PUBLISHED DOCUMENT

PD CEN ISO/TR 10400:2011

Petroleum and natural gas industries — Equations and calculations for the properties of casing, tubing, drill pipe and line pipe used as casing or tubing

ICS 75.180.10

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National foreword

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The UK participation in its preparation was entrusted by Technical Committee PSE/17, Materials and equipment for petroleum, petrochemical and natural gas industries, to Panel PSE/17/-/5, Casing, tubing and drill pipe.

A list of organizations represented on this committee can be obtained on request to its secretary.

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Petroleum and natural gas industries - Equations and calculations for the properties of casing, tubing, drill pipe and line pipe used as casing or tubing (ISO/TR 10400:2007)

Industries du pétrole et du gaz naturel - Équations et calculs relatifs aux propriétés des tubes de cuvelage, des tubes de production, des tiges de forage et des tubes de conduites utilisés comme tubes de cuvelage et tubes de production (ISO/TR 10400:2007)

Erdöl- und Erdgasindustrie - Formeln und Berechnungen der Eigenschaften von Futterrohren, Steigrohren, Bohrgestängen und Leitungsrohren (ISO/TR 10400:2007)

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Foreword

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Foreword

ISO (the International Organization for Standardization) is a worldwide federation of national standards bodies (ISO member bodies). The work of preparing International Standards is normally carried out through ISO technical committees. Each member body interested in a subject for which a technical committee has been established has the right to be represented on that committee. International organizations, governmental and non-governmental, in liaison with ISO, also take part in the work. ISO collaborates closely with the International Electrotechnical Commission (IEC) on all matters of electrotechnical standardization.

International Standards are drafted in accordance with the rules given in the ISO/IEC Directives, Part 2.

The main task of technical committees is to prepare International Standards. Draft International Standards adopted by the technical committees are circulated to the member bodies for voting. Publication as an International Standard requires approval by at least 75 % of the member bodies casting a vote.

In exceptional circumstances, when a technical committee has collected data of a different kind from that which is normally published as an International Standard ("state of the art", for example), it may decide by a simple majority vote of its participating members to publish a Technical Report. A Technical Report is entirely informative in nature and does not have to be reviewed until the data it provides are considered to be no longer valid or useful.

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ISO/TR 10400 was prepared by Technical Committee ISO/TC 67, *Materials, equipment and offshore structures for petroleum, petrochemical and natural gas industries*, Subcommittee SC 5, *Casing, tubing and drill pipe*.

This first edition of ISO/TR 10400 cancels and replaces ISO 10400:1993, which has been technically revised.

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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. Either or both the load and resistance may be modified by a design factor.

Both deterministic and probabilistic (synthesis method) approaches to performance properties are addressed in this Technical Report. The deterministic approach uses specific geometric and material property values to calculate a single performance property value. The synthesis 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 Technical Report. Rather, this text serves to identify useful equations for obtaining the resistance of a tubular to specified loads, independent of their origin. This Technical Report provides limit state equations (see annexes) which are useful for determining the resistance of an individual sample whose geometry and material properties are given, and design equations which 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 reader.

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Petroleum and natural gas industries — Equations and calculations for the properties of casing, tubing, drill pipe and line pipe used as casing or tubing

1 Scope

This Technical Report illustrates the equations and templates necessary to calculate the various pipe properties given in International Standards, including

— 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, and

— 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 ISO 11960 or API 5CT, ISO 11961 or API 5D, and ISO 3183 or 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 Technical Report (e.g. cold rotary straightened pipe). Pipe modified by cold working after production, such as expandable tubulars and coiled tubing, is beyond the scope of this Technical Report.

Application of performance property equations in this Technical Report 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, etc., with the closest appropriate casing/tubing product. Similar caution should be exercised when using the performance equations for drill pipe.

This Technical Report and the equations contained herein relate the input pipe manufacturing parameters in ISO 11960 or API 5CT, ISO 11961 or API 5D, and ISO 3183 or API 5L to expected pipe performance. The design equations in this Technical Report are not to be understood as a manufacturing warrantee. Manufacturers are typically licensed to produce tubular products in accordance with manufacturing specifications which 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 Technical Report is not a design code. It only provides equations and templates for calculating the properties of tubulars intended for use in downhole applications. This Technical Report 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.

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All equations and listed values for performance properties in this Technical Report assume a benign environment and material properties conforming to ISO 11960 or API 5CT, ISO 11961 or API 5D and ISO 3183 or 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 Technical Report.

Throughout this Technical Report tensile stresses are positive.

2 Conformance

2.1 Normative references

In the interests of worldwide application of this Technical Report, ISO/TC 67 has decided, after detailed technical analysis, that certain of the normative documents listed in Clause 3 and prepared by ISO/TC 67 or other ISO Technical Committees are interchangeable in the context of the relevant requirement with the relevant document prepared by the American Petroleum Institute (API), the American Society for Testing and Materials (ASTM) or the American National Standards Institute (ANSI). These latter documents are cited in the running text following the ISO reference and preceded by or, for example, “ISO XXXX or API YYYY”. Application of an alternative normative document cited in this manner will lead to technical results different from the use of the preceding ISO reference. However, both results are acceptable and these documents are thus considered interchangeable in practice.

2.2 Units of measurement

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

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

3 Normative references

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

ISO 3183:2007, *Petroleum and natural gas industries — Steel pipe for pipeline transportation systems*

ISO 10405, *Petroleum and natural gas industries — Care and use of casing and tubing*

ISO 11960:2004, *Petroleum and natural gas industries — Steel pipes for use as casing or tubing for wells*

ISO 11961, *Petroleum and natural gas industries — Steel drill pipe*

ISO 13679, *Petroleum and natural gas industries — Procedures for testing casing and tubing connections*

ANSI-NACE International Standard TM0177, *Laboratory Testing of Metals for Resistance to Sulfide Stress Cracking and Stress Corrosion Cracking in H_2S Environments*

API 5B, *Threading, Gauging and Thread Inspection of Casing, Tubing, and Line Pipe Threads (US Customary Units)*

API RP 579, *Recommended Practice for Fitness-for-Service*, January 2000

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API RP 5C1, *Recommended Practice for Care and Use of Casing and Tubing*

API RP 5C5, *Recommended Practice on Procedures for Testing Casing and Tubing Connections*

API 5CT, *Specification for Casing and Tubing*

API 5D, *Specification for Drill Pipe*

API 5L:2004, *Specification for Line Pipe*

BS 7910, *Guide to methods for assessing the acceptability of flaws in metallic structures*

4 Terms and definitions

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

4.1
Cauchy stress
true stress
force applied to the surface of a body divided by the current area of that surface

4.2
coefficient of variance
dimensionless measure of the dispersion of a random variable, calculated by dividing the standard deviation by the mean

4.3
design equation
equation which, 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.

4.4
deterministic
approach which assumes 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.

4.5
ductile rupture
failure of a tube due to internal pressure and/or axial tension in the plastic deformation range

4.6
e
Euler's constant
2,718 281 828

4.7
effective stress
combination of pressure and axial stress used in this Technical Report to simplify equations

NOTE Effective stress as used in this Technical Report does not introduce a distinct, physically defined stress quantity. Effective stress is a dependent quantity, which is determined as a combination of axial stress, internal pressure, external pressure and pipe dimensions, and provides a convenient grouping of these terms in some equations. The effective stress is sometimes called the Lubinski fictitious stress.

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4.8
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

4.9
engineering stress
force applied to the surface of a body divided by the original area of that surface

4.10
fracture pressure
internal pressure at which a tube fails due to propagation of an imperfection

4.11
inspection threshold
maximum size of a crack-like imperfection which is defined to be acceptable by the inspection system

4.12
J-integral
measure of the intensity of the stress-strain field near the tip of a crack

4.13
label 1
dimensionless designation for the size or specified outside diameter that may be used when ordering pipe

4.14
label 2
dimensionless designation for the mass per unit length or wall thickness that may be used when ordering pipe

4.15
limit state equation
equation which, 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.

4.16
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.

4.17
mass
label used to represent wall thickness of tube cross section for a given pipe size

4.18
pipe body yield
stress state necessary to initiate yield at any location in the pipe body

4.19
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.

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4.20
probabilistic method
approach which uses distributions of geometric and material property values to calculate a distribution of performance property values

4.21
synthesis method
probability approach which addresses the uncertainty and likely values of pipe performance properties by using distributions of geometric and material property values

NOTE These distributions are combined with a limit state equation to determine the statistical distribution of a performance property. The performance distribution in combination with a defined lower percentile determines the final design equation.

4.22
template
procedural guide consisting of equations, test methods and measurements for establishing design performance properties

4.23
TPI
threads per inch

NOTE 1 thread per inch = 0,039 4 threads per millimetre; 1 thread per millimetre = 25,4 threads per inch.

4.24
true stress-strain curve
plot of Cauchy stress (ordinate) vs. logarithmic strain (abscissa)

4.25
yield
permanent, inelastic deformation

4.26
yield stress bias
ratio of actual yield stress to specified minimum yield stress

5 Symbols

A hand-tight standoff

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 ISO 3183 or API 5L

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

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

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

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A_x maximum diameter at the extreme-line pin seal tangent point

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. 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)

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

B specified inside diameter of the extreme-line connection, in accordance with API 5B

B_c empirical constant in historical API collapse equation

B_f maximum bearing face diameter, special bevel, in accordance with ISO 11960 or 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 centreline 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_i inside diameter of extreme-line box upset, in accordance with API 5B

D_{max} maximum pipe outside diameter

D_{min} minimum pipe outside diameter

D_p extreme-line pin critical section outside diameter; $D_p = H_x + \delta - \varphi$

D_4 major diameter, in accordance with API 5B

d pipe inside diameter, $d = D - 2t$

d_b inside diameter of the critical section of the extreme-line box; $d_b = I_x + 2h_x - \Delta + \theta$

d_{iu} inside diameter of pin upset, in accordance with ISO 11960 or API 5CT

d_j extreme-line specified joint inside diameter, made up

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

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E	Young's modulus
E_c	pitch diameter, at centre 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	axial force
F_{eff}	effective axial force
F_c	empirical constant in historical API collapse equation
F_{YAPI}	axial force at yield, historical API equation
f	degrees of freedom $= N_t - 1$
$f(\bar{x})$	joint probability density function of the variables in $\bar{x}$
f_{rn}	root truncation of the pipe thread of API line pipe threads, as follows: 0,030 mm (0.001 2 in) for 27 TPI, 0,046 mm (0.001 8 in) for 18 TPI, 0,061 mm (0.002 4 in) for 14 TPI, 0,074 mm (0.002 9 in) for 11-1/2 TPI, 0,104 mm (0.004 1 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_{yax}	equivalent yield strength in the presence of axial stress
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

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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 is the thread height of a round-thread equivalent Vee thread, as follows:
0,815 mm (0.032 1 in) for 27 TPI,
1,222 mm (0.048 1 in) for 18 TPI,
1,755 mm (0.069 1 in) for 14 TPI,
1,913 mm (0.075 3 in) for 11-1/2 TPI,
2,199 6 mm (0.086 60 in) for 10 TPI,
2,749 6 mm (0.108 25 in) for 8 TPI

Ht_{des} decrement factor, as given in Table F.9

Ht_{ult} a decrement factor

H_x maximum extreme-line root diameter at last perfect pin thread

h_B buttress thread height: 1,575 for SI units, 0.062 for USC units

h_n stress-strain curve shape factor

h_s round thread height

h_x minimum box thread height for extreme-line casing, as follows:
1,52 mm (0.060 in) for 6 TPI
2,03 mm (0.080 in) for 5 TPI

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: 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

I_x minimum extreme-line crest diameter of box thread at Plane H

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

J_{Ic} fracture resistance of the material

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J_{Imat} 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_{Imat} 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 ISO 13679 or API RP 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. The value of k_a can be established for a specific pipe material based on testing

k_B variable intermediate term in ISO 13679 or API RP 5C5 representation of von Mises yield criterion

k_C variable intermediate term in ISO 13679 or API RP 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 and 625 000 for USC units

$k_{e\ uls}$ calibration factor for ultimate elastic collapse, 1,089

k_i factor used to determine minimum wall thickness for transverse impact specimens:
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 and 1/12 for USC units

k_m mass correction factor, 1,000 for carbon steel, 0,989 for martensitic chromium steel

k_n stress conversion factor, equal to $1{,}18 \times 10^{-4}$ MPa^{-1} for SI units and 8.12×10^{-7} psi^{-1} for USC units

k_{pi} upper quadrant geometry factor in ISO 13679 or API RP 5C5 representation of von Mises yield criterion

k_{po} lower quadrant geometry factor in ISO 13679 or API RP 5C5 representation of von Mises yield criterion

k_{wall} factor to account for the specified manufacturing tolerance of the pipe wall. For example, for a tolerance of –12,5 %, $k_{wall} = 0{,}875$

k_{wpe} mass per unit length conversion factor, equal to 0,024 661 5 for SI units and 10.69 for USC units

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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, 0,991 1
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 ISO 11960 or API 5CT
L_{iu}	length of pin upset, in accordance with ISO 11960 or 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	specified outside diameter of the extreme-line connection; 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 ISO 11960 or 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 ISO 11960 or API 5CT
m_{prt}	pin mass removed by threading
m_{xbu}	extreme-line pin upset mass
m_{xpu}	extreme-line pin upset mass
m_{xrt}	extreme-line mass removed by threading and recessing
m_{rt}	mass removed by threading

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mu	model uncertainty
N	number of thread turns make-up
N_L	coupling length, in accordance with ISO 11960 or 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
O_x	minimum diameter at the extreme-line box seal tangent point
ov	ovality
P_j	joint strength
p	thread pitch 3,175 mm (0.125 in) for round thread casing 5,080 mm (0.200 in) for buttress thread casing
p_c	collapse pressure
p_{ci}	collapse pressure in the presence of internal pressure
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_{iL}	internal pressure at leak
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

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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

$p_{y\ Tresca}$ Tresca yield pressure for collapse

$p_{y\ ult}$ ultimate yield collapse term

$p_{y\ vme}$ von Mises yield pressure for collapse

Q diameter of coupling recess, in accordance with API 5B

r radial coordinate, $(d/2) \leqslant r \leqslant (D/2)$

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, 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

$t_{c\ ave}$ actual average pipe wall thickness

$t_{c\ max}$ maximum pipe wall thickness

$t_{c\ min}$ minimum pipe wall thickness

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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 ISO 11960 or 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

$\vec{x}$ vector of random variables

Z_p correction factor for variation in t/D from average

Δ taper drop in extreme-line pin perfect thread length
6,43 mm (0.253 in) for 6 TPI
5,79 mm (0.228 in) for 5 TPI

α sensitivity factor

β first-order reliability index

δ extreme-line taper rise between Plane H and Plane J, as follows:
0,89 mm (0.035 in) for 6 TPI
0,81 mm (0.032 in) for 5 TPI

φ one-half of the maximum extreme-line seal interference, $\varphi = (A_x - O_x)/2$

ε_{eng} engineering strain

ε_{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

ε_{ln} logarithmic strain

ε_{ymn} strain at which specified minimum yield strength is determined

μ mean

μ_{ec} mean calculated eccentricity as a percent, [eccentricity = 100 $(t_{c\,max} - t_{c\,min})/t_{c\,ave}$]

μ_{fy} mean calculated f_y as a percent

μ_{ov} mean calculated ovality as a percent, [ovality = 100 $(D_{max} - D_{min})/D_{ave}$]

μ_{rs} mean calculated residual stress (compression at ID face is negative)

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ν Poisson's ratio

π circumference of a circle divided by its diameter, assigned a value of 3,141 6

ϕ_f probability of failure

θ one-half of the extreme-line maximum thread interference, $\theta = (H_x - I_x)/2$

θ_p the proportion of the population not included

σ standard deviation

σ_a component of axial stress not due to bending

σ_b component of axial stress due to bending

σ_c true (Cauchy) stress

σ_e equivalent stress

σ_{eff} effective stress

σ_f fibre stress corresponding to the percent of specified yield strength as given in Table 12

σ_h circumferential or hoop stress

σ_{pmx} maximum principal stress

σ_r radial stress

σ_{res} residual stress

σ_{th} threshold stress

σ_{ye} equivalent yield strength in the presence of axial tension

σ_{ymne} equivalent minimum yield strength in the presence of axial tension

τ_{ha} torsional (shear) stress

6 Triaxial yield of pipe body

6.1 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

a) radial and circumferential stress as determined by the Lamé Equations for a thick cylinder,

b) uniform axial stress due to all sources except bending,

c) axial bending stress for a Timoshenko beam,

d) 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.

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6.2 Assumptions and limitations

6.2.1 General

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

6.2.2 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.

6.2.3 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.

6.2.4 No residual stress

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

6.2.5 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 Clause 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.

6.3 Data requirements

The following input data are required to complete the calculation for triaxial yield of the pipe body:

c tube curvature, the inverse of the radius of curvature to the centreline of the pipe;

D specified pipe outside diameter;

E Young's modulus;

F_a axial force;

f_{ymn} specified minimum yield strength;

k_{wall} factor to account for the specified manufacturing tolerance of the pipe wall. For example, for a tolerance of –12,5 %, $k_{wall} = 0{,}875$;

p_i internal pressure;

p_o external pressure;

T applied torque;

t specified pipe wall thickness.

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6.4 Design equation for triaxial yield of pipe body

The onset of yield is defined as

$$\sigma_e = f_{ymn} \tag{1}$$

where $\sigma_e < f_{ymn}$ corresponds to elastic behaviour, and

f_{ymn} is the specified minimum yield strength;

σ_e is the equivalent stress.

The equivalent stress is defined as

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

with

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

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

$$\sigma_a = F_a/A_p \tag{5}$$

$$\sigma_b = \pm M_b r/I = \pm Ecr \tag{6}$$

$$\tau_{ha} = Tr/J_p \tag{7}$$

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 centreline of the pipe;

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. For example, 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;

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r is the radial coordinate, $(d/2) \leqslant r \leqslant (D/2)$ for σ_a, σ_b and τ_{ha}, $(d_{wall}/2) \leqslant r \leqslant (D/2)$ for σ_r and σ_h;

T is the applied torque;

t is the specified pipe wall thickness;

σ_a is the component of axial stress not due to bending;

σ_b is the component of axial stress due to bending;

σ_e is the equivalent stress;

σ_h is the circumferential or hoop stress;

σ_r is the radial stress;

τ_{ha} is the torsional (shear) stress.

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 centre of tube curvature than the centreline of the pipe experience compressive bending stress. Points in the pipe cross section farther from the centre of tube curvature than the centreline of the pipe experience tensile bending stress.

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 °/30 m (°/100 ft). If, therefore, the required units for c are radians/metre, and c is expressed in °/30 m (°/100 ft), the right-hand side of Equation (6) should be multiplied by the constant $\pi/(180 \times 30) = 5{,}817\ 8 \times 10^{-4}$ rad-m/°–30 m [$\pi/(180 \times 100) = 1{,}645\ 3 \times 10^{-4}$ rad-ft/°–100 ft].

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 σ_e should be used in Equation (1).

The purpose of the design equation is to determine the stress state which 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 which comes about naturally as part of the pipe manufacturing process.

6.5 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 clause, minding the limitations given in 6.2.

6.6 Example calculations

6.6.1 Initial yield of pipe body, Lamé Equation for pipe when external pressure, bending and torsion are zero

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.

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6.6.1.1 Yield design equation, special case for capped-end conditions

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.)

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

$$p_{\mathrm{iYLc}} = f_{\mathrm{ymn}}/\{(3\,D^4 + d_{\mathrm{wall}}{}^4)/(D^2 - d_{\mathrm{wall}}{}^2)^2 + d^4/(D^2 - d^2)^2 - 2d^2 d_{\mathrm{wall}}{}^2/[(D^2 - d^2)\,(D^2 - d_{\mathrm{wall}}{}^2)]\}^{1/2} \qquad (8)$$

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 $k_{\mathrm{wall}}\,t$, $d_{\mathrm{wall}} = D - 2k_{\mathrm{wall}}\,t$;

f_{ymn} is the specified minimum yield strength;

k_{wall} is the factor to account for the specified manufacturing tolerance of the pipe wall. For example, for a tolerance of –12,5 %, $k_{\mathrm{wall}} = 0{,}875$;

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

t is the specified pipe wall thickness.

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).

6.6.1.2 Yield design equation, special case for internal pressure with zero axial load

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.

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

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

where

D is the specified pipe outside diameter;

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

f_{ymn} is the specified minimum yield strength;

k_{wall} is the factor to account for the specified manufacturing tolerance of the pipe wall. For example, for a tolerance of –12,5 %, $k_{\mathrm{wall}} = 0{,}875$;

p_{iYLo} is the internal pressure at yield for an open-end 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, Equations (1) and (2).

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6.6.2 Initial yield pressure of pipe body, historical API equation

6.6.2.1 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.

6.6.2.2 Historical, one-dimensional yield pressure design equation

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

$$p_{\mathrm{iYAPI}} = [2 f_{\mathrm{ymn}} (k_{\mathrm{wall}} t)/D] \tag{10}$$

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. For example, for a tolerance of –12,5 %, $k_{\mathrm{wall}} = 0{,}875$;

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 may be derived (by methods other than that used by Barlow) (see 6.2).

6.6.3 Pipe body yield strength, historical API equation

6.6.3.1 General

Pipe body yield strength is the axial load required 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:

$$F_{\mathrm{YAPI}} = f_{\mathrm{ymn}} A_{\mathrm{p}} \tag{11}$$

where

A_{p} is the area of the pipe cross section, $A_{\mathrm{p}} = \pi/4\ (D^2 - d^2)$;

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.

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6.6.4 Yield in the absence of bending and torsion

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 kg/dm^3 (9 lb/gal). The internal 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 mm (9.625 in), of wall thickness 13,84 mm (0.545 in) and have $k_{wall} = 0{,}875$. Assume the mass density of steel is 7,85 kg/dm^3 (0.283 3 lb/in^3). Compute the yield state of the uppermost cross section of the tube when the tube is at a depth of 3 000 m (9 842.5 ft). The tube has a minimum yield stress of 551,6 MPa (80 000 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	1 995 717	N	448 655	lb
p_i	31,73	MPa	4 601	psi
p_o	31,73	MPa	4 601	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^2	15.55	in^2
I	66 920 762	mm^4	160.8	in^4
J_p	133 841 524	mm^4	321.6	in^4
σ_a	199,01	MPa	28 859	psi
Inner radius				
σ_b	0	MPa	0	psi
σ_h	–31,73	MPa	–4 602	psi
σ_r	–31,73	MPa	–4 602	psi
τ_{ha}	0	MPa	0	psi
σ_e, σ_b+	230,74	MPa	33 461	psi
σ_e, σ_b–	230,74	MPa	33 461	psi
Outer radius				
σ_b	0	MPa	0	psi
σ_h	–31,73	MPa	–4 602	psi
σ_r	–31,73	MPa	–4 602	psi
τ_{ha}	0	MPa	0	psi
σ_e, σ_b+	230,74	MPa	33 461	psi
σ_e, σ_b–	230,74	MPa	33 461	psi

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Since both bending and torsion are zero in this example, there is no need to calculate σ_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.

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

7 Ductile rupture of the pipe body

7.1 General

The equations for ductile rupture pertain to the actual failure of the pipe body due to internal pressure. While the yield equations of Clause 6 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 which fails the pipe body with loss of internal pressure integrity.

The equations for ductile rupture depend on the minimum physical wall thickness, the pipe outer diameter, the maximum depth of imperfections which 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. The selection of the basic equation, the application of the basic equation to ISO/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 three interlinked concepts:

a) an equation for equilibrium-plasticity based rupture of a pipe with known physical wall thickness and diameter;

b) reduction in performance due to wall loss in proportion to depths of imperfections which may not be detected by the manufacturing and inspection system;

c) 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 in Annex B.

7.2 Assumptions and limitations

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.

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 which is buckled (nor to pipe in a dogleg).

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7.3 Data requirements

7.3.1 General

The following input data are required 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. 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 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 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} factor to account for the specified manufacturing tolerance of the pipe wall. For example, 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.

7.3.2 Determination of the hardening index

In the absence of stress-strain information, the following values of n are suggested.

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
C95	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 following correlation.

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$$n = 0{,}1693 - k_n f_y \quad (13)$$

where f_y has units of MPa (psi) and

f_y is the yield strength of a representative tensile specimen;

k_n is the stress conversion factor, equal to $1{,}18 \times 10^{-4}$ MPa^{-1} for SI units and 8.12×10^{-7} 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.

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 Oil Country Tubular Goods (OCTG). However, if a high-hardening material such as duplex steel is used, it is important to determine n to avoid a non-conservative rupture strength prediction. Values of n for these materials can be as high as 0,30.

7.3.3 Determination of the burst strength factor, k_a

7.3.3.1 General

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.

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.

7.3.3.2 Analytical method for determining k_a

The following procedure should be used.

a) 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 analysed.

b) Use the finite element model to simulate applying internal pressure to the pipe model in order to calculate the value of the J-integral of the pipe as a function of internal pressure (see B.7.2 for examples).

c) 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 [1] for the precisely defined test methodology.

d) 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}.

e) 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 $(1\text{-}k_a a/t)$ 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.

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f) Repeat the above calculation using the finite element model results for the crack with 12,5 % depth.

g) Average the two values of k_a from steps e) and f) and use the average in the rupture equation.

7.3.3.3 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 precracking. In general, the finite element approach combined with the J_{1c} measurement is easier to pursue.

7.3.3.4 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.

7.4 Design equation for capped-end ductile rupture

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. For example, 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.

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

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 which 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 (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.

7.5 Adjustment for the effect of axial tension and external pressure

7.5.1 General

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 tension or compression. 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 formulation of the von Mises or Tresca yield surface in terms of axial stress, radial stress and hoop stress.

Moreover, rupture is the prevailing failure mechanism only if the axial tension is not too large. For large axial tension and smaller internal over-pressure, 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 tension, 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.

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7.5.2 Design equation for ductile rupture under combined loads

In the presence of external pressure and axial tension or compression different from capped-end conditions, the general equation for ductile rupture is

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

with

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

where

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

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

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

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

$$p_{ref} = ½\,(p_{ref\,M} + p_{ref\,T}),\ \text{used in Fig. 1} \quad (21)$$

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

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

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

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. 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_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. 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. For example, 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;

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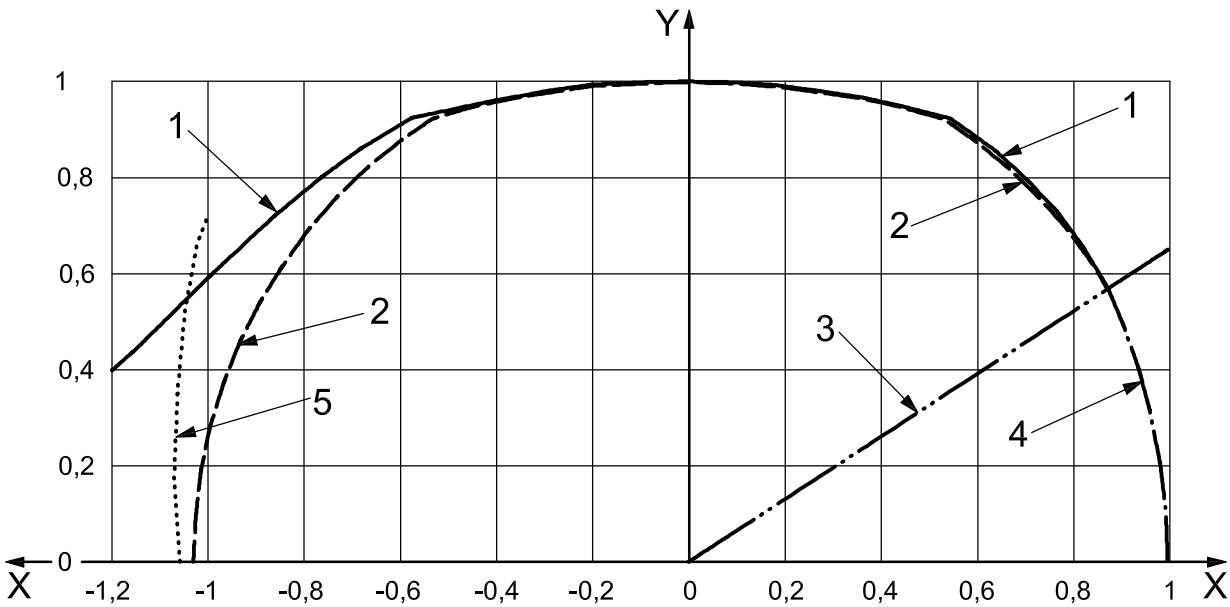
p_{iRa} is p_{iR} adjusted for axial load and external pressure;

p_o is the external pressure;

t is the specified pipe wall thickness.

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



Key

X effective axial tension (F_{eff}/F_{uts})

Y pressure differential $(p_i - p_o)/p_{ref}$

1 rupture (exact)

2 rupture [Equation (15)]

3 transition

4 necking [Equation (26)]

5 wrinkling

Figure 1 — Illustration of the effect of effective axial tension and external pressure on ductile rupture

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

The rupture equation is valid, i.e. rupture occurs before necking, when

$$F_{eff}/F_{uts} \leqslant (\sqrt{3}/2)^{1-n} \tag{25}$$

7.5.3 Design equation for ductile necking under combined loads

In the presence of internal and external pressure, the general equation for ductile necking is

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

where

$$F_a = \pi\, t\, (D - t)\, \sigma_a \tag{27}$$

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

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

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

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

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

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.

The necking equation is valid, i.e. necking occurs before rupture, when

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

7.5.4 Boundary between rupture and necking

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

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

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

7.5.5 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.

7.6 Example calculations

7.6.1 Ductile rupture of an end-capped pipe

For an end-capped pipe under pressure load, the effective axial load is zero and Equations (13) and (14) 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.

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.

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

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Table 3 — Example calculations — Ductile rupture of an end-capped pipe

Term	SI		USC	
	Value	Units	Value	Units
Load				
F_{eff}	0	N	0	lb
Geometry				
D	177,8	mm	7	in
t	11,51	mm	0.453	in
k_{wall}	0,875		0.875	
Material				
E	206 842	MPa	30 000 000	psi
f_{umn}	862	MPa	125 000	psi
Calculations				
n	0,08		0.08	
a_N	0,575	mm	0.022 7	in
k_a	1		1	
p_{iR}	99,7	MPa	14 460	psi

7.6.2 Ductile rupture for a given true axial load

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.

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.

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

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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	206 842	MPa	30 000 000	psi
f_{umn}	862	MPa	125 000	psi
Calculations				
n	0,08		0.08	
a_N	0,575 8	mm	0.022 7	in
F_{uts}	5 180 423	N	1 164 663	lb
p_{uts}	97,22	MPa	14 100	psi
$P_{ref\ T}$	91,98	MPa	13 340	psi
$P_{ref\ M}$	107,43	MPa	15 581	psi
k_R	0,939 01		0.939 01	
p_M	92,47	MPa	13 412	psi
p_{iRa}	92,22	MPa	13 376	psi

8 External pressure resistance

8.1 General

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.

The equations in Clause 8 are taken directly from Reference [2]. The collapse equations presented here were originally developed in USC units, and should only be used in these units.

8.2 Assumptions and limitations

The following limitation applies to the design equations for external pressure resistance:

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

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8.3 Data requirements

The following input data are required 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.

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

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 (35).

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

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.

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 Equation (36) as follows:

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

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.

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

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

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Table 5 — Yield collapse pressure equation range

Grade [a]	D/t range [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
C95, 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

[a] 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.

[b] The D/t range values were calculated from Equations (36), (44) or (49), (45) or (50), and (46) or (51).

8.4.3 Plastic collapse pressure equation

The minimum collapse pressure for the plastic range of collapse is calculated by Equation (37):

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

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.

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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 Equation (38):

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

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 [a]	A_c	B_c	C_c psi	D/t range [b]
H40	2.950	0.046 5	754	16.40 to 27.01
–50	2.976	0.051 5	1 056	15.24 to 25.63
J55, K55	2.991	0.054 1	1 206	14.81 to 25.01
–60	3.005	0.056 6	1 356	14.44 to 24.42
–70	3.037	0.061 7	1 656	13.85 to 23.38
C75, E75	3.054	0.064 2	1 806	13.60 to 22.91
L-N-80	3.071	0.066 7	1 955	13.38 to 22.47
C90	3.106	0.071 8	2 254	13.01 to 21.69
C95, T95, X95	3.124	0.074 3	2 404	12.85 to 21.33
–100	3.143	0.076 8	2 553	12.70 to 21.00
P105, G105	3.162	0.079 4	2 702	12.57 to 20.70
P110	3.181	0.081 9	2 852	12.44 to 20.41
–120	3.219	0.087 0	3 151	12.21 to 19.88
Q125	3.239	0.089 5	3 301	12.11 to 19.63
–130	3.258	0.092 0	3 451	12.02 to 19.40
S135	3.278	0.094 6	3 601	11.92 to 19.18
–140	3.297	0.097 1	3 751	11.84 to 18.97
–150	3.336	0.102 1	4 053	11.67 to 18.57
–155	3.356	0.104 7	4 204	11.59 to 18.37
–160	3.375	0.107 2	4 356	11.52 to 18.19
–170	3.412	0.112 3	4 660	11.37 to 17.82
–180	3.449	0.117 3	4 966	11.21 to 17.47

[a] 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.

[b] The D/t range values and equation factors were calculated from Equations (36), (38), (44) or (49), (45) or (50), (46) or (51), (47) or (52), and (48) or (53).

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8.4.4 Transition collapse pressure equation

The minimum collapse pressure for the plastic to elastic transition zone p_T is calculated by Equation (39):

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

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.

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 Equation (40):

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

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.

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Table 7 — Equation factors and D/t range for transition collapse

Grade [a]	F_c	G_c	D/t range [b]
H40	2.063	0.032 5	27.01 to 42.64
–50	2.003	0.034 7	25.63 to 38.83
J55, K55	1.989	0.036 0	25.01 to 37.21
–60	1.983	0.037 3	24.42 to 35.73
–70	1.984	0.040 3	23.38 to 33.17
C75, E75	1.990	0.041 8	22.91 to 32.05
L-N-80	1.998	0.043 4	22.47 to 31.02
C90	2.017	0.046 6	21.69 to 29.18
C95, T95, X95	2.029	0.048 2	21.33 to 28.36
–100	2.040	0.049 9	21.00 to 27.60
P105, G105	2.053	0.051 5	20.70 to 26.89
P110	2.066	0.053 2	20.41 to 26.22
–120	2.092	0.056 5	19.88 to 25.01
Q125	2.106	0.058 2	19.63 to 24.46
–130	2.119	0.059 9	19.40 to 23.94
S135	2.133	0.061 5	19.18 to 23.44
–140	2.146	0.063 2	18.97 to 22.98
–150	2.174	0.066 6	18.57 to 22.11
–155	2.188	0.068 3	18.37 to 21.70
–160	2.202	0.070 0	18.19 to 21.32
–170	2.231	0.073 4	17.82 to 20.60
–180	2.261	0.076 9	17.47 to 19.93

[a] 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.

[b] The D/t range values and equation factors were calculated from Equations (36), (38), (44) or (49), (45) or (50), (46) or (51), (47) or (52), and (48) or (53).

8.4.5 Elastic collapse pressure equation

The minimum collapse pressure for the elastic range of collapse is calculated by Equation (41):

$$p_E = 46.95 \times 10^6/[(D/t)\,(D/t - 1)^2] \tag{41}$$

where

D is the specified pipe outside diameter;

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.

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Table 8 — D/t range for elastic collapse

Grade [a]	D/t range [b]
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
C95, 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

[a] 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.

[b] The D/t range values were calculated from Equations (40), (44) or (49), and (45) or (50).

8.4.6 Collapse pressure under axial tension stress

The collapse resistance of casing in the presence of an axial stress is calculated by modifying the yield stress to an axial stress equivalent grade according to Equation (42):

$$f_{yax} = \{[1 - 0.75(\sigma_a/f_{ymn})^2]^{1/2} - 0.5\ \sigma_a/f_{ymn}\}\, f_{ymn} \tag{42}$$

where

f_{yax} is the equivalent yield strength in the presence of axial stress;

f_{ymn} is the specified minimum yield strength;

σ_a is the component of axial stress not due to bending.

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Collapse resistance equation factors and D/t ranges for the axial stress equivalent grade are then calculated by means of Equations (36), (38), (40), (44) or (49), (45) or (50), (46) or (51), (47) or (52), and (48) or (53). Using equation factors for the axial stress equivalent grade, collapse resistance under axial stress is calculated by means of Equations (35), (37), (41) and (39).

API collapse resistance equations are not valid for the yield strength of axial stress equivalent grade (f_{yax}) less than 24 000 psi.

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

8.4.7 Effect of internal pressure on collapse

The external pressure equivalent of external pressure and internal pressure is determined by means of Equation (43). The equation is based on the internal pressure acting on the inside diameter and the external pressure acting on the outside diameter.

$$p_{ci} = p_c + (1 - 2\,t/D)\,p_i \tag{43}$$

where

D is the specified pipe outside diameter;

p_c is the collapse pressure;

p_{ci} is the collapse pressure in the presence of internal pressure;

p_i is the internal pressure;

t is the specified pipe wall thickness.

The value p_c is the collapse resistance calculated neglecting internal pressure, but accounting for any axial load as described in 8.4.6. Equation (43) was taken from Reference [56].

8.5 Equations for empirical constants

8.5.1 General

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 (8.5.2) and USC (8.5.3) units.

8.5.2 SI units

$$A_c = 2{,}876\,2 + 0{,}154\,89 \times 10^{-3} f_{ymn} + 0{,}448\,09 \times 10^{-6} f_{ymn}^2 - 0{,}162\,11 \times 10^{-9} f_{ymn}^3 \tag{44}$$

$$B_c = 0{,}026\,233 + 0{,}73402 \times 10^{-4} f_{ymn} \tag{45}$$

$$C_c = -3{,}212\,5 + 0{,}030\,867 f_{ymn} - 0{,}15204 \times 10^{-5} f_{ymn}^2 + 0{,}778\,10 \times 10^{-9} f_{ymn}^3 \tag{46}$$

$$F_c = 3{,}237 \times 10^5\,[(3\,B_c/A_c)/(2 + B_c/A_c)]^3/\{f_{ymn}\,[(3\,B_c/A_c)/(2 + B_c/A_c) - B_c/A_c][1 - (3\,B_c/A_c)/(2 + B_c/A_c)]^2\} \tag{47}$$

$$G_c = F_c\,B_c/A_c \tag{48}$$

where

A_c is the empirical constant in historical API collapse equation;

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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.

8.5.3 USC units

$$A_c = 2.876\,2 + 0.106\,79 \times 10^{-5} f_{ymn} + 0.213\,01 \times 10^{-10} f_{ymn}^2 - 0.531\,32 \times 10^{-16} f_{ymn}^3 \quad (49)$$

$$B_c = 0.026\,233 + 0.506\,09 \times 10^{-6} f_{ymn} \quad (50)$$

$$C_c = -465.93 + 0.030\,867 f_{ymn} - 0.104\,83 \times 10^{-7} f_{ymn}^2 + 0.369\,89 \times 10^{-13} f_{ymn}^3 \quad (51)$$

$$F_c = 46.95 \times 10^6 [(3 B_c/A_c)/(2 + B_c/A_c)]^3/\{f_{ymn} [(3 B_c/A_c)/(2 + B_c/A_c) - B_c/A_c][1 - (3 B_c/A_c)/(2 + B_c/A_c)]^2\} \quad (52)$$

$$G_c = F_c B_c/A_c \quad (53)$$

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

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 ISO 11960 or API 5CT. Application of these relations outside the range of yield strengths and D/t ratios contained in ISO 11960 or 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 ISO 11960 or API 5CT. However, the API 5L inventory of line pipe contains D/t ratios that often exceed casing D/t ratios significantly.

For line pipe having a yield strength and D/t falling within the limits of the sizes and thicknesses listed in ISO 11960 or API 5CT, 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.

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8.7 Example calculations

Calculate the collapse pressure of size Label 1: 7, mass Label 2: 26, grade P110 casing with axial stress of 11 000 psi and internal pressure 1 000 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				
σ_a	75,842	MPa	11 000	psi
p_i	6,895	MPa	1 000	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_{yax}	717,7	MPa	104 087	psi
A_c	3.158			
B_c	0.078 9			
C_c	18,444	MPa	2 675	psi
F_c	2.051			
G_c	0.051 2			
Equation	Plastic			
p_c	42,1	MPa	6 110	psi
p_{ci}	48,3	MPa	7 010	psi

9 Joint strength

9.1 General

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.

In this and other clauses, the following abbreviations may be employed:

— BC, buttress thread and coupling;

— BC SC, buttress thread with special clearance coupling;

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— EU, external upset end;

— EU SC, external upset end with special clearance coupling;

— LC, long thread and coupling (round thread);

— NU, non-upset end;

— STC, short thread and coupling (round thread);

— XC, extreme-line casing.

9.2 API casing connection tensile joint strength

9.2.1 General

The following tensile joint strength performance properties apply to casing connections manufactured in accordance with API 5B and ISO 11960 or API 5CT.

9.2.2 Round thread casing joint strength

9.2.2.1 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 jumpout or pullout, 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.

9.2.2.2 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.

9.2.2.3 Data requirements

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

D specified pipe outside diameter, inches;

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

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

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

t specified pipe wall thickness, inches.

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

A hand-tight standoff;

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

f_{umc} actual tensile strength of a representative tensile specimen from the coupling, psi;

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H is the thread height of a round thread equivalent Vee thread, 2,199 6 mm (0.086 60 in) for 10 TPI, 2,749 6 mm (0.1082 5 in) for 8 TPI;

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

s_{rn} root truncation of the pipe thread of round threads, 0,36 mm (0.014 in) for 10 TPI, 0,43 mm (0.017 in) for 8 TPI;

T_d taper (on diameter), 0,062 5 mm/mm (0.062 5 in/in);

W specified coupling outside diameter, in accordance with ISO 11960 or API 5CT.

9.2.2.4 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:

$$P_j = 0.95\, A_{jp} f_{umnp} \text{ (fracture strength)} \tag{54}$$

or

$$P_j = 0.95\, A_{jp} L_{et}[(0.74 D^{-0.59} f_{umnp})/(0.5 L_{et} + 0.14 D) + f_{ymnp}/(L_{et} + 0.14 D)] \quad \text{(pull-out strength)} \tag{55}$$

or

$$P_j = 0.95\, A_{jc} f_{umc} \quad \text{(coupling fracture strength)} \tag{56}$$

where

$$A_{jp} = \pi/4\, [(D - 0.142\,5)^2 - d^2] \tag{57}$$

$$A_{jc} = \pi/4\, (W^2 - d_1^2) \tag{58}$$

$$d_1 = E_1 - (L_1 + A)T_d + H - 2s_{rn} \tag{59}$$

and

A is the hand-tight standoff;

A_{jc} is the area of the coupling cross section, square inches;

A_{jp} is the area of the pipe cross section under the last perfect thread, square inches;

D is the specified pipe outside diameter, inches;

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

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

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

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

f_{umc} is the specified minimum tensile strength of the coupling, psi;

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

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H is the thread height of a round thread equivalent Vee thread, 2,199 6 mm (0.0866 0 in) for 10 TPI, 2,749 6 mm (0.1082 5 in) for 8 TPI;

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

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

P_j is the joint strength, pounds;

s_{rn} is the root truncation of the pipe thread of round threads, 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, inches;

T_d is the taper (on diameter), 0,062 5 mm/mm (0.062 5 in/in);

W is the specified coupling outside diameter, in accordance with ISO 11960 or API 5CT.

9.2.3 Buttress thread casing joint strength

9.2.3.1 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.

9.2.3.2 Assumptions and limitations

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

— the buttress thread does not fail by pull-out. Note that this assumption is contradicted by some test data for larger D/t ratios;

— the effect of internal and external pressure is ignored;

— the effect of casing curvature is 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.

9.2.3.3 Data requirements

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

D specified pipe outside diameter, inches;

E_7 pitch diameter, in accordance with API 5B;

f_{umnc} specified minimum tensile strength of the coupling, psi;

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

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

h_B buttress thread height: 1,575 for SI units, 0.062 for USC units;

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