Ensemble	Gaussian	Table F.3	Table F.3	n/a	n/a
	Potential governing case	Gaussian	n/a	n/a	Minimum yield	15 ksi between 0.5 %- exceedence 204xcee dances limits
Average WT <sup>b</sup>	Ensemble	Gaussian	1.0069	0.0259	n/a	n/a
Average OD <sup>c, d</sup>	Ensemble	Gaussian	1.0059	0.00181	n/a	n/a
Ovality Eccentricity Residual stress	Potential governing case <sup>e, f</sup>	Gaussian	n/a	n/a	See footnotes	See footnotes
Model uncertainty <sup>g</sup>	Ensemble	Gaussian	0.9991	0.0670	n/a	n/a

<sup>a</sup> Design strength calculated for both cases, and the lower values taken.  
<sup>b</sup> Normalized by specified wall thickness.  
<sup>c</sup> Normalized by specified outside diameter.  
<sup>d</sup> Low sensitivity factor (Figure F.15).  
<sup>e</sup>  $H_{t\text{des}} = 0.2$  allows for one severe or two mild governing cases.  
<sup>f</sup> Ensemble PDFs used for calculation of  $k_{e\text{des}}$  and  $k_{y\text{des}}$ .  
<sup>g</sup> Q&T pipe dataset.

## F.5.4 Results

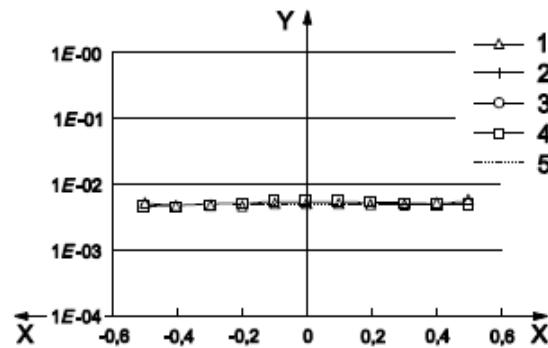
### F.5.4.1 Down-rating Factors

Down-rating factors (Table F.8) were calculated for seamless pipe, for cold and hot rotary straightening, for a TRL of 0.5 %.

**Table F.8—Down-rating Factors**

Grade	CRS		HRS	
	$k_{e\text{des}}$	$k_{y\text{des}}$	$k_{e\text{des}}$	$k_{y\text{des}}$
H40	0.830	0.910	n/a	n/a
J/K55	0.830	0.890	n/a	n/a
L80	0.825	0.855	0.825	0.865
L80 13Cr	0.825	0.830	0.825	0.840
N80 Type 1	0.825	0.870	n/a	n/a
N80 Q&T	0.825	0.870	0.825	0.870
C90	n/a	n/a	0.825	0.850
R95	0.825	0.840	0.825	0.855
T95	n/a	n/a	0.825	0.855
P110	0.825	0.855	0.825	0.855
Q125	n/a	n/a	0.825	0.850

For the sake of simplicity,  $k_{e\text{des}} = 0.825$  was adopted for all grades. Figure F.10 shows predicted failure probability for the new design equations. It is nearly constant, in contrast to the highly variable reliability for the previous equations (Figure F.7). Moreover, it is likewise nearly constant for all grades and straightening methods.



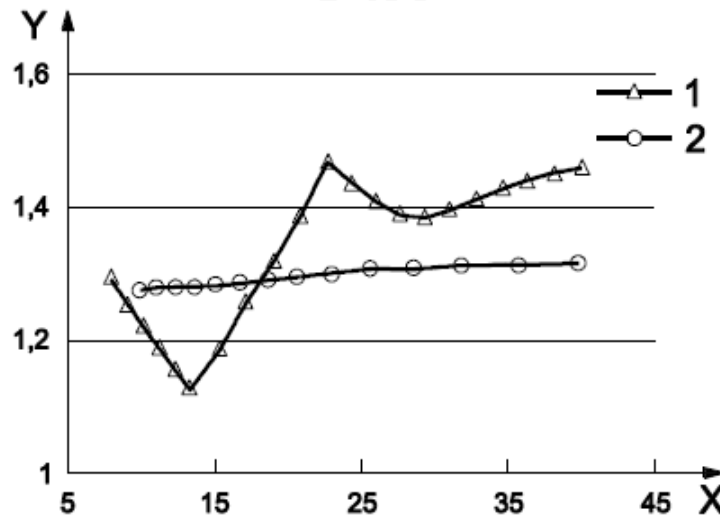
**Key**

X	log (yield/elastic strength)	3	P110
Y	predicted failure probability	4	Q125
1	J/K55	5	TRL
2	L80		

NOTE Seamless CRS, TRL = 0.005.

**Figure F.10—Predicted Failure Probabilities for Probabilistic Method**

Figure F.11 shows design margin, as predicted ULS strength/design strength, for both the old and new design equations. The new equations give a much more uniform margin across the dataspace, and it is this that is giving the more uniform predicted failure probabilities.



**Key**

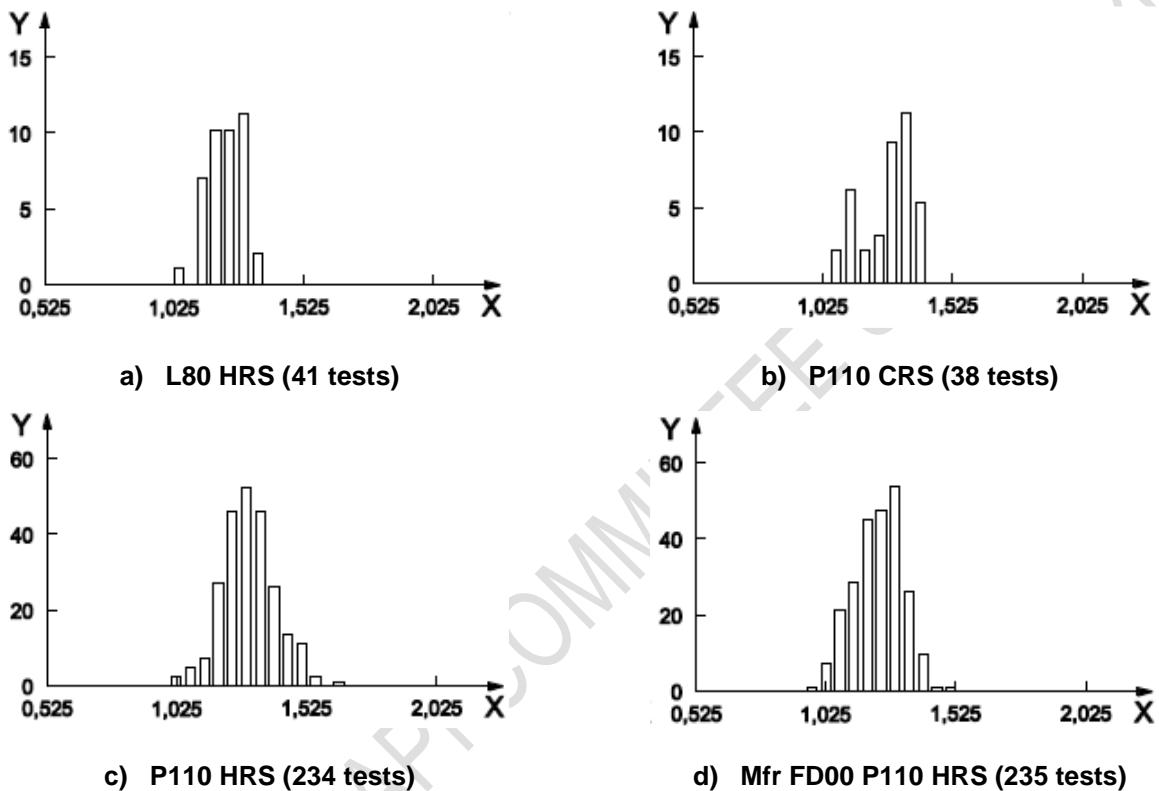
X	specified outside diameter/specified wall thickness
Y	margin
1	API 5C3 (6 <sup>th</sup> Edition)
2	probabilistic method (this annex)

NOTE L80 seamless CRS.

**Figure F.11—Comparison of Design Margin**

#### F.5.4.2 Comparison with Collapse Test Data

Figures F.12 and F.13 compare the proposed design collapse strengths against collapse test data (Table F.2). The smaller datasets are not statistically significant and do not support firm conclusions. However, the larger datasets show that the necessary TRL is being achieved.

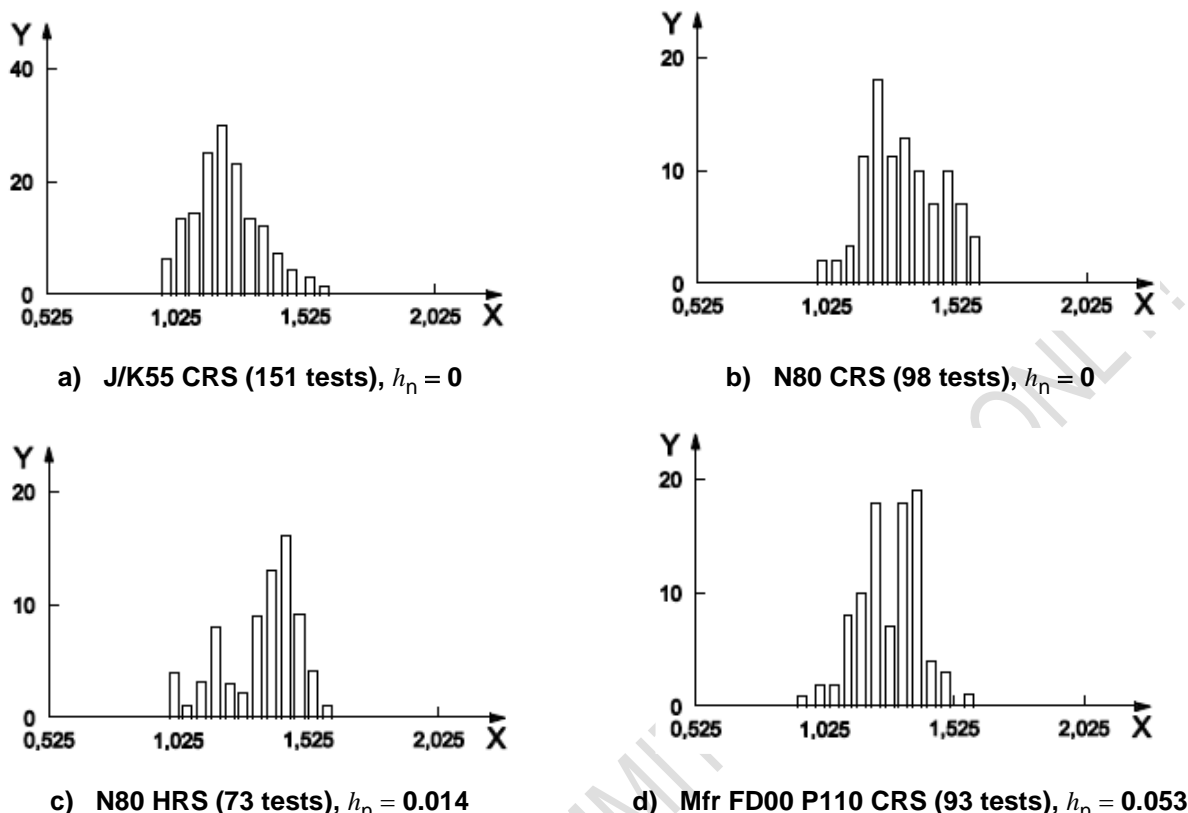


#### Key

X actual/design collapse strength  
Y occurrences

**Figure F.12—Comparison of Design Collapse Strengths with Collapse Test Data (Sharp-kneed Stress-strain Curves),  $h_n = 0$**





#### Key

X actual/design collapse strength  
Y occurrences

**Figure F.13—Calibration of  $h_n$  via Design Strength (Rounded-kneed Stress-strain Curves)**

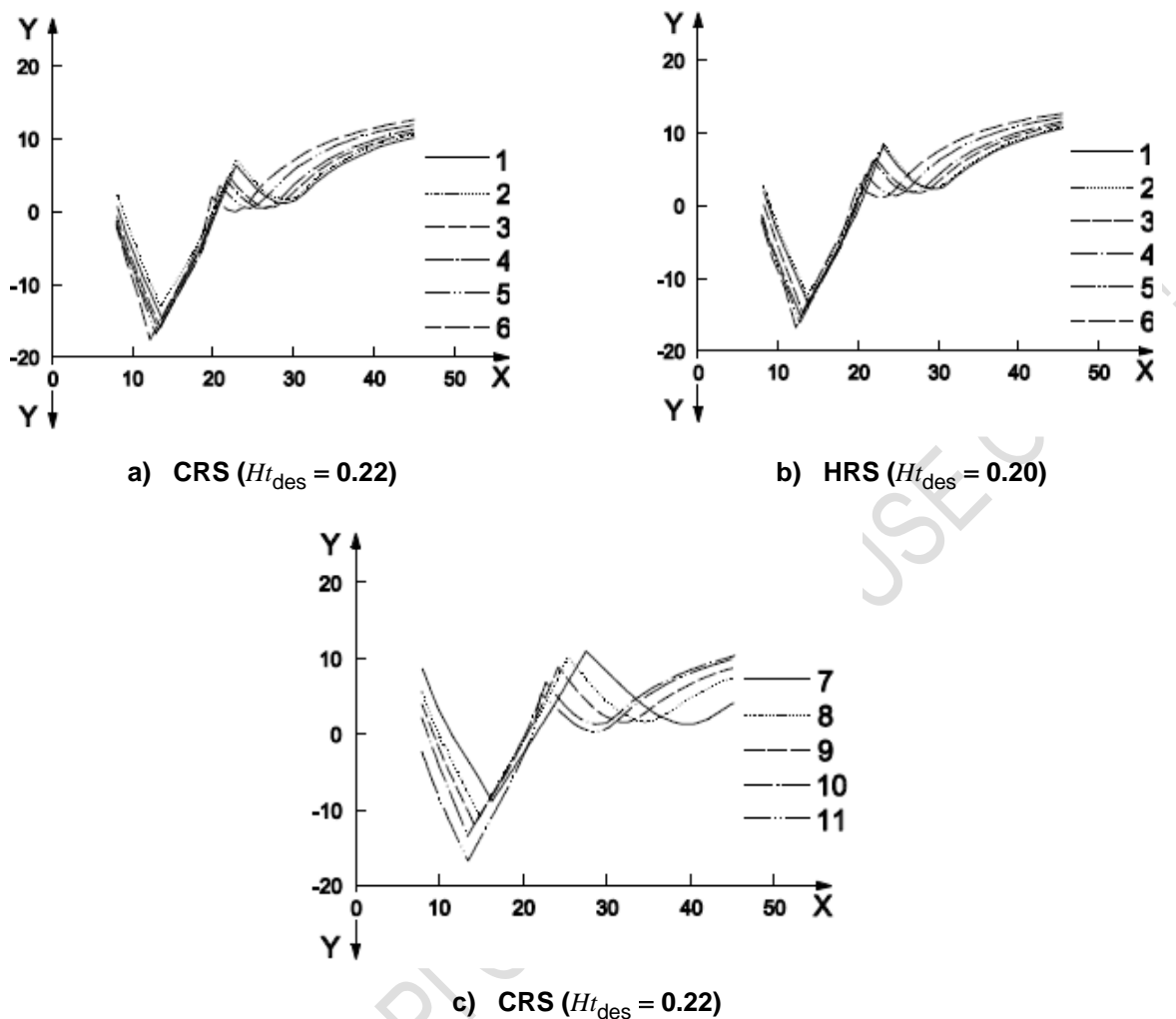
It was, however, found necessary to omit six tests for 10-3/4 in. 40.5 lb/ft N80 with high residual stresses from the test data catalogue, as all six tests gave low outliers ( $0.73 < \text{actual/design strength} < 1.01$ ). It is therefore suggested that Equation (F.1) does not apply to very thin wall pipe [ $\log_{10}(p_y/p_e) > 0.4$ ] with very high compressive residual stress ( $rs/f_y < -0.5$ ).

#### F.5.4.3 Comparison with API 5C3 (6<sup>th</sup> Edition)

Figure F.14 shows the change in design collapse strength with respect to the API 5C3 (6<sup>th</sup> Edition) values, for seamless pipe and a TRL of 0.005. A positive value denotes an increase in strength with respect to API 5C3 (6<sup>th</sup> Edition), and a negative value denotes a reduction in strength.

For Q&T CRS product, the new design strengths are 13 % to 17 % lower than the old values at the high-risk peak ( $D/t = 12$  to  $13$ ), and 2 % to 7 % greater at the low-risk trough ( $D/t = 20$  to  $23$ ). For Q&T HRS, the new strengths are also 13 % to 17 % lower at the high-risk peak, and 3 % to 8 % greater at the low-risk trough.

For non-Q&T product, the new strengths are 9 % to 17 % less than the old values at the high-risk peak ( $D/t = 13$  to  $16$ ), and 5 % to 11 % greater at the low-risk trough ( $D/t = 23$  to  $28$ ).



#### Key

X	specified outside diameter/specified wall thickness	6	Q125
Y	strength difference, %, with respect to <b>API 5C3 (6<sup>th</sup> Ed.)</b>	7	H40
1	L80	8	J/K55
2	N80 Q&T	9	L80 13Cr
3	C90	10	N80 Type 1
4	R95, T95		
5	P110		

**Figure F.14—Comparison of Previous and Revised Design Strengths**

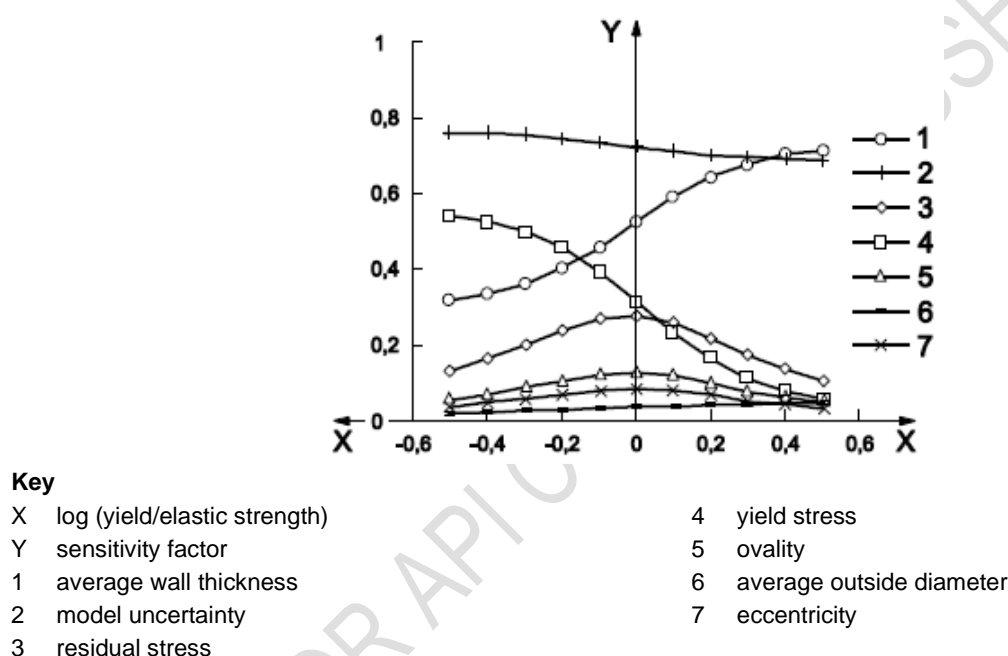
#### F.5.4.4 Sensitivity Factors

Figure F.15 shows sensitivity factors (also called alpha values) versus dataspace position, for a TRL of 0.005. They are a measure of the sensitivity of failure probability to each input variable [52], [81], [103] and may therefore be employed by manufacturers and users to gauge the effect of changes in production quality on safety. The values given are for L80 seamless CRS and the ensemble production quality statistics (Tables F.3 and F.4).

**NOTE** In the ULS equation, eccentricity, ovality, and residual stress have empirically derived coefficients; the accuracy of their sensitivity factors is therefore likely to be rather lower than for the other variables, which have a stronger theoretical basis.

Model uncertainty is the dominant variable over most of the dataspace, with  $\alpha = 0.76$  to  $0.69$ ; this means that the remaining limitations of the collapse strength equations have a rather greater effect than all the other input variables put together. At the left-hand (yield) end of the dataspace, yield stress has the next largest effect, with  $\alpha = 0.54$ : this is expected, given the form of the yield collapse equation and the various coefficients of variance (Table F.4). As one goes rightwards across the dataspace, the contribution (and alpha) of each variable changes. At the right-hand (elastic) end of the dataspace, wall thickness has narrowly the greatest effect, with  $\alpha = 0.71$ , and variables other than model uncertainty having negligible influence.

Comparison analyses have shown that sensitivity factors vary significantly by manufacturer and product. Mills are therefore encouraged to develop their own values, using the methods described in Annex H.



**Figure F.15—Sensitivity Factors**

## F.6 Summary

### F.6.1 Design Equation (Governing Case PDFs)

The minimum (or design) collapse strength for pipe under external pressure only is given by Equation (F.22). For pipe under external pressure and axial force and/or internal pressure, see F.4.2.2.

$$p_{\text{des}} = \{(p_{\text{e des}} + p_{\text{y des}}) - [(p_{\text{e des}} - p_{\text{y des}})^2 + 4p_{\text{e des}} p_{\text{y des}} H t_{\text{des}}]^{1/2}\} / [2(1 - H t_{\text{des}})] \quad (\text{F.22})$$

where

$$p_{\text{e des}} = 0.825 \times 2E / [(1 - \nu^2) (D/t) ((D/t) - 1)^2] \quad (\text{F.23})$$

$$p_{\text{y des}} = k_{\text{y des}} \times 2f_{\text{ymn}} (t/D) [1 + t/(2D)] \quad (\text{F.24})$$

and

- $D$  is the specified outside diameter;
- $E$  is Young's modulus,  $30 \times 10^6$  psi =  $206.9 \times 10^9$  N/m<sup>2</sup>;
- $f_{ymn}$  is the specified minimum yield strength;
- $H_{t\text{ des}}$  is a decrement factor: 0.22 for CRS product and 0.20 for HRS;
- $k_{y\text{ des}}$  is the down-rating factor for design yield collapse (Table F.9);
- $p_{\text{des}}$  is the design collapse pressure;
- $p_{e\text{ des}}$  is the design elastic collapse term;
- $p_{y\text{ des}}$  is the design yield collapse term;
- $t$  is the specified wall thickness;
- $\nu$  is Poisson's ratio, 0.28.

**Table F.9—Values of  $H_{t\text{ des}}$  and  $k_{y\text{ des}}$**

Grade <sup>a</sup>	Cold Rotary Straightened		Hot Rotary Straightened	
	$H_{t\text{des}}$ <sup>b</sup>	$K_{y\text{des}}$ <sup>b</sup>	$H_{t\text{des}}$ <sup>b</sup>	$K_{y\text{des}}$ <sup>b</sup>
H40	0.22	0.910	Not applicable <sup>c</sup>	
J55	0.22	0.890	Not applicable <sup>c</sup>	
K55	0.22	0.890	Not applicable <sup>c</sup>	
L80	0.22	0.855	0.20	0.865
L80 13Cr	0.22	0.830	0.20	0.840
N80 type 1	0.22	0.870	Not applicable <sup>c</sup>	
N80 Q&T	0.22	0.870	0.20	0.870
C90	Not applicable <sup>d</sup>		0.20	0.850
R95	0.22	0.840	0.20	0.855
T95	Not applicable <sup>d</sup>		0.20	0.855
P110	0.22	0.855	0.20	0.855
Q125	Not applicable <sup>d</sup>		0.20	0.850
<sup>a</sup> The data for the specified grade should be used; do not interpolate for actual yield stress.				
<sup>b</sup> $H_{t\text{des}}$ and $k_{y\text{des}}$ are dimensionless and can be used with any consistent units system.				
<sup>c</sup> Hot rotary straightening not normally used for this grade.				
<sup>d</sup> API 5CT does not allow cold rotary straightening.				

## F.6.2 Assumptions and Limitations

Equation (F.22) is based on a ULS equation, pipe dimension and stress statistics (that is, the statistical variation of measured pipe OD, wall thickness, yield stress, and so forth), and a TRL.

The ULS equation predicts the pipe failure pressure; that is, it does not include a safety factor. It was chosen to give the best fit to collapse test results for 2986 samples of Q&T pipe, manufactured between 1977 and

2000. All tests were conducted at room temperature; therefore, the collapse design factor should allow for any use at elevated temperatures.

Equation (F.22) (the design equation) was developed from the ULS equation using pipe dimension and stress statistics from at least 14 API mills, mostly for modern pipe, to satisfy a TRL of 0.005 for moderate production worst cases. This is broadly consistent with the TRL specified for plastic collapse in **API 5C3 (6<sup>th</sup> Edition)**.

The production quality statistics were based on worldwide API production, and thus the predicted reliability level for any individual mill and product can differ from 0.005. Annex G and Annex H give procedures for the calculation of case-specific design collapse strengths from collapse test data and pipe dimension and stress statistics, respectively.

The design strengths assume rounded-kneed stress-strain curves for CRS product, and sharp-kneed stress-strain curves for HRS product. Therefore, the CRS design strengths are slightly conservative for pipe with sharp-kneed stress-strain curves, and the HRS design strengths are slightly unconservative for pipe with rounded-kneed stress-strain curves.

Equation (F.22) does not directly check the onset of yield and therefore does not cover sour service. If **necessary**, yield onset should be checked separately using the von Mises equations in 6.4.

## F.6.3 Example

### F.6.3.1 Design Collapse Strength

Calculate the design collapse strength for 9-5/8 in. 53.5 lb/ft L80 CRS product. For 9-5/8 in. 53.5 lb/ft,  $t = 0.545$  in., and for L80 CRS,  $k_{y\text{ des}} = 0.855$  and  $H_{t\text{ des}} = 0.22$  (Table F.9), so in USC units:

$$p_{e\text{ des}} = 0.825 \times 2 \times 30 \times 10^6 / [(1 - 0.28^2) (9.625/0.545) ((9.625/0.545) - 1)^2] = 10,957 \text{ psi} \quad (\text{F.25})$$

$$p_{y\text{ des}} = 0.855 \times 2 \times 80,000 (0.545/9.625) \{1 + [0.545/(2 \times 9.625)]\} = 7965 \text{ psi} \quad (\text{F.26})$$

$$p_{\text{des}} = \{(10,957 + 7965) - [(10,957 - 7965)^2 + (4 \times 10,957 \times 7965 \times 0.22)]^{1/2}\} / [2 (1 - 0.22)]$$

$$= 6194 \text{ psi} \quad (\text{F.27})$$

### F.6.3.2 Design Collapse Strength under Combined Loads

Repeat the example in F.6.3.1 for an internal pressure of 5000 psi, applied together with a tensile axial stress of 20,000 psi. From F.4.2.2:

$$\Delta p_{y\text{ T des}} = 2 \times 0.855 \times 80,000 \times 0.06002 = 8211 \text{ psi} \quad (\text{F.28})$$

$$A_o = \pi \times 9.625^2 / 4 = 72.76 \text{ in.}^2 \quad (\text{F.29})$$

$$A_i = \pi \times 8.535^2 / 4 = 57.21 \text{ in.}^2 \quad (\text{F.30})$$

$$A_s = 72.76 - 57.21 = 15.55 \text{ in.}^2 \quad (\text{F.31})$$

$$F_{y\text{ des}} = 0.855 \times 80,000 \times 15.55 = 1063 \text{ kip} \quad (\text{F.32})$$

$$F_{ax} = 20,000 \times 15.55 = 310.9 \text{ kip} \quad (\text{F.33})$$

If a root-finding routine is available,  $p_o$  can be obtained as the root of [see Equation (F.14)]:

$$f(p_o) = \Delta p_{y \text{ vme des}}(p_o) - p_o + p_i = 0 \quad (\text{F.34})$$

Otherwise, iteration can be started by assuming  $p_o \cong \Delta p_{y \text{ T des}} + p_i = 8211 + 5000 = 13,211$  psi. Then, from Equation (F.15):

$$F_{\text{eff}} = 310,900 - (5000 \times 57.21) + (13,211 \times 72.76) = 986.1 \text{ kip} \quad (\text{F.35})$$

In Equation (F.14):

$$\Delta p_{y \text{ vme des}} = (4/3^{1/2}) 0.855 \times 80,000 [0.545/(9.625 - 0.545)] [1 - (986.1/1063)^2]^{1/2} = 3549 \text{ psi} \quad (\text{F.36})$$

The starting value for the next iteration is obtained as follows:

$$p_{o \text{ new}} = 0.5 (p_o + \Delta p_{y \text{ vme des}} + p_i) = 0.5 (13,211 + 3549 + 5000) = 10,880 \text{ psi} \quad (\text{F.37})$$

The remainder of the iterations are given in Table F.10 below. Other iterative methods (e.g. Newton-Raphson) can be used if desired.

**Table F.10—Iteration for  $p_o$**

$p_o$ psi	$F_{\text{eff}}$ kip	$\Delta p_{y \text{ vme des}}$ psi	$p_{o \text{ new}}$ psi
13,211	986.1	3549	10,880
10,880	816.5	6074	10,977
10,977	823.6	5998	10,988
10,988	824.3	5990	10,989
10,989	824.4	5989	10,989

$\Delta p_{y \text{ vme des}} < \Delta p_{y \text{ T des}}$ , hence  $\Delta p_{y \text{ des}} = \Delta p_{y \text{ vme des}} = 5989$  psi, and

$$\begin{aligned} \Delta p_{\text{des}} &= \{(10,957 + 5989) - [(10,957 - 5989)^2 + (4 \times 10,957 \times 5989 \times 0.22)]^{1/2}\} / [2 (1 - 0.22)] \\ &= 5043 \text{ psi} \end{aligned} \quad (\text{F.38})$$

$$p_{o \text{ des}} = 5043 + 5000 = 10,043 \text{ psi} \quad (\text{F.39})$$

## Annex G (informative)

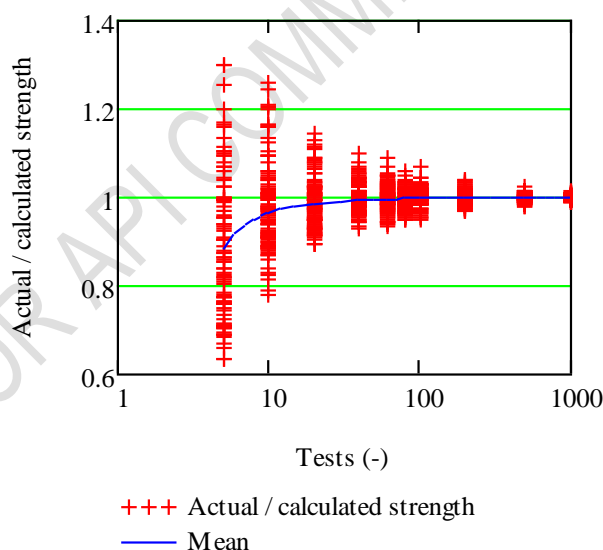
### Calculation of Design Collapse Strength from Collapse Test Data

#### G.1 Introduction

The design collapse strengths in Annex F were calculated for API pipe only, and do not apply to non-API cases such as HC pipe or special sour service grades (e.g. [Grade C110](#)). This annex describes the calculation procedure used to obtain design collapse strengths for non-API product, using collapse test data. Manufacturers may also, if desired, develop design collapse strengths for API pipe. All case-specific collapse strengths should be substantiated by provision, upon request, of collapse test datasets.

Design collapse strength calculated from test data is subject to statistical uncertainty. The uncertainty increases as dataset size  $n$  reduces. For very large datasets ( $n \geq 1000$ ), its effect is negligible, and design collapse strength may be calculated as in G.3. For smaller datasets ( $n < 1000$ ), the effect is significant, and design collapse strength should be calculated as in G.4. In both cases, it is assumed that the collapse test data are homogeneous; that is, the mean and dispersion are constant during production.

As sample size reduces, the small dataset method (see G.4) becomes subject to high scatter, and will on average over-estimate design collapse strength. Figure G.1 shows a typical example. The use of datasets of 30 or more tests will [minimise this scatter](#).



**Figure G.1 — Example of Accuracy of Small Dataset Method vs. Sample Size**

For small datasets ( $n < 1000$ ), the design collapse strength is calculated for a 95 % confidence of achieving the 0.5 % target reliability level (TRL) (see G.4.1). Section G.4.2 describes how to determine design strengths for other confidence levels and TRLs, if desired. **In this event, the parameters used shall be clearly indicated and agreed.**

For large datasets ( $n \geq 1000$ ), the design collapse strength is calculated for the 0.5 % TRL (see G.3.1). **Section G.3.2 describes how to determine design strengths for other TRLs, if desired. In this event, the TRL used shall be clearly indicated and agreed.**

## G.2 Collapse Test Data

All collapse test data should be for the size/weight/grade combination and manufacturing process for which design collapse strengths are to be calculated. The manufacturing process is deemed to include forming process, heat treatment, and rotary straightening. The minimum length of the test specimen should be as follows:

eight times the specified diameter for specified diameters of 9-5/8 and below;

seven times the specified diameter for specified diameters of 10-3/4 and above.

## G.3 Large Datasets

### G.3.1 Calculation method for specified target reliability level (TRL)

The design collapse strength should be calculated from Equation (G.1):

$$P_{\text{des } 0.5\%} = \mu_s - 2.576 \sigma_s \quad (\text{G.1})$$

where

$P_{\text{des } 0.5\%}$  is the design collapse strength, for a TRL of 0.5 %;

$\mu_s$  is the mean of collapse test dataset;

$\sigma_s$  is the standard deviation of collapse test dataset =  $[\sum_{i=1}^n (p_{\text{ult } i} - \mu_s)^2 / (n - 1)]^{1/2}$ .

This method should not be used for  $n < 1000$ .

### G.3.2 Calculation method for other target reliability levels (TRLs)

For TRLs other than 0.5%, the design collapse strength should be calculated from Equation (G.2):

$$P_{\text{des } \alpha} = \mu_s + \Phi^{-1}(\alpha) \sigma_s \quad (\text{G.2})$$

where

$P_{\text{des } \alpha}$  is the design collapse strength, for a target reliability level (TRL) of  $\alpha$ ;

$\mu_s$  is the mean of collapse test dataset;

$\Phi^{-1}$  is the inverse standard normal CDF (that is, the inverse normal CDF with zero mean and unit SD), evaluated at  $\alpha$ ;

$\alpha$  is the target reliability level;

$\sigma_s$  is the standard deviation of collapse test dataset =  $[\sum_{i=1}^n (p_{\text{ult } i} - \mu_s)^2 / (n - 1)]^{1/2}$ .

This method should not be used for  $n < 1000$ .



## G.4 Small Datasets

### G.4.1 Calculation Method

The design collapse strength should be calculated from Equation (G.3):

$$P_{\text{des } 0.5\% \text{ } 95\%} = \mu_s - F \sigma_s \quad (\text{G.3})$$

where

$P_{\text{des } 0.5\% \text{ } 95\%}$  is the 95 % confidence design collapse strength, for a TRL of 0.5 %;

$\mu_s$  is the mean of collapse test dataset;

$\sigma_s$  is the standard deviation of collapse test dataset =  $[\sum_{i=1}^n (p_{\text{ult } i} - \mu_s)^2 / (n - 1)]^{1/2}$ ;

$n$  is the number of collapse tests;

$F$  is a correction factor for dataset size, Table G.1; values not tabulated may be obtained as:

$$F = 24.32720 - 57.45545 \log_{10} n + 72.10244 (\log_{10} n)^2 - 52.72779 (\log_{10} n)^3 + 23.64113 (\log_{10} n)^4 - 6.41648 (\log_{10} n)^5 + 0.96953 (\log_{10} n)^6 - 0.06267 (\log_{10} n)^7 \quad (\text{G.4})$$

$P_{\text{des } 0.5\% \text{ } 95\%}$  has a 95 % probability of being lower than the large-dataset value ( $n = \infty$ ). Equation (G.4) is valid for  $10 \leq n \leq 1000$ . It should not be used for  $n < 10$ .

This method does not address the average over-estimation of design collapse strength shown in Figure G.1.

**Table G.1—Dataset Size Factor  $F$  (TRL = 0.5 %)**

$n$	$F$	$n$	$F$	$n$	$F$	$n$	$F$
3	11.628	16	3.812	70	3.051	300	2.786
4	7.748	18	3.710	80	3.016	400	2.756
5	6.313	20	3.628	90	2.987	500	2.736
6	5.566	25	3.478	100	2.963	600	2.722
7	5.103	30	3.376	110	2.942	700	2.710
8	4.787	35	3.300	120	2.925	800	2.701
9	4.556	40	3.242	130	2.910	900	2.694
10	4.378	45	3.195	140	2.896	1000	2.688
12	4.122	50	3.157	150	2.884	$\infty$	2.576
14	3.944	60	3.096	200	2.837		

Figure G.2 shows how reduction in design strength varies with  $n$  and dataset coefficient of variance ( $\text{COV} = \sigma_s / \mu_s$ ) for 95% confidence of a 0.5% TRL, and Figure G.3 shows the variation with  $n$  and TRL. Note that the results do not imply that mean collapse strength reduces with  $n$ ; rather, the decrease in design

strength allows for sampling uncertainty as  $n$  reduces. In particular, small datasets usually underestimate standard deviation.

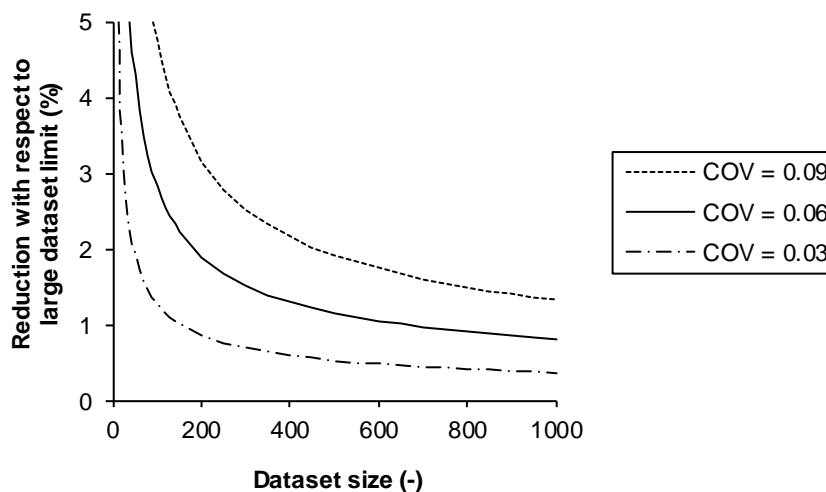


Figure G.2 — Reduction in design strength vs. dataset size (95% confidence of 0.5% TRL)

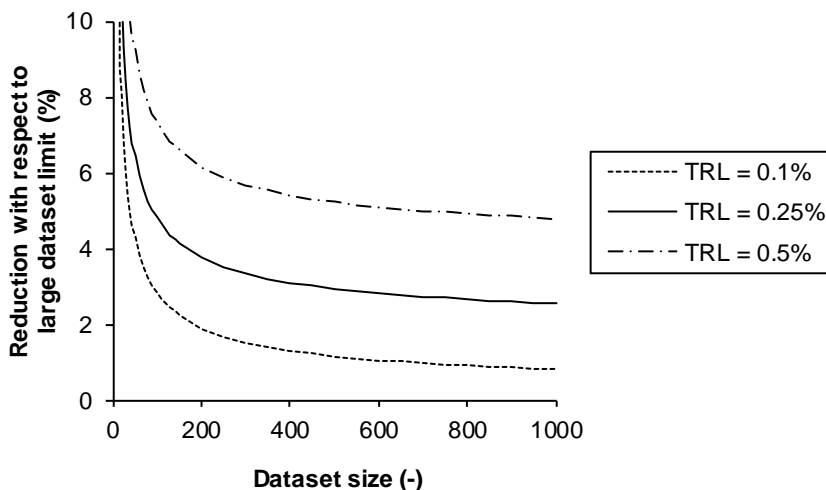


Figure G.3 — Reduction in design strength vs. dataset size (COV = 0.06, 95% confidence)

#### G.4.2 Calculation method for other probability levels

For probability levels other than 95 % confidence of the 0.5 % TRL, the correction factor for dataset size is calculated as follows. Denote the TRL as  $\alpha$ , and the confidence level as  $\gamma$ . Calculate  $K_\alpha$  from Equation (G.5):

$$K_{\alpha} = \Phi^{-1}(\alpha) \quad (G.5)$$

where

$K_{\alpha}n^{0.5}$  is the non-centrality parameter;

$n$  is the number of samples;

$\Phi^{-1}$  is the inverse standard normal CDF (that is, the inverse normal CDF with zero mean and unit SD);

$\alpha$  is the target reliability level.

The non-central Student t deviate is obtained from Equation (G.6) as the solution of:

$$\gamma = P_{\text{ncst}}(t, n-1, -K_{\alpha}n^{0.5}) \quad (G.6)$$

where

$\gamma$  is the confidence level;

$P_{\text{ncst}}$  is the non-central Student t CDF;

$t$  is the non-central Student t deviate;

$n$  is the number of samples;

$K_{\alpha}n^{0.5}$  is the non-centrality parameter.

$P_{\text{ncst}}$  is most easily evaluated using the algorithm of Lenth<sup>[76]</sup>. The correction factor is then determined from Equation (G.7):

$$F = t n^{-0.5} \quad (G.7)$$

where

$F$  is the correction factor;

$t$  is the non-central Student t deviate;

$n$  is the number of samples.

### G.4.3 Derivation

As dataset size reduces, the uncertainty in sample mean and standard deviation (SD) increases, and it becomes necessary to treat them as random variables. For a Gaussian population, sample mean is Gaussian-distributed, and sample variance is chi-squared<sup>[72]</sup>; therefore, the minimum value is distributed as noncentral Student  $t$ <sup>[77], [69], [89]</sup>. The values in Table G.1 were calculated using Lenth's algorithm for the noncentral  $t$  CDF<sup>[76]</sup>, and checked via numerical integration of the PDF expressions of Rinne<sup>[93]</sup> and Wolfram<sup>[107]</sup> for  $n \leq 150$ , and using the Gaussian approximation of Eisenhart et al.<sup>[61]</sup> for  $n \geq 200$ .

#### G.4.4 Example

Sixty-eight collapse tests were performed for a production run of 7 in. 23.0 lb/ft HC95 HRS. The aim points and process conditions were held steady during production. The sample mean and standard deviation were 6.609 and 0.3318 ksi, respectively. Calculate the design collapse strength for the specified probability levels 218 levels.

From Equation (G.4):

$$F = 24.32720 - 57.45545 \log_{10} 68 + 72.10244 (\log_{10} 68)^2 - 52.72779 (\log_{10} 68)^3 + 23.64113 (\log_{10} 68)^4 - 6.41648 (\log_{10} 68)^5 + 0.96953 (\log_{10} 68)^6 - 0.06267 (\log_{10} 68)^7 = 3.059 \quad (\text{G.8})$$

From Equation (G.3):

$$p_{\text{des } 0.5\% \text{ } 95\%} = 6.609 - 3.059 \times 0.3318 = 5.594 \text{ ksi} \quad (\text{G.9})$$

This compares with 5.754 ksi (a 2.9 % increase) had the design strength been calculated without the correction for dataset size.

Now re-calculate the design strength for a 95 % confidence of achieving a 0.1 % target reliability level, for the same number of tests.

From Equation (G.5):

$$K_{\alpha} = \Phi^{-1}(0.1\%) = -3.090 \quad (\text{G.10})$$

From Equation (G.6), either by iteration or using a mathematical spreadsheet:

$$P_{\text{ncst}}(30.12, 67, 3.090 \times 68^{0.5}) = 0.95 = \gamma \quad (\text{G.11})$$

giving  $t = 30.12$ . From Equation (G.7):

$$F = 30.12 \times 68^{-0.5} = 3.652 \quad (\text{G.12})$$

Finally, from Equation (G.3):

$$p_{\text{des } 0.1\% \text{ } 95\%} = 6.609 - 3.652 \times 0.3318 = 5.397 \text{ ksi}$$

## Annex H (informative)

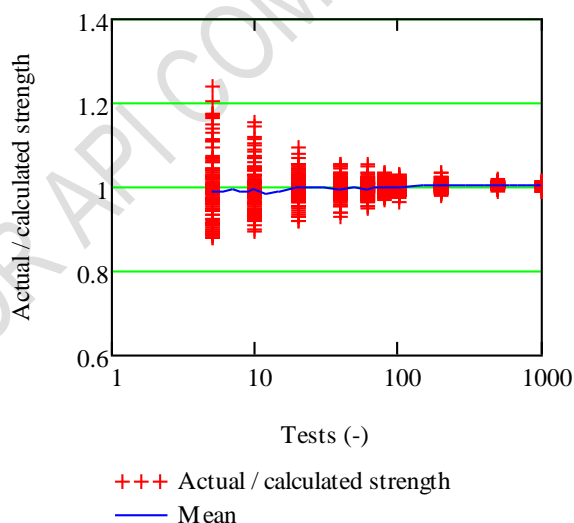
### Calculation of Design Collapse Strengths from Production Quality Data

#### H.1 Introduction

The design collapse strengths in Annex F were calculated for API pipe only and do not apply to non-API cases such as HC pipe or special sour service grades (e.g. Grade C110). This Annex describes the calculation procedure used to obtain design collapse strengths for non-API product, using production quality data. Manufacturers may also, if desired, develop design collapse strengths for API pipe. All case-specific collapse strengths should be substantiated by provision, upon request, of current production quality statistics and model uncertainty data.

Design collapse strength calculated from measurement data is subject to statistical uncertainty. The uncertainty increases as dataset size  $n$  reduces. For very large datasets ( $n \geq 1000$ ), its effect is negligible, and design collapse strength may be calculated as in H.3.2 and H.3.4.1. For smaller datasets ( $n < 1000$ ), the effect is significant, and design collapse strength should be calculated as in H.3.3 and H.3.4.2. In both cases, it is assumed that the production quality data are homogeneous; that is, the means and dispersions are constant during production.

As sample size reduces, the small dataset method (see H.3.3 and H.3.4.2) becomes subject to high scatter, and will on average slightly over-estimate design collapse strength. Figure H.1 shows a typical example. The use of datasets of 20 or more sets of measurements will minimise this scatter.



**Figure H.1 — Example of accuracy of small dataset method vs. sample size**

The calculation procedure is in two main parts, namely:

measurement and statistical characterization of the parameters determining collapse strength (average outside diameter, average wall thickness, eccentricity, ovality, yield stress, residual stress, and model uncertainty);

probabilistic analysis to determine the down-rating factors that satisfy the appropriate safety level.

These are described, in turn, below.

## H.2 Production Quality Data

### H.2.1 Representativeness

Data representativeness should be as specified in Table H.1 below. Measurement of each parameter should be in accordance with Annex I.

**Table H.1—Data Representativeness**

Parameter	Data Should Be for the Applicable
Average outside diameter	Forming process <sup>a</sup>
Average wall thickness	Forming process <sup>a</sup>
Eccentricity	Forming process <sup>a</sup>
Ovality	Forming process <sup>a</sup>
Yield stress	Grade, heat treatment, and rotary straightening type
Residual stress	Rotary straightening type
Collapse pressure	<sup>b</sup>
<sup>a</sup> Pipe geometry data are not required for the sizes and weights for which design collapse strengths are to be calculated, but the range of sizes and weights chosen should be representative of the product. <sup>b</sup> Collapse strengths are only used to develop model uncertainty data, and therefore the collapse tests need not be for the manufacturing process (forming process, heat treatment, and rotary straightening) for which design collapse strengths are to be calculated; however, the dataset used should be representative of the product.	

### H.2.2 Data Analysis

#### H.2.2.1 Pipe Dimensions and Stresses

Average outside diameter, average wall thickness, yield stress, and residual stress are then generalized via the use of bias factors, as follows:

average outside diameter: actual value/specified outside diameter;

average wall thickness: actual value/specified wall thickness;

yield stress: actual value/specified minimum yield stress;

residual stress: actual value/actual yield stress.

The mean and coefficient of variance (standard deviation/mean) of each bias factor is then calculated. Eccentricity and ovality are already in bias form; hence, the mean and coefficient of variance are obtained directly from the measured values.

#### H.2.2.2 Model Uncertainty

Model uncertainty is obtained by calculating:

the predicted collapse pressure for each collapse test sample, using the Klever-Tamano ULS equation [Equation (F.1)] with the measured pipe dimensions and stresses;

actual/predicted collapse pressure for each sample;

the mean and coefficient of variance of actual/predicted collapse pressure for the collapse test dataset.

### H.3 Probabilistic Analysis

#### H.3.1 Analysis Method

##### H.3.1.1 Probabilistic Analysis

A recognized technique (e.g. FORM, SORM, Monte Carlo) should be used.

##### H.3.1.2 Ultimate Collapse Strength Equation

The ultimate collapse strength should be calculated from Equation (F.21). More general forms of the Klever-Tamano equation (see SPE 90904) may be used if desired, but values of additional factors should be properly substantiated by calibration against a statistically significant amount of collapse test data.

##### H.3.1.3 Design Collapse Strength Equation

The design collapse strength should be calculated from Equation (F.5), with the decrement factor  $H_{t_{des}}$  obtained using:

$$H_{t_{des}} = 0.127\mu_{ov} + 0.0039\mu_{ec} - 0.440(\mu_{rs}/\mu_{fy}) + h_n \quad (H.1)$$

where

$h_n$  is the stress-strain curve shape coefficient;

$\mu_{ec}$  is the mean eccentricity, in percent,  $ec = 100(t_{c \max} - t_{c \min})/t_{c \text{ ave}}$ ;

$\mu_{fy}$  is the mean actual yield strength;

$\mu_{ov}$  is the mean ovality, in percent,  $ov = 100(D_{\max} - D_{\min})/D_{\text{ave}}$ ;

$\mu_{rs}$  is the mean residual stress (compression at ID face is negative).

and  $h_n = 0.017$  for CRS product, and  $h_n = 0$  for HRS product.

##### H.3.1.4 Target Reliability Level

The TRL should be 0.005.

#### H.3.2 Data—Large Datasets

Input variable data should be as specified in Table H.2 below.

Potential governing case data may be used instead of ensemble data if **appropriate** (see F.3.4 and F.5.3). In this event, the probability distribution and PDF parameters should generally be for an individual lot or batch. The distribution chosen should be justified by plotting the data frequency distribution onto probability scales [52], [103].

**Table H.2—Probability Data—Large Datasets**

Parameter	Probability Distribution	PDF Parameters
Average outside diameter	Gaussian	Deterministic, $\mu$ , $\sigma$ = as calculated in H.2.2.1
Average wall thickness	Gaussian	Deterministic, $\mu$ , $\sigma$ = as calculated in H.2.2.1
Eccentricity	Two-parameter Weibull <sup>a</sup>	Deterministic, $B$ , $C$ = as calculated in Equations (H.2) and (H.3)
Ovality	Two-parameter Weibull <sup>a</sup>	Deterministic, $B$ , $C$ = as calculated in Equations (H.2) and (H.3)
Yield stress	Gaussian	Deterministic, $\mu$ , $\sigma$ = as calculated in H.2.2.1
Residual stress	Gaussian	Deterministic, $\mu$ , $\sigma$ = as calculated in H.2.2.1
Model uncertainty	Gaussian	Deterministic, $\mu$ , $\sigma$ = as calculated in H.2.2.2
<sup>a</sup> For worst case PDF data with COV < 0.2, the Gaussian distribution is generally applicable.		

For two-parameter Weibull distributions, the PDF parameters should be calculated from:

$$\text{Shape parameter } C \text{ obtained as solution of } \{\Gamma[1 + (2/C)]\}/\{\Gamma[1 + (1/C)]\}^2 - 1 - (\sigma/\mu)^2 = 0 \quad (\text{H.2})$$

$$\text{Scale parameter } B = \mu/\Gamma [1 + (1/C)] \quad (\text{H.3})$$

where

$\mu$  is the mean;

$\sigma$  is the standard deviation;

$\Gamma$  denotes the gamma function [40].

Equation (H.2) can be solved by iteration, or by using the root-finding function in a mathematical spreadsheet.

### H.3.3 Data—Small Datasets

Input variable data should be as specified in Table H.3 below [72]. Large-dataset PDFs (Table H.2) may be used for any variables with  $n \geq 1000$ .

Potential governing case data can be used instead of ensemble data if **appropriate** (see F.3.4 and F.5.3). In this event, the probability distribution and PDF parameters should generally be for an individual lot or batch. The distribution chosen should be justified by plotting the data frequency distribution onto probability scales [52], [103].



**Table H.3—Probability Data—Small Datasets**

Parameter	Probability Distribution	PDF Parameters
Average OD	Gaussian	Random
Average OD: mean	Gaussian	$\mu = \mu_s$ (as calculated in H.2.2.1), $\sigma = \sigma_s/n^{0.5}$
Average OD: SD	Gaussian <sup>a</sup>	$\mu = \sigma_s$ (as calculated in H.2.2.1), $\sigma = \sigma_s/(2n)^{0.5}$
Average WT	Gaussian	Random
Average WT: mean	Gaussian	$\mu = \mu_s$ (as calculated in H.2.2.1), $\sigma = \sigma_s/n^{0.5}$
Average WT: SD	Gaussian <sup>a</sup>	$\mu = \sigma_s$ (as calculated in H.2.2.1), $\sigma = \sigma_s/(2n)^{0.5}$
Eccentricity	Two-parameter Weibull <sup>b</sup>	Random
Eccentricity: scale parameter	Gaussian	$\mu_B = B$ from Equations (H.4) and (H.5), $\sigma B$ from Equations (H.6) to (H.9)
Eccentricity: shape parameter	Log-normal	$\mu_C = C$ from Equation (H.4), $\sigma C$ from Equations (H.5) to (H.10)
Ovality	Two-parameter Weibull <sup>b</sup>	Random
Ovality: scale parameter	Gaussian	$\mu_B = B$ from Equations (H.4) and (H.5), $\sigma B$ from Equations (H.6) to (H.9)
Ovality: shape parameter	Log-normal	$\mu_C = C$ from Equation (H.4), $\sigma C$ from Equations (H.5) to (H.10)
Yield stress	Gaussian	Random
Yield stress: mean	Gaussian	$\mu = \mu_s$ (as calculated in H.2.2.1), $\sigma = \sigma_s/n^{0.5}$
Yield stress: SD	Gaussian <sup>a</sup>	$\mu = \sigma_s$ (as calculated in H.2.2.1), $\sigma = \sigma_s/(2n)^{0.5}$
Residual stress	Gaussian	Random
Residual stress: mean	Gaussian	$\mu = \mu_s$ (as calculated in H.2.2.1), $\sigma = \sigma_s/n^{0.5}$
Residual stress: SD	Gaussian <sup>a</sup>	$\mu = \sigma_s$ (as calculated in H.2.2.1), $\sigma = \sigma_s/(2n)^{0.5}$
Model uncertainty	Gaussian	Random
Model uncertainty: mean	Gaussian	$\mu = \mu_s$ (as calculated in H.2.2.2), $\sigma = \sigma_s/n^{0.5}$
Model uncertainty: SD	Gaussian <sup>a</sup>	$\mu = \sigma_s$ (as calculated in H.2.2.2), $\sigma = \sigma_s/(2n)^{0.5}$

<sup>a</sup> Strictly,  $n\sigma_s^2/\sigma^2$  is chi-squared distributed, where  $\sigma_s$  = sample SD,  $\sigma$  = process SD; but as  $\sigma$  is unknown, this does not enable PDF calculation; in practice, a Gaussian PDF can be used for  $n \geq 20$ , as the chi-squared PDF tends to the Gaussian PDF for large  $n$ .

<sup>b</sup> For worst case PDF data with COV < 0.2, the Gaussian distribution is generally applicable.

The sampling uncertainties for two-parameter Weibull variables can be calculated as below [79].  $C$  is obtained as the solution of Equation (H.4):

$$(1/C) + (1/n) [\sum_{i=1}^n \ln(x_i)] - [\sum_{i=1}^n x_i^C \ln(x_i)] [\sum_{i=1}^n x_i^C]^{-1} = 0 \quad (\text{H.4})$$

where

$n$  is the dataset size;

$x_i$  is the measurements;

and  $\sum_{i=1}^n$  denotes a sum taken over the terms  $i = 1, 2, \dots, n$ .

Equation (H.4) can be solved by iteration or by using the root-finding function in a mathematical spreadsheet.

$$B = [(\sum_{i=1}^n x_i^C)/n]^{1/C} \quad (H.5)$$

$$\partial^2 \ln L / \partial B^2 = CB^{-2} [n - (C + 1) B^{-C} \sum_{i=1}^n x_i^C] \quad (H.6)$$

$$\partial^2 \ln L / \partial C^2 = -nC^{-2} - B^{-C} \{ \sum_{i=1}^n x_i^C [\ln(x_i)]^2 - \ln(B) [2 \sum_{i=1}^n x_i^C \ln(x_i) - \ln(B) \sum_{i=1}^n x_i^C] \} \quad (H.7)$$

$$\partial^2 \ln L / \partial B \partial C = -nB^{-1} + B^{-(C+1)} \{ C \sum_{i=1}^n x_i^C \ln(x_i) + [1 - C \ln(B)] \sum_{i=1}^n x_i^C \} \quad (H.8)$$

$$\sigma B = \{ \text{abs}[\partial^2 \ln L / \partial C^2 (\partial^2 \ln L / \partial B^2 \partial^2 \ln L / \partial C^2 - (\partial^2 \ln L / \partial B \partial C)^2)^{-1}] \}^{0.5} \quad (H.9)$$

$$\sigma C = \{ \text{abs}[\partial^2 \ln L / \partial B^2 (\partial^2 \ln L / \partial B^2 \partial^2 \ln L / \partial C^2 - (\partial^2 \ln L / \partial B \partial C)^2)^{-1}] \}^{0.5} \quad (H.10)$$

### H.3.4 Analysis Procedure

#### H.3.4.1 Large Datasets

To calculate design strengths for a range of pipe sizes, the method should be as follows.

- Calculate the PDF parameters for each input variable, for measurements as specified in H.2.
- For a given grade, heat treatment, and rotary straightening type, calculate the  $D/t$  implied by dataspace positions  $\log_{10}(p_y/p_e)$  of  $-0.5$  to  $+0.5$  by intervals of  $0.2$ , with  $p_e$  and  $p_y$  from Equations (F.6) and (F.7), respectively.
- Calculate the design collapse strength for each dataspace position, using Equation (F.5) with the specified values of the input variables, and assumed values of  $k_{e \text{ des}}$  and  $k_{y \text{ des}}$ . The decrement factor  $H_{t \text{ des}}$  is obtained from Equation (H.1).
- Calculate the predicted failure probability ( $\phi_f$ ) for each dataspace position, for a deterministic load  $L_{\text{nom}}$  given by the design collapse strength in each case. Either correlated or independent variable analysis may be used; the former is more accurate but more complicated, whereas the latter is simpler but slightly conservative.  $\phi_f$  is the probability of collapse strength being less than the design strength. Plot  $\phi_f$  against dataspace position.
- Iterate over steps c) and d), adjusting  $k_{e \text{ des}}$  and  $k_{y \text{ des}}$  so as to obtain the flattest possible reliability response over the dataspace, and an average  $\phi_f$  within  $\pm 10\%$  of the TRL.
- Use the final values of  $k_{e \text{ des}}$  and  $k_{y \text{ des}}$  in Equation (F.5), with  $H_{t \text{ des}}$  as calculated in step c), to obtain design collapse strengths for the desired pipe sizes and masses.

If only a single pipe size is of interest, the design strength can be calculated by setting  $D$  and  $t$  to the appropriate values, and calculating  $\phi_f$  for a range of deterministic loads  $L_{\text{nom}}$ . The design strength is the value of  $L_{\text{nom}}$  to give  $\phi_f$  equal to the TRL.

#### H.3.4.2 Small Datasets

This approach should be used whenever the smallest dataset contains fewer than 1000 samples. For simplicity, the method for a single pipe case and 95% confidence of the TRL is described.

- a) Calculate the PDF parameters for each input variable sample, for measurements as specified in H.2, and the relevant values of  $D$  and  $t$ .
- b) For the input variables with fewer than 1000 samples, determine the sampling uncertainties of each PDF parameter, as described in H.3.3. For input variables with  $n > 1000$ , the PDF parameters can be taken as deterministic, with the values calculated in a) above.
- c) Use the sampling uncertainties to develop a minimum of 10,000 random realizations for each PDF parameter. This can be done by the inverse transform method [52], [81], [103] or by using the random-variates generation facility in a mathematical spreadsheet.
- d) Calculate the predicted failure probability ( $\phi_f$ ) for a given deterministic load  $L_{nom}$ , for each realization of PDF parameters. Either correlated or independent variable analysis can be used; the former is more accurate but more complicated, whereas the latter is simpler but slightly conservative. The sample correlation coefficients can be used as an approximation to the correlations for each realization; this gives reasonable results in practice.
- e) Plot the frequency of occurrence of  $\phi_f$ ; that is, its probability distribution.
- f) Interpolate the cumulative probability distribution for the 95 % confidence value.
- g) Repeat steps d) to f) for a range of load levels  $L_{nom}$ , chosen to make the 95 % confidence  $\phi_f$  bracket the TRL.
- h) The design strength is the value of  $L_{nom}$  to give a 95 % confidence  $\phi_f$  equal to the TRL. This can be obtained by interpolation.

## H.4 Example

### H.4.1 Large Datasets

#### H.4.1.1 Introduction

This example uses the P110 HRS data for mill FD00 of Annex F. The calculation steps are as in H.3.4. First, all the measurement datasets will be assumed to contain more than 1000 samples, such that calculation of sampling uncertainty is not necessary.

#### H.4.1.2 Single Pipe Size

Determine the design collapse strength for 9-5/8 in. 47 lb/ft P110 HRS from mill FD00.

- e) First, we determine the input variable PDF parameters. These are given in Tables F.2, F.3, and F.4 and are reproduced in Table H.4 below.

**Table H.4—Input Variable PDFs (Bias and COV Format)**

Variable	Mean	COV	Distribution	Units
Average OD	1.0071	0.00189	Gaussian	—
Average WT	1.0068	0.0217	Gaussian	—
Yield stress	1.160	0.0354	Gaussian	—
Ovality	0.241	0.338	Two-parameter Weibull	%
Eccentricity	5.170	0.317	Two-parameter Weibull	%
Residual/yield	−0.142	0.189	Gaussian	—
Model uncertainty	0.9681	0.0543	Gaussian	—

- f) Some of the PDF parameters are in dimensionless (generalized) format and shall be converted to means and standard deviations (SDs) for the pipe size of interest. The dimensional means are obtained as specified value  $\times$  bias, that is:

mean average outside diameter =  $9.625 \times 1.0071 = 9.693$  in.

mean average wall thickness =  $0.472 \times 1.0068 = 0.475$  2 in.

mean yield stress =  $110 \times 1.160 = 127.6$  ksi

the residual stress data are normalized by yield and need to be converted to a stress proper, as follows:

$$\mu_{rs} \cong \mu_{rs/f_y} \times \mu_{f_y} = \mu_{rs/f_y} \times (\mu_{f_y}/f_{ymn}) \times f_{ymn} = -0.142 \times 1.160 \times 110 = -18.12 \text{ ksi} \quad (\text{H.11})$$

$$\text{COV}_{rs} \cong (\text{COV}_{rs/f_y}^2 - \text{COV}_{f_y}^2)^{0.5} \text{ (assuming SRSS)} = (0.189^2 - 0.0354^2)^{0.5} = 0.186 \quad (\text{H.12})$$

The SDs are obtained as dimensional mean  $\times$  COV. Table H.5 summarizes the various values.

**Table H.5—Input Variable PDFs (Mean and SD Format, 9-5/8 in. 47 lb/ft P110 HRS)**

Variable	Mean	SD	Distribution	Units
Average OD	9.693	0.01832	Gaussian	in.
Average WT	0.4752	0.01031	Gaussian	in.
Yield stress	127.6	4.517	Gaussian	ksi
Ovality	0.2407	0.08146	Two-parameter Weibull	%
Eccentricity	5.170	1.639	Two-parameter Weibull	%
Residual stress	-18.12	3.364	Gaussian	ksi
Model uncertainty	0.9681	0.05257	Gaussian	—

- g) Second, calculate predicted failure probability  $\phi_f$  for the Table H.5 PDF data and a given deterministic load  $L_{nom}$ . Taking  $L_{nom} = 6100$  psi gives  $\phi_f = 4.51 \times 10^{-3}$ , calculated using FORM. This is too low, so  $L_{nom}$  is increased until  $\phi_f$  brackets the TRL, as shown in Table H.6 below.

**Table H.6—Case-specific Calibration for Mill FD00 (9-5/8 in. 47 lb/ft P110 HRS)—Large Datasets**

$L_{nom}$ psi	$\phi_f$
6100	$4.511 \times 10^{-3}$
6110	$4.769 \times 10^{-3}$
6120	$5.039 \times 10^{-3}$

- h) The design collapse strength is obtained by interpolation as 6119 psi.

#### H.4.1.3 Multiple Pipe Sizes

Determine the design collapse strength of P110 HRS product from mill FD00, for a variety of pipe sizes.

The analysis is similar in many respects to the single pipe size case, but the pipe dimensions (average OD and WT) are now generalized by using a range of dataspace positions. Failure probability and design collapse strength (as down-rating factors  $k_{e\text{ des}}$  and  $k_{y\text{ des}}$ ) are then calculated for each dataspace position, as described below.

- a)  $D/t$  is calculated for each dataspace position, using Equations (F.6) and (F.7) with the specified values of  $f_{ymn}$ ,  $E$  and  $\nu$ . Equations (F.6) and (F.7) cannot be solved directly for  $D/t$  for a given value of  $p_y/p_e$ , and the solution is therefore obtained either by iteration or by using a root-solver routine in a mathematical spreadsheet.

The iterative solution for  $\log_{10}(p_y/p_e) = -0.5$  is given for illustration. For  $D/t = 10.4$ :

$$p_y = (2 \times 110,000/10.4) [1 + (0.5/10.4)] = 22,171 \quad (\text{H.13})$$

$$p_e = 2 \times 30 \times 10^6 / [(1 - 0.28^2) \times 10.4 \times (10.4 - 1)^2] = 70,847 \quad (\text{H.14})$$

$$\log_{10}(p_y/p_e) = \log_{10}(22,171/70,847) = -0.5045 \quad (\text{H.15})$$

Table H.7 shows the remaining iterations, and Table H.8 the final  $D/t$  value for each dataspace position.

**Table H.7—Iteration for  $D/t$**

$D/t$	$p_y$ psi	$p_e$ psi	$\log_{10}(p_y/p_e)$
10.40	22,171	70,847	-0.5045
10.50	21,950	68,702	-0.4955
10.44	22,082	69,978	-0.5009
10.45	22,060	69,764	-0.5000

**Table H.8— $D/t$  for Each Dataspace Position**

$\log_{10}(p_y/p_e)$	$D/t$	$p_y$ psi	$p_e$ psi
-0.5	10.45	22,060	69,764
-0.3	12.95	17,644	35,205
-0.1	16.10	14,090	17,738
0.1	20.07	11,237	8926
0.3	25.06	8954	4488
0.5	31.35	129	2255

- b) The mean outside diameter and wall thickness are obtained as specified value  $\times$  bias, for a given specified OD; for instance, for 9-5/8 in. and  $\log_{10}(p_y/p_e) = -0.5$ :

mean average outside diameter =  $9.625 \times 1.0071 = 9.693$  in.

mean average wall thickness =  $(9.625/10.45) \times 1.0068 = 0.9273$  in.

standard deviation of average wall thickness =  $0.9273 \times 0.0217 = 0.02012$  in.

The remainder of the PDF parameters are as before (Table H.5).

- c) The nominal load  $L_{nom}$  is then calculated for each dataspace position, using Equation (F.5) with the specified values of  $D$ ,  $t$ ,  $f_{ymn}$ ,  $E$ , and  $\nu$ , or equivalently the values of  $p_e$  and  $p_y$  from step a), a decrement factor  $Ht_{des}$  obtained as [see Equation (F.4) and Table H.4]:

$$Ht_{des} = (0.127 \times 0.214) + (0.0039 \times 5.170) - [0.440 \times (-0.142)] + 0 = 0.1132 \quad (H.16)$$

and initial values for  $k_{e des}$  and  $k_{y des}$ ; 0.825 and 0.91 are usually reasonable choices. Table H.9 shows the results.

NOTE The values of  $L_{nom}$  for iterations 2 and 3 are calculated as part of step e); provided for convenience.

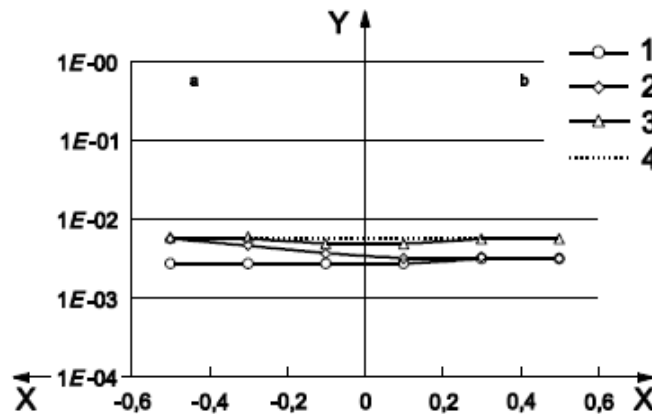
**Table H.9—Nominal Loads**

	$L_{nom}$		
Iteration	1	2	3
$k_{e des}$	0.825	0.825	0.840
$k_{y des}$	0.910	0.925	0.925
$\log_{10}(p_y/p_e)$			
−0.5	19,012	19,302	19,328
−0.3	14,440	14,637	14,682
−0.1	10,183	10,282	10,358
0.1	6251	6279	6362
0.3	3422	3428	3483
0.5	1783	1785	1816

- d) Predicted failure probability  $\phi_f$  under a deterministic load of  $L_{nom}$  is then calculated for each dataspace position, using probabilistic analysis implementing Equation (F.21).  $\phi_f$  is the probability of collapse strength being less than  $L_{nom}$ . Figure H.1 shows  $\phi_f$  versus dataspace position, calculated using FORM.

NOTE The curves for iterations 2 and 3 are calculated as part of step e), but are given here for brevity.

- e) The best-fit values of  $k_{e des}$  and  $k_{y des}$  are then determined by iteration. Figure H.1 shows that  $\phi_f$  for iteration 1 is well below the chosen TRL, with an average of  $2.59 \times 10^{-3}$ . Raising  $k_{e des}$  will increase  $L_{nom}$  in the elastic range, and thus likewise increase  $\phi_f$ . This will tilt the right-hand (elastic) end of the curve upwards. Similarly, increasing  $k_{y des}$  will tilt the left-hand (yield) end of the curve upwards. It is desired to lift both ends of the curve, but for illustration each will be done one separately. For iteration 2,  $k_{y des}$  is raised to 0.925; this increases  $L_{nom}$  as shown in Table H.9, and  $\phi_f$  as in Figure H.1. The yield end of the curve is now in about the right place. The elastic end is still too low, however, as is the average  $\phi_f$  ( $3.53 \times 10^{-3}$ ). For iteration 3,  $k_{e des}$  is increased to 0.84. This gives the best overall fit to the TRL, with an average  $\phi_f$  of  $4.83 \times 10^{-3}$ .



#### Key

X log (yield/elastic strength)

Y probability of failure

1 iteration 1

2 iteration 2

3 iteration 3

4 target reliability

a yield

b elastic

NOTE Manufacturer FD00, P110 seamless HRS.

**Figure H.2—Case-specific Calibration for Mill FD00 (Multiple Pipe Sizes)**

- f) The final values of  $k_{e\ des}$  and  $k_{y\ des}$  are then used to develop design collapse ratings for the desired sizes and weights, as shown in Table H.10. Equation (F.5) is used, together with the specified values of  $D$ ,  $t$ ,  $f_{ymn}$ ,  $E$  and  $\nu$ , and  $H_{t\ des}$  from Equation (H.1). The **API 5C3 (6<sup>th</sup> Edition)** ratings are given for comparison. Note that the rating for 9-5/8 in. 47 lb/ft P110 HRS (6106 psi) is very slightly lower than the value given by the single-size method (6119 psi [H.4.1.2]); the difference is because the rating has been calculated as a best fit to the TRL line (Figure H.1), rather than for the exact TRL as before.

**Table H.10—Mill-specific Design Ratings**

Size, Weight, and Grade	API <b>5C3 (6<sup>th</sup> Ed.)</b> Rating psi	Design Rating psi	Increase %
20 in. 94 lb/ft P110 HRS	516	590	14.3
13-3/8 in. 72 lb/ft P110 HRS	2880	3125	8.5
9-5/8 in. 47 lb/ft P110 HRS	5300	6106	15.2
7 in. 32 lb/ft P110 HRS	10,780	11,158	3.5
NOTE The design ratings should not be used for CRS pipe.			

## H.4.2 Small Datasets

Determine the design collapse strength for 9-5/8 in. 47 lb/ft P110 HRS from mill FD00, for the actual dataset sizes.

Determination of input variable PDF parameters. These are as before (Table H.5), but now the parameters are treated as random variables, rather than deterministic.

Sampling uncertainties. Using the methods of H.3.3 gives values as Table H.12.

Random realizations of PDF parameters. Table H.11 gives the first and last few realizations. They were obtained using the random deviates generation facility in a mathematical spreadsheet.

**Table H.11—Random Realizations of PDF Parameters (9-5/8 in. 47 lb/ft P110 HRS)**

Variable	Parameter	Distribution	Realization					
			1	2	3	...	9999	10,000
Average OD in.	Mean	Gaussian	9.6928	9.6925	9.6927	...	9.6933	9.6953
	SD	Gaussian	0.01871	0.02082	0.01819	...	0.01908	0.01903
Average WT in.	Mean	Gaussian	0.4762	0.4758	0.4734	...	0.4741	0.4742
	SD	Gaussian	0.01039	0.01048	0.00962	...	0.00992	0.00956
Yield stress ksi	Mean	Gaussian	126.2	127.0	126.9	...	127.1	126.9
	SD	Gaussian	4.300	2.839	4.336	...	4.996	4.911
Ovality %	<i>B</i>	Gaussian	0.2628	0.2670	0.2763	...	0.2712	0.2765
	<i>C</i>	Log-normal	3.219	3.229	3.251	...	3.213	3.240
Eccentricity %	<i>B</i>	Gaussian	5.706	5.739	5.829	...	5.847	5.543
	<i>C</i>	Log-normal	3.524	3.590	3.579	...	3.438	3.392
Residual stress ksi	Mean	Gaussian	-17.69	-17.37	-17.91	...	-18.31	-18.24
	SD	Gaussian	3.493	3.251	3.559	...	3.204	3.333
Model uncertainty	Mean	Gaussian	0.9608	0.9619	0.9675	...	0.9718	0.9648
	SD	Gaussian	0.04822	0.04796	0.05816	...	0.04933	0.05664

**Table H.12—Sampling Uncertainties**

Variable	Samples	Gaussian		Two-parameter Weibull		Units
		Mean	SD	<i>B</i>	<i>C</i>	
		Mean SD	Mean SD	Mean SD	Mean SD	
Average OD	203	9.693 0.001286	0.01832 0.000909	—	—	in.
Average WT	132	0.4752 0.000898	0.01031 0.000635	—	—	in.
Yield stress	46	127.6 0.6660	4.517 0.4709	—	—	ksi
Ovality	204	—	—	0.2689 0.005747	3.276 0.04123	%
Eccentricity	194	—	—	5.745 0.1171	3.510 0.06634	%
Residual stress	54	-18.12 0.4578	3.364 0.3237	—	—	ksi



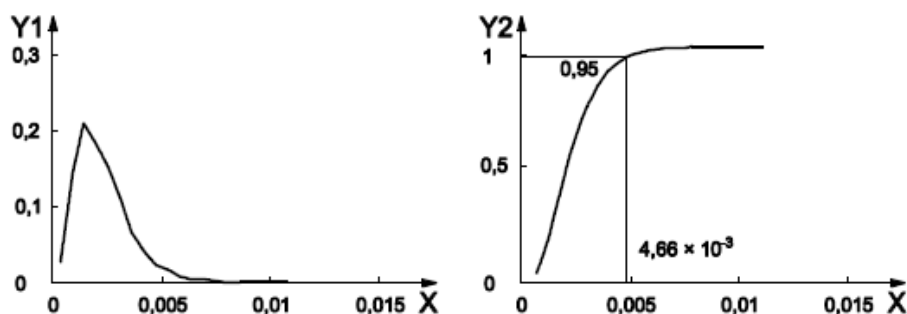
Model uncertainty	75	0.9681 0.006070	0.05257 0.004292	—	—	—
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- i) The predicted failure probability ( $\phi_f$ ) is then calculated for each realization. Table H.13 gives  $\phi_f$  for the first and last few realizations, for a deterministic load  $L_{nom}$  of 5960 psi. All values were calculated using FORM.

**Table H.13—Predicted Failure Probability for Each Realization ( $L_{nom} = 5960$  psi)**

Realization	$\phi_f$
1	$1.641 \times 10^{-3}$
2	$1.488 \times 10^{-3}$
3	$3.947 \times 10^{-3}$
...	...
9999	$1.516 \times 10^{-3}$
10,000	$3.357 \times 10^{-3}$

- j) The probability distribution of  $\phi_f$  (Figure H.2) is determined by the counting the number of occurrences of  $\phi_f$  in each interval. The 95 % confidence value is taken; this is obtained by interpolating the cumulative density.



**a) Probability distribution function (PDF)**

**b) Cumulative distribution function (CDF)**

**Key**

X probability of failure

Y1 probability density

Y2 cumulative probability

NOTE Manufacturer FD00, 9-5/8 in. 47 lb/ft P110 HRS.

**Figure H.3—PDF and CDF of Predicted Failure Probability ( $L_{nom} = 5960$  psi)**

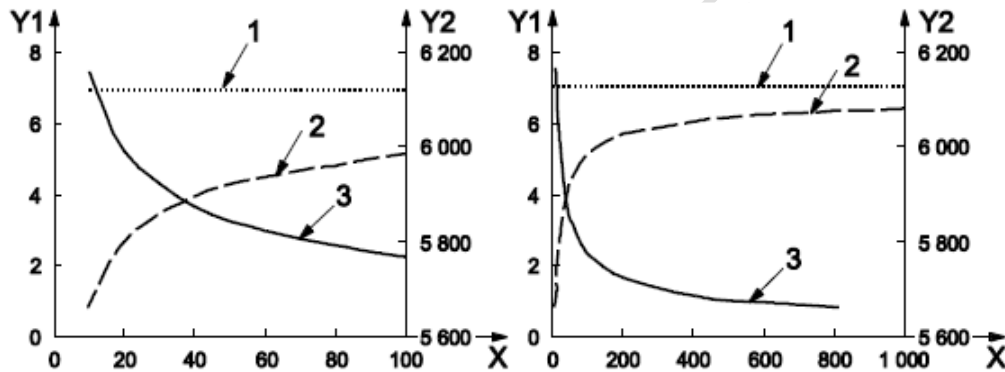
- k) For  $L_{nom} = 5960$  psi, the 95 % confidence failure probability ( $\phi_{f,0.95}$ ) is  $4.66 \times 10^{-3}$  (Figure H.2). This is slightly too low, and  $L_{nom}$  shall therefore be increased. Table H.14 shows  $L_{nom}$  versus  $\phi_{f,0.95}$ . The

design strength is the value of  $L_{nom}$  to make  $\phi_{f 0.95}$  equal the TRL. This is obtained by interpolation as 5973 psi, some 2.4 % lower than the large-dataset value.

**Table H.14—Case-specific Calibration for Mill FD00 (9-5/8 in. 47 lb/ft P110 HRS)—Small Datasets**

$L_{nom}$ psi	$\phi_{f 0.95}$
5960	$4.663 \times 10^{-3}$
5970	$4.919 \times 10^{-3}$
5980	$5.186 \times 10^{-3}$

Figure H.3 shows how collapse rating varies with dataset size, for the hypothetical case of all input variable dataset sizes being equal. The curves are specific to the given case and should not be used for general guidance.



**Key**

- X dataset size
- Y1 reduction with respect to large dataset limit, %
- Y2 collapse rating, psi
- 1 large dataset limit
- 2 rating
- 3 reduction

**Figure H.4—Reduction in Design Strength vs Dataset Size (Mill FD00, 9-5/8 in. 47 lb/ft P110 HRS)**

## Annex I

(~~informative~~normative)

### Collapse Test Procedure

#### I.1 Introduction

~~To be acceptable for API use, collapse tests should be conducted as described below.~~ Collapse tests shall be conducted as described below for testing at ambient temperature.

Unless otherwise agreed, collapse tests shall be conducted to failure.

#### I.2 Test Specimen

The minimum length of the collapse test specimen ~~should be as follows:~~ submitted to pressure, shall be eight times the specified outside diameter ( $D$ ). For sizes greater than 9-5/8 in., seven times the specified outside diameter ( $D$ ) may be used if limited by test apparatus.

~~— eight times the specified diameter ( $D$ ) for specified diameters of 9-5/8 in. and below;~~

~~— seven times the specified diameter ( $D$ ) for specified diameters greater than 9-5/8 in.~~

In addition, ~~to the length of the collapse test specimen, additional material should be allocated for the residual stress and tensile test specimens (see Figure I.1)~~ shall be taken from the same length as the collapse specimen and should be adjacent to the collapse specimen if possible.

#### I.3 Test Apparatus

The test apparatus ~~should~~ shall apply the test pressure to the full specimen length as defined in I.2. ~~Unless specified, it should the test apparatus shall~~ not impose radial or axial restraints on the specimen, either mechanically or hydraulically, and ~~should~~ shall not apply pressure to the inside surface of the specimen. ~~For combined collapse and axial load tests, the apparatus should maintain the axial load within  $\pm 1$  % of the target value during application of external pressure.~~

The test chamber ~~should~~ shall be equipped with a maximum reading pressure-measuring device that is open to the test chamber during the test. ~~The device should be certified by the manufacturer to be accurate within 0.5 % of the full scale reading.~~

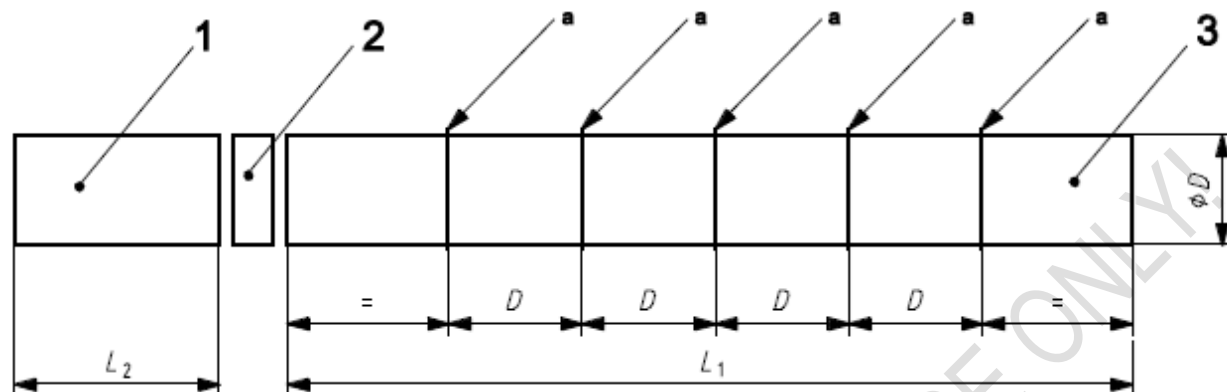
The pressure-measuring device should be equipped with a damping system to bleed pressure slowly upon specimen collapse. The ~~pressure-measuring device~~ should ~~shall~~ be calibrated in accordance with the frequency specified in API 5CT (for hydrostatic testing) and with the accuracy specified in API 5C5 ~~at six monthly intervals or more frequently if there is reason to doubt its accuracy. The error within its working range should not exceed 1 %.~~

#### I.4 Measurements prior to Collapse Testing

##### I.4.1 General

~~Pipe geometry, yield stress, and residual stress should be accurately measured prior to collapse testing, as described below.~~ Pipe geometry shall be measured prior to collapse testing as shown in Figure I.1.

Unless otherwise agreed, yield stress and residual stress shall be measured from adjacent samples in accordance with I.2.



#### Key

1 residual stress test specimen

2 tensile test specimen

3 collapse test specimen

$D$  specified outside diameter

$L_1$  minimum length of collapse test specimen

$L_2$  minimum length of residual stress test specimen

<sup>a</sup> Five equally spaced locations at which average outside diameter, average wall thickness, and ovality are measured, and eccentricity is calculated from wall thickness measurements.

**Figure I.1—Measurements prior to Collapse Testing**

## I.4.2 Pipe Geometry

### I.4.2.1 General

Average outside diameter, average wall thickness, ovality, and eccentricity ~~should~~ shall be ~~measured at five equally spaced locations, as shown in Figure I.1. Measurements and calculations for each location should be determined~~ as described in I.4.2.2 to I.4.2.5.

Alternative measurement methods may be used provided that the number of sections and measurement sampling along the sample axis is equal or greater than the one described in Figure I.1. Values reported in Table I.1 shall be determined after reducing the sampling to the same position and number of determinations described in I.4.2.2 to I.4.2.5. If measurement data from the full pipe length is used, the traceability of the measurements and sample position relative to the full pipe length shall be recorded.

### I.4.2.2 Average Outside Diameter

~~The average outside diameter should be measured with a pi-tape~~ shall be determined at five equally spaced locations as shown in Figure I.1. In case of a dispute, pi-tape (wrap tape) measurements shall govern.

### I.4.2.3 Average Wall Thickness

Thicknesses shall be measured and recorded to a minimum accuracy of 0.1 mm (0.004 in.).

**I.4.2.4** —Wall thickness ~~should~~ **shall** be ~~measured~~ **determined** at eight equally spaced positions (that is, at 45° intervals) for the five equally spaced locations, as shown in Figure I.1, and the average ~~taken~~ **calculated**. ~~Thicknesses should be measured and recorded to a minimum accuracy of 0.1 mm.~~

#### **I.4.2.5** **I.4.2.4** Ovality

Ovality ~~should~~ **shall** be ~~measured~~ **determined** at five equally spaced locations, as shown in Figure I.1 with an API ovality gauge or equivalent. Readings ~~should~~ **shall** be taken over ~~all circumferential positions~~ **the entire circumference**; measurements at equally spaced intervals (e.g. 45°) are not acceptable. Ovality ~~should be calculated at each location is defined, in percent,~~ as  $100 (D_{\max} - D_{\min}) / D_{\text{ave}}$ , where  $D_{\text{ave}}$  is the average outside diameter from I.4.2.2. In case of dispute, pipe ovality gauge measurements shall govern.

#### **I.4.2.6** **I.4.2.5** Eccentricity

Eccentricity ~~should~~ **shall** be calculated at five equally spaced locations, as shown in Figure I.1. Eccentricity at each location is defined, in percent, as  $100 (t_{c \max} - t_{c \min}) / t_{c \text{ ave}}$ , where  $t_{c \max}$  and  $t_{c \min}$  are, respectively, the maximum and minimum wall thicknesses from ~~the eight circumferential measurements of~~ I.4.2.3, and  $t_{c \text{ ave}}$  is the average wall thickness from 1.4.2.3.

### **I.4.3 Yield Stress**

Unless otherwise agreed, a tensile test ~~should~~ **shall** be conducted for each collapse test sample ~~using the split ring method~~. Unless otherwise agreed, the tensile specimen ~~should~~ **shall** be taken from pipe adjacent to the end of the collapse test specimen, as shown in Figure I.1. Tensile testing ~~should~~ **shall** be in accordance with **API 5CT**.

If the pipe is flame-cut to obtain material for a tensile test specimen, the specimen ~~should~~ **shall** not be prepared from areas including the heat-affected zone.

### **I.4.4 Residual Stress**

#### **I.4.4.1 Measurement and Calculation**

Unless otherwise agreed, residual stress ~~should~~ **shall** be measured for each collapse test sample, using the split ring method. The testing procedure shall be in accordance with ASTM E1928, except as stated herein. Unless otherwise agreed, the ring ~~should~~ **shall** be taken from pipe adjacent to the end of the collapse test specimen, as shown in Figure I.1. Sample lengths of at least twice the specified outside diameter ~~are necessary to accurately~~ ( $L/D \geq 2$ ) shall be used to measure residual stress <sup>[94] [123]</sup>; ~~shorter samples give lower predicted residual stresses.~~ Accordingly, two alternative approaches may be used, namely:

full length ( $L/D \geq 2$ ) specimens;

shorter specimens ( $2 > L/D \geq 0.5$ ), with the apparent residual stress corrected using a product-specific calibration curve for the effect of sample length. I.4.4.2 gives instructions for the preparation and use of such curves.

The calibration curve may be used for all subsequent collapse tests for the given manufacturing process. If any relevant part of the process changes (tempering temperature, straightening method, and so forth), testing should be repeated and the curve recalculated.

Testing should be in accordance with **ASTM E1928-99**, except as noted above. Circumferential residual stress at the inside surface,  $\sigma_{\text{res}}$ , ~~is~~ **Residual stress,  $\sigma_{\text{res}}$ , shall be** calculated as follows:

$$\sigma_{\text{res}} = E t_{\text{c-ave}} t_{\text{eff}} (1/D_{\text{ac}} - 1/D_{\text{bc}}) / (1 - \nu^2) \quad (1.1)$$

where

$D_{\text{ac}}$  is the average outside diameter after cutting;

$D_{\text{bc}}$  is the average outside diameter before cutting;

$E$  is Young's modulus,  $206.9 \times 10^9 \text{ N/m}^2$  ( $30 \times 10^6 \text{ psi}$ );

$t_{\text{c-aveeff}}$  is the average actual wall thickness;

$\nu$  is Poisson's ratio, 0.28.

This results in a negative ~~sign-on-~~ $\sigma_{\text{res}}$  residual stress if the pipe springs open (~~compression at ID face~~), and a positive ~~sign-on-~~ $\sigma_{\text{res}}$  stress if the pipe springs shut (~~tension at ID face~~). This is consistent with the sign convention used in Annex F (compression at ID face is negative). ~~The circumferential residual stress calculated in accordance with Equation (1.1) is assumed to be linearly distributed through the pipe wall from the inside surface to the outside surface.~~

#### I.4.4.2—Correction for Specimen Length

##### I.4.4.2.1—General

~~Correction curves should be based on test results for a total of twenty specimens of lengths from 0.5D to 2.0D. The slit ring method should be used, and residual stress should be calculated as described in I.4.4.1. The specimen lengths and cutting sequence should be as shown in Figure I.2. All specimens should be cut from a single pipe, as residual stress is approximately constant along each pipe but varies between pipes.~~

**CONTACT YOUR API SC5 STAFF REP IF NEEDED**

#### Key

$L$ —length of test specimen       $D$ —outside diameter      1 to 20 order of cutting specimens

**Figure I.2—Order of Cutting Specimens from Test Pipe**

~~The mean apparent residual stress should be calculated for each  $L/D$  and divided by the mean residual stress for  $L/D = 2$  to determine predicted/actual stress versus specimen length, a plot of which is the correction curve. The line can be assumed to become horizontal at  $L/D = 2$ . Figure I.3 gives an example of curve preparation. Separate curves should be prepared for each grade and heat treatment type (e.g. normalized N80 and quenched and tempered N80 are separate cases).~~

##### I.4.4.2.2—Example of Use

~~A slit ring specimen of length 0.75D gives an apparent residual stress of 23.56 ksi. Reference to a correction curve previously prepared for the relevant grade and heat treatment (Figure I.3) gives a correction factor of 0.804. The actual residual stress is therefore  $23.56/0.804 = 29.3 \text{ ksi}$ . Figure I.3 is for illustration only, and should not be used for any other purpose.~~

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#### Key

$X$ —specimen length/outside diameter

$Y$ —predicted residual stress/actual residual stress

For a given grade and heat treatment, slit ring testing of a single pipe gives predicted residual stresses as follows:

	L/D			
	0.5	1.0	1.5	2.0
Predicted residual stress	-27.60	-31.28	-36.43	-37.35
	-27.97	-33.49	-37.54	-38.64
	-24.66	-34.22	-35.14	-35.33
	-25.76	-30.91	-34.22	-34.78
	-28.70	-29.81	-33.12	-37.90
Mean	-26.94	-31.94	-35.29	-36.80
Predicted/actual stress	-26.94/-36.80	-31.94/-36.80	-35.29/-36.80	-36.80/-36.80
Ratio	0.732	0.868	0.959	1

Figure I.3—Example of Preparation of Residual Stress Correction Curve

#### I.4.4.3 Axial Force

When a tubular is bent laterally into a circular arc, the pressures apply a lateral uniform distributed load (UDL) of  $(p_i A_i - p_o A_o) / R$ .

For small deflections, the curvature is defined as  $1 / R \equiv d^2 y / dx^2$ . Thus, this term can be grouped with the tension term  $E d^4 y / dx^4$  in the governing differential equation.

For bending and buckling, the tubular therefore acts as if it were loaded by the effective axial force,  $F_{eff} = F_a - p_i A_i + p_o A_o$ . It should be seen as a convenient grouping of terms, which determines the structural response; it does not exist as a physical axial force.

### I.5 Test Procedure

The exterior surface of the specimen ~~should~~ shall be hydraulically loaded at a rate ~~sufficiently slow as to~~ permit reading of the collapse pressure within the specified accuracy. ~~Tests may be conducted either with or without axial stress. If the former~~ When axial stress is specified, the axial load ~~should~~ shall be applied first, and held constant during pressure loading.

### I.6 Data Reporting

~~The data reported should be as shown in Table I.1~~ The information provided in Table I.1 shall be reported. If information designated in I.4.3 to I.4.4 is not collected by agreement, this shall be noted in the report. If a specimen does not collapse, the maximum pressure reached during testing shall be held for not less than 5 s and shall be reported and denoted as "Proof" instead of "Collapse". ~~The pipe geometry properties (average outside diameter, average wall thickness, ovality, and eccentricity) should be the average of the values for the five circumference locations.~~

For I.4.2 results, the pipe geometry properties (average outside diameter, average wall thickness, ovality, and eccentricity) shall be the average of the values of, determined according to I.4.2. to I.4.2.5 I.4.2.1 and Figure I.1, the five circumference locations taken over 4 times the specified diameter. Data should be provided in electronic format if possible.

**Table I.1—Format for Reporting Collapse Test Data**

Grade	Nominal Wall Thickness in.	Forming Process <sup>a</sup>	Heat Treat- ment <sup>b</sup>	Yield Stress ksi	Average OD <sup>c</sup> in.	Ovality <sup>c, d</sup> %	Average Wall Thickness <sup>ac</sup> in.	Eccen- tricity <sup>c, e</sup> %	Straight- ening <sup>f</sup>	Residual Stress <sup>g</sup> ksi	L/D <sup>h</sup>	Axial Stress <sup>i</sup> ksi	Collapse Pressure ksi	Test Result <sup>j</sup>
P110	0.362	SR	QT	117.9	7.047	0.085	0.364	12.4	CG	-27.9	11.2	12.1	8.17	Collapse
P110	0.362	SR	QT	120.6	7.044	0.241	0.366	6.8	CG	-25.6	11.2	11.4	7.74	Proof
P110	0.362	SR	QT	118.2	7.046	0.185	0.363	9.0	CG	-28.1	11.2	0	7.57	Proof
P110	0.362	SR	QT	115.1	7.046	0.170	0.366	22.5	CG	-24.2	11.2	0	7.65	Collapse
P110	0.362	SR	QT	113.4	7.043	0.071	0.363	9.9	CG	-34.7	11.2	0	8.31	Proof

**NOTE** Data may be entered in metric units if desired.

**a** Forming process:  
 —SP = seamless, pilger mill; SM = seamless, plug mill; SR = seamless, retained or floating mandrel mill; SS = seamless, stretch reduced; SE = seamless, hot expanded.  
 —WE = welded, electric weld; WL = welded, laser weld; WS = welded, submerged arc weld.

**b** Type of heat treatment/cold working:  
 —AR = as rolled; CR = control rolled; NR = normalized; NT = quenched and tempered; QT = quenched and tempered; CP = cold worked, pilgered; CD = cold worked, drawn.

**c** Average of values for the circumferences.

**d** Ovality = 100 (maximum OD – minimum OD)/average OD.

**e** Eccentricity = 100 (maximum WT – minimum WT)/average WT.

**f** Type of straightening:  
 —NS = not straightened; CG = cold gag straightened; CX = cold cross-rolled straightened; CS = cold straightened and stress-relieved; HR = hot rotary straightened.

**g** Compression at ID face is negative. Hence enter residual stress as negative if the pipe springs open after cutting, and positive if it springs shut.

**h** L = specimen length; D = nominal diameter.

**i** Tension is positive.

**j** Test result: Proof = specimen did not collapse; Collapse = specimen collapsed



<u>Nominal</u>	<u>Weight</u>	<u>lb/ft</u>	
		<u>26.0</u>	
		<u>26.0</u>	
		<u>26.0</u>	
		<u>26.0</u>	
		<u>26.0</u>	
<u>Nominal</u>	<u>OD</u>	<u>in.</u>	
		<u>7.000</u>	
		<u>7.000</u>	
		<u>7.000</u>	
		<u>7.000</u>	
		<u>7.000</u>	

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## Annex J (informative)

### Discussion of Equations for Joint Strength

#### J.1 Introduction

Joint strength is a measure of the structural integrity of a threaded connection and does not include consideration of leak resistance. For casing applications, where installation of the tubular string is considered permanent, the limit load may be based on either yield or fracture/pull-out of the connector. For tubing applications, where the tubular string may be repeatedly recovered from and re-installed in the wellbore, the limit load is usually based on yield of the connector.

#### J.2 Design Equations for Tensile Joint Strength of API Casing Connections

##### J.2.1 General

The following tensile joint strength performance properties apply to casing connections manufactured in accordance with API 5B and **API 5CT**. API 5B design values were originally developed in USC units and appear in some of the equations. It is suggested that users perform these calculations in USC units and then convert the result to SI units if needed.

##### J.2.2 Round Thread Casing Joint Strength

###### J.2.2.1 Limit State Equation

Round thread casing joint strength is calculated (in USC units) by taking the minimum of the fracture strength and pull-out strength of the connection:

$$P_j = A_{jp} f_{up} \text{ (fracture strength)} \quad (J.1)$$

or

$$P_j = A_{jp} L_{et} [(0.74D - 0.59 f_{up}) / (0.5L_{et} + 0.14D) + f_{yp} / (L_{et} + 0.14D)] \text{ (pull-out strength)} \quad (J.2)$$

where

$$A_{jp} = \pi/4 [(D - 0.1425)^2 - d^2] \quad (J.3)$$

and

$A_{jp}$  is the area of the pipe cross section under the last perfect thread, in square inches;

$D$  is the specified pipe outside diameter, in inches;

$d$  is the pipe inside diameter,  $d = D - 2t$ , in inches;

$f_{up}$  is the tensile strength of a representative tensile specimen from the pipe body, in pounds per square inch;

$f_{yp}$  is the yield strength of a representative tensile specimen from the pipe body, in pounds per square inch;

$L_{et}$  is the engaged thread length,  $[= L_4 - M]$  for nominal make-up, in accordance with API 5B, in inches;

$P_j$  is the joint strength, in pounds;

$t$  is the specified pipe wall thickness, in inches.

### J.2.2.2 Design Equation

Round thread casing design joint strength is calculated (in USC units) by taking the minimum of the fracture strength and pull-out strength of the connection:

$$P_j = 0.95 A_{jp} f_{umnp} \text{ (fracture strength)} \quad (J.4)$$

or

$$P_j = 0.95 A_{jp} L_{et} [(0.74D^{-0.59} f_{umnp}) / (0.5L_{et} + 0.14D) + f_{ymnp} / (L_{et} + 0.14D)] \text{ (pull-out strength)} \quad (J.5)$$

where

$$A_{jp} = \pi/4 [(D - 0.1425)^2 - d^2] \quad (J.6)$$

and

$A_{jp}$  is the area of the pipe cross section under the last perfect thread, in square inches;

$D$  is the specified pipe outside diameter, in inches;

$d$  is the pipe inside diameter,  $d = D - 2t$ , in inches;

$f_{umnp}$  is the specified minimum tensile strength of the pipe body, in pounds per square inch;

$f_{ymnp}$  is the specified minimum yield strength of the pipe body, in pounds per square inch;

$L_{et}$  is the engaged thread length  $[= L_4 - M]$  for nominal make-up, in accordance with API 5B, in inches;

$P_j$  is the joint strength, in pounds;

$t$  is the specified pipe wall thickness, in inches.

### J.2.2.3 Background

Derivation of the limit state equations is covered in [ASME Paper No. 64-PET-1](#). The equations are based on the results of an API-sponsored test program consisting of tension tests of 162 joints of round-thread casing in grades K55, N80, and P110, covering a range of wall thicknesses in 114.3 mm, 127.0 mm, 139.7 mm, 168.3 mm, 177.8 mm, 244.5 mm, and 273.0 mm diameters, using both short and long threads where called for by the size and grade tested. Fourteen tests failed by fracture of the pipe and 148 tests failed by pull-out. The fracture strength Equation (J.4) agrees satisfactorily with the 14 test fractures. The pull-out strength Equation (J.5) is based on analytical considerations and was adjusted to fit the data by

statistical methods. The analytical procedure comprehended coupling properties, but it was found by analysis of the current group of tests that the coupling was noncritical for standard coupling dimensions as listed in API 5B. Subsequent testing established that these equations are also applicable to J55 casing.

The factor 0.95 in Equations (J.4) and (J.5) originates in the statistical error of a multiple-regression equation with adjustment to permit the use of minimum properties in place of average properties.

#### J.2.2.4 Coupling Fracture Strength

##### J.2.2.4.1 Limit State Equation

Should the coupling dimensions be such that coupling fracture strength can be lower than either the fracture strength or pull-out strength of the pipe body, the coupling fracture strength can be determined from:

$$P_j = A_{jc} f_{uc} \quad (J.7)$$

where

$A_{jc}$  is the area of the coupling cross section,  $A_{jc} = \pi/4 (W^2 - d_1^2)$ , in square inches;

$d_1$  is the diameter at the root of the coupling thread at the end of the pipe in the power-tight position, in inches;

$f_{uc}$  is the tensile strength of a representative tensile specimen from the coupling, in psi;

$P_j$  is the joint strength, in pounds;

$W$  is the specified coupling outside diameter, in accordance with **API 5CT**, in inches;

and

$$d_1 = E_1 - (L_1 + pA)T_d + H - 2s_m, \text{ in inches} \quad (J.8)$$

where

$A$  is the hand-tight standoff, in turns;

$E_1$  is the pitch diameter at the hand-tight plane, in accordance with API 5B, in inches;

$H$  is the thread height of a round thread equivalent Vee thread, as follows:  
2.1996 mm (0.08660 in.) for 10 TPI  
2.7496 mm (0.10825 in.) for 8 TPI;

$L_1$  is the length from the end of the pipe to the hand-tight plane, in accordance with API 5B, in inches;

$p$  is the thread pitch, in inches;

$s_m$  is the root truncation of the pipe thread of round threads, as follows:  
0.36 mm (0.014 in.) for 10 TPI  
0.43 mm (0.017 in.) for 8 TPI;

$T_d$  is the taper (on diameter): 0.0625 mm/mm (0.0625 in./in.).

#### J.2.2.4.2 Design Equation

Should the coupling dimensions be such that coupling fracture strength can be lower than either the fracture strength or pull-out strength of the pipe thread, the coupling fracture strength (joint strength  $P_j$ ) can be determined from:

$$P_j = 0.95 A_{jc} f_{umc}, \text{ in pounds} \quad (J.9)$$

where

$A_{jc}$  is the area of the coupling cross section,  $A_{jc} = \pi/4 (W^2 - d_1^2)$ , in square inches;

$d_1$  is the diameter at the root of the coupling thread at the end of the pipe in the power-tight position, in inches;

$f_{umc}$  is the tensile strength of a representative tensile specimen from the coupling, in psi;

$W$  is the specified coupling outside diameter, in accordance with API 5CT, in inches;

and

$$d_1 = E_1 - (L_1 + pA)T_d + H - 2s_m, \text{ in inches} \quad (J.10)$$

where

$A$  is the hand-tight standoff, in turns;

$E_1$  is the pitch diameter at the hand-tight plane, in accordance with API 5B, in inches;

$H$  is the thread height of a round thread equivalent Vee thread, as follows:  
2.1996 mm (0.08660 in.) for 10 TPI  
2.7496 mm (0.10825 in.) for 8 TPI;

$L_1$  is the length from the end of the pipe to the hand-tight plane, in accordance with API 5B, in inches;

$p$  is the thread pitch, in inches;

$s_m$  is the root truncation of the pipe thread of round threads, as follows:  
0.36 mm (0.014 in.) for 10 TPI  
0.43 mm (0.017 in.) for 8 TPI;

$T_d$  is the taper (on diameter): 0.0625 mm/mm (0.0625 in./in.).

### J.2.3 Buttress Thread Casing Joint Strength

#### J.2.3.1 Limit State Equation

Buttress thread casing joint strength is calculated by taking the minimum of the pipe thread strength and the coupling thread strength:

$$P_j = A_p f_{up} [1.008 - 0.0396(1.083 - f_{yp}/f_{up})D] \text{ (pipe thread strength)} \quad (J.11)$$

or

$$P_j = A_{jc} f_{uc} \text{ (coupling thread strength)} \quad (J.12)$$

where

$A_{jc}$  is the area of the coupling cross section,  $A_{jc} = \pi/4 (W^2 - d_1^2)$ , in square inches;

$A_p$  is the area of the pipe cross section,  $A_p = \pi/4 (D^2 - d^2)$ , in square inches;

$D$  is the specified pipe outside diameter, in inches;

$d$  is the pipe inside diameter,  $d = D - 2t$ , in inches;

$d_1$  is the diameter at the root of the coupling thread at the end of the pipe in the power-tight position, in inches;

$f_{uc}$  is the tensile strength of a representative tensile specimen from the coupling, in pounds per square inch;

$f_{up}$  is the tensile strength of a representative tensile specimen from the pipe body, in pounds per square inch;

$f_{yp}$  is the yield strength of a representative tensile specimen from the pipe body, in pounds per square inch;

$P_j$  is the joint strength, in pounds;

$T$  is the specified pipe wall thickness, in inches;

$W$  is the specified coupling outside diameter, in accordance with **API 5CT**, in inches;

and

$$d_1 = E_7 - (L_7 + I_B) T_d + h_B, \text{ in inches} \quad (J.13)$$

where

$E_7$  is the pitch diameter, in accordance with API 5B, in inches;

$h_B$  is the buttress thread height: 1.575 mm for SI units, 0.062 in. for USC units;

$I_B$  is the length from the face of the buttress thread coupling to the base of the triangle in the hand-tight position: 10.16 mm (0.400 in.) for Label 1: 4-1/2, 12.70 mm (0.500 in.) for sizes between Label 1: 5 and Label 1: 13-3/8, inclusive, and 9.52 mm (0.375 in.) for sizes greater than Label 1: 13-3/8;

$L_7$  is the length of perfect threads, in accordance with API 5B, in inches;

$T_d$  is the taper (on diameter): 0.0625 mm/mm (0.0625 in./in.).

### J.2.3.2 Design Equation

Buttress thread casing design joint strength is calculated by taking the minimum of the pipe thread strength and the coupling thread strength:

$$P_j = 0.95A_p f_{umnp} [1.008 - 0.0396(1.083 - f_{ymnp}/f_{umnp})D] \text{ (pipe thread strength)} \quad (J.14)$$

or

$$P_j = 0.95A_{jc} f_{umnc} \text{ (coupling thread strength)} \quad (J.15)$$

where

- $A_{jc}$  is the area of the coupling cross section,  $A_{jc} = \pi/4 (W^2 - d_1^2)$ , in square inches;
- $A_p$  is the area of the pipe cross section,  $A_p = \pi/4 (D^2 - d^2)$ , in square inches;
- $D$  is the specified pipe outside diameter, in inches;
- $d$  is the pipe inside diameter,  $d = D - 2t$ , in inches;
- $d_1$  is the diameter at the root of the coupling thread at the end of the pipe in the power-tight position, in inches;
- $f_{umnc}$  is the specified minimum tensile strength of the coupling, in pounds per square inch;
- $f_{umnp}$  is the specified minimum tensile strength of the pipe body, in pounds per square inch;
- $f_{ymnp}$  is the specified minimum yield strength of the pipe body, in pounds per square inch;
- $P_j$  is the joint strength, in pounds;
- $t$  is the specified pipe wall thickness, in inches;
- $W$  is the specified coupling outside diameter, in accordance with **API 5CT**, in inches;

and

$$d_1 = E_7 - (L_7 + I_B) T_d + h_B, \text{ in inches} \quad (J.16)$$

where

- $E_7$  is the pitch diameter, in accordance with API 5B, in inches;
- $h_B$  is the buttress thread height: 1.575 mm for SI units, 0.062 in. for USC units;
- $I_B$  is the length from the face of the buttress thread coupling to the base of the triangle in the hand-tight position, as follows:  
 10.16 mm (0.400 in.) for Label 1: 4-1/2  
 12.70 mm (0.500 in.) for sizes between Label 1: 5 and Label 1: 13-3/8, inclusive  
 9.52 mm (0.375 in.) for sizes greater than Label 1: 13-3/8;
- $L_7$  is the length of perfect threads, in accordance with API 5B, in inches;
- $T_d$  is the taper (on diameter); for buttress threads,  $T_d$  is 0.0625 mm (0.0625 in./in.) for sizes Label 1: 13-3/8 and smaller and 0.0833 mm/mm (0.0833 in./in.) for sizes greater than Label 1: 13-3/8.

### J.2.3.3 Background

The buttress joint strength equations based on a regression analysis of 151 tests of buttress thread casing ranging in size from Label 1: 4-1/2 to Label 1: 20 outside diameter and in strength levels from 275.8 MPa to 1034.2 MPa minimum yield.

## J.2.4 API Tubing Connections Tensile Joint Strength

### J.2.4.1 General

The following tensile joint strength performance properties apply to tubing connections manufactured in accordance with API 5B and **API 5CT**.

### J.2.4.2 Non-upset Tubing

#### J.2.4.2.1 General

Non-upset tubing joint strength is calculated as the product of the yield strength and the area of the pipe cross section under the last perfect thread. The areas of the critical sections of regular tubing couplings and special-clearance couplings are, in all instances, greater than the governing critical areas of the pipe part of the joint and do not affect the strength of the joint.

#### J.2.4.2.2 Limit State Equation

The joint strength in tension of non-upset tubing is defined by Equation (J.19):

$$P_j = f_y \left\{ \pi/4 [(D_4 - 2h_s)^2 - d^2] \right\} \quad (J.17)$$

where

$D$  is the specified pipe outside diameter, in inches;

$d$  is the pipe inside diameter,  $d = D - 2t$ , in inches;

$D_4$  is the major diameter, in accordance with API 5B, in inches;

$f_y$  is the yield strength of a representative tensile specimen, in psi;

$h_s$  is the round thread height, as follows:

1.3122 mm (0.05560 in.) for 10 TPI

1.8098 mm (0.07125 in.) for 8 TPI;

$P_j$  is the joint strength, in pounds;

$t$  is the specified pipe wall thickness, in inches.

#### J.2.4.2.3 Design Equation

The design joint strength in tension of non-upset tubing is defined by Equation (J.20):

$$P_j = f_{ymn} \left\{ \pi/4 [(D_4 - 2h_s)^2 - d^2] \right\} \quad (J.18)$$

where



$D$  is the specified pipe outside diameter, in inches;

$d$  is the pipe inside diameter,  $d = D - 2t$ , in inches;

$D_4$  is the major diameter, in accordance with API 5B, in inches;

$F_{ymn}$  is the specified minimum yield strength, in psi;

$h_s$  is the round thread height, as follows:

1.3122 mm (0.05560 in.) for 10 TPI

1.8098 mm (0.07125 in.) for 8 TPI;

$P_j$  is the joint strength, in pounds;

$t$  is the specified pipe wall thickness, in inches.

### J.2.4.3 Upset Tubing

#### J.2.4.3.1 General

Upset tubing joint strength is calculated as the product of the yield strength and the area of the body of the pipe. The area of the section under the last perfect thread of API upset tubing is greater than the area of the body of the pipe. The areas of the critical sections of regular tubing couplings, special-clearance couplings, and the box of integral-joint tubing are, in all instances, greater than the governing critical areas of the pipe part of the joint and do not affect the strength of the joint.

#### J.2.4.3.2 Limit State Equation

The joint strength in tension of upset tubing is defined by Equation (J.21):

$$P_j = f_y [\pi/4 (D^2 - d^2)] \quad (J.19)$$

where

$d$  is the pipe inside diameter,  $d = D - 2t$ , in inches;

$D$  is the specified pipe outside diameter, in inches;

$f_y$  is the yield strength of a representative tensile specimen, in psi;

$P_j$  is the joint strength, in pounds;

$t$  is the specified pipe wall thickness, in inches.

#### J.2.4.3.3 Design Equation

The design joint strength in tension of upset tubing is defined by the following expression:

$$P_j = f_{ymn} [\pi/4 (D^2 - d^2)] \quad (J.20)$$

where

$d$  is the pipe inside diameter,  $d = D - 2t$ , in inches;

$D$  is the specified pipe outside diameter, in inches;

$f_{ymn}$  is the specified minimum yield strength, in psi;

$P_j$  is the joint strength, in pounds;

$t$  is the specified pipe wall thickness, in inches.

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## Annex K (informative)

### Tables of Calculated Performance Properties in SI Units

#### K.1 Introduction

All listed values for performance properties in this annex assume a benign environment and material properties conforming to **API 5CT**. Other environments may require additional analyses, such as that outlined in Annex D.

#### K.2 List of Tables Contained in Annex K and Accompanying Notes

##### Table K.1—Performance Property Calculations—External and Internal Pressure for Casing (SI Units)

NOTE 1 All performance properties values in this document assume a benign environment and material properties conforming to **API 5CT**; other environments may require additional analyses, such as that outlined in Annex D.

NOTE 2 Calculation results for N80 are repeated for each size and linear mass; the first set represents pipe manufactured using the nonquenched and tempered process; the second set represents pipe manufactured using the quenched and tempered process.

NOTE 3 Calculation results for P110 are repeated for each size and linear mass; the first set represents product inspected to a 12.5 % calibration notch; the second set represents product inspected to a 5 % calibration notch.

NOTE 4 The designation L80\* includes grades L80 Type 1 and L80 13Cr.

NOTE 5 The minimum internal yield pressure is the lowest of internal yield pressure of the pipe or the internal yield pressure of the coupling; the interface pressure at the E1 plane for round thread casing or at the E7 plane for buttress thread casing may be less than the minimum internal yield pressure for the connection (see Table K.2).

NOTE 6 The collapse resistance values are based on the historical **API 5C3 (6<sup>th</sup> Edition)** or **ISO 10400:1993** equations (see Section 8).

##### Table K.2—Performance Property—Interface Pressure of Casing Connections (SI Units)

NOTE 1 All performance properties values in this document assume a benign environment and material properties conforming to **API 5CT**; other environments may require additional analyses, such as that outlined in Annex D.

NOTE 2 The interface pressure at the E1 plane for round thread casing or at the E7 plane for buttress thread casing may be less than the minimum internal yield pressure for the connection (see Table K.1).

##### Table K.3—Performance Property Calculations—Axial Tension of Casing Pipe Body and Connections (SI Units)

NOTE 1 All performance properties values in this document assume a benign environment and material properties conforming to **API 5CT**; other environments may require additional analyses, such as that outlined in Annex D.

NOTE 2 Calculation results for M65 and N80 are repeated for each size and linear mass; the first set represents pipe manufactured using the nonquenched and tempered process; the second set represents pipe manufactured using the quenched and tempered process; for a particular grade, the axial tension properties are not affected by the heat treat process.

NOTE 3 Calculation results for P110 are repeated for each size and linear mass; the first set represents product inspected to a 12.5 % calibration notch; the second set represents product inspected to a 5 % calibration notch; for a particular grade the axial tension properties are not affected by the different inspection calibration standards.

NOTE 4 The designation L80\* includes grades L80 Type 1 and L80 13Cr.

NOTE 5 Some joint strengths for the connections are greater than the corresponding pipe body yield strength.

NOTE 6 For J55 and K55 casing, the next higher grade coupling is L80; for N80 Q&T casing, the next higher grade coupling is P110; for P110 casing, the next higher grade coupling is Q125; no higher grade couplings have been established for the other grades.

#### **Table K.4—Performance Property Calculations—External and Internal Pressure for Tubing (SI Units)**

NOTE 1 All performance properties values in this document assume a benign environment and material properties conforming to **API 5CT**; other environments may require additional analyses, such as that outlined in Annex D.

NOTE 2 Calculation results for N80 are repeated for each size and linear mass; the first set represents pipe manufactured using the nonquenched and tempered process; the second set represents pipe manufactured using the quenched and tempered process.

NOTE 3 Calculation results for P110 are repeated for each size and linear mass; the first set represents product inspected to a 12.5 % calibration notch; the second set represents product inspected to a 5 % calibration notch.

NOTE 4 The designation L80\* includes grades L80 Type 1 and L80 13Cr.

NOTE 5 The minimum internal yield pressure is the lowest of internal yield pressure of the pipe or the internal yield pressure of the coupling.

NOTE 6 The collapse resistance values are based on the historical **API 5C3 (6<sup>th</sup> Edition) or ISO 10400:1993** equations (see Section 8).

#### **Table K.5—Performance Property Calculations—Axial Tension for Tubing Pipe Body and Connections (SI Units)**

NOTE 1 All performance properties values in this document assume a benign environment and material properties conforming to **API 5CT**; other environments may require additional analyses, such as that outlined in Annex D.

NOTE 2 Calculation results for N80 are repeated for each size and linear mass; the first set represents pipe manufactured using the nonquenched and tempered process; the second set represents pipe manufactured using the quenched and tempered process; for a particular grade, the axial tension properties are not affected by the heat treat process.

NOTE 3 Calculation results for P110 are repeated for each size and linear mass; the first set represents product inspected to a 12.5 % calibration notch; the second set represents product inspected to a 5 % calibration notch; for a particular grade, the axial tension properties are not affected by the different inspection calibration standards.

NOTE 4 The designation L80\* includes grades L80 Type 1 and L80 13Cr.

## Annex L (informative)

### Tables of Calculated Performance Properties in USC Units

#### L.1 Introduction

All listed values for performance properties in this annex assume a benign environment and material properties conforming to **API 5CT**. Other environments may require additional analyses, such as that outlined in Annex D.

#### L.2 List of Tables Contained in Annex L and Accompanying Notes

##### Table L.1—Performance Property Calculations—External and Internal Pressure for Casing (USC Units)

NOTE 1 All performance properties values in this document assume a benign environment and material properties conforming to **API 5CT**; other environments may require additional analyses, such as that outlined in Annex D.

NOTE 2 Calculation results for N80 are repeated for each size and weight; the first set represents pipe manufactured using the nonquenched and tempered process; the second set represents pipe manufactured using the quenched and tempered process.

NOTE 3 Calculation results for P110 are repeated for each size and weight; the first set represents product inspected to a 12.5 % calibration notch; the second set represents product inspected to a 5 % calibration notch.

NOTE 4 The designation L80\* includes grades L80 Type 1 and L80 13Cr.

NOTE 5 The minimum internal yield pressure is the lowest of internal yield pressure of the pipe or the internal yield pressure of the coupling; the interface pressure at the E1 plane for round thread casing or at the E7 plane for buttress thread casing may be less than the minimum internal yield pressure for the connection (see Table L.2).

NOTE 6 The collapse resistance values are based on the historical **API 5C3 (6<sup>th</sup> Edition)** or **ISO 10400:1993** equations (see Section 8).

##### Table L.2—Performance Property—Interface Pressure of Casing Connections (USC Units)

NOTE 1 All performance properties values in this document assume a benign environment and material properties conforming to **API 5CT**; other environments may require additional analyses, such as that outlined in Annex D.

NOTE 2 The interface pressure at the E1 plane for round thread casing or at the E7 plane for buttress thread casing may be less than the minimum internal yield pressure for the connection (see Table L.1).

##### Table L.3—Performance Property Calculations—Axial Tension of Casing Pipe Body and Connections (USC Units)

NOTE 1 All performance properties values in this document assume a benign environment and material properties conforming to **API 5CT**; other environments may require additional analyses, such as that outlined in Annex D.

NOTE 2 Calculation results for N80 are repeated for each size and weight; the first set represents pipe manufactured using the nonquenched and tempered process; the second set represents pipe manufactured using the quenched and tempered process; for a particular grade, the axial tension properties are not affected by the heat treat process.

NOTE 3 Calculation results for P110 are repeated for each size and weight; the first set represents product inspected to a 12.5 % calibration notch; the second set represents product inspected to a 5 % calibration notch; for a particular grade, the axial tension properties are not affected by the different inspection calibration standards.

NOTE 4 The designation L80\* includes grades L80 Type 1 and L80 13Cr.

NOTE 5 Some joint strengths for the connections are greater than the corresponding pipe body yield strength.

NOTE 6 For J55 and K55 casing, the next higher grade coupling is L80; for N80 Q&T casing, the next higher grade coupling is P110; for P110 casing, the next higher grade coupling is Q125; no higher grade couplings have been established for the other grades.

#### **Table L.4—Performance Property Calculations—External and Internal Pressure for Tubing (USC Units)**

NOTE 1 All performance properties values in this document assume a benign environment and material properties conforming to **API 5CT**; other environments may require additional analyses, such as that outlined in Annex D.

NOTE 2 Calculation results for N80 are repeated for each size and weight; the first set represents pipe manufactured using the nonquenched and tempered process; the second set represents pipe manufactured using the quenched and tempered process.

NOTE 3 Calculation results for P110 are repeated for each size and weight; the first set represents product inspected to a 12.5 % calibration notch; the second set represents product inspected to a 5 % calibration notch.

NOTE 4 The designation L80\* includes grades L80 Type 1 and L80 13Cr.

NOTE 5 The minimum internal yield pressure is the lowest of internal yield pressure of the pipe or the internal yield pressure of the coupling.

NOTE 6 The collapse resistance values are based on the historical **API 5C3 (6<sup>th</sup> Edition)** or ISO 10400:1993 equations (see Section 8).

#### **Table L.5—Performance Property Calculations—Axial Tension for Tubing Pipe Body and Connections (USC Units)**

NOTE 1 All performance properties values in this document assume a benign environment and material properties conforming to **API 5CT**; other environments may require additional analyses, such as that outlined in Annex D.

NOTE 2 Calculation results for N80 are repeated for each size and weight; the first set represents pipe manufactured using the nonquenched and tempered process; the second set represents pipe manufactured using the quenched and tempered process; for a particular grade, the axial tension properties are not affected by the heat treat process.

NOTE 3 Calculation results for P110 are repeated for each size and weight; the first set represents product inspected to a 12.5 % calibration notch; the second set represents product inspected to a 5 % calibration notch; for a particular grade, the axial tension properties are not affected by the different inspection calibration standards.

NOTE 4 The designation L80\* includes grades L80 Type 1 and L80 13Cr.

## Annex M (informative)

### Temperature Effect on Performance Properties

#### M.1 Introduction

The purpose of this annex is to provide a way to derate material properties when considering elevated temperature. The temperature range considered for this section is from 24°C (75°F) to 232°C (450°F). This is applicable to API 5CT quenched and tempered products (except L80 9Cr and 13Cr) used in applications from 24°C (75°F) to 232°C (450°F). The equations in this annex are applicable to the API chemistry types within the sample set evaluated and do not distinguish between carbon-manganese, chrome-molybdenum, or other alloy types; the provided factors may not apply to proprietary material grades.

It should be noted that these derating factors were determined from an empirical dataset from 2011, 2013, and 2015, as the actual material derating factor may vary based on the steel chemistry and final material grade. Therefore, physical tests may be appropriate to determine a more precise derating factor for a specific material grade when using the equations below.

The technical basis for using derated material properties in external pressure calculations has not yet been established.

#### M.2 Yield Strength

##### M.2.1 Derating Factor

The yield strength derating formula for USC units has a factor of 0.00034; this factor is the calculated slope from an analyzed data set. The factor is converted into SI units by the function  $0.00034 \times (9/5) = 0.00061$ .

##### M.2.2 Limitations

The yield strength derating factor should be applied when considering effects of temperature used to derate between ambient and elevated temperatures. This should be used on any equation in which the yield strength is a direct input variable used for calculations.

##### M.2.3 Yield Strength Calculation

For quenched and tempered API 5CT products (except L80 Type 9Cr and L80 Type 13Cr) used in applications from an ambient temperature of 24°C (75°F) to an elevated temperature of 232°C (450°F), performance properties that are a function of yield strength should be calculated using the specified minimum yield strength at elevated temperature equations M.1 (SI units) or M.2 (US Customary units).

$$f_{\text{yet}} = f_{\text{ymn}}(1 - 6.1 \times 10^{-4}(T_E - 24 \text{ }^{\circ}\text{C})) \quad (\text{M.1})$$

$$f_{\text{yet}} = f_{\text{ymn}}(1 - 3.4 \times 10^{-4}(T_E - 75 \text{ }^{\circ}\text{F})) \quad (\text{M.2})$$

where

$f_{\text{yet}}$	yield strength at elevated temperature;
$f_{\text{ymn}}$	specified minimum yield strength;
$T_E$	application temperature (°C or °F).

## M.3 Young's Modulus

### M.3.1 General

The Young's modulus derating formula has a factor of 0.00016; this factor is the calculated slope from the Boiler Pressure Vessel Code (BPVC). A comparison was conducted between the analyzed data set vs. the BPVC data. The results trended the same, which indicated that as more data is collected from the API material grades it would become more aligned with the BPVC. The factor of 0.00016 was obtained according to significant digit convention and converted into SI units by the function  $0.00016 \times (9/5) = 0.00030$ .

#### Young's Modulus Calculation

For quenched and tempered API 5CT products (except L80 Type 9Cr and L80 Type 13Cr) used in applications from an ambient temperature of 24°C (75°F) to an elevated temperature of 232°C (450°F), performance properties that are a function of Young's modulus should be calculated using the Young's modulus at elevated temperature in equations M.3 (SI units) or M.4 (US Customary units).

$$E_{ET} = E_{Ambient} (1 - 3.0 \times 10^{-4} (T_E - 24^\circ\text{C})) \quad (\text{M.3})$$

$$E_{ET} = E_{Ambient} (1 - 1.6 \times 10^{-4} (T_E - 75^\circ\text{F})) \quad (\text{M.4})$$

where

$E_{ET}$	Young's modulus at elevated temperature;
$E_{Ambient}$	Young's modulus at ambient temperature;
$T_E$	application temperature (°C or °F).



## Bibliography

- [1] ASTM E1152-95, *Test Method for Determining J-R Curves*
- [2] API Bulletin 5C3, *Bulletin on Formulas and Calculations for Casing, Tubing, Drill Pipe, and Line Pipe Properties*, October 1994, 6th Edition
- [3] ASTM E1928-99, *Standard Practice for Estimating the Approximate Residual Circumferential Stress in Straight Thin-walled Tubing*
- [4] Burk, J.D., *Fracture Resistance of Casing Steels for Deep Gas Wells*, J. Metals, January 1985, pp. 65–70
- [5] Anderson, T.L., *Fracture Mechanics, Fundamentals and Applications*, 2nd Edition, CRC Press, Boca Raton, FL, 1995
- [6] Boresi, A.P., Sidebottom, O.M., *Advanced Mechanics of Materials*, John Wiley and Sons, 1985
- [7] Bulloch, J.H., *Stress Corrosion Cracking of Low Alloy Steels in Natural Seawater Environment—The Influence of Carbon Level*, Theoretical and Applied Fracture Mechanics, 16 (1991), pp. 1–17
- [8] NACE Paper No. 06130, *Results of Combined Axial Tension/Compression and Internal Pressure Testing of Mini-Pipe Specimens in H<sub>2</sub>S Environment: Discoveries of Material Response to Triaxial Stress with Non-Uniform Exposure to H<sub>2</sub>S*, 2006
- [9] SPE 97577, *Combined Axial Tension/Compression and Internal Pressure Testing of Mini-Pipe Specimens in H<sub>2</sub>S Environment to Determine Three Dimensional (Triaxial) Stress States which Produce Crack Initiation Failure: Explanation of the New Test Fixture, Mini-Pipe Specimen, and Preliminary Test Results*, 2005
- [10] NACE Paper No. 232, *Evaluation of the Resistance of Some Highly Alloyed Stainless Steels to Stress Corrosion Cracking in Hot Chloride Solutions under High Pressures of CO<sub>2</sub> and H<sub>2</sub>S*, 1985
- [11] NACE Paper No. 179, *Variations in the Microstructure of Seamless Pipe and their Relationship to Sulfide Stress Cracking*, 1979
- [12] NACE Paper No. 53, *Variations in Environmental Cracking Resistance of Thick-Walled Low Alloy Steel Tubulars*, 1988
- [13] Fung, Y.C., *Foundations of Solid Mechanics*, Prentice-Hall, 1965
- [14] Galambos, T.V. and Ravindra, M.K., *Properties of Steel for Use in LRFD*, J. Structural Division, ASCE, September 1978
- [15] Grobner, P.J., Sponseller, D.L. and Diesburg, D.E., *Effect of Molybdenum Content on the Sulfide Stress Cracking Resistance of AISI-Type 4130 Steel with 0.035 % Cb*, Corrosion, June 1979, pp. 240–249
- [16] Higdon, A., Ohlsen, E.H., Stiles, W.B., Weese, J. A. and Riley, W.F., Higdon, *Mechanics of Materials*, 3rd Edition, John Wiley and Sons, 1976
- [17] Hutchinson, J.W., *Fundamentals of the Phenomenological Theory of Nonlinear Fracture Mechanics*, J. Appl. Mech., 50, December 1983, pp. 1042–1051

- [18] Kamat, S.V., Malakondaiah, G., Srinivas, M., Marthanda Murthy, J. and Rama Rao, P., *Effect of Alloying Additions on KISSC on Ultrahigh Strength NiSiCr Steel*, Bull. Material Science, 17 (6), November 1994, pp. 633–641
- [19] Kane, R.D. and Greer, J.B., *Sulfide Stress Cracking of High-Strength Steels in Laboratory and Oilfield Environments*, J. Pet. Tech., November 1977, pp. 1483–1488
- [20] SPE 102585, *Formulas for Rupture, Necking, and Wrinkling of OCTG Under Combined Loads*, September 2006
- [21] SPE 48329, *Analytical Burst Strength Prediction of OCTG With and Without Defects*, 1998
- [22] EPRI Report No. NP-1931, *An Engineering Approach to Elastic-Plastic Fracture Mechanics*, Electric Power Research Institute, 1981
- [23] MacLennan, I. and Hancock, J.W., *Constraint-based Failure Assessment Diagrams*, Int. J. Press. Ves. & Piping, 64, 1995, pp. 287–298
- [24] Meguid, S.A., *Engineering Fracture Mechanics*, Elsevier Science Publishers, 1989
- [25] SPE 48327, *Burst Pressure Prediction of Thin-Walled, Ductile Tubulars Subjected to Axial Load*, 1998
- [26] Popov, E.P., *Introduction to Mechanics of Solids*, Prentice Hall, 1968
- [27] Sokolnikoff, I.S., *Mathematical Theory of Elasticity*, McGraw Hill, 1956
- [28] SPE 48330, *Accounting for Flaws in the Burst Strength of OCTG*, 1998
- [29] NACE Paper No. 1074, *Understanding the Size Effect in NACE TM0177 Method D (DCB) Testing and Implications for Users*, 2001
- [30] Tada, H., Paris, P.C. and Irwin, G.R., *The Stress Analysis of Cracks Handbook*, 2nd Edition, Paris Productions, St. Louis, 1985
- [31] SPE 48320, *The Development of a Risk-Based Burst Design Framework for Well Casing and Tubing*, 1998
- [32] Timoshenko, S.P. and Goodier, J.N., *Theory of Elasticity*, McGraw Hill, 3rd Edition, 1970
- [33] Watkins, M. and Vaughn, G.A., *Effects of H<sub>2</sub>S Partial Pressure on the Sulfide Stress Cracking Resistance of Steel*, Materials Performance, January 1986, pp. 44–48
- [34] NACE Paper No. 50, *Microstructure—The Critical Variable Controlling the SSC Resistance of Low Alloy Steels*, March 1995
- [35] SPE 29458, *Collapse and Post Collapse Behaviour of Tubulars: A Simple Approach*, April 1995
- [36] Abramowitz, M. and Stegun, I.A., *Handbook of Mathematical Functions*, Dover, 1972
- [37] SPE 48331, *On the Development of Reliability-Based Design Rules for Casing Collapse*, May 1998
- [38] SPE 85112, *On the Calibration of Design Collapse Strengths for Quenched and Tempered Pipe*, September 2003
- [39] Ang, A.H.-S. and Tang, W.H., *Probability Concepts in Engineering Planning and Design, Vol. II: Decision, Risk and Reliability*, John Wiley and Sons, 1984

- [40] Calladine, C.R., *Theory of Shell Structures*, Cambridge University Press, 1983
- [41] Clinedinst, W.O., *A Rational Expression for the Critical Collapsing Pressure of Pipe under External Pressure*, Drilling and Production Practice, presented at API Chicago meeting, 1939, Pub. API, 1940
- [42] Clinedinst, W.O., *Collapse Resistance of Pipe*, PhD dissertation, Century University, 1985
- [43] DEAE-64 project report, CTR 03: Data Analysis, rev. 02, February 1997
- [44] De Winter, P.E. et al., *Collapse Behaviour of Submarine Pipelines*, In: *Shell Structures: Stability and Strength*, Elsevier, 1985
- [45] Ditlevsen, O. and Madsen, H.O., *Structural Reliability Methods*, John Wiley and Sons, 1996
- [46] Eisenhart, C. et al., *Techniques of Statistical Analysis*, McGraw-Hill, 1947
- [47] Haagsma, S.C. and Schaap, D., *Collapse Resistance of Submarine Pipelines Studied*, Oil and Gas Journal, February 1981
- [48] Hohenbichler, M. and Rackwitz, R., *Non-normal Dependent Vectors in Structural Safety*, J. Engineering Mechanics Division, ASCE, 107, 1981
- [49] Holmquist, J.L. and Nadai, A., *A Theoretical and Experimental Approach to the Problem of Collapse of Deep-Well Casing*, Drilling and Production Practice, 1939, Pub. API, 1940
- [50] Huang, N.C. and Pattillo, P.D., *The Effect of Length: Diameter Ratio on Collapse of Casing*, Trans. ASME, J. Pressure Vessel Technology, 106, May 1984
- [51] SPE 26317, *An Improved Design Equation for Tubular Collapse*, October 1993
- [52] Jianzeng, H. and Taihe, S., *Equations Calculate Collapse Pressures for Casing Strings*, Oil and Gas Journal, January 2001
- [53] Johnson, N.L. and Welch, B.L., *Applications of the Non-central t-distribution*, Biometrika, 31, 1939
- [54] SPE 48332, *A Reliability Approach to the Design of OCTG Tubulars against Collapse*, May 1998
- [55] Kendall, M.G. and Stuart, A., *The Advanced Theory of Statistics, Vol. 1: Distribution Theory*, Charles Griffin, 1958
- [56] Kendrick, S., *The Technical Basis of the External Pressure Section of BS 5500*, Trans. ASME, J. Pressure Vessel Technology, 106, May 1984
- [57] SPE 90904, *A New OCTG Strength Equation for Collapse under Combined Loads*, September 2004
- [58] SPE 28327, *A New Equation for Elasto-Plastic Collapse Strength of Thick-Walled Casing*, September 1994
- [59] Lenth, R.V., *Cumulative Distribution Function of the Non-central t-distribution*, Statistical Algorithms, 38 (1), 1989
- [60] Lieberman, G.J., *Tables for One-sided Statistical Tolerance Limits*, Industrial Quality Control, 14 (10), April 1958
- [61] Liu, P.-L. and der Kiureghian, A., *Multivariate Distribution Models with Prescribed Marginals and Covariances*, Prob. Engineering Mechanics, 1 (2), 1986

- [62] Mann, N.R. et al., *Methods for Statistical Analysis of Reliability and Life Data*, John Wiley and Sons, 1974
- [63] Mehdizadeh, P., *Casing Collapse Performance*, Trans. ASME, J. Engineering for Industry, 98, August 1976
- [64] Melchers, R.E., *Structural Reliability Analysis and Prediction*, 2nd Edition, John Wiley and Sons, 1999
- [65] Murphey, C.E. and Langner, C.G., *Ultimate Pipe Strength under Bending, Collapse and Fatigue*, Proc. 4th International Symposium on OMAE, February 1985
- [66] Nataf, A., *Détermination des Distributions dont les Marges sont Données*, Comptes Rendus de l'Académie des Sciences, 225, 1962
- [67] Nippon Steel Corporation, *Experimental Study on the Effect of Axial Loading on Collapse Resistance of Casing*, January 1982
- [68] SCR-13, Owen, D.B., *Tables of Factors for One-Sided Tolerance Limits for a Normal Distribution*, Sandia Corporation, April 1958
- [69] Panarelli, J.E. and Hodge, P.G., *Interaction of Pressure, End Load, and Twisting Moment for a Rigid-Plastic Circular Tube*, Trans. ASME, J. Appl. Mechanics, September 1963
- [70] Pattillo, P.D. and Huang, N.C., *Collapse of Oil Well Casing with Ovality*, Trans. ASME, J. Energy Resources Technol., 107, March 1985
- [71] Rackwitz, R., *Practical Probabilistic Approach to Design*, Bulletin 112, Comité Européen du Béton, 1976
- [72] Rinne, H., *Taschenbuch der Statistik*, Verlag Harri Deutsch, 1997
- [73] Sachs, G. and Espey, G., *A New Method for Determination of Stress Distribution in Thin Walled Tubing*, American Inst. Mining and Met. Engrs. Metals Technology Technical Publication 1384, October 1941; also in Hetenyi, M. (Ed.), *Handbook of Experimental Stress Analysis*, John Wiley and Sons, 1957, p. 466 et seq.
- [74] ASME Paper No. 77-PVP-57, *Plastic Collapse of Oval Straight Tubes under External Pressure*, 1977
- [75] Bulletin No. 329, *A Study of the Collapsing Pressure of Thin-Walled Cylinders*, University of Illinois Engineering Experiment Station, 1941
- [76] Sumitomo Metal Industries, *On the Collapse Resistance of Commercial Casings*, November 1981
- [77] Sumitomo Metal Industries, *Experimental Study on the Effect of Axial Loading on Collapse Resistance of Casing*, December 1981
- [78] Tamano, T. et al., *A New Empirical Equation for Collapse Resistance of Commercial Casing*, J. Energy Resources Technol., ASME, 1983
- [79] Thoft-Christensen, P. and Baker, M.J., *Structural Reliability Theory and Its Applications*, Springer-Verlag, 1982
- [80] Timoshenko, S.P. and Gere, J.M., *Theory of Elastic Stability*, McGraw-Hill International, 2nd Edition, 1963

- [81] Tokimasa, K. and Tanaka, K., FEM Analysis of the Collapse Strength of a Tube, Trans. ASME, J. Pressure Vessel Technol., May 1986
- [82] Verner, E.A. et al., Collapse of Thick Wall Pipe in Ultra Deep Water, Proc. Conf. on Pipelines in Adverse Environments, ASCE, November 1983
- [83] Wolfram, S., Mathematica Technical Manual, online resource at mathworld.wolfram.com, 2005
- [84] Yeh, M.K. and Kyriakides, S., On the Collapse of Inelastic Thick-Walled Tubes under External Pressure, Proc. 4th International Symposium on OMAE, February 1985
- [85] Hald, A., Statistical Theory with Engineering Applications, John Wiley and Sons, 1952
- [86] George, W., Snedecor, Statistical Methods, Iowa State College Press, 1956
- [87] ASME Paper No. 64-PET-1, Strength of Threaded Joints for Steel Pipe, , October 1964
- [88] Crampton, D.K., Internal stress and season cracking in brass tubes, Trans. AIME, 89, 1930, pp. 233-255
- [89] Loxley, E.M., The measurement and detection of residual stresses in cold drawn tubes, PhD thesis, Sheffield University, May 1955
- [90] Barlow, P., *On the force excited by hydraulic pressure in a Bramah press; the resisting power of the cylinder, and rules for computing the thickness of metal for presses of various powers and dimensions*, Trans. Institution of Civil Engineers, 1, 1836, pp. 133-139
- [91] ~~Tresca, H., Memoire sur l'écoulement des solides à de forte pressions, Comptes Rendus Acad. Sci. Paris, 1, 1864~~
- [92] ~~Mises, R.V., Mechanik der festen körper im plastisch-deformablen zustand, Nachrichten von der Gesellschaft der Wissenschaften zu Göttingen, Mathematisch-Physikalische Klasse, 1913, pp. 582-592~~
- [93][91] Chakrabarti, S.K. and Frampton, R.E., Review of riser analysis techniques, Applied Ocean Research, 4 (2), 1982, pp. 73-90
- [94][92] Goodman, J., Mechanics applied to engineering, 8th ed., Longmans, Green and Co., 1914
- [95][93] API Recommended Practice 1111, Design, Construction, Operation, and Maintenance of Offshore Hydrocarbon Pipelines (*Limit State Design*)
- [96][94] DNVGL-ST-F101, *Submarine Pipeline Systems*
- [97][95] API Recommended Practice 579, *Fitness-for-Service*, **First Edition**
- [98][96] API Recommended Practice 5C1, *Care and Use of Casing and Tubing*
- [99][97] API Recommended Practice 5C5, *Procedures for Testing Casing and Tubing Connections*
- [100][98] \_\_\_\_\_ API Specification 5B, *Threading, Gauging, and Inspection of Casing, Tubing, and Line Pipe Threads*
- [101][99] \_\_\_\_\_ API Specification 5CT, *Casing and Tubing*
- [102][100] \_\_\_\_\_ API Specification 5DP, *Drill Pipe*

~~[103]~~~~[101]~~ \_\_\_\_\_ API Specification 5L:2004, *Specification for Line Pipe*

~~[104]~~~~[102]~~ \_\_\_\_\_ BS 7910, Guide to methods for assessing the acceptability of flaws in metallic structures

~~[105]~~~~[103]~~ \_\_\_\_\_ ISO 3183:2007, Petroleum and natural gas industries — Steel pipe for pipeline transportation systems

~~[106]~~~~[104]~~ \_\_\_\_\_ ISO 10400:1993, Petroleum and natural gas industries — Formulae and calculation for casing, tubing, drill pipe and line pipe properties

~~[107]~~~~[105]~~ \_\_\_\_\_ ISO 10405, Petroleum and natural gas industries — Care and use of casing and tubing

~~[108]~~~~[106]~~ \_\_\_\_\_ ISO 11960:2004, Petroleum and natural gas industries — Steel pipes for use as casing or tubing for wells

~~[109]~~~~[107]~~ \_\_\_\_\_ ISO 11961, Petroleum and natural gas industries — *Steel drill pipe*

~~[110]~~~~[108]~~ \_\_\_\_\_ ISO 13679, *Petroleum and natural gas industries — Procedures for testing casing and tubing connections*

~~[111]~~~~[109]~~ \_\_\_\_\_ ~~[AAA] Long, X. and Moore, P., The Study on Residual Stresses in OCTG and Its Effect on Pipe Sour Cracking (SSC) Performances, NACE 2018, Paper 10919, Phoenix, USA.~~