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**Erdöl- und Erdgasindustrie –
Berechnung der Wanddicke von Heizrohren in Erdölraffinerien
(ISO 13704:2007);
Englische Fassung EN ISO 13704:2007**

Petroleum, petrochemical and natural gas industries –
Calculation of heater-tube thickness in petroleum refineries (ISO 13704:2007);
English version EN ISO 13704:2007

Industries du pétrole, de la pétrochimie et du gaz naturel –
Calcul de l'épaisseur des tubes de fours de raffineries de pétrole (ISO 13704:2007);
Version anglaise EN ISO 13704:2007

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Nationales Vorwort

Dieses Dokument (EN ISO 13704:2007) wurde vom Technischen Komitee ISO/TC 67 „Materials, equipment and offshore structures for petroleum, petrochemical and natural gas industries“, Subkomitee SC 6 „Processing equipment and systems“, dessen Sekretariat vom AFNOR (Frankreich) gehalten wird, in Zusammenarbeit mit dem Technischen Komitee CEN/TC 12 „Materialien, Ausrüstungen und Offshore-Bauwerke für die Erdöl-, petrochemische und Erdgasindustrie“, dessen Sekretariat vom AFNOR (Frankreich) gehalten wird, erarbeitet.

Es handelt sich dabei um die unveränderte Übernahme von ISO 13704:2007.

Das zuständige deutsche Gremium ist der NA 109-00-01 AA „Materialien, Ausrüstung und Offshore-Bauwerke für die Erdöl-, petrochemische und Erdgasindustrie“ im Normenausschuss Erdöl- und Erdgasgewinnung (NÖG).

Diese Europäische Norm enthält unter Berücksichtigung des DIN-Präsidialbeschlusses 1/2004 nur die englische Originalfassung der ISO-Norm.

Diese Norm enthält neben den gesetzlichen Einheiten auch die Einheiten „Btu“, „°F“, „ft“, „inch“, „lb“, „lbf“, „psi“, „°R“, die im Deutschen Normenwerk nicht zugelassen sind. Es wird jedoch ausdrücklich darauf hingewiesen, dass die Anwendung dieser Einheiten im nationalen amtlichen und geschäftlichen Verkehr aufgrund des Gesetzes über Einheiten im Messwesen nicht zulässig ist.

Umrechnung:

$$1 \text{ Btu} = 1\,055 \text{ J}$$

$$^{\circ}\text{C} = (5/9) (^{\circ}\text{F} - 32)$$

$$1 \text{ ft} = 0,304\,8 \text{ m}$$

$$1 \text{ inch} = 25,4 \text{ mm}$$

$$1 \text{ lb} = 0,453\,592\,37 \text{ kg}$$

$$1 \text{ lbf} = 4,448\,222 \text{ N}$$

$$1 \text{ psi} = 6\,894,757 \text{ Pa}$$

$$^{\circ}\text{C} = (5/9) (^{\circ}\text{R} - 491,67)$$

English Version

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Erdöl- und Erdgasindustrie – Berechnung der Wanddicke
von Heizrohren in Erdölraffinerien
(ISO 13704:2007)

This European Standard was approved by CEN on 3 November 2007.

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Foreword

This document (EN ISO 13704:2007) has been prepared by Technical Committee ISO/TC 67 “Materials, equipment and offshore structures for petroleum and natural gas industries” in collaboration with Technical Committee CEN/TC 12 “Materials, equipment and offshore structures for petroleum, petrochemical and natural gas industries” the secretariat of which is held by AFNOR.

This European Standard shall be given the status of a national standard, either by publication of an identical text or by endorsement, at the latest by May 2008, and conflicting national standards shall be withdrawn at the latest by May 2008.

According to the CEN/CENELEC Internal Regulations, the national standards organizations of the following countries are bound to implement this European Standard: Austria, Belgium, Bulgaria, Cyprus, Czech Republic, Denmark, Estonia, Finland, France, Germany, Greece, Hungary, Iceland, Ireland, Italy, Latvia, Lithuania, Luxembourg, Malta, Netherlands, Norway, Poland, Portugal, Romania, Slovakia, Slovenia, Spain, Sweden, Switzerland and the United Kingdom.

Endorsement notice

The text of ISO 13704:2007 has been approved by CEN as a EN ISO 13704:2007 without any modification.

1 Scope

This International Standard specifies the requirements and gives recommendations for the procedures and design criteria used for calculating the required wall thickness of new tubes and associated component fittings for petroleum-refinery heaters. These procedures are appropriate for designing tubes for service in both corrosive and non-corrosive applications. These procedures have been developed specifically for the design of refinery and related process-fired heater tubes (direct-fired, heat-absorbing tubes within enclosures). These procedures are not intended to be used for the design of external piping.

This International Standard does not give recommendations for tube retirement thickness; Annex A describes a technique for estimating the life remaining for a heater tube.

2 Terms and definitions

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

2.1

actual inside diameter

D_i

inside diameter of a new tube

NOTE The actual inside diameter is used to calculate the tube skin temperature in Annex B and the thermal stress in Annex C.

2.2

component fitting

fitting connected to the fired heater tubes

EXAMPLES Return bends, elbows, reducers.

NOTE 1 There is a distinction between standard component fittings and specially designed component fittings; see 4.9.

NOTE 2 Typical material specifications for standard component fittings are ASTM A 234, ASTM A 403 and ASTM B 366.

2.3

corrosion allowance

δ_{CA}

additional material thickness added to allow for material loss during the design life of the component

2.4

design life

t_{DL}

operating time used as a basis for tube design

NOTE The design life is not necessarily the same as the retirement or replacement life.

2.5**design metal temperature** T_d

tube-metal or skin temperature used for design

NOTE This is determined by calculating the maximum tube metal temperature ($T_{\max}$ in Annex B) or the equivalent tube metal temperature (T_{eq} in 2.8) and adding an appropriate temperature allowance (see 2.16). A procedure for calculating the maximum tube metal temperature from the heat-flux density is included in Annex B. When the equivalent tube metal temperature is used, the maximum operating temperature can be greater than the design metal temperature. When the equivalent tube metal temperature is used to determine the design metal temperature, this design metal temperature is only applicable to the rupture design. It is necessary to develop a separate design metal temperature applicable to the elastic design. The design metal temperature applicable to the elastic design is the maximum calculated tube metal temperature among all operating cases plus the appropriate temperature allowance.

2.6**elastic allowable stress** σ_{el}

allowable stress for the elastic range

See 5.2.

2.7**elastic design pressure** p_{el}

maximum pressure that the heater coil can sustain for short periods of time

NOTE This pressure is usually related to relief-valve settings, pump shut-in pressures, etc.

2.8**equivalent tube metal temperature** T_{eq}

calculated constant metal temperature that in a specified period of time produces the same creep damage as does a changing metal temperature

NOTE In 4.8 the equivalent tube metal temperature concept is described in more detail. It provides a procedure to calculate the equivalent tube metal temperature based on a linear change of tube metal temperature from start-of-run to end-of-run.

2.9**inside diameter** D_i^*

inside diameter of a tube with the corrosion allowance removed; used in the design calculations

NOTE The inside diameter of an as-cast tube is the inside diameter of the tube with the porosity and corrosion allowances removed.

2.10**minimum thickness** δ_{min}

minimum required thickness of a new tube, taking into account all appropriate allowances

NOTE See Equation (5).

2.11**outside diameter** D_o

outside diameter of a new tube

2.12

rupture allowable stress

σ_r

allowable stress for the creep-rupture range

See 4.4.

2.13

rupture design pressure

p_r

maximum operating pressure that the coil section can sustain during normal operation

2.14

rupture exponent

n

parameter used for design in the creep-rupture range

NOTE See figures in Annexes E and F.

2.15

stress thickness

δ_σ

thickness, excluding all thickness allowances, calculated from an equation that uses an allowable stress

2.16

temperature allowance

T_A

part of the design metal temperature that is included for process- or flue-gas mal-distribution, operating unknowns, and design inaccuracies

NOTE The temperature allowance is added to the calculated maximum tube metal temperature or to the equivalent tube metal temperature to obtain the design metal temperature (see 2.5).

3 General design information

3.1 Information required

The design parameters (design pressures, design fluid temperature, corrosion allowance and tube material) shall be defined. In addition, the following information shall be furnished:

- a) design life of the heater tube;
- b) whether the equivalent-temperature concept is to be applied and, if so, the operating conditions at the start and at the end of the run;
- c) temperature allowance (see ISO 13705), if any;
- d) corrosion fraction (if different from that shown in Figure 1);
- e) whether elastic-range thermal-stress limits are to be applied.

If any of items a) to e) are not furnished, use the following applicable parameters:

- design life equal to 100 000 h;
- design metal temperature based on the maximum metal temperature (the equivalent-temperature concept shall not apply);

- temperature allowance equal to 15 °C (25 °F);
- corrosion fraction given in Figure 1;
- elastic-range thermal-stress limits.

3.2 Limitations for design procedures

3.2.1 The allowable stresses are based on a consideration of yield strength and rupture strength only; plastic or creep strain has not been considered. Using these allowable stresses can result in small permanent strains in some applications; however, these small strains do not affect the safety or operability of heater tubes.

3.2.2 No considerations are included for adverse environmental effects, such as graphitization, carburization or hydrogen attack. Limitations imposed by hydrogen attack can be developed from the Nelson curves in API 941 ^[1].

3.2.3 These design procedures have been developed for seamless tubes. They are not applicable to tubes that have a longitudinal weld. ISO 13705 allows only seamless tubes.

3.2.4 These design procedures have been developed for thin tubes (tubes with a thickness-to-outside-diameter ratio, $\delta_{\min}/D_o$, of less than 0,15). Additional considerations can apply to the design of thicker tubes.

3.2.5 No considerations are included for the effects of cyclic pressure or cyclic thermal loading.

3.2.6 Limits for thermal stresses are provided in Annex C. Limits for stresses developed by mass, supports, end connections and so forth are not discussed in this International Standard.

3.2.7 Most of the Larson-Miller parameter referenced curves in 5.6 are not Larson-Miller curves in the traditional sense but are derived from the 100 000 h rupture strength as explained in Clause H.3. Consequently, the curves might not provide a reliable estimate of the rupture strength for a design life that is less than 20 000 h or more than 200 000 h.

3.2.8 The procedures in this International Standard have been developed for systems in which the heater tubes are subject to an internal pressure that exceeds the external pressure. There are some cases in which a heater tube can be subject to a greater external pressure than the internal pressure. This can occur, for example, in vacuum heaters or on other types of heaters during shutdown or trip conditions, especially when a unit is cooling or draining, forming a vacuum inside the heater tubes. Conditions where external pressures exceed the internal pressures can govern heater-tube wall thickness. Determination of this (i.e. vacuum design) is not covered in this International Standard. In the absence of any local or national codes that can apply, it is recommended that a pressure vessel code, such as ASME VIII (Division 1, UG-28) or EN 13445, be used, as such codes also address external pressure designs.

4 Design

4.1 General

There is a fundamental difference between the behaviour of carbon steel in a hot-oil heater tube operating at 300 °C (575 °F) and that of chromium-molybdenum steel in a catalytic-reformer heater tube operating at 600 °C (1 110 °F). The steel operating at the higher temperature creeps, or deforms permanently, even at stress levels well below the yield strength. If the tube metal temperature is high enough for the effects of creep to be significant, the tube eventually fails due to creep rupture, although no corrosion or oxidation mechanism is active. For the steel operating at the lower temperature, the effects of creep are non-existent or negligible. Experience indicates that, in this case, the tube lasts indefinitely, unless a corrosion or an oxidation mechanism is active.

Since there is a fundamental difference between the behaviour of the materials at these two temperatures, there are two different design considerations for heater tubes: elastic design and creep-rupture design. Elastic design is design in the elastic range, at lower temperatures, in which allowable stresses are based on the yield strength (see 4.3). Creep-rupture design (which is referred to below as rupture design) is the design for the creep-rupture range, at higher temperatures, in which allowable stresses are based on the rupture strength (see 4.4).

The temperature that separates the elastic and creep-rupture ranges of a heater tube is not a single value; it is a range of temperatures that depends on the alloy. For carbon steel, the lower end of this temperature range is about 425 °C (800 °F); for type 347 stainless steel, the lower end of this temperature range is about 590 °C (1 100 °F). The considerations that govern the design range also include the elastic design pressure, the rupture design pressure, the design life and the corrosion allowance.

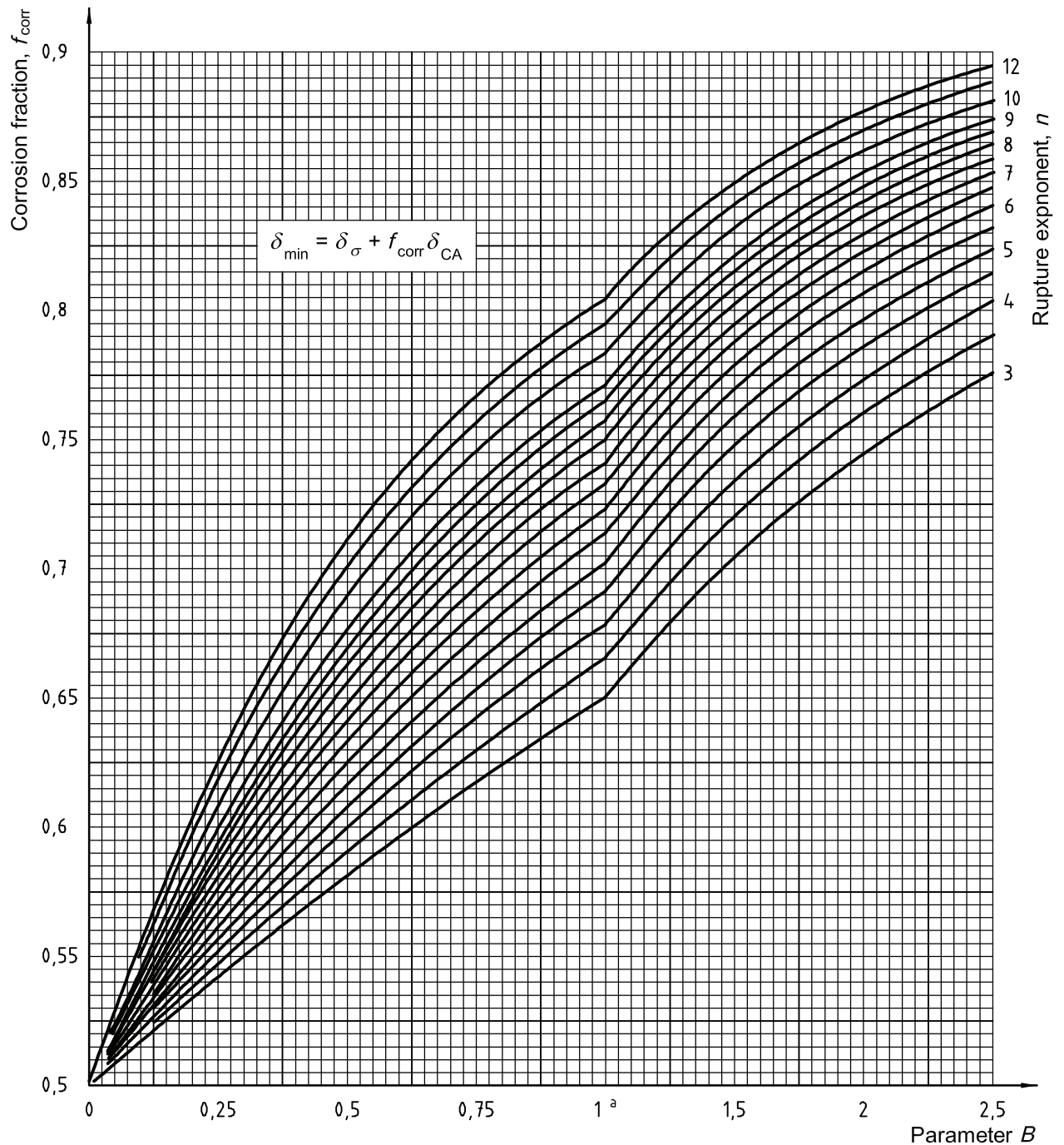
The rupture design pressure is never more than the elastic design pressure. The characteristic that differentiates these two pressures is the relative length of time over which they are sustained. The rupture design pressure is a long-term loading condition that remains relatively uniform over a period of years. The elastic design pressure is usually a short-term loading condition that typically lasts only hours or days. The rupture design pressure is used in the rupture design equation, since creep damage accumulates as a result of the action of the operating, or long-term, stress. The elastic design pressure is used in the elastic design equation to prevent excessive stresses in the tube during periods of operation at the maximum pressure.

The tube shall be designed to withstand the rupture design pressure for long periods of operation. If the normal operating pressure increases during an operating run, the highest pressure shall be taken as the rupture design pressure.

In the temperature range near or above the point where the elastic and rupture allowable stress curves cross, both elastic and rupture design equations are to be used. The larger value of $\delta_{\min}$ should govern the design (see 4.5). A sample calculation that uses these methods is included in Clause 6. Calculation sheets (see Annex D) are available for summarizing the calculations of minimum thickness and equivalent tube metal temperature.

The allowable minimum thickness of a new tube is given in Table 1.

All of the design equations described in Clause 4 are summarized in Table 2.



Key

$$\delta_{\sigma} = \frac{p_r D_o}{2\sigma_r + p_r}$$

δ_{CA} is the corrosion allowance

D_o is the outside diameter

σ_r is the rupture allowable stress

p_r is the rupture design pressure

$$B = \delta_{\text{CA}} / \delta_{\sigma}$$

^a Note change of scale at $X = 1$.

Figure 1 — Corrosion fraction

4.2 Equation for stress

In both the elastic range and the creep-rupture range, the design equation is based on the mean-diameter equation for stress in a tube. In the elastic range, the elastic design pressure, p_{el} , and the elastic allowable stress, σ_{el} , are used. In the creep-rupture range, the rupture design pressure, p_r , and the rupture allowable stress, σ_r , are used.

The mean-diameter equation gives a good estimate of the pressure that produces yielding through the entire tube wall in thin tubes (see 3.2.4 for a definition of thin tubes). The mean-diameter equation also provides a good correlation between the creep rupture of a pressurized tube and a uniaxial test specimen. It is, therefore, a good equation to use in both the elastic range and the creep-rupture range^{[16], [17], [18], [19]}. The mean-diameter equation for stress is as given in Equation (1):

$$\sigma = \frac{p}{2} \left(\frac{D_o}{\delta} - 1 \right) = \frac{p}{2} \left(\frac{D_i}{\delta} + 1 \right) \quad (1)$$

where

σ is the stress, expressed in megapascals [pounds per square inch¹⁾];

p is the pressure, expressed in megapascals (pounds per square inch);

D_o is the outside diameter, expressed in millimetres (inches);

D_i is the inside diameter, expressed in millimetres (inches), including the corrosion allowance;

δ is the thickness, expressed in millimetres (inches).

The equations for the stress thickness, δ_σ , in 4.3 and 4.4 are derived from Equation (1).

4.3 Elastic design (lower temperatures)

The elastic design is based on preventing failure by bursting when the pressure is at its maximum (that is, when a pressure excursion has reached p_{el} near the end of the design life after the corrosion allowance has been used up. With the elastic design, δ_σ and δ_{min} (see 4.6) are calculated as given in Equations (2) and (3):

$$\delta_\sigma = \frac{p_{el} D_o}{2\sigma_{el} + p_{el}} \text{ or } \delta_\sigma = \frac{p_{el} D_i^*}{2\sigma_{el} - p_{el}} \quad (2)$$

$$\delta_{min} = \delta_\sigma + \delta_{CA} \quad (3)$$

where

D_i^* is the inside diameter, expressed in millimetres (inches), with corrosion allowance removed;

σ_{el} is the elastic allowable stress, expressed in megapascals (pounds per square inch), at the design metal temperature.

1) The unit "pounds per square inch (psi)" is referred to as "pound-force per square inch (lbf/in²)" in ISO/IEC 80000.

4.4 Rupture design (higher temperatures)

The rupture design is based on preventing failure by creep rupture during the design life. With the rupture design, δ_σ and $\delta_{\min}$ (see 4.6) are calculated from Equations (4) and (5):

$$\delta_\sigma = \frac{p_r D_o}{2\sigma_r + p_r} \text{ or } \delta_\sigma = \frac{p_r D_i^*}{2\sigma_r - p_r} \quad (4)$$

$$\delta_{\min} = \delta_\sigma + f_{\text{corr}} \delta_{\text{CA}} \quad (5)$$

where

σ_r is the rupture allowable stress, expressed in megapascals (pounds per square inch), at the design metal temperature and the design life;

f_{corr} is the corrosion fraction, given as a function of B and n in Figure 1;

$$B = \delta_{\text{CA}} / \delta_\sigma$$

n is the rupture exponent at the design metal temperature (shown in the figures given in Annexes E and F).

The derivation of the corrosion fraction is described in Annex G. It is recognized in this derivation that stress is reduced by the corrosion allowance; correspondingly, the rupture life is increased.

Equations (4) and (5) are suitable for heater tubes; however, if special circumstances require that the user choose a more conservative design, a corrosion fraction of unity ($f_{\text{corr}} = 1$) may be specified.

4.5 Intermediate temperature range

At temperatures near or above the point where the curves of σ_{el} and σ_r intersect in the figures given in Annexes E and F, either elastic or rupture considerations govern the design. In this temperature range, it is necessary to apply both the elastic and the rupture designs. The larger value of $\delta_{\min}$ shall govern the design.

4.6 Minimum allowable thickness

The minimum thickness, $\delta_{\min}$, of a new tube (including the corrosion allowance) shall not be less than that shown in Table 1. For ferritic steels, the values shown are the minimum allowable thicknesses of schedule 40 average wall pipe. For austenitic steels, the values are the minimum allowable thicknesses of schedule 10S average wall pipe. (Table 5 shows which alloys are ferritic and which are austenitic.) The minimum allowable thicknesses are 0,875 times the average thicknesses. These minima are based on industry practice. The minimum allowable thickness is not the retirement or replacement thickness of a used tube.

4.7 Minimum and average thicknesses

The minimum thickness, $\delta_{\min}$, is calculated as described in 4.3 and 4.4. Tubes that are purchased to this minimum thickness have a greater average thickness. A thickness tolerance is specified in each ASTM specification. For most of the ASTM specifications shown in the figures given in Annexes E and F, the tolerance on the minimum thickness is $\left(\begin{smallmatrix} 0 \\ +28 \end{smallmatrix} \right)$ % for hot-finished tubes and $\left(\begin{smallmatrix} 0 \\ +22 \end{smallmatrix} \right)$ % for cold-drawn tubes. This is equivalent to tolerances on the average thickness of $\pm 12,3$ % and $\pm 9,9$ %, respectively. The remaining ASTM specifications require that the minimum thickness be greater than 0,875 times the average thickness, which is equivalent to a tolerance on the average thickness of $+ 12,5$ %.

With a $\left(\begin{smallmatrix} 0 \\ +28 \end{smallmatrix} \right)$ % tolerance, a tube that is purchased to a 12,7 mm (0,500 in) minimum-thickness specification has the following average thickness:

$$(12,7)(1 + 0,28/2) = 14,5 \text{ mm (0,570 in)}$$

To obtain a minimum thickness of 12,7 mm (0,500 in) in a tube purchased to a $\pm 12,5$ % tolerance on the average thickness, the average thickness shall be specified as follows:

$$(12,7)/(0,875) = 14,5 \text{ mm (0,571 in)}$$

All thickness specifications shall indicate whether the specified value is a minimum or an average thickness. The tolerance used to relate the minimum and average wall thicknesses shall be the tolerance given in the ASTM specification to which the tubes are purchased.

Table 1 — Minimum allowable thickness of new tubes

Tube outside diameter		Minimum thickness			
		Ferritic steel tubes		Austenitic steel tubes	
mm	(in)	mm	(in)	mm	(in)
60,3	(2,375)	3,4	(0,135)	2,4	(0,095)
73,0	(2,875)	4,5	(0,178)	2,7	(0,105)
88,9	(3,50)	4,8	(0,189)	2,7	(0,105)
101,6	(4,00)	5,0	(0,198)	2,7	(0,105)
114,3	(4,50)	5,3	(0,207)	2,7	(0,105)
141,3	(5,563)	5,7	(0,226)	3,0	(0,117)
168,3	(6,625)	6,2	(0,245)	3,0	(0,117)
219,1	(8,625)	7,2	(0,282)	3,3	(0,130)
273,1	(10,75)	8,1	(0,319)	3,7	(0,144)

4.8 Equivalent tube metal temperature

In the creep-rupture range, the accumulation of damage is a function of the actual operating tube metal temperatures (TMTs). For applications in which there are significant differences between start-of-run and end-of-run TMTs, a design based on the maximum temperature can be excessive, since the actual operating TMT is usually less than the maximum.

For a linear change in metal temperature from start of run, T_{sor} , to end of run, T_{eor} , an equivalent tube metal temperature, T_{eq} , can be calculated as shown in Equation (6). A tube operating at the equivalent tube metal temperature sustains the same creep damage as one that operates from the start-of-run to end-of-run temperatures.

$$T_{\text{eq}} = T_{\text{sor}} + f_T (T_{\text{eor}} - T_{\text{sor}}) \quad (6)$$

where

T_{eq} is the equivalent tube metal temperature, expressed in degrees Celsius (Fahrenheit);

T_{sor} is the tube metal temperature, expressed in degrees Celsius (Fahrenheit), at start of run;

T_{eor} is the tube metal temperature, expressed in degrees Celsius (Fahrenheit), at end of run;

f_T is the temperature fraction given in Figure 2.

The derivation of the temperature fraction is described in Annex G. The temperature fraction is a function of two parameters, V and N , as given in Equations (7) and (8):

$$V = n_0 \left(\frac{\Delta T^*}{T_{\text{sor}}^*} \right) \ln \left(\frac{A}{\sigma_0} \right) \quad (7)$$

$$N = n_0 \left(\frac{\Delta \delta}{\delta_0} \right) \quad (8)$$

where

n_0 is the rupture exponent at T_{sor} ;

ΔT^* is the temperature change, equal to $T_{\text{eor}} - T_{\text{sor}}$, expressed in kelvin [degrees Rankine²], during the operating period;

$T_{\text{sor}}^* = T_{\text{sor}} + 273 \text{ K } (T_{\text{sor}} + 460 \text{ }^\circ\text{R})$;

$\ln$ is the natural logarithm;

$\Delta \delta$ is the change in thickness, equal to $\phi_{\text{corr}} t_{\text{op}}$, expressed in millimetres (inches), during the operating period;

ϕ_{corr} is the corrosion rate, expressed in millimetres per year (in inches per year);

t_{op} is the duration of operating period, expressed in years;

δ_0 is the initial thickness, expressed in millimetres (inches), at the start of the run;

σ_0 is the initial stress, expressed in megapascals (pounds per square inch), at the start of the run, using Equation (1);

A is the material constant, expressed in megapascals (pounds per square inch).

The constant A is given in Table 3. The significance of the material constant is explained in Clause G.5.

2) Rankine is a deprecated unit.

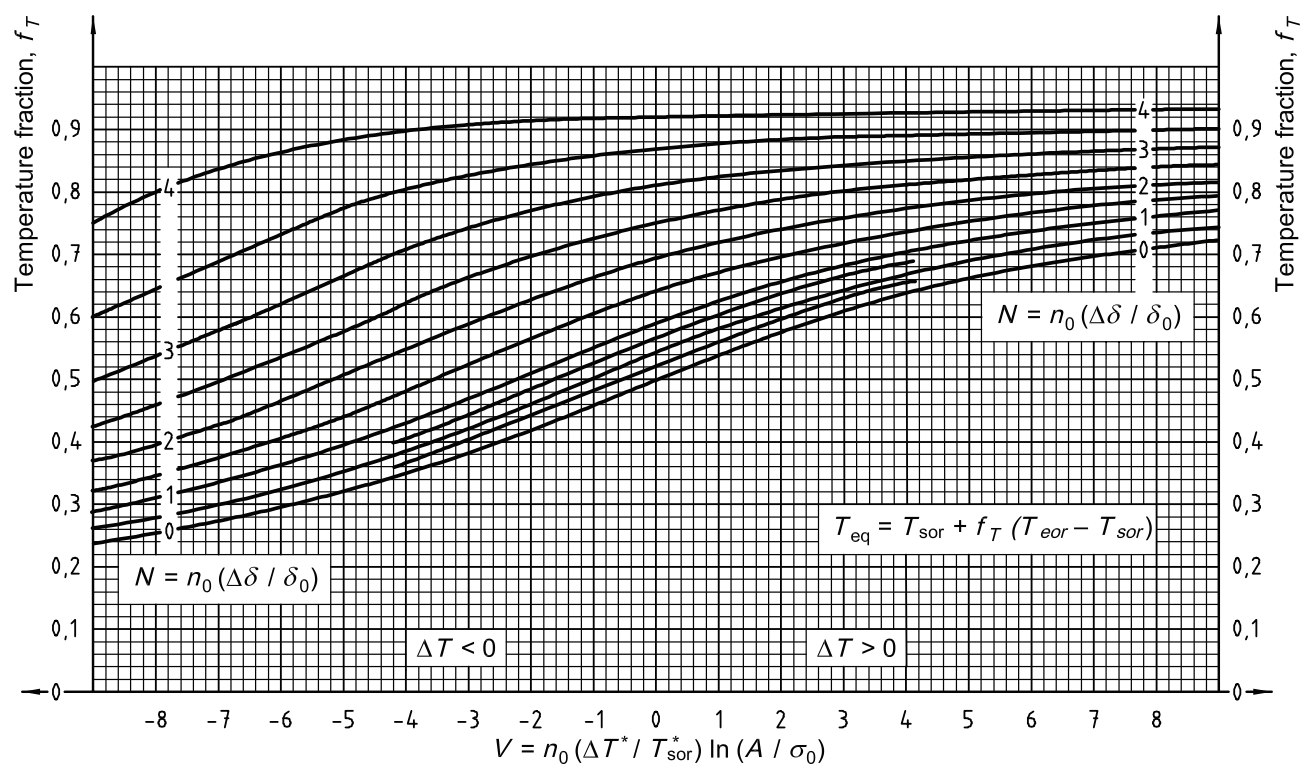


Figure 2 — Temperature fraction

Table 2 — Summary of working equations

Elastic design (lower temperatures):

$$\delta_{\sigma} = \frac{p_{el} D_o}{2\sigma_{el} + p_{el}} \quad \text{or} \quad \delta_{\sigma} = \frac{p_{el} D_i^*}{2\sigma_{el} - p_{el}} \quad (2)$$

$$\delta_{min} = \delta_{\sigma} + \delta_{CA} \quad (3)$$

Rupture design (higher temperatures):

$$\delta_{\sigma} = \frac{p_r D_o}{2\sigma_r + p_r} \quad \text{or} \quad \delta_{\sigma} = \frac{p_r D_i^*}{2\sigma_r - p_r} \quad (4)$$

$$\delta_{min} = \delta_{\sigma} + f_{corr} \delta_{CA} \quad (5)$$

where

- δ_{σ} is the stress thickness, expressed in millimetres (inches);
- p_{el} is the elastic design gauge pressure, expressed in megapascals (pounds per square inch);
- p_r is the rupture design gauge pressure, expressed in megapascals (pounds per square inch);
- D_o is the outside diameter, expressed in millimetres (inches);
- D_i^* is the inside diameter, expressed in millimetres (inches), with the corrosion allowance removed;
- σ_{el} is the elastic allowable stress, expressed in megapascals (pounds per square inch), at the design metal temperature;
- σ_r is the rupture allowable stress, expressed in megapascals (pounds per square inch), at the design metal temperature and design life;
- δ_{min} is the minimum thickness, expressed in millimetres (inches), including corrosion allowance;
- δ_{CA} is the corrosion allowance, expressed in millimetres (inches);
- f_{corr} is the corrosion fraction, given in Figure 1 as a function of B and n , where $B = \delta_{CA} / \delta_{\sigma}$;
- n is the rupture exponent at the design metal temperature.

Equivalent tube metal temperature:

$$T_{eq} = T_{sor} + f_T (T_{eor} - T_{sor}) \quad (6)$$

where

- ΔT^* ($= T_{eor} - T_{sor}$) is the temperature change, expressed in kelvin (degrees Rankine), during the operating period;
- T_{sor} is the tube metal temperature, expressed in degrees Celsius (Fahrenheit), at the start of the run;
- T_{eor} is the tube metal temperature, expressed in degrees Celsius (Fahrenheit), at the end of the run;
- $T_{sor}^* = T_{sor} + 273 \text{ K } (T_{sor} + 460 \text{ }^{\circ}\text{R})$;
- A is the material constant, expressed in megapascals (pounds per square inch) from Table 3;
- σ_0 is the initial stress, expressed in megapascals (pounds per square inch), at the start of the run using Equation (1);
- $\Delta \delta$ ($= \phi_{corr} t_{op}$) is the change in thickness, expressed in millimetres (inches), during the operating period;
- δ_0 is the initial thickness, expressed in millimetres (inches), at the start of the run;
- ϕ_{corr} is the corrosion rate, expressed in millimetres per year (inches per year);
- t_{op} is the duration, expressed in years, of the operating period.

Table 3 — Material constant for temperature fraction

Material	Type or grade	Constant <i>A</i>	
		MPa	(psi)
Low-carbon steel	—	$7,46 \times 10^5$	$(1,08 \times 10^8)$
Medium-carbon steel	B	$2,88 \times 10^5$	$(4,17 \times 10^7)$
C-½Mo steel	T1 or P1	$2,01 \times 10^7$	$(2,91 \times 10^9)$
1-¼Cr-½Mo steel	T11 or P11	$5,17 \times 10^7$	$(7,49 \times 10^9)$
2-¼Cr-1Mo steel	T22 or P22	$8,64 \times 10^5$	$(1,25 \times 10^8)$
3Cr-1Mo steel	T21 or P21	$2,12 \times 10^6$	$(3,07 \times 10^8)$
5Cr-½Mo steel	T5 or P5	$5,49 \times 10^5$	$(7,97 \times 10^7)$
5Cr-½Mo-Si steel	T5b or P5b	$2,88 \times 10^5$	$(4,18 \times 10^7)$
7Cr-½Mo steel	T7 or P7	$1,64 \times 10^5$	$(2,37 \times 10^7)$
9Cr-1Mo steel	T9 or P9	$7,54 \times 10^6$	$(1,09 \times 10^9)$
9Cr-1Mo V steel	T91 or P91	$2,23 \times 10^6$	$(3,24 \times 10^8)$
18Cr-8Ni steel	304 or 304H	$1,55 \times 10^6$	$(2,25 \times 10^8)$
16Cr-12Ni-2Mo steel	316 or 316H	$1,24 \times 10^6$	$(1,79 \times 10^8)$
16Cr-12Ni-2Mo steel	316L	$1,37 \times 10^6$	$(1,99 \times 10^8)$
18Cr-10Ni-Ti steel	321	$1,32 \times 10^6$	$(1,92 \times 10^8)$
18Cr-10Ni-Ti steel	321H	$2,76 \times 10^5$	$(4,00 \times 10^7)$
18Cr-10Ni-Nb ^a steel	347 or 347H	$1,23 \times 10^6$	$(1,79 \times 10^8)$
Ni-Fe-Cr	Alloy 800H / 800HT	$1,03 \times 10^5$	$(1,50 \times 10^7)$
25Cr-20Ni	HK40	$2,50 \times 10^5$	$(3,63 \times 10^7)$
^a Formerly called columbium, Cb.			

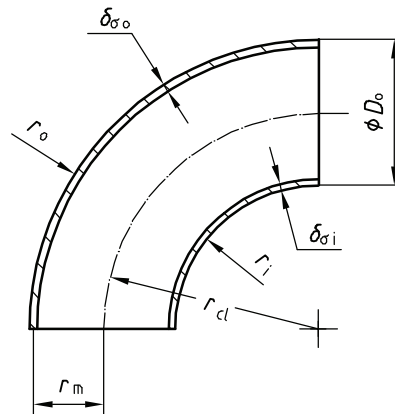
The temperature fraction and the equivalent temperature shall be calculated for the first operating cycle. In applications that involve very high corrosion rates, the temperature fraction for the last cycle is greater than that for the first. In such cases, the calculation of the temperature fraction and the equivalent temperature should be based on the last cycle.

If the temperature change from start-of-run to end-of-run is other than linear, a judgment shall be made regarding the use of the value of f_T given in Figure 2.

Note that the calculated thickness of a tube is a function of the equivalent temperature, which, in turn, is a function of the thickness (through the initial stress). A few iterations can be necessary to arrive at the design. (See the sample calculation in 6.4.)

4.9 Component fittings

Component fittings manufactured in accordance with ASME B16.9 are considered suitable for use at the pressure-temperature ratings specified therein. Other component fittings shall be specially designed in accordance with this subclause.



r_o = outer radius

r_i = inner radius

For other symbols, see text below Equation (9).

Figure 3 — Return bend and elbow geometry

The stress variations in a return bend or elbow (see Figure 3) are far more complex than in a straight tube. The hoop stresses at the inner radius of a return bend are higher than in a straight tube of the same thickness. It might be necessary for the minimum thickness at the inner radius to be greater than the minimum thickness of the attached tube.

Because fabrication processes for forged return bends generally result in greater thickness at the inner radius, the higher stresses at the inner radius can be sustained without failure in most situations.

The hoop stress σ_i , expressed in megapascals (pounds per square inch), along the inner radius of the bend is given by Equation (9):

$$\sigma_i = \frac{2r_{cl} - r_m}{2(r_{cl} - r_m)} \sigma \quad (9)$$

where

r_{cl} is the centre line radius of the bend, expressed in millimetres (inches);

r_m is the mean radius of the tube, expressed in millimetres (inches);

σ is the stress, expressed in megapascals (pounds per square inch), given by Equation (1).

The hoop stress σ_o , expressed in megapascals (pounds per square inch), along the outer radius is given by Equation (10):

$$\sigma_o = \frac{2r_{cl} + r_m}{2(r_{cl} + r_m)} \sigma \quad (10)$$

Using the approximation that r_m is almost equal to $D_o/2$, Equation (9) can be solved for the stress thickness at the inner radius. For design, the stress thickness is given by Equation (11).

$$\delta_{\sigma i} = \frac{D_o p}{2N_i \sigma + p} \quad (11)$$

where

$\delta_{\sigma i}$ is the stress thickness, expressed in millimetres (inches), at the inner radius;

$$N_i = \frac{4 \frac{r_{cl}}{D_o} - 2}{4 \frac{r_{cl}}{D_o} - 1} \quad (12)$$

σ is the allowable stress, expressed in megapascals (pounds per square inch) at the design metal temperature.

NOTE 1 p represents both elastic design pressure and rupture design pressure.

The return bend thickness evaluations shall be made using both elastic design pressure and rupture design pressure, and the governing thicknesses shall be the larger values at the inner and outer radii.

Using the approximation given above, Equation (10) can be solved for the stress thickness at the outer radius. For elastic design, the stress thickness is as given in Equation (13):

$$\delta_{\sigma o} = \frac{D_o p}{2N_o \sigma + p} \quad (13)$$

where

$\delta_{\sigma o}$ is the stress thickness, expressed in millimetres (inches), at the outer radius;

$$N_o = \frac{4 \frac{r_{cl}}{D_o} + 2}{4 \frac{r_{cl}}{D_o} + 1} \quad (14)$$

σ is the allowable stress, expressed in megapascals (pounds per square inch), at the design metal temperature.

NOTE 2 p represents both elastic design pressure and rupture design pressure.

The return bend thickness evaluations shall be made using both elastic design pressure and rupture design pressure, and the governing thicknesses shall be the larger values at the inner and outer radii.

The minimum thickness, $\delta_{\sigma i}$, at the inside radius and the minimum thickness, $\delta_{\sigma o}$, at the outside radius shall be calculated using Equations (11) and (13). The corrosion allowance, δ_{CA} , shall be added to the minimum calculated thickness.

The minimum thickness along the neutral axis of the bend shall be the same as for a straight tube.

5 Allowable stresses

5.1 General

The allowable stresses for various heater-tube alloys are plotted against design metal temperature in Figures E.1 to E.19 (SI units) and Figures F.1 to F.19 [US Customary (USC) units]. The values shown in these

figures are recommended only for the design of heater tubes. These figures show two different allowable stresses, the elastic allowable stress and the rupture allowable stress. The bases for these allowable stresses are given in 5.2 and 5.3 (see also 3.2.3).

5.2 Elastic allowable stress

The elastic allowable stress, σ_{el} , is two-thirds of the yield strength at temperature for ferritic steels and 90 % of the yield strength at temperature for austenitic steels. The data sources for the yield strength are given in Annex H.

If a different design basis is desired for special circumstances, the user shall specify the basis, and the alternative elastic allowable stress shall be developed from the yield strength.

5.3 Rupture allowable stress

The rupture allowable stress, σ_r , is 100 % of the minimum rupture strength for a specified design life. Annex H defines the minimum rupture strength and provides the data sources. The 20 000-h, 40 000-h, 60 000-h and 100 000-h rupture allowable stresses were developed from the Larson-Miller parameter curves for the minimum rupture strength shown on the right-hand side of Figures E.1 to E.19 (Figures F.1 to F.19). For a design life other than those shown, the corresponding rupture allowable stress shall be developed from the Larson-Miller parameter curves for the minimum rupture strength (see 5.6).

If a different design basis is desired, the user shall specify the basis, and the alternative rupture allowable stress shall be developed from the Larson-Miller parameter curves for the minimum or average rupture strength. If the resulting rupture allowable stress is greater than the minimum rupture strength for the design life, the effects of creep on the tube design equation should be considered.

5.4 Rupture exponent

Figures E.1 to E.19 (Figures F.1 to F.19) show the rupture exponent, n , as a function of the design metal temperature. The rupture exponent is used for design in the creep-rupture range (see 4.4). The meaning of the rupture exponent is discussed in Clause H.4.

5.5 Yield and tensile strengths

Figures E.1 to E.19 (Figures F.1 to F.19) also show the yield and tensile strengths. These curves are included only for reference. Their sources are given in Annex H.

5.6 Larson-Miller parameter curves

On the right-hand side of Figures E.1 to E.19 (Figures F.1 to F.19) are plots of the minimum and the average 100 000-h rupture strengths against the Larson-Miller parameter. The Larson-Miller parameter is calculated from the design metal temperature, T_d , and the design life, t_{DL} , as given in Equations (15) and (16):

When T_d is expressed in degrees Celsius:

$$(T_d + 273) (C_{LM} + \lg t_{DL}) \times 10^{-3} \quad (15)$$

When T_d is expressed in degrees Fahrenheit:

$$(T_d + 460) (C_{LM} + \lg t_{DL}) \times 10^{-3} \quad (16)$$

The Larson-Miller constant, C_{LM} , is stated in the curves. (See Clause H.3 for a detailed description of these curves).

The curves for minimum and average rupture strength can be used to calculate remaining tube life, as shown in Annex A.

The plot of the minimum rupture strength against the Larson-Miller parameter is included so that the rupture allowable stress can be determined for any design life. The curves shall not be used to determine rupture allowable stresses for temperatures higher than the limiting design metal temperatures shown in Table 4 and Figures E.1 to E.19 (Figures F.1 to F.19). Furthermore, the curves can give inaccurate rupture allowable stresses for tube life less than 20 000 h or greater than 200 000 h (see Clause H.3).

5.7 Limiting design metal temperature

The limiting design metal temperature for each heater-tube alloy is given in Table 4. The limiting design metal temperature is the upper limit of the reliability of the rupture strength data. Higher temperatures, i.e. up to 30 °C (50 °F) below the lower critical temperature, are permitted for short-term operating conditions, such as those that exist during steam-air decoking or regeneration. Operation at higher temperatures can result in changes in the alloy's microstructure. Lower critical temperatures for ferritic steels are shown in Table 4. Austenitic steels do not have lower critical temperatures. Other considerations can require lower operating-temperature limits, such as oxidation, graphitization, carburization, and hydrogen attack. These factors shall be considered when furnace tubes are designed.

Table 4 — Limiting design metal temperature for heater-tube alloys

Materials	Type or grade	Limiting design metal temperature		Lower critical temperature	
		°C	(°F)	°C	(°F)
Carbon steel	B	540	(1 000)	720	(1 325)
C-½Mo steel	T1 or P1	595	(1 100)	720	(1 325)
1¼Cr-½Mo steel	T11 or P11	595	(1 100)	775	(1 430)
2¼Cr-1Mo steel	T22 or P22	650	(1 200)	805	(1 480)
3Cr-1Mo steel	T21 or P21	650	(1 200)	815	(1 500)
5Cr-½Mo steel	T5 or P5	650	(1 200)	820	(1 510)
5Cr-½Mo-Si steel	T5b or P5b	705	(1 300)	845	(1 550)
7Cr-½Mo steel	T7 or P7	705	(1 300)	825	(1 515)
9Cr-1Mo steel	T9 or P9	705	(1 300)	825	(1 515)
9Cr-1Mo-V steel	T91 or P91	650 ^a	(1 200 ^a)	830	(1 525)
18Cr-8Ni steel	304 or 304H	815	(1 500)	—	—
16Cr-12Ni-2Mo steel	316 or 316H	815	(1 500)	—	—
16Cr-12Ni-2Mo steel	316L	815	(1 500)	—	—
18Cr-10Ni-Ti steel	321 or 321H	815	(1 500)	—	—
18Cr-10Ni-Nb steel	347 or 347H	815	(1 500)	—	—
Ni-Fe-Cr	Alloy 800H/800HT	985 ^a	(1 800 ^a)	—	—
25Cr-20Ni	HK40	1 010 ^a	(1 850 ^a)	—	—

^a This is the upper limit on the reliability of the rupture strength data (see Annex H); however, these materials are commonly used for heater tubes at higher temperatures in applications where the internal pressure is so low that rupture strength does not govern the design.

5.8 Allowable stress curves

Figures E.1 to E.19 provide the elastic allowable stress and the rupture allowable stress in SI units for most common heater-tube alloys. Figures F.1 to F.19 show the same data in USC units.

The sources for these curves are provided in Annex H. The figure number for each alloy is shown in Table 5.

Table 5 — Index to allowable stress curves

Steel type	Figure number	Alloy
Ferritic	E.1 (F.1)	Low-carbon steel (A 161, A 192)
	E.2 (F.2)	Medium-carbon steel (A 53B, A 106B, A 210A-1)
	E.3 (F.3)	C-½Mo
	E.4 ^a (F.4)	1¼Cr-½Mo
	E.5 ^a (F.5)	2¼Cr-1Mo
	E.6 ^a (F.6)	3Cr-1Mo
	E.7 ^a (F.7)	5Cr-½Mo
	E.8 (F.8)	5Cr-½Mo-Si
	E.9 ^a (F.9)	7Cr-½Mo
	E.10 ^a (F.10)	9Cr-1Mo
	E.11 (F.11)	9Cr-1Mo-V
Austenitic	E.12 (F.12)	18Cr-8Ni (304 and 304H)
	E.13 (F.13)	16Cr-12Ni-2Mo (316 and 316H)
	E.14 (F.14)	16Cr-12Ni-2Mo (316L)
	E.15 (F.15)	18Cr-10Ni-Ti (321)
	E.16 (F.16)	18Cr-10Ni-Ti (321H)
	E.17 (F.17)	18Cr-10Ni-Nb (347 and 347H)
	E.18 (F.18)	Ni-Fe-Cr (Alloy 800H/800HT)
	E.19 (F.19)	25Cr-20Ni (HK40)

^a Broken lines on these figures indicate the elastic allowable stresses for the A 200 grades. These figures do not show the yield strengths of the A 200 grades. The yield strengths of the A 200 grades are 83 % of the yield strengths shown. The tensile strengths, rupture allowable stresses, rupture strengths, and rupture exponents for the A 200 grades are the same as for the A 213 and A 335 grades.

6 Sample calculations

6.1 Elastic design

The following example illustrates the use of design equations for the elastic range. Suppose the following information is given (the USC unit conversions in parentheses are approximate):

Material = 18Cr-10Ni-Nb, type 347 stainless steel

$$D_o = 168,3 \text{ mm (6,625 in)}$$

$$p_{el} = 6,2 \text{ MPa gauge (900 psig)}$$

$$T_d = 425 \text{ °C (800 °F)}$$

$$\delta_{CA} = 3,2 \text{ mm (0,125 in)}$$

From Figure E.17 (SI units) or Figure F.17 (USC units):

$$\sigma_{el} = 124 \text{ MPa (18 100 psi)}$$

$$\sigma_y = 138 \text{ MPa (20 110 psi)}$$

Using Equations (2) and (3):

$$\delta_{\sigma} = \frac{(6,2)(168,3)}{2(125) + 6,2} = 4,0 \text{ mm}$$

$$\delta_{\min} = 4,0 + 3,2 = 7,2 \text{ mm}$$

In USC units:

$$\delta_{\sigma} = \frac{(900)(6,625)}{2(18\,250) + 900} = 0,159 \text{ in}$$

$$\delta_{\min} = 0,159 + 0,125 = 0,284 \text{ in}$$

This design calculation is summarized in the calculation sheet in Figure 4.

CALCULATION SHEET SI units (USC units)		
Heater _____	Plant _____	Refinery _____
Coil _____	Material Type 347	ASTM Spec. A 213
Calculation of minimum thickness	Elastic design	Rupture design
Outside diameter, mm (in)	$D_o = 168,3 (6,625)$	$D_o =$
Design pressure, gauge, MPa (psi)	$p_{el} = 6,2 (900)$	$p_r =$
Maximum or equivalent metal temperature, °C (°F)	$T_{\max} =$	$T_{\max} =$
Temperature allowance, °C (°F)	$T_A =$	$T_A =$
Design metal temperature, °C (°F)	$T_d = 425 (800)$	$T_d =$
Design life, h	—	$t_{DL} =$
Allowable stress at T_d , Figures E.1 to E.19 (Figures F.1 to F.19), MPa (psi)	$\sigma_{el} = 125 (18\,250)$	$\sigma_r =$
Stress thickness, Equation (2) or (4), mm (in)	$\delta_{\sigma} = 4,04 (0,159)$	$\delta_{\sigma} =$
Corrosion allowance, mm (in)	$\delta_{CA} = 3,2 (0,125)$	$\delta_{CA} =$
Corrosion fraction, Figure 1, $n = B =$	—	$f_{corr} =$
Minimum thickness, Equations (3) or (5), mm (in)	$\delta_{\min} = 7,2 (0,284)$	$\delta_{\min} =$

Figure 4 — Sample calculation for elastic design

6.2 Thermal-stress check (for elastic range only)

The thermal stress, σ_T , in the tube designed in accordance with 6.1 shall be checked using the following values for the variables in the equations given in Annex C:

$$\alpha = 1,81 \times 10^{-5} \text{ K}^{-1} (10,05 \times 10^{-6} \text{ R}^{-1})$$

(thermal expansion coefficient taken from Table C-3 of Reference [20]);

$$E = 1,66 \times 10^5 \text{ MPa} (24,1 \times 10^6 \text{ psi})$$

(modulus of elasticity taken from Table C-6 of Reference [20]);

$$\nu = 0,3 \quad (\text{Poisson's ratio value commonly used for steels});$$

$$q_o = 63,1 \text{ kW/m}^2 [20\,000 \text{ Btu/(h}\cdot\text{ft}^2)] \quad (\text{assumed heat-flux density});$$

$$\lambda_s = 20,6 \text{ W/(m}\cdot\text{K)} [11,9 \text{ Btu/(h}\cdot\text{ft}\cdot^\circ\text{F)}] \quad (\text{thermal conductivity}).$$

Using SI units in Equation (C.2):

$$X = \left[\frac{(1,81) (1,66)}{4 (1 - 0,3)} \right] \left[\frac{(63,1) (168,3)}{(20,6)} \right]$$

$$X = 553,2 \text{ MPa}$$

Using USC units in Equation (C.2):

$$X = \left[\frac{(10,05) (24,1)}{4 (1 - 0,3)} \right] \left[\frac{(20\,000) (6,625)}{(11,9) (12)} \right]$$

$$X = 8,026 \times 10^4 \text{ psi}$$

The thickness calculated in 6.1 is the minimum. The average thickness shall be used in the thermal-stress calculation. The average thickness (see 4.7) is calculated as follows:

In SI units:

$$(7,2) (1 + 0,14) = 8,2 \text{ mm}$$

In USC units:

$$(0,284) (1 + 0,14) = 0,324 \text{ in}$$

The actual inside diameter is calculated as follows:

In SI units:

$$D_i = 168,3 - 2(8,2) = 151,9 \text{ mm}$$

$$y = 168,3/151,9 = 1,108$$

where y is the ratio of outside diameter to actual inside diameter, D_o/D_i .

In USC units:

$$D_i = 6,625 - 2(0,324) = 5,977 \text{ in}$$

$$y = 6,625/5,977 = 1,108$$

The term in brackets in Equation (C.1) is calculated as follows:

$$\frac{2(1,108)^2}{(1,108)^2 - 1} \ln(1,108) - 1 = 0,106$$

Using Equation (C.1), the maximum thermal stress, $\sigma_{T\max}$, is calculated as follows:

$$\sigma_{T\max} = (553,2) (0,106)$$

$$\sigma_{T\max} = 58,6 \text{ MPa}$$

In USC units:

$$\sigma_{T\max} = (8,026 \times 10^4) (0,106)$$

$$\sigma_{T\max} = 8\,508 \text{ psi}$$

The limits for this stress for austenitic steels are given by Equations (C.4) and (C.6), in which the yield strength is 140 MPa (20 200 psi).

$$\sigma_{T,\text{lim}1} = [2,7 - 0,9(1,108)] (140)$$

$$\sigma_{T,\text{lim}1} = 238 \text{ MPa}$$

$$\sigma_{T,\text{lim}2} = (1,8) (140)$$

$$\sigma_{T,\text{lim}2} = 252 \text{ MPa}$$

In USC units:

$$\sigma_{T,\text{lim}1} = [2,7 - 0,9(1,108)] (20\,200)$$

$$\sigma_{T,\text{lim}1} = 34\,400 \text{ psi}$$

$$\sigma_{T,\text{lim}2} = (1,8) (20\,200)$$

$$\sigma_{T,\text{lim}2} = 36\,360 \text{ psi}$$

Since the maximum thermal stress is less than these limits, the design is acceptable.

If a thicker tube is specified arbitrarily (as Schedule 80S can be in this example), the actual average tube thickness shall be used in calculating the thermal stress and its limits as follows:

The inside diameter of a 6-in Schedule 80S tube is as follows:

$$D_i = 146,3 \text{ mm}$$

therefore

$$y = 168,3/146,3 = 1,150$$

In USC units:

$$D_i = 5,761 \text{ in}$$

$$y = 6,625/5,761 = 1,150$$

The term in brackets in Equation (C.1) is calculated as follows:

$$\frac{2 (1,150)^2}{(1,150)^2 - 1} \ln (1,150) - 1 = 0,146$$

Using Equation (C.1), the maximum thermal stress is calculated as follows:

$$\sigma_{T\max} = (553,2) (0,146)$$

$$\sigma_{T\max} = 80,9 \text{ MPa}$$

In USC units:

$$\sigma_{T\max} = (8,026 \times 10^4) (0,146)$$

$$\sigma_{T\max} = 11\,718 \text{ psi}$$

The average thickness of this tube is 11,0 mm (0,432 in), so the minimum thickness is calculated as follows:

$$\delta_{\min} = \frac{11,0}{1+0,14} = 9,6 \text{ mm}$$

In USC units:

$$\delta_{\min} = \frac{0,432}{1+0,14} = 0,379 \text{ in}$$

Using Equation (C.9), the stress is calculated as follows:

$$\sigma_{pm} = \frac{6,2}{2} \left(\frac{168,3}{9,6} - 1 \right) = 51,2 \text{ MPa}$$

In USC units:

$$\sigma_{pm} = \frac{900}{2} \left(\frac{6,625}{0,379} - 1 \right) = 7\,416 \text{ psi}$$

The thermal-stress limit based on the primary plus secondary stress intensity is calculated using Equation (C.14). Using the values above, this limit is calculated as follows:

$$\sigma_{T,\lim1} = (2,7 \times 140) - (1,15 \times 51,2)$$

$$\sigma_{T,\lim1} = 319,1 \text{ MPa}$$

In USC units:

$$\sigma_{T,\lim1} = (2,7 \times 20\,200) - (1,15 \times 7\,416)$$

$$\sigma_{T,\lim2} = 46\,010 \text{ psi}$$

The thermal-stress ratchet limit is calculated using Equation (C.19). In this case, the limit is as follows:

$$\sigma_{T,\lim2} = 4[(1,35 \times 140) - 51,2]$$

$$\sigma_{T,\lim2} = 551,2 \text{ MPa}$$

In USC units:

$$\sigma_{T,\lim2} = 4[(1,35 \times 20\,200) - 7\,416]$$

$$\sigma_{T,\lim2} = 79\,416 \text{ psi}$$

The thermal stress in the thicker tube is well below these limits.

6.3 Rupture design with constant temperature

A modification of the example in 6.1 illustrates how the design equations are used for the creep-rupture range. Suppose the tube described in 6.1 is designed for the following conditions:

$$T_d = 705\text{ °C (1 300 °F)}$$

$$t_{DL} = 100\,000\text{ h}$$

$$p_r = 5,8\text{ MPa gauge (840 psig)}$$

From Figure E.17 (SI units) or Figure F.17 (USC units).

$$\sigma_r = 37,3\text{ MPa (5 450 psi)}$$

Using Equation (4):

In SI units:

$$\delta_\sigma = \frac{(5,8)(168,3)}{2(37,3) + 5,8} = 12,1\text{ mm}$$

In USC units:

$$\delta_\sigma = \frac{(840)(6,625)}{2(5\,450) + 840} = 0,474\text{ in}$$

From this:

In SI units:

$$B = \frac{3,2}{12,1} = 0,264$$

In USC units:

$$B = \frac{0,125}{0,474} = 0,264$$

From Figure E.17 (SI units) or Figure F.17 (USC units)

$$n = 4,4$$

With these values for B and n , use Figure 1 to obtain the following corrosion fraction:

$$f_{\text{corr}} = 0,558$$

Hence, using Equation (5):

In SI units:

$$\delta_{\text{min}} = 12,1 + (0,558 \times 3,2)$$

$$\delta_{\text{min}} = 13,9\text{ mm}$$

In USC units:

$$\delta_{\min} = 0,474 + (0,558 \times 0,125)$$

$$\delta_{\min} = 0,544 \text{ in}$$

To confirm that this is an appropriate design, the elastic design is checked using the elastic design pressure instead of the rupture design pressure. Using Equations (2) and (3) with the conditions given above:

In SI units:

$$\sigma_{\text{el}} = 113 \text{ MPa}$$

$$\delta_{\sigma} = \frac{(6,2)(168,3)}{2(113) + 6,2} = 4,5 \text{ mm}$$

$$\delta_{\min} = 4,5 + 3,2 = 7,7 \text{ mm}$$

In USC units:

$$\sigma_{\text{el}} = 16\,400 \text{ psi}$$

$$\delta_{\sigma} = \frac{(900)(6,625)}{2(16\,400) + 900} = 0,177 \text{ in}$$

$$\delta_{\min} = 0,177 + 0,125 = 0,302 \text{ in}$$

Since $\delta_{\min}$ based on rupture design is greater, it governs the design. This design calculation is summarized on the calculation sheet in Figure 5.

CALCULATION SHEET SI units (USC units)		
Heater _____	Plant _____	Refinery _____
Coil _____	Material Type 347	ASTM Spec. A 213
Calculation of minimum thickness	Elastic design	Rupture design
Outside diameter, mm (in)	$D_o = 168,3 (6,625)$	$D_o = 168,3 (6,625)$
Design pressure, gauge, MPa (psi)	$p_{\text{el}} = 6,2 (900)$	$p_r = 5,8 (840)$
Maximum or equivalent metal temperature, °C (°F)	$T_{\text{max}} =$	$T_{\text{max}} =$
Temperature allowance, °C (°F)	$T_A =$	$T_A =$
Design metal temperature, °C (°F)	$T_d = 705 (1\,300)$	$T_d = 705 (1\,300)$
Design life, h	—	$t_{\text{DL}} = 100\,000$
Allowable stress at T_d , Figures E.1 to E.19 (Figures F.1 to F.19), MPa (psi)	$\sigma_{\text{el}} = 113 (16\,400)$	$\sigma_r = 37,3 (5\,450)$
Stress thickness, Equation (2) or (4), mm (in)	$\delta_{\sigma} = 4,5 (0,177)$	$\delta_{\sigma} = 12,0 (0,474)$
Corrosion allowance, mm (in)	$\delta_{\text{CA}} = 3,18 (0,125)$	$\delta_{\text{CA}} = 3,18 (0,125)$
Corrosion fraction, Figure 1, $n = 4,4$; $B = 0,264$	—	$f_{\text{corr}} = 0,558$
Minimum thickness, Equation (3) or (5), mm (in)	$\delta_{\min} = 7,67 (0,302)$	$\delta_{\min} = 13,82 (0,544)$

Figure 5 — Sample calculation for rupture design (constant temperature)

6.4 Rupture design with linearly changing temperature

Suppose the tube described in 6.3 operates in a service for which the estimated tube metal temperature varies from 635 °C (1 175 °F) at the start of run to 690 °C (1 275 °F) at the end of run. Assume that the run lasts a year, during which the thickness changes by about 0,33 mm (0,013 in).

Assume that the initial minimum thickness is 8,0 mm (0,315 in); therefore, using Equation (1), the initial stress is as follows:

In SI units:

$$\sigma_0 = \frac{5,8}{2} \left(\frac{168,3}{8,0} - 1 \right) = 58,1 \text{ MPa}$$

In USC units:

$$\sigma_0 = \frac{840}{2} \left(\frac{6,625}{0,315} - 1 \right) = 8\,413 \text{ psi}$$

At the start-of-run temperature, $n_0 = 4,8$. From Table 3, A is $1,23 \times 10^6$ MPa ($1,79 \times 10^8$ psi). The parameters for the temperature fraction are, therefore, as follows:

In SI units:

$$V = 4,8 \left(\frac{55}{908} \right) \ln \left(\frac{1,23 \times 10^6}{58,1} \right) = 2,9$$

$$N = 4,8 \left(\frac{0,33}{8,0} \right) = 0,2$$

In USC units:

$$V = 4,8 \left(\frac{100}{1635} \right) \ln \left(\frac{1,79 \times 10^8}{8\,413} \right) = 2,9$$

$$N = 4,8 \left(\frac{0,013}{0,315} \right) = 0,2$$

From Figure 2, $f_T = 0,62$, and the equivalent temperature is calculated using Equation (6) as follows:

In SI units:

$$T_{eq} = 635 + (0,62 \times 55) = 669 \text{ °C}$$

In USC units:

$$T_{eq} = 1\,175 + (0,62 \times 100) = 1\,237 \text{ °F}$$

A temperature allowance of 15 °C (25 °F) is added to yield a design temperature of 684 °C (1 262 °F), which is rounded up to 685 °C (1 265 °F). Using this temperature to carry out the design procedure illustrated in 6.3 yields the following:

In SI units:

$$\delta_{\sigma} = 9,9 \text{ mm}$$

$$\delta_{\min} = 9,9 + (0,572 \times 3,2)$$

$$\delta_{\min} = 11,7 \text{ mm}$$

In USC units:

$$\delta_{\sigma} = 0,388 \text{ in}$$

$$\delta_{\min} = 0,388 + (0,572 \times 0,125)$$

$$\delta_{\min} = 0,460 \text{ in}$$

This thickness is different from the 8,0 mm (0,315 in) thickness that was initially assumed. Using this thickness, the initial stress is calculated as follows:

In SI units:

$$\sigma_0 = \frac{5,7}{2} \left(\frac{168,3}{11,7} - 1 \right) = 38,8 \text{ MPa}$$

In USC units:

$$\sigma_0 = \frac{840}{2} \left(\frac{6,625}{0,460} - 1 \right) = 5\,629 \text{ psi}$$

With this stress, the temperature-fraction parameters V and N become the following:

In SI units:

$$V = 4,8 \left(\frac{55}{908} \right) \ln \left(\frac{1,23 \times 10^6}{38,8} \right) = 3,0$$

$$N = 4,8 \left(\frac{0,33}{11,7} \right) = 0,1$$

In USC units:

$$V = 4,8 \left(\frac{100}{1\,635} \right) \ln \left(\frac{1,79 \times 10^8}{5\,629} \right) = 3,0$$

$$N = 4,8 \left(\frac{0,013}{0,460} \right) = 0,1$$

Using these values in Figure 2, $f_T = 0,62$, the value that was determined in the first calculation. Since the temperature fraction did not change, further iteration is not necessary. This design calculation is summarized in the calculation sheet in Figure 6.

CALCULATION SHEET SI units (USC units)		
Heater _____	Plant _____	Refinery _____
Coil _____	Material Type 347	ASTM Spec. A 213
Calculation of minimum thickness	Elastic design	Rupture design
Outside diameter, mm (in)	$D_o =$	$D_o = 168,3 (6,625)$
Design pressure, gauge, MPa (psi)	$p_{el} =$	$p_r = 5,8 (840)$
Maximum or equivalent metal temperature, °C (°F)	$T_{eq} =$	$T_{eq} = 669 (1\ 237)$
Temperature allowance, °C (°F)	$T_A =$	$T_A = 15 (25)$
Design metal temperature, °C (°F)	$T_d =$	$T_d = 685 (1\ 265)$
Design life, h	—	$t_{DL} = 100\ 000$
Allowable stress at T_d , Figures E.1 to E.19, (Figures F.1 to F.19) MPa (psi)	$\sigma_{el} =$	$\sigma_r = 46,6 (6\ 750)$
Stress thickness, Equation (2) or (4), mm (in)	$\delta_\sigma =$	$\delta_\sigma = 9,85 (0,388)$
Corrosion allowance, mm (in)	$\delta_{CA} =$	$\delta_{CA} = 3,18 (0,125)$
Corrosion fraction, Figure 1, $n = 4,5$; $B = 0,322$	—	$f_{corr} = 0,572$
Minimum thickness, Equation (3) or (5), mm (in)	$\delta_{min} =$	$\delta_{min} = 11,68 (0,460)$
Calculation of equivalent tube metal temperature		
Duration of operating period, years	$t_{op} =$	$1,0$
Metal temperature, start of run, °C (°F)	$T_{sor} =$	$635 (1\ 175)$
Metal temperature, end of run, °C (°F)	$T_{eor} =$	$690 (1\ 275)$
Temperature change during operating period, K (°R)	$\Delta T^* =$	$55 (100)$
Metal absolute temperature, start of run, K (°R)	$T_{sor}^* =$	$908 (1\ 635)$
Thickness change during operating period, mm (in)	$\Delta\delta =$	$0,33 (0,013)$
Assumed initial thickness, mm (in)	$\delta_0 =$	$8,00 (0,315)$
Corresponding initial stress, Equation (1), MPa (psi)	$\sigma_0 =$	$58,1 (8\ 413)$
Material constant, Table 3, MPa (psi)	$A =$	$1,23 \times 10^6 (1,79 \times 10^8)$
Rupture exponent at T_{sor} , Figures E.1 to E.19 (Figures F.1 to F.19)	$n_0 =$	$4,8$
Temperature fraction, Figure 2, $V = 2,9$; $N = 0,2$	$f_T =$	$0,62$
Equivalent metal temperature, Equation (6), °C (°F)	$T_{eq} =$	$669 (1\ 237)$

Figure 6 — Sample calculation for rupture design (changing temperature)

Annex A (informative)

Estimation of remaining tube life

A.1 General

Figures E.1 to E.19 (Figures F.1 to F.19) and the considerations made in Annex G have applications other than for the design of new tubes. They can also be used to help answer re-rating and retirement questions about existing tubes that have operated in the creep-rupture range. This annex describes how tube damage and remaining life can be estimated. Because of the uncertainties involved in these calculations, decisions about tube retirement should not be based solely on the results of these calculations. Other factors such as tube thickness or diameter-strain measurements should be primary considerations in decisions about tube retirement.

There are three primary areas of uncertainty in these calculations. First, it is necessary to estimate the accumulated tube damage (the life fraction used up) based on the operating history, i.e. the influence from the operating pressure, the tube-metal temperature, and the corrosion rate, of the tube. The uncertainties in these factors, particularly the temperature, can have a significant effect on the estimate. Second, knowledge of the actual rupture strength of a given tube is not precise. The example calculation in Clause A.2 demonstrates the effects of this uncertainty. Finally, it is necessary to consider the tube-damage rule as described in Clause G.2. However, as mentioned in Clause G.2, the limitations of this hypothesis are not well understood. In spite of all these uncertainties, the estimation that is made using the procedure described in this annex can provide information that assists in making decisions about tube re-rating and retirement.

The essence of this calculation procedure can be outlined as follows. The operating history is divided into periods of time during which the pressure, metal temperature, and corrosion rate are assumed constant. For each of these periods, the life fraction used up is calculated. The sum of these calculated life fractions is the total accumulated tube damage. The fraction remaining is calculated by subtracting this sum from unity. Finally, the remaining life fraction is transformed into an estimate of the expected life at specified operating conditions.

A more detailed life-assessment evaluation for heater tubes operating in the creep-rupture range can be found in API RP 579:2000, Appendix F.

A.2 Estimation of accumulated tube damage

Since the concepts required to estimate damage are developed elsewhere in this International Standard, they are not repeated here. The calculation procedure can best be explained by working through an example. For this example, the following conditions are assumed:

Material:	18Cr-10Ni-Nb (type 347) stainless steel;
Outside diameter:	168,3 mm (6,625 in);
Initial minimum thickness:	6,8 mm (0,268 in).

It is also assumed that the operating history of the tube can be approximated as shown in Table A.1. (The SI conversions are approximate.)