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

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

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Contents

Page

Foreword

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

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

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

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

This second edition cancels and replaces the first edition (ISO 13704:2001), which has been technically revised.

Petroleum, petrochemical 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 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, $\delta_{\text{min}}/D_{\text{o}}$, 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.

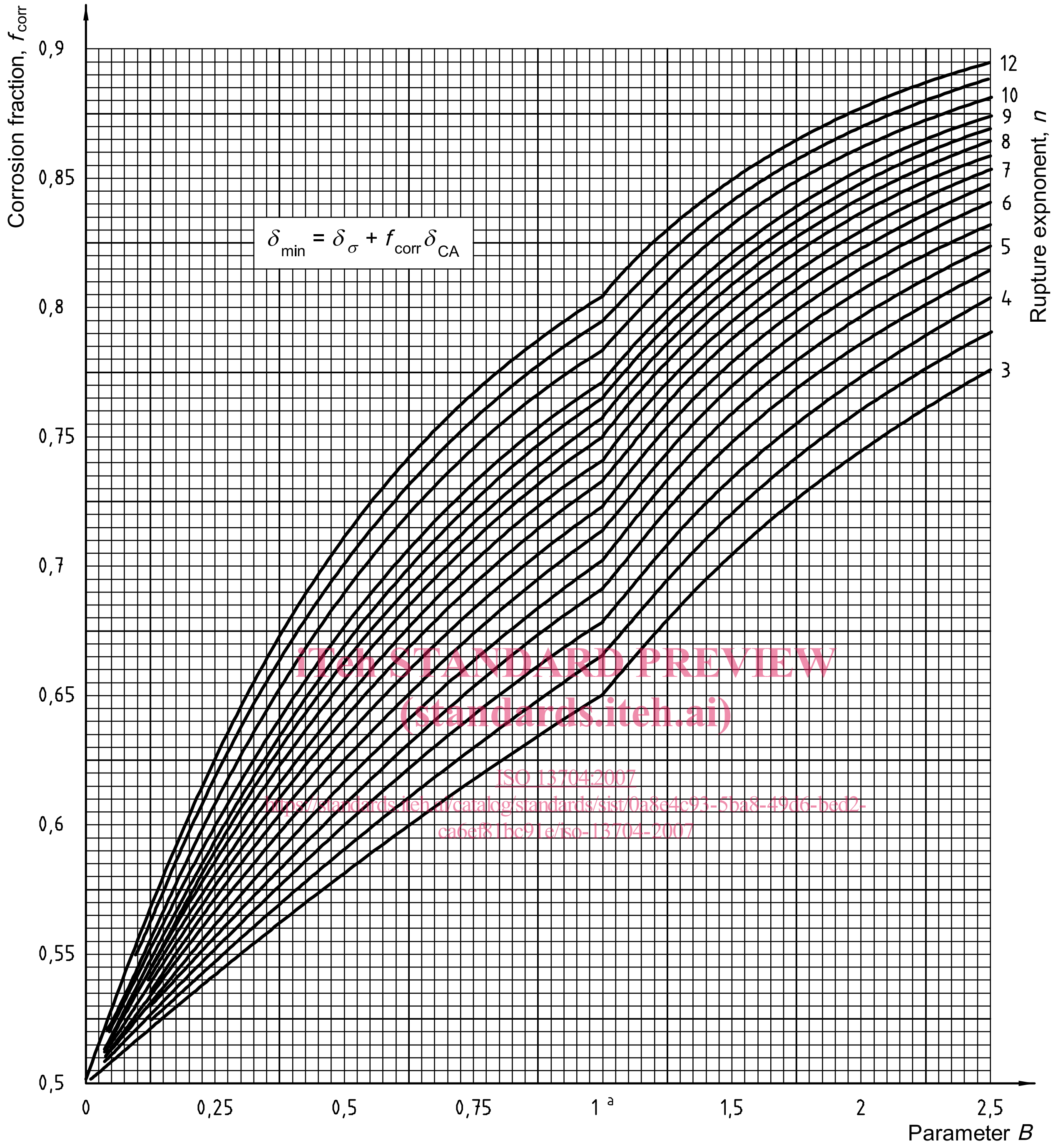
The rupture design pressure is never more than the elastic design pressure. The characteristic that differentiates these two pressures is the relative length of time over which they are sustained. The rupture design pressure is a long-term loading condition that remains relatively uniform over a period 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 δ_{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

$\delta_{\sigma} = \dfrac{p_{\mathrm{r}} D_{\mathrm{o}}}{2\sigma_{\mathrm{r}} + p_{\mathrm{r}}}$

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

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

[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):

$$\sigma = \frac{p}{2}\left(\frac{D_{\mathrm{o}}}{\delta} - 1\right) = \frac{p}{2}\left(\frac{D_{\mathrm{i}}}{\delta} + 1\right) \tag{1}$$

where

σ is the stress, expressed in megapascals [pounds per square inch[1]];

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):

$$\delta_{\sigma} = \frac{p_{\mathrm{el}} D_{\mathrm{o}}}{2\sigma_{\mathrm{el}} + p_{\mathrm{el}}} \text{ or } \delta_{\sigma} = \frac{p_{\mathrm{el}} D_{\mathrm{i}}^{*}}{2\sigma_{\mathrm{el}} - p_{\mathrm{el}}} \tag{2}$$

$$\delta_{\mathrm{min}} = \delta_{\sigma} + \delta_{\mathrm{CA}} \tag{3}$$

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.

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[1] The unit “pounds per square inch (psi)” is referred to as “pound-force per square inch (lbf/in^2)” 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):

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

$$\delta_{\min} = \delta_\sigma + f_{\text{corr}}\delta_{\text{CA}} \tag{5}$$

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 ± 12,3 % and ± 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 $\left({}^{0}_{+28}\right)$ % tolerance, a tube that is purchased to a 12,7 mm (0,500 in) minimum-thickness specification has the following average thickness:

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

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

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

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.

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

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):

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

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

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 Rankine[2]], during the operating period;

T^*_{sor} $= T_{\text{sor}} + 273$ K ($T_{\text{sor}} + 460$ °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.

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[2] Rankine is a deprecated unit.

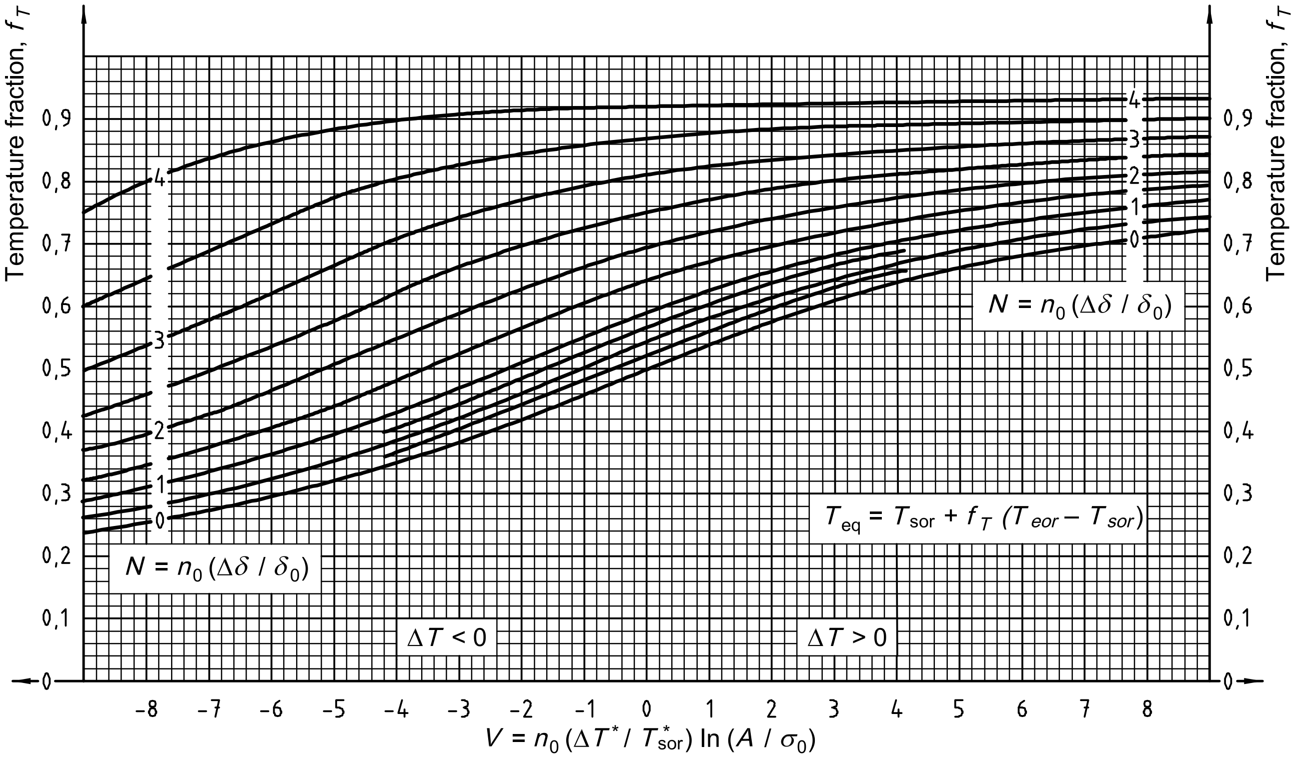

Figure 2 — Temperature fraction