

Petroleum and natural gas industries — Calculation of heater-tube thickness in petroleum refineries

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National foreword

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Petroleum and natural gas industries — Calculation of heater-tube thickness in petroleum refineries

*Industries du pétrole et du gaz naturel — Calcul de l'épaisseur des tubes
de fours de raffineries de pétrole*



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Foreword

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ISO 13704 was prepared by Technical Committee ISO/TC 67, *Materials, equipment and offshore structures for petroleum and natural gas industries*, Subcommittee SC 6, *Processing equipment and systems*.

Annexes C, E and F form an integral part of this International Standard. Annexes A, B, D, G and H are for information only.

Introduction

This International Standard is based on API standard 530 ^[30], fourth edition, October 1996.

Petroleum and natural gas industries — Calculation of heater-tube thickness in petroleum refineries

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 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 International Standard, 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

corrosion allowance

δ_{CA}

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

2.3

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.

2.4

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 2.7) and adding an appropriate temperature allowance (see 2.15). 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 higher than the design metal temperature.

2.5

elastic allowable stress

σ_{el}

allowable stress for the elastic range (see 5.2)

NOTE See 3.2.3 for information about tubes that have longitudinal welds.

2.6
elastic design pressure

p_{el}
maximum pressure that the heater coil will sustain for short periods of time

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

2.7
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 linearly changing metal temperature (see 4.8)

2.8
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.

2.9
minimum thickness

δ_{min}
minimum required thickness of a new tube, taking into account all appropriate allowances [see equation (5)]

2.10
outside diameter

D_o
outside diameter of a new tube

2.11
rupture allowable stress

σ_r
allowable stress for the creep-rupture range (see 4.4)

NOTE See 3.2.3 for information about tubes that have longitudinal welds.

2.12
rupture design pressure

p_r
maximum operating pressure that the coil section will sustain during normal operation

2.13
rupture exponent

n
parameter used for design in the creep-rupture range

See figures in annexes E and F.

2.14
stress thickness

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

2.15**temperature allowance** T_A

part of the design metal temperature that is included for process- or flue-gas maldistribution, 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 2.4).

3 General design information**3.1 Information required**

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

- a) the design life of the heater tube;
- b) whether the equivalent-temperature concept is to be applied, and if so, furnish the operating conditions at the start and at the end of the run;
- c) the temperature allowance, if any;
- d) the 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:

- f) a design life equal to 100 000 h;
- g) a design metal temperature based on the maximum metal temperature (the equivalent-temperature concept shall not apply);
- h) a temperature allowance equal to 15 °C (25 °F);
- i) the corrosion fraction given in Figure 1;
- j) the 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 might result in small permanent strains in some applications; however, these small strains will 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 RP 941 [15].

3.2.3 These design procedures have been developed for seamless tubes. When they are applied to tubes that have a longitudinal weld, the allowable stress values should be multiplied by the appropriate joint efficiency factor. Joint efficiency factors shall not be applied to circumferential welds.

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 may 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 The design loading includes only internal pressure. 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 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 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.

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 will creep, or deform 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 will eventually fail due to creep rupture, although no corrosion or oxidation mechanism is active. For the steel operating at the lower temperature, the effects of creep will be non-existent or negligible. Experience indicates that in this case the tube will last 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.

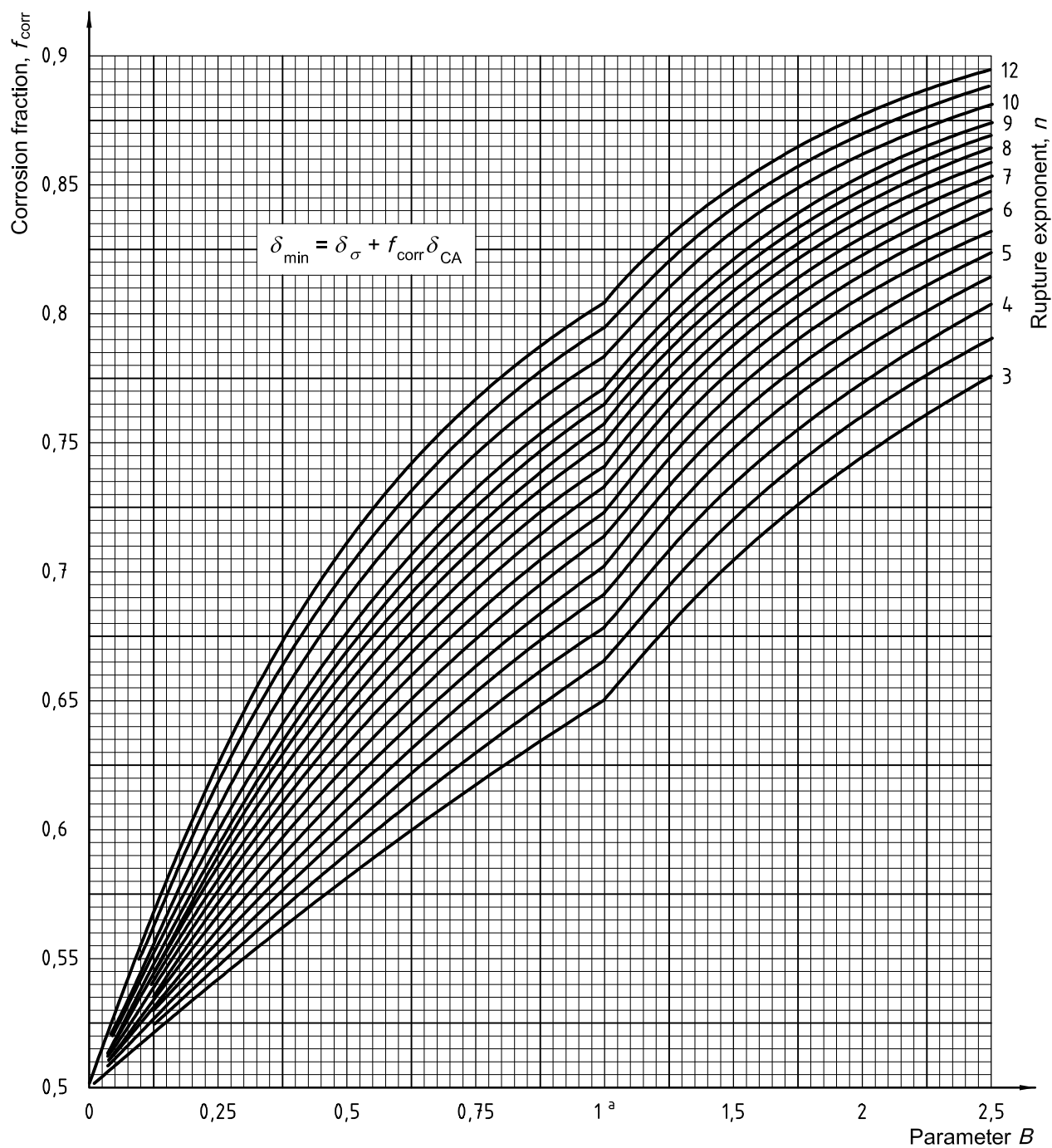
The rupture design pressure is usually less than the elastic design pressure. The characteristic that differentiates these two pressures is the relative length of time over which they are sustained. The rupture design pressure is a long-term loading condition that remains relatively uniform over a period 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 this clause are summarized in Table 2.



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

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

D_o is the outside diameter

σ_r is the rupture allowable stress

δ_{CA} is the corrosion allowance

p_r is the rupture design pressure

n is the rupture exponent

^a Note change of scale.

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 will produce 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] and [19]. The mean diameter equation for stress is as follows:

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

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 follows:

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

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

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 31.

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 as follows:

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

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

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;

where

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

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.

This design equation is 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 will govern the design. In this temperature range, both the elastic and rupture designs are to be applied. 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 will 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 $\left(+28\right)^0$ % tolerance, a tube that is purchased to a 12,7 mm (0,500 in) minimum-thickness specification will have 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 will be 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 temperature. For applications in which there is a significant difference between start-of-run and end-of-run metal temperatures, a design based on the maximum temperature might be excessive, since the actual operating temperature will usually be 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 below. A tube operating at the equivalent tube metal temperature will sustain 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 :

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

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

where

n_0 is the rupture exponent at T_{sor} ;

ΔT^* ($= T_{\text{eor}} - T_{\text{sor}}$) is the temperature change, expressed in kelvins (degrees Rankine), during operating period, K ($^{\circ}\text{R}$);

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

$\ln$ is the natural logarithm;

$\Delta \delta = \phi_{\text{corr}} t_{\text{op}}$ is the change in thickness, 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 start of 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 G.5.

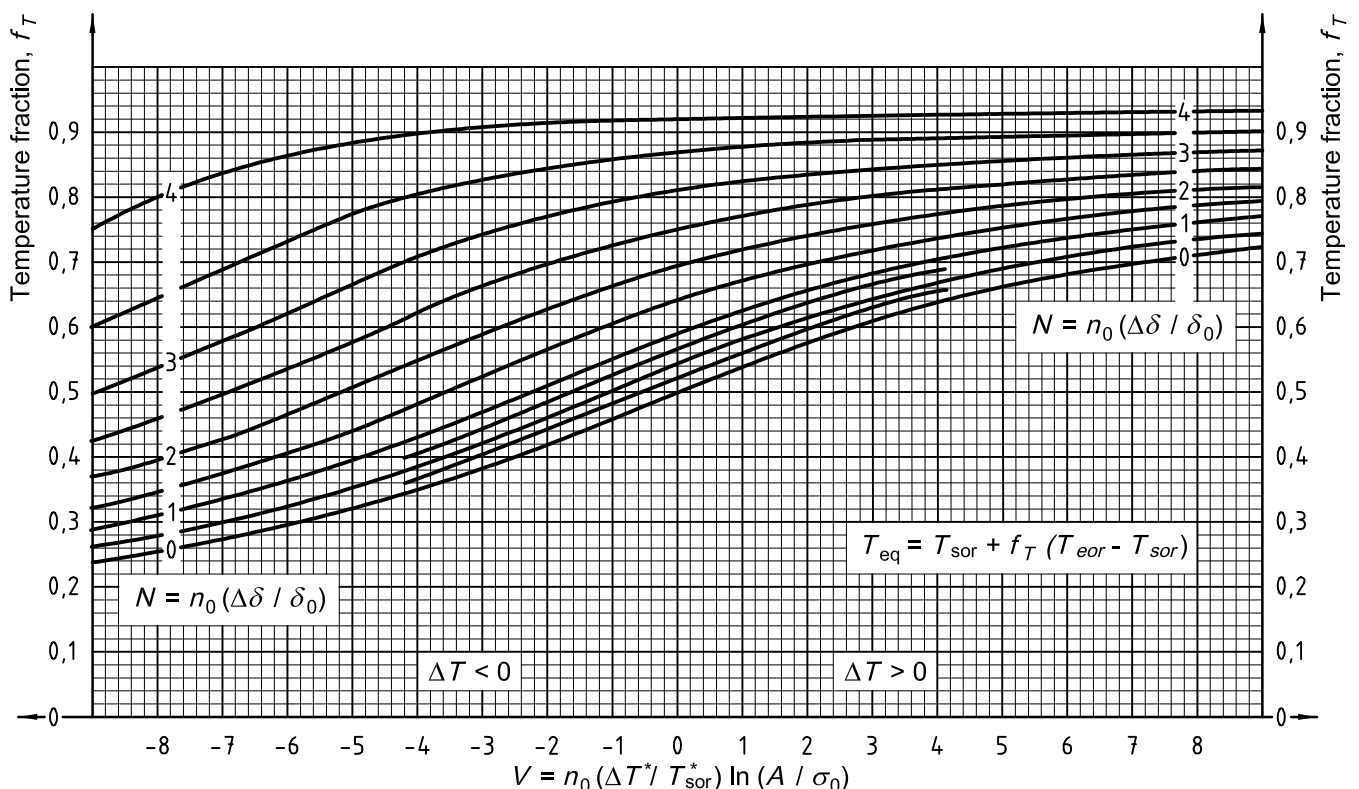


Figure 2 — Temperature fraction

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

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

 n is the rupture exponent at the design metal temperature**Equivalent tube metal temperature:**

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

 ΔT^* ($= T_{eor} - T_{sor}$) is the temperature change, expressed in kelvins (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

where

 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 A	
		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.			

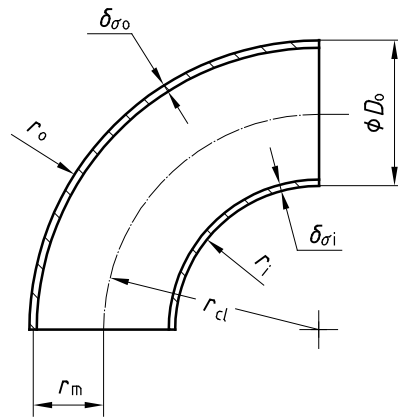
The temperature fraction and the equivalent temperature shall be calculated for the first operating cycle. In applications that involve very high corrosion rates, the temperature fraction for the last cycle will be greater than that for the first. In such cases, the calculation of the temperature fraction 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 might be necessary to arrive at the design. (See the sample calculation in 6.4.).

4.9 Return bends and elbows

The following design procedure shall be applied to austenitic stainless steel return bends and elbows (see Figure 3) located in the firebox and operating in the elastic range. In this situation, the allowable stress does not vary much with temperature. This design procedure may also be applied in other situations, if applicable.



r_o = outer radius

r_i = inner radius

Figure 3 — Return bend and elbow geometry

The stress variations in a return bend or elbow 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. In the situation defined above, the minimum thickness at the inner radius might need 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, expressed in megapascals (pounds per square inch), along the inner radius of the bend, σ_i , is given by:

$$\sigma_i = \frac{2r_{cl} - r_m}{2(r_{cl} - r_m)} \sigma \quad (7)$$

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, expressed in megapascals (pounds per square inch), along the outer radius σ_o is given by:

$$\sigma_o = \frac{2r_{cl} + r_m}{2(r_{cl} + r_m)} \sigma \quad (8)$$

Using the approximation that r_m is almost equal to $D_o/2$, equation (7) can be solved for the stress thickness at the inner radius. For elastic design the stress thickness is given by equation (9).

$$\delta_{\sigma_i} = \frac{D_o p_{el}}{2N_i \sigma_{el} + p_{el}} \quad (9)$$

where

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

$$N_i = \frac{4 \frac{r_{cl}}{D_o} - 2}{4 \frac{r_{cl}}{D_o} - 1} \quad (10)$$

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

The design metal temperature shall be the estimated temperature at the inner radius plus an appropriate temperature allowance.

Using the approximation given above, equation (8) can be solved for the stress thickness at the outer radius. For elastic design the stress thickness is as follows:

$$\delta_{\sigma o} = \frac{D_o p_{el}}{2N_o \sigma_{el} + p_{el}} \quad (11)$$

where

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

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

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

The design metal temperature shall be the estimated temperature at the outer radius plus an appropriate temperature allowance.

The minimum thickness at the inside radius, $\delta_{\sigma i}$, and outside radius, $\delta_{\sigma o}$, shall be calculated using equation (9) and equation (11). 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.

This design procedure is for return bends and elbows located in the firebox that may operate at temperatures close to that of the tubes. This procedure might not be applicable to these fittings if they are located in header boxes since they will operate at lower temperatures. Other considerations, such as hydrostatic test pressure, could govern the design of fittings located in header boxes.

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 in annex E (SI units) and Figures F.1 to F.19 in annex F (US customary 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 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 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 follows.

When T_d is expressed in degrees Celsius:

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

When T_d is expressed in degrees Fahrenheit:

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

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

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 could give inaccurate rupture allowable stresses for times less than 20 000 h or greater than 200 000 h (see H.3).

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

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 may 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 US customary 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

	Figure number	Alloy
Ferritic steels	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 steels	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 US customary unit conversions in parentheses are approximate):

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

D_o = 168,3 mm (6,625 in)

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

T_d = 425 °C (800 °F)

δ_{CA} = 3,2 mm (0,125 in)

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

σ_{el} = 125 MPa (18 250 psi)

σ_y = 140 MPa (20 200 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 US customary 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 (US customary 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, equation (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 according to 6.1 shall be checked using the equations given in annex C as follows:

$$\alpha = 1,81 \times 10^{-5} \text{ K}^{-1} (10,05 \times 10^{-6} \text{ }^{\circ}\text{R}^{-1}) \text{ (thermal expansion coefficient taken from Table C-3 of reference [20] of the Bibliography);}$$

$$E = 1,66 \times 10^5 \text{ MPa} (24,1 \times 10^6 \text{ psi}) \text{ (modulus of elasticity taken from Table C-6 of reference [20] of the Bibliography);}$$

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

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

$$\lambda_s = 20,6 \text{ W/(m}\cdot\text{K)} [11,9 \text{ Btu/(h}\cdot\text{ft}\cdot^\circ\text{F)}] \text{ (thermal conductivity taken from Table 1 of reference [21]).}$$

Using equation (C.2):

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

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

In US customary units:

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

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

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:

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

In US customary units:

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

The actual inside diameter is calculated as follows:

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

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

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

In US customary 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:

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

$$= 58,6 \text{ MPa}$$

In US customary units:

$$\begin{aligned}\sigma_{T\max} &= (8,026 \times 10^4) (0,106) \\ &= 8\,508 \text{ psi}\end{aligned}$$

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).

$$\begin{aligned}\sigma_{T\lim1} &= [2,7 - 0,9(1,108)] (140) \\ &= 238 \text{ MPa}\end{aligned}$$

$$\begin{aligned}\sigma_{T\lim2} &= (1,8) (140) \\ &= 252 \text{ MPa}\end{aligned}$$

In US customary units:

$$\begin{aligned}\sigma_{T\lim1} &= [2,7 - 0,9(1,108)] (20\,200) \\ &= 34\,400 \text{ psi}\end{aligned}$$

$$\begin{aligned}\sigma_{T\lim2} &= (1,8) (20\,200) \\ &= 36\,360 \text{ psi}\end{aligned}$$

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

If a thicker tube is specified arbitrarily (as Schedule 80S might 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 US customary 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:

$$\begin{aligned}\sigma_{T\max} &= (553,2) (0,146) \\ &= 80,9 \text{ MPa}\end{aligned}$$

In US customary units:

$$\begin{aligned}\sigma_{T\max} &= (8,026 \times 10^4) (0,146) \\ &= 11\,718 \text{ psi}\end{aligned}$$

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 US customary units:

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

Using equation (C.7), 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 US customary 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.9). Using the values above, this limit is calculated as follows:

$$\begin{aligned}\sigma_{T\lim1} &= (2,7 \times 140) - (1,15 \times 51,2) \\ &= 319,1 \text{ MPa}\end{aligned}$$

In US customary units:

$$\begin{aligned}\sigma_{T\lim1} &= (2,7 \times 20\,200) - (1,15 \times 7\,416) \\ &= 46\,010 \text{ psi}\end{aligned}$$

The thermal-stress ratchet limit is calculated using equation (C.12). In this case, the limit is as follows:

$$\begin{aligned}\sigma_{T\lim2} &= 4[(1,35 \times 140) - 51,2] \\ &= 551,2 \text{ MPa}\end{aligned}$$

In US customary units:

$$\begin{aligned}\sigma_{T\lim2} &= 4[(1,35 \times 20\,200) - 7\,416] \\ &= 79\,416 \text{ psi}\end{aligned}$$

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 to be designed for the following conditions:

$$T_d = 705 \text{ }^{\circ}\text{C} \text{ (1 300 }^{\circ}\text{F)}.$$

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

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

From Figure E.17 (SI units) or Figure F.17 (US customary units).

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

Using equation (4):

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

In US customary units:

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

From this,

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

In US customary units:

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

From Figure E.17 (SI units) or Figure F.17 (US customary units)

$$n = 4,4$$

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

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

Hence, using equation (5),

$$\begin{aligned} \delta_{\min} &= 12,1 + (0,558 \times 3,2) \\ &= 13,9 \text{ mm} \end{aligned}$$

In US customary units:

$$\begin{aligned} \delta_{\min} &= 0,474 + (0,558 \times 0,125) \\ &= 0,544 \text{ in} \end{aligned}$$

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:

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

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

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

In US customary 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 (US customary 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, MPa (psi) gauge	$p_{\text{el}} = 6,2 (900)$	$p_r = 5,8 (840)$	
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 will operate 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 will change 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 will be as follows:

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

In US customary 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:

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

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

In US customary 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:

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

In US customary 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:

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

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

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

In US customary units:

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

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

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

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

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

In US customary 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:

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

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

In US customary units:

$$V = 4,8 \left(\frac{100}{1635} \right) \ln \left(\frac{1,79 \times 10^8}{5\,629} \right) = 3,0$$

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

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 (US customary 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, MPa (psi) gauge	$p_{el} =$	$p_r = 5,8 (840)$
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 A.2 demonstrates the effects of this uncertainty. Finally, it is necessary to consider the tube damage rule as described in G.2. However, as mentioned in 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 might provide information that will assist 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 in 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.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.)

The operating periods need not be of uniform length. In an actual heater, neither the operating pressure nor the metal temperature is uniform. Nonetheless, for this calculation, they are assumed to be uniform during each period. The values chosen for each period should represent typical values. The choice of the length of the operating period will depend on the extent of the variation of the pressure and temperature.

It is necessary to approximate the operating history for the tube thickness. This history can usually be developed from thickness measurements made before the initial start-up and during routine heater-tube inspections. For all of these estimates, it is assumed that the outside diameter remains constant.

Table A.1 — Approximation of the operating history

Operation period	Duration ^a	Operating gauge pressure		Tube metal temperature		Minimum thickness			
						Beginning		End	
	a	MPa	(psi)	°C	(°F)	mm	(in)	mm	(in)
1	1,3	3,96	(575)	649	(1 200)	6,81	(0,268)	6,40	(0,252)
2	0,6	4,27	(620)	665	(1 230)	6,40	(0,252)	6,20	(0,244)
3	2,1	4,07	(590)	660	(1 220)	6,20	(0,244)	5,51	(0,217)
4	2,0	4,34	(630)	665	(1 230)	5,51	(0,217)	4,83	(0,190)

^a "a" is the international unit symbol for "year".

This information can be used to calculate the life fractions shown in Table A.2.

For tube undergoing corrosion, an equation similar to equation (G.8) can be developed for the life fraction; however, this is not necessary since sufficient accuracy can be achieved for this calculation by using the average stress for each period (that is, the average of the stress at the beginning and at the end of the operating period).

The minimum and average Larson-Miller values in Table A.2 are determined from the average stress using the Larson-Miller parameter curves for minimum and average rupture strength in Figures E.1 to E.17. For this example, Figure E.17 was used.

With these Larson-Miller values and the metal temperature for each period, the expression for the Larson-Miller parameter was solved for the rupture time. This expression is at the top of Figures E.1 to E.19. Since this expression gives the rupture time in hours, the value needs converting to years. The resulting times based on the minimum rupture strength and the average rupture strength are shown in Table A.2.

The following example illustrates how to calculate the minimum-strength rupture time, t_r , for the first operating period. The equation to be solved is as follows:

$$19,06 = (649 + 273)(15 + \lg t_r) \times 10^{-3}$$

In US customary units:

$$34,32 = (1\,200 + 460)(15 + \lg t_r) \times 10^{-3}$$

or

$$\lg t_r = 5,67$$

$$t_r = 4,73 \times 10^5 \text{ h}$$

$$= 54,0 \text{ a}$$

The life fractions are simply the duration of the operating period divided by the rupture time that corresponds to that period. Using the minimum-strength rupture time calculated above, the fraction for the first line in Table A.2 is $1,3/54,0 = 0,02$. The accumulated damage is the sum of the fractions.

The effect of the uncertainty about the rupture strength is evident in Table A.2. If the actual rupture strength of this tube is in the lower part of the scatter band (near the minimum rupture strength), then 64 % of the tube life has been used. If the actual strength is in the middle of the scatter band (near the average rupture strength), then only 23 % of the tube life has been used. If the actual rupture strength is higher, even less of the tube life has been used.

The effect of the uncertainty about the operating temperature can also be evaluated. Suppose the actual metal temperature of this tube were 5 °C (9 °F) higher than that shown in Table A.1. To estimate the effect of this difference, the life-fraction calculations in Table A.2 have been made with the slightly higher temperature. The corresponding accumulated damage fractions are 0,81 and 0,28, respectively. These should be compared with the values 0,64 and 0,23, which were calculated first.

Table A.2 — Life fractions for each period

Operating period	Average stress		Larson-Miller values				Rupture time based on minimum strength		Rupture time based on average strength	
			minimum		average					
	MPa	psi	°C	(°F)	°C	(°F)	a	Life fraction	a	Life fraction
1	48,52	(7 045)	19,06	(34,32)	19,48	(35,09)	54,0	0,02	154,8	0,01
2	54,91	(7 973)	18,83	(33,90)	19,25	(34,66)	13,1	0,05	35,8	0,02
3	56,66	(8 213)	18,77	(33,80)	19,19	(34,55)	15,0	0,14	42,1	0,05
4	68,78	(9 985)	18,41	(33,15)	18,83	(33,90)	4,7	0,43	13,1	0,15
					Accumulated damage =			0,64		0,23

A.3 Estimation of remaining tube life

As in A.2, this calculation procedure is best explained using an example. The example used is summarized in Tables A.1 and A.2. The life fraction remaining for this tube is as follows:

Minimum rupture strength: $1 - 0,64 = 0,36$

Average rupture strength: $1 - 0,23 = 0,77$

These fractions should be converted to the expected life under the specified operating conditions.

The following related questions can be asked at this point:

- What is the estimated life at a given operating pressure, metal temperature, and corrosion rate?
- For a specified operating pressure and corrosion rate, what temperature limit should be imposed for the tube to last a minimum period of time?
- How much should the operating pressure or metal temperature be reduced to extend the expected life by a given percentage?

Not all of these questions are answered in this annex, but the method used to develop the answers should be clear from the following example.

For this example, the expected operating conditions are as follows:

Operating gauge pressure = 4,27 MPa (620 psi)

Metal temperature = 660 °C (1 220 °F)

Corrosion rate = 0,33 mm/a (0,013 in/a)

From these values a table of future-life fractions can be developed as shown in Table A.3 for the minimum rupture strength and in Table A.4 for the average rupture strength. As before, the average stress is the average of the stresses at the beginning and end of each operating period.

Since the tube in the example is undergoing corrosion, the life estimation should be calculated in steps. For this example, a 1-year step was used. As can be seen from the two tables, the estimated life of this tube is between 1,5 a and 4,5 a. If the rupture strength were in the upper part of the scatter band (above the average rupture strength), the estimated life would be even longer.

For tubes that are not undergoing corrosion, estimating the life is easier. The rupture life is calculated as above from the anticipated stress and temperature. The estimated remaining life is simply the fraction remaining multiplied by the rupture life. In these cases, tables such as Tables A.3 and A.4 are not required.

The example given above describes a way to answer Question a), posed at the beginning of this clause: What is the estimated life for a specified set of operating conditions? Question b), concerning the temperature limit that should be imposed for a specified pressure, corrosion rate, and minimum life, can be answered as follows. The pressure and corrosion rate can be used to calculate an average stress from which a Larson-Miller value can be found using the curves in Figures E.1 to E.19. With this value and a rupture life calculated by dividing the required life by the remaining life fraction, the Larson-Miller parameter equation can be solved for the maximum temperature. The other questions can be answered in similar ways.

Table A.3 — Future life fractions, minimum strength

Time a	Minimum thickness		Average stress		Minimum Larson-Miller value		Rupture time a	Fraction	Remaining Fraction
	mm	(in)	MPa	(psi)	°C	(°F)			
0	4,83	(0,190)	—	—	—	—	—	—	0,36
1	4,50	(0,177)	74,99	(10 896)	18,25	(32,85)	4,1	0,24	0,12
1,5	4,34	(0,171)	79,19	(11 497)	18,14	(32,66)	3,1	0,16	−0,04

Table A.4 — Future life fractions, average strength

Time a	Minimum thickness		Average stress		Minimum Larson-Miller value		Rupture time a	Fraction	Remaining fraction
	mm	(in)	MPa	(psi)	°C	(°F)			
0	4,83	(0,190)	—	—	—	—	—	—	0,77
1	4,50	(0,177)	74,99	(10 896)	18,66	(33,60)	11,4	0,09	0,68
2	4,17	(0,164)	80,87	(11 753)	18,53	(33,35)	8,2	0,12	0,56
3	3,84	(0,151)	87,74	(12 752)	18,37	(33,07)	5,5	0,18	0,38
4	3,51	(0,138)	95,84	(13 932)	18,22	(32,80)	3,8	0,26	0,12
4,5	3,35	(0,132)	102,76	(14 940)	18,07	(32,53)	2,6	0,19	−0,07

Annex B (informative)

Calculation of maximum radiant section tube skin temperature

B.1 General

This annex provides a procedure for calculating the maximum radiant section tube metal (skin) temperature. Correlations for estimating the fluid-film heat-transfer coefficient are given in B.2. A method for estimating the maximum local heat flux density is given in B.3. The equations used to calculate the maximum tube skin temperature and the temperature distribution through the tube wall are described in B.4. The sample calculation in B.5 demonstrates the use of these equations.

B.2 Heat-transfer coefficient

A value necessary for calculating the maximum tube metal temperature is the fluid heat-transfer coefficient at the inside wall of the tube. Although the following correlations are extensively used and accepted in heater design, they have inherent inaccuracies associated with all simplified correlations that are used to describe complex relationships.

For single-phase fluids, the heat-transfer coefficient is calculated by one of the two equations below, in which Re is the Reynolds number and Pr is the Prandtl number. No correlation is included for the heat-transfer coefficient in laminar flow, since this flow regime is rare in process heaters. There is inadequate information for reliably determining the inside coefficient in laminar flow for oil in tube sizes that are normally used in process heaters.

From reference [35], for liquid flow with $Re > 10\,000$:

$$K_l = 0,023 \left(\frac{\lambda_{f,Tb}}{D_i} \right) Re^{0,8} Pr^{0,33} \left(\frac{\mu_{f,Tb}}{\mu_{f,Tw}} \right)^{0,14} \quad (B.1)$$

From reference [36], for vapour flow with $Re > 15\,000$:

$$K_v = 0,021 \left(\frac{\lambda_{f,Tb}}{D_i} \right) Re^{0,8} Pr^{0,4} \left(\frac{T_b}{T_w} \right)^{0,5} \quad (B.2)$$

$$Re = \frac{D_i q_{mA}}{\mu_{f,Tb}} \quad (B.3)$$

$$Pr = \frac{c_p \mu_{f,Tb}}{\lambda_{f,Tb}} \quad (B.4)$$

where

- K_l is the heat-transfer coefficient, in $W/(m^2 \cdot K)$ [$Btu/(h \cdot ft^2 \cdot ^\circ F)$], for the liquid phase;
- K_v is the heat-transfer coefficient, in $W/(m^2 \cdot K)$ [$Btu/(h \cdot ft^2 \cdot ^\circ F)$], for the vapour phase;
- $\lambda_{f,Tb}$ is the thermal conductivity, in $W/(m \cdot K)$ [$Btu/(h \cdot ft^2 \cdot ^\circ F)$], of fluid at bulk temperature;
- D_i is the inside diameter, expressed in metres (feet), of the tube;

- $\mu_{f,Tb}$ is the absolute viscosity, in Pa·s [lb/(ft·h)], of fluid at bulk temperature;
- $\mu_{f,Tw}$ is the absolute viscosity, in Pa·s [lb/(ft·h)], of fluid at wall temperature;
- T_b is the absolute bulk temperature, expressed in kelvins (degrees Rankine), of vapour;
- T_w is the absolute wall temperature, expressed in kelvins (degrees Rankine), of vapour;
- q_{mA} is the areic mass flow rate, in kg/(m²·s) [lb/(ft²·h)], of the fluid;
- c_p is the specific heat capacity, in J/(kg·K) [Btu/(lb·°R)], of the fluid at bulk temperature.

All of the material properties except $\mu_{f,Tw}$ are evaluated at the bulk fluid temperature. To convert absolute viscosity in millipascal-seconds to pounds per foot per hour, multiply $\mu_{f,Tw}$ by 2,42.

For two-phase flows, the heat-transfer coefficient can be approximated using the following equation:

$$K_{2p} = K_l w_l + K_v w_v \quad (\text{B.5})$$

where

- K_{2p} is the heat-transfer coefficient, in W/(m²·K) [Btu/(h·ft²·°F)], for two phases;
- w_l is the mass fraction of the liquid;
- w_v is the mass fraction of the vapour.

The liquid and vapour heat-transfer coefficients, K_l and K_v , should be calculated using the mixed-phase areic mass flow rate but using the liquid and vapour material properties, respectively.

NOTE In two-phase flow applications where dispersed flow or mist flow regimes occur due to entrainment of tiny liquid droplets in the vapour (e.g. towards the outlet of vacuum heaters), the heat transfer coefficient can be calculated using the correlation for vapour phase using equation (B.2), based on the total flow rate, rather than approximated by equation (B.5).

B.3 Maximum local heat flux density

The average heat flux density in the radiant section of a heater (or in a zone of the radiant section) is equal to the duty in the section or zone divided by the total outside surface area of the coil in the section or zone. The maximum local heat flux density at any point in the coil can be estimated from the average heat flux density. The maximum local heat flux density is used with the equations in B.4 to calculate the maximum metal temperature.

Local heat flux densities vary considerably throughout a heater because of nonuniformities around and along each tube. Circumferential variations result from variations in the radiant heat flux density produced by shadings of other tubes or from the placement of the tubes next to a wall. Conduction around the tubes and convection flows of flue gases tend to reduce the circumferential variations in the heat flux density. The longitudinal variations result from the proximity to burners and variations in the radiant firebox and the bulk fluid temperatures. In addition to variations in the radiant section, the tubes in the shock section of a heater can have a high convective heat flux density.

The maximum heat flux density at any point in a coil can be estimated as follows:

$$q_{R,max} = F_{cir} F_L F_T q_{R,ave} + q_{conv} \quad (\text{B.6})$$

where

- $q_{R,max}$ is the maximum radiant heat flux density, in W/m² [Btu/(h·ft²)], for the outside surface;
- F_{cir} is the factor accounting for circumferential heat-flux-density variations;

F_L is the factor accounting for longitudinal heat-flux-density variations;

F_T is the factor accounting for the effect of tube metal temperature on the radiant heat flux density;

$q_{R,ave}$ is the average radiant heat flux density, in W/m^2 [$Btu/(h \cdot ft^2)$], for the outside surface;

q_{conv} is the average convective heat flux density, W/m^2 [$Btu/(h \cdot ft^2)$], for the outside surface.

The circumferential variation factor, F_{cir} , is given as a function of tube spacing and coil geometry in Figure B.1. The factor given by this figure is the ratio of the maximum local heat flux density at the fully exposed face of a tube to the average heat flux density around the tube. This figure was developed from considerations of radiant heat transfer only. As mentioned above, influences such as conduction around the tube and flue gas convection act to reduce this factor. Since these influences are not included in this calculation, the calculated value will be somewhat higher than the actual maximum heat flux density.

The longitudinal variation factor, F_L , is not easy to quantify. Values between 1,0 and 1,5 are most often used. In a firebox that has a very uniform distribution of heat flux density, a value of 1,0 can be appropriate. Values greater than 1,5 can be appropriate in a firebox that has an extremely uneven distribution of heat flux density (for example, a long or a tall, narrow firebox with burners in one end only).

The tube metal temperature factor, F_T , will be less than 1,0 near the coil outlet or in areas of maximum tube metal temperature. It will be greater than 1,0 in areas of lower tube metal temperatures. For most applications, the factor can be approximated as follows:

$$F_T = \left(\frac{T_{g,ave}^{*4} - T_{tm}^{*4}}{T_{g,ave}^{*4} - T_{tm,ave}^{*4}} \right) \quad (B.7)$$

where

$T_{g,ave}^*$ is the average flue-gas temperature, expressed in kelvins (degrees Rankine), in the radiant section;

T_{tm}^* is the tube metal temperature, expressed in kelvins (degrees Rankine), at the point under consideration;

$T_{tm,ave}^*$ is the average tube metal temperature, expressed in kelvins (degrees Rankine), in the radiant section.

The convective heat flux density in most parts of a radiant section is usually small compared with the radiant heat flux density. In the shock section, however, the convective heat flux density can be significant; it should therefore be added to the radiant heat flux density when the maximum heat flux density in the shock section is estimated.

B.4 Maximum tube metal temperature

In addition to the heat-transfer coefficient and the maximum heat flux density, the temperature profile of the fluid in the coil is necessary for calculating the maximum tube metal temperature in the radiant section of the heater. This profile, which is often calculated by the heater supplier, defines the variation of the bulk fluid temperature through the heater coil. For operation at or near design, the design profile can be used. For operation significantly different from design, a bulk temperature profile shall be developed.

Once the bulk fluid temperature is known at any point in the coil, the maximum tube metal temperature can be calculated as follows:

$$T_{max} = T_{bf} + \Delta T_{ff} + \Delta T_{coke} + \Delta T_{tw} \quad (B.8)$$

$$\Delta T_{ff} = \frac{q_{R,max}}{K_{ff}} \left(\frac{D_o}{D_i - 2\delta_{coke}} \right) \quad (B.9)$$

$$\Delta T_{coke} = \frac{q_{R,max} \delta_{coke}}{\lambda_{coke}} \left(\frac{D_o}{D_i - \delta_{coke}} \right) \quad (B.10)$$

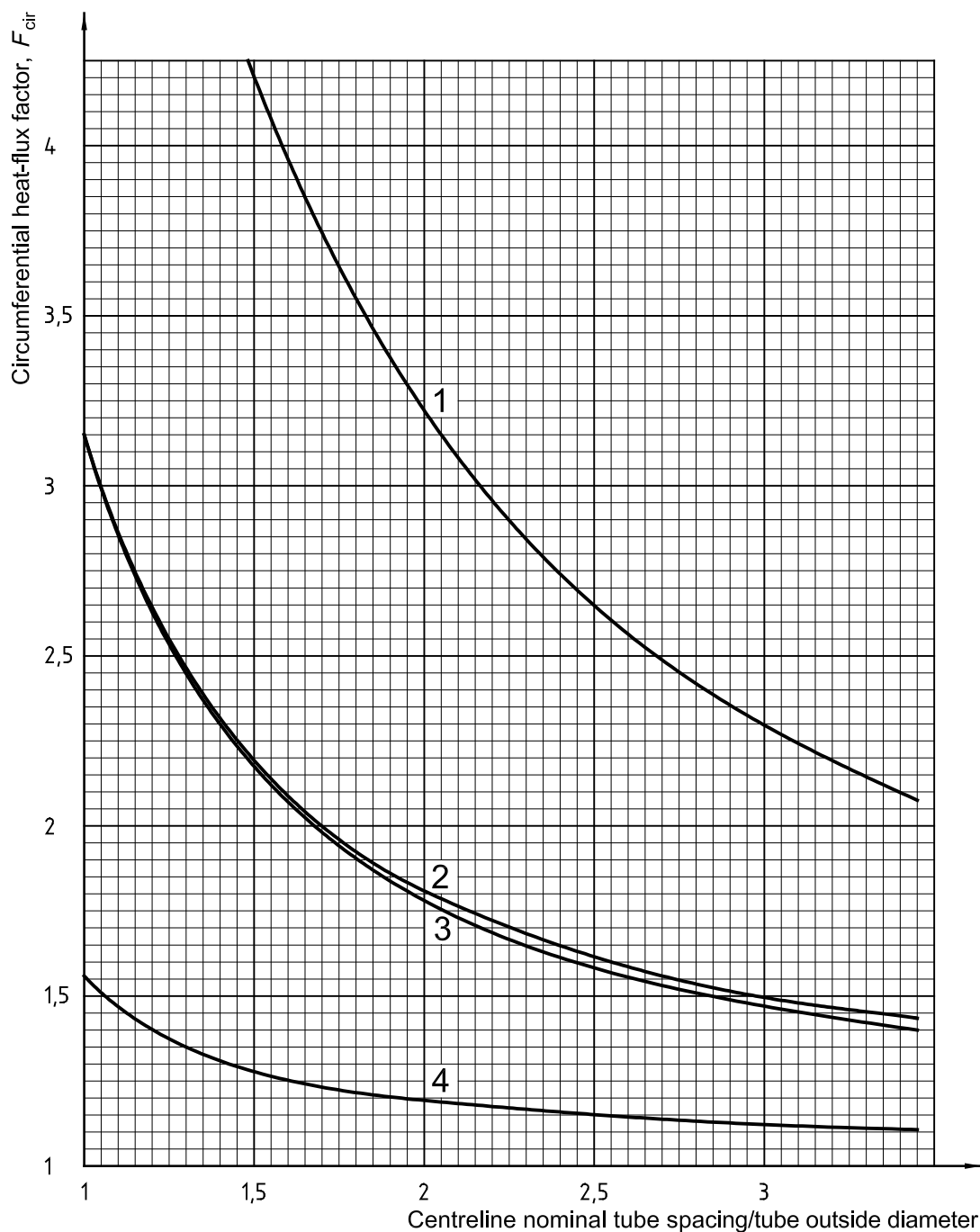
$$\Delta T_{tw} = \frac{q_{R,max} \delta_{t,ave}}{\lambda_{tm}} \left(\frac{D_o}{D_o - \delta_{t,ave}} \right) \quad (B.11)$$

where

- T_{max} is the maximum tube metal temperature, expressed in degrees Celsius (Fahrenheit);
- T_{bf} is the bulk fluid temperature, expressed in degrees Celsius (Fahrenheit);
- ΔT_{ff} is the temperature difference across the fluid film, expressed in degrees Celsius (Fahrenheit);
- ΔT_{coke} is the temperature difference across coke or scale, expressed in degrees Celsius (Fahrenheit);
- ΔT_{tw} is the temperature difference across the tube wall, expressed in degrees Celsius (Fahrenheit);
- $q_{R,max}$ is the maximum radiant heat flux density, in $W/(m^2 \cdot K)$ [$Btu/(h \cdot ft^2 \cdot ^\circ F)$], for the outside surface;
- K_{ff} is the fluid-film heat-transfer coefficient, in $W/(m^2 \cdot K)$ [$Btu/(h \cdot ft^2 \cdot ^\circ F)$];
- D_o is the outside diameter, expressed in metres (feet), of the tube;
- D_i is the inside diameter, expressed in metres (feet), of the tube;
- δ_{coke} is the coke and/or scale thickness, expressed in metres (feet);
- λ_{coke} is the thermal conductivity of coke or scale, in $W/(m^2 \cdot K)$ ($Btu/h \cdot ft \cdot ^\circ F$);
- $\delta_{t,ave}$ is the average tube thickness, expressed in metres (feet);
- λ_{tm} is the thermal conductivity, in $W/(m \cdot K)$ [$Btu/(h \cdot ft \cdot ^\circ F)$], of the tube metal.

In equations (B.10) and (B.11), the denominators within the parentheses are the mean diameters of the coke layer and tube, respectively. The effect of coke or scale on the tube metal temperature can be estimated using equation (B.10).

The thermal conductivity of the tube material used in equation (B.11) should be evaluated at the average tube wall temperature. For cast tubes, the nominal as-cast thickness should be used for $\delta_{t,ave}$ in equation (B.11).



Key

- 1 Curve 1 = double row against wall, triangular spacing
- 2 Curve 2 = double row with equal radiation from both sides and two diameters between rows, equilateral spacing
- 3 Curve 3 = single row against wall
- 4 Curve 4 = single row with equal radiation from both sides

These curves are valid when used with a tube-centre-to-refractory-wall spacing of 1,5 times the nominal tube diameter. Any appreciable variation from this spacing should be given special consideration.

NOTE 1 These curves do not take into consideration the convection heat transfer to the tubes, circumferential heat transfer by conduction through the tube wall, or variations in heat flux density in different zones of the radiant section.

NOTE 2 These curves are based on the work of H. C. Hottel, as reported on page 69 of reference [35].

Figure B.1 — Ratio of maximum local to average heat flux

B.5 Sample calculation

The following sample calculation demonstrates how to use the equations given in the previous clauses.

NOTE Differences in results between SI and US customary calculations for dimensionless numbers are due to the significant figures used in the dimension conversions.

In the heater under consideration, the medium carbon steel tubes are in a single row against the wall. Other aspects of the heater configuration are as follows:

Tube spacing = 203,2 mm (= 0,667 ft = 8,0 in);

$D_o = 114,3 \text{ mm}$ (= 0,375 ft = 4,5 in);

$\delta_{t,ave} = 6,4 \text{ mm}$ (= 0,020 8 ft = 0,25 in);

$D_i = 101,6 \text{ mm}$ (= 0,333 ft = 4,0 in);

$\delta_{coke} = 0 \text{ mm}$ (0 in);

$\lambda_{tm} = 42,2 \text{ W/(m}\cdot\text{K)}$ [24,4 Btu/(h·ft·°F)] at an assumed tube metal temperature of 380 °C (720 °F).

The flow in the tubes is two-phase with 10 % mass vapour. Other operating conditions are as follows:

Flow rate (total liquid plus vapour) = 6,3 kg/s (50 000 lb/h)

$T_b = 271 \text{ °C}$ (520 °F).

$q_{R,ave} = 31\,546 \text{ W/m}^2$ [10 000 Btu/(h·ft²)]

The properties of the liquid at the bulk temperature are as follows:

$\mu_{f,Tb} = 2,0 \times 10^{-3} \text{ Pa}\cdot\text{s}$ [4,84 lb/(h·ft)]

$\lambda_{f,Tb} = 0,116\,3 \text{ W/(m}\cdot\text{K)}$ [0,067 2 Btu/(h·ft·°F)]

$c_{p,f} = 2\,847 \text{ J/(kg}\cdot\text{K)}$ [0,68 Btu/(lb·°F)].

The properties of the vapour at the bulk temperature are as follows:

$\mu_{v,Tb} = 7,0 \times 10^{-6} \text{ Pa}\cdot\text{s}$ [0,017 lb/(ft·h)].

$\lambda_{v,Tb} = 0,034\,6 \text{ W/(m}\cdot\text{K)}$ [0,020 Btu/(h·ft·°F)]

$c_{p,v} = 2\,394 \text{ J/(kg}\cdot\text{K)}$ [0,572 Btu/(lb·°F)].

From the inside diameter, the flow area is equal to $8,107 \times 10^{-3} \text{ m}^2$ (0,087 3 ft²). Using the total flow rate:

$$q_{mA} = 6,3 / (8,107 \times 10^{-3})$$

$$= 777,1 \text{ kg/(m}^2\cdot\text{s)}$$

In US customary units:

$$q_{mA} = (50\,000 / 0,087\,3)$$

$$= 5,73 \times 10^5 \text{ lb/(h}\cdot\text{ft}^2)$$

The Reynolds number [equation (B.3)] is as follows:

For liquid

$$Re = \frac{(0,1016) (777,1)}{0,002} = 3,95 \times 10^4$$

In US customary units:

$$Re = \frac{(0,333) (5,73 \times 10^5)}{4,84} = 3,94 \times 10^4$$

For vapour

$$Re = \frac{(0,1016) (777,1)}{7,0 \times 10^{-6}} = 1,13 \times 10^7$$

In US customary units:

$$Re = \frac{(0,333) (5,73 \times 10^5)}{0,017} = 1,12 \times 10^7$$

The Prandtl number [equation (B.4)] is as follows:

For liquid

$$Pr = \frac{(2\,847) (0,002)}{0,116\,3} = 49,0$$

In US customary units:

$$Pr = \frac{(0,68) (4,84)}{0,067\,2} = 49,0$$

For vapour

$$Pr = \frac{(2\,395) (7,0 \times 10^{-6})}{0,034\,6} = 0,485$$

In US customary units:

$$Pr = \frac{(0,572) (0,017)}{0,020} = 0,486$$

Assume that for the liquid

$$\left(\frac{\mu_{f,Tb}}{\mu_{f,Tw}} \right)^{0,14} = 1,1$$

Assume that for the vapour

$$\left(\frac{T_b}{T_w} \right)^{0,5} = 0,91$$

These assumptions will be checked later. Using equation (B.1),

$$K_I = 0,023 \left(\frac{\lambda_{f,Tb}}{D_i} \right) (3,94 \times 10^4)^{0,8} (49,0)^{0,33} (1,1)$$

$$= 433,8 \left(\frac{\lambda_{f,Tb}}{D_i} \right)$$

Using equation (B.2)

$$K_V = 0,021 \left(\frac{\lambda_{f,Tb}}{D_i} \right) (1,12 \times 10^7)^{0,8} (0,486)^{0,4} (0,91)$$

$$= 6\,242 \left(\frac{\lambda_{f,Tb}}{D_i} \right)$$

Hence

$$K_I = 433,8 \left(\frac{0,116\,3}{0,101\,6} \right) = 497\, \text{W/m}^2 \cdot \text{K}$$

$$K_V = 6\,242 \left(\frac{0,034\,6}{0,101\,6} \right) = 2\,126\, \text{W/m}^2 \cdot \text{K}$$

In US customary units:

$$K_I = 433,8 \left(\frac{0,067\,2}{0,333} \right) = 87,5\, \text{Btu/h} \cdot \text{ft}^2 \cdot ^\circ\text{F}$$

$$K_V = 6\,242 \left(\frac{0,020}{0,333} \right) = 375\, \text{Btu/h} \cdot \text{ft}^2 \cdot ^\circ\text{F}$$

The two-phase heat-transfer coefficient can then be calculated using equation (B.5):

$$K_{2p} = (0,90)K_I + (0,10)K_V$$

$$= (0,90)(497) + (0,10)(2\,126)$$

$$= 659,9\, \text{W/(m}^2 \cdot \text{K)}$$

In US customary units:

$$K_{2p} = (0,90)(87,5) + (0,10)(375)$$

$$= 116,3\, \text{Btu/(h} \cdot \text{ft}^2 \cdot ^\circ\text{F)}$$

The ratio of tube spacing to tube diameter is as follows:

$$\frac{203,2}{114,3} = 1,78$$