INTERNATIONAL STANDARD

ISO 12241

Third edition
2022-06

Thermal insulation for building equipment and industrial installations — Calculation rules

Isolation thermique des équipements de bâtiments et des installations industrielles — Méthodes de calcul

Reference number
ISO 12241:2022(E)

ISO copyright office
CP 401 • Ch. de Blandonnet 8
CH-1214 Vernier, Geneva
Phone: +41 22 749 01 11
Email: copyright@iso.org
Website: www.iso.org

Published in Switzerland

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.

The procedures used to develop this document and those intended for its further maintenance are described in the ISO/IEC Directives, Part 1. In particular, the different approval criteria needed for the different types of ISO documents should be noted. This document was drafted in accordance with the editorial rules of the ISO/IEC Directives, Part 2 (see www.iso.org/directives).

Attention is drawn to the possibility that some of the elements of this document may be the subject of patent rights. ISO shall not be held responsible for identifying any or all such patent rights. Details of any patent rights identified during the development of the document will be in the Introduction and/or on the ISO list of patent declarations received (see www.iso.org/patents).

Any trade name used in this document is information given for the convenience of users and does not constitute an endorsement.

For an explanation of the voluntary nature of standards, the meaning of ISO specific terms and expressions related to conformity assessment, as well as information about ISO's adherence to the World Trade Organization (WTO) principles in the Technical Barriers to Trade (TBT), see www.iso.org/iso/foreword.html.

This document was prepared by Technical Committee ISO/TC 163, *Thermal performance and energy use in the built environment*, Subcommittee SC 2, *Calculation methods,* in collaboration with the European Committee for Standardization (CEN) Technical Committee CEN/TC 89, *Thermal performance of buildings and building components*, in accordance with the Agreement on technical cooperation between ISO and CEN (Vienna Agreement).

This third edition cancels and replaces the second edition (ISO 12241:2008), which has been technically revised.

The main changes are as follows:

— how to calculate the convective part of the external surface coefficient of heat transfer;

— how to introduce thermal bridges in the general heat loss calculation;

— provides detailed data along with the method for calculating fittings (thermal bridges), only informative.

Any feedback or questions on this document should be directed to the user's national standards body. A complete listing of these bodies can be found at www.iso.org/members.html.

Introduction

Methods relating to conduction are direct mathematical derivations from Fourier's law of heat conduction, so no significant difference in the formulae used in the member countries exists. For convection and radiation, however, there are no methods in practical use that are mathematically traceable to Newton's law of cooling or the Stefan-Boltzman law of thermal radiation, without some empirical element. For convection in particular, many different formulae have been developed, based on laboratory data. Different formulae have become popular in different countries, and no exact means are available to select between these formulae.

Within the limitations given below, these methods can be applied to most types of industrial, thermal-insulation, heat-transfer problems.

a) These methods do not take into account the permeation of air and the transmittance of thermal radiation through transparent media.

b) The formulae in these methods require for their solution that some system variables be known, given, assumed or measured. In all cases, the accuracy of the results depends on the accuracy of the input variables. This document contains no guidelines for accurate measurement of any of the variables. However, it does contain guides that have proven satisfactory for estimating some of the variables for many industrial thermal systems.

c) When the steady-state calculations are used in a changing thermal environment (process equipment operating year-round, outdoors, for example), it is necessary to use local weather data based on yearly averages or yearly extremes of the weather variables (depending on the nature of the particular calculation) for the calculations in this document.

d) In particular, the user should not infer from the methods of this document that either insulation quality or avoidance of dew formation can be reliably assured based on minimal, simple measurements and application of the basic calculation methods given here. For most industrial heat flow surfaces, there is no isothermal state (no one, homogeneous temperature across the surface), but rather a varying temperature profile. Furthermore, the heat flow through a surface at any point is a function of several variables that are not directly related to insulation quality. Among others, these variables include ambient temperature, movement of the air, roughness and emissivity of the heat flow surface, and the radiation exchange with the surroundings (which often vary widely). For calculation of dew formation, variability of the local humidity is an important factor.

e) Except inside buildings, the average temperature of the radiant background seldom corresponds to the air temperature, and measurement of background temperatures, emissivity and exposure areas is beyond the scope of this document. For these reasons, neither the surface temperature nor the temperature difference between the surface and the air can be used as a reliable indicator of insulation performance or avoidance of dew formation.

Clauses 4 and 5 of this document give the methods used for industrial thermal insulation calculations not covered by more specific standards.

Clauses 6 and 7 of this document are adaptations of the general formula for specific applications of calculating heat flow, temperature drop, and freezing times in pipes and other vessels. Thermal insulation to heating/cooling systems such as a boiler and refrigerator are not dealt with by this document.

Annexes A and B of this document are for information only.

Thermal insulation for building equipment and industrial installations — Calculation rules

1 Scope

This document gives rules for the calculation of heat-transfer-related properties of building equipment and industrial installations, predominantly under steady-state conditions. This document also gives a simplified approach for the calculation of thermal bridges.

2 Normative references

The following documents are referred to in the text in such a way that some or all of their content constitutes requirements of this document. For dated references, only the edition cited applies. For undated references, the latest edition of the referenced document (including any amendments) applies.

ISO 7345, *Thermal performance of buildings and building components — Physical quantities and definitions*

ISO 9346, *Hygrothermal performance of buildings and building materials — Physical quantities for mass transfer — Vocabulary*

ISO 13787, *Thermal insulation products for building equipment and industrial installations — Determination of declared thermal conductivity*

ISO 13788, *Hygrothermal performance of building components and building elements — Internal surface temperature to avoid critical surface humidity and interstitial condensation — Calculation methods*

ISO 23993, *Thermal insulation products for building equipment and industrial installations — Determination of design thermal conductivity*

3 Terms, definitions and symbols

3.1 Terms and definitions

For the purposes of this document, the terms and definitions given in ISO 7345, ISO 9346, ISO 13787 and ISO 23993 and the following apply.

ISO and IEC maintain terminology databases for use in standardization at the following addresses:

— ISO Online browsing platform: available at https://www.iso.org/obp

— IEC Electropedia: available at https://www.electropedia.org/

3.1.1
thermally separated end disc
end disc used so that the extremities and end caps are not in contact with the object

Note 1 to entry: This construction is used to avoid thermal bridges and the risk of damaging vapour retarders or pipe tracing [15].

3.2 Symbols

Table 1 gives the definition and unit of symbols used in this document.

Table 1 — Definition and unit of symbol

Symbol	Definition	Unit
A	area	m^2
A_s	solar absorption coefficient	
a	length of a rectangle	m
a_r	temperature factor	K^3
b	width of a rectangle	m
C_r	radiation coefficient	$W/(m^2 \cdot K^4)$
c_p	specific heat capacity at constant pressure	$J/(kg \cdot K)$
D	diameter	m
d	thickness	m
d_R	insulation layer thickness of the pipe	m
F	overall conversion factor for thermal conductivity	
Gr	Grashof number	
H	height	m
h	surface coefficient of heat transfer	$W/(m^2 \cdot K)$
J_s	solar radiation	W/m^2
K	thermal bridge coefficient	W/K
L	length	m
l	characteristic length	m
l_i	insulation box inside length	m
m	mass	kg
$\dot{m}$	mass flow rate	kg/s
Nu	Nusselt number	
P	perimeter	m
p	pressure	Pa
p_a	water vapour pressure	Pa
Pr	Prandtl number	
q	density of heat flow rate	W/m^2 or W/m
R	thermal resistance	$m^2 \cdot K/W$ or $m \cdot K/W$ or K/W
Re	Reynolds number	
S	space inside the insulation box	
T	thermodynamic temperature	K
t	time	s
U	thermal transmittance	$W/(m^2 \cdot K)$ or $W/(m \cdot K)$ or W/K
w	velocity of the air or other fluid	m/s
x	Bolt length + 20 mm	mm
α	coefficient of longitudinal temperature drop	m^{-1}
α'	coefficient of cooling time	s^{-1}
Δh	latent heat	J/kg
ε	emissivity	
Φ	heat flow rate	W
λ	thermal conductivity	$W/(m \cdot K)$
λ_d	declared thermal conductivity	$W/(m \cdot K)$
λ_D	design thermal conductivity	$W/(m \cdot K)$

Table 1 *(continued)*

Symbol	Definition	Unit
θ	Celsius temperature	°C
θ_b	point of measurement of the temperature at the fin base	°C
ρ	density	kg/m^3
φ	relative humidity	%
σ	Stefan-Boltzmann constant (see Reference [8])	$W/(m^2 \cdot K^4)$
ν	kinematic viscosity of air or other fluid	m^2/s
Δ	difference	
ΔA	equivalent area	m^2
ΔL	equivalent length	m
$\Delta\lambda$	extra conductivity due to regularly placed components in the insulation system	$W/(m \cdot K)$

3.3 Subscripts

Table 2 gives the definition of subscripts used in this document.

Table 2 — Definition of subscripts

A	valve	i	interior (internal)
a	ambient,	in	initial
anc	anchor	Ka	insulation box
av	average	l	linear
B	thermal bridge	lab	laboratory
c	cooling	lam	laminar flow
cv	convection	MRT	mean radiant temperature
cr	critical	P	pump
cs	cross section	p	pipe
d	duct	r	radiation
E	soil	ref	reference
e	exterior (external)	s	surface
ef	effective	sat	saturated vapour
en	entrance	se	exterior surface
ex	exit	si	interior surface
f	fluid	sph	spherical
fa	frontal of the fin	sq	per square
fas	fastener	T	total
FEM	Finite Element Method	tb	insulation related thermal bridge
fi	final	tur	turbulent flow
fin	fin	V	vertical
fl	flange	v	vessel
forced	forced	W	wall
fr	freezing	w	water
free	free	wp	start freezing
H	horizontal		

4 Calculation rules and formulae of heat transfer

4.1 Fundamental formulae for heat transfer

4.1.1 General

The formulae given in Clause 4 apply only to the case of heat transfer in steady state, i.e. to the case where temperatures remain constant in time at any point of the medium considered. The design thermal conductivity is temperature-dependent; see Figure 1, dashed line. However, in this document, the design value for the mean temperature for each layer shall be used.

4.1.2 Thermal conduction

Thermal conduction normally describes molecular heat transfer in solids, liquids, and gases under the effect of a temperature gradient.

It is assumed in the calculation that a temperature gradient exists in one direction only and that the temperature is constant in planes perpendicular to it.

The density of heat flow rate, q, for a plane wall in the x-direction is given by Formula (1):

$$q = \lambda_D \cdot \frac{d\theta}{dx} \tag{1}$$

For a single layer, Formulae (2), (3) and (4) are given:

$$q = \frac{\lambda_D}{d} \cdot (\theta_{\text{si}} - \theta_{\text{se}}) \tag{2}$$

or

$$q = \left(\frac{\theta_{\text{si}} - \theta_{\text{se}}}{R} \right) \tag{3}$$

and

$$R = \frac{d}{\lambda_D} \tag{4}$$

where

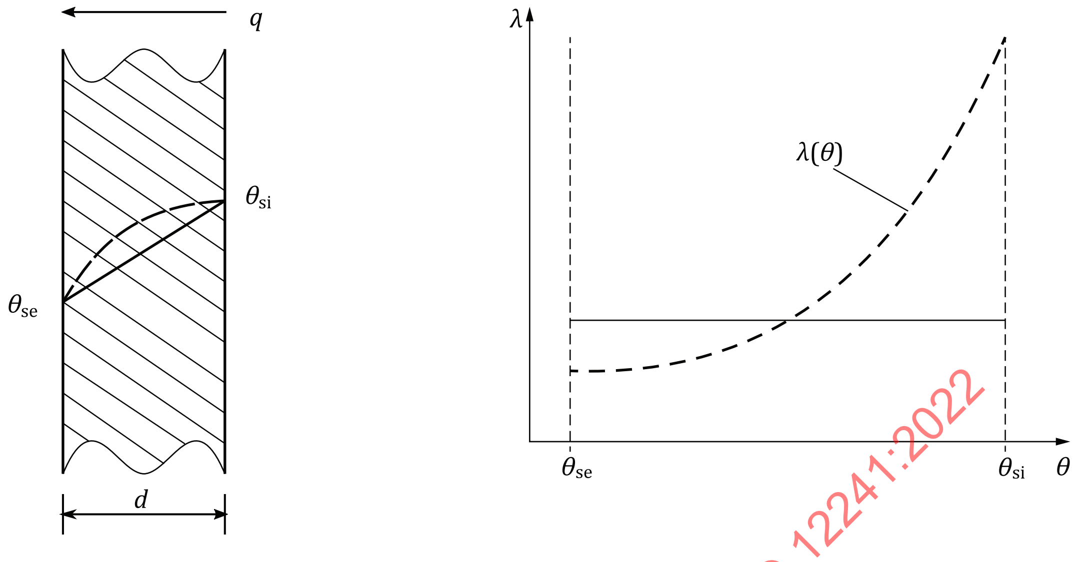
λ_D is the design thermal conductivity of the insulation product or system, expressed in W/(m·K);

d is the thickness of the plane wall, expressed in m;

θ_{si} is the temperature of the internal surface, expressed in °C;

θ_{se} is the temperature of the external surface, expressed in °C;

R is the thermal resistance of the wall, expressed in m^2·K/W.



a) Temperature distribution in a single-layer

b) Thermal conductivity as function of the temperature

NOTE The dashed curve in Figure 1a), represents the temperature variation in a wall, considering that the thermal conductivity depends on the temperature, such as the dashed curve in Figure 1b). In case that the thermal conductivity is considered as temperature-independent (the solid line in Figure 1b), the variation of the temperature inside a wall is represented by the straight line in Figure 1a).

Figure 1 — Temperature distribution

For a multi-layer wall (see Figure 2), q is calculated according to Formula (3), where R is the thermal resistance of the multi-layer wall, as given in Formula (5):

$$R = \sum_{j=1}^{n} \frac{d_j}{\lambda_{D_j}} \quad (5)$$

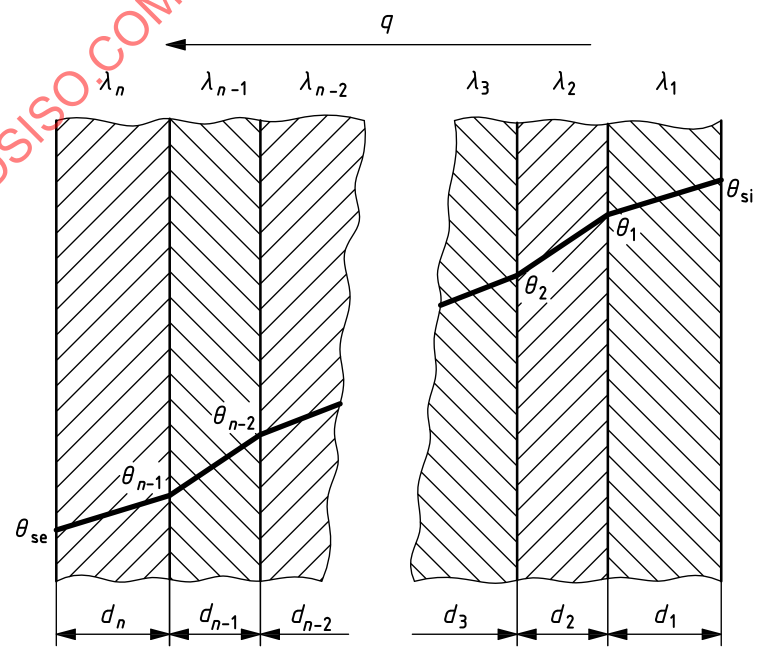


Figure 2 — Temperature distribution in a multi-layer wall

The linear density of heat flow rate, q_l, of a single-layer hollow cylinder (see Figure 3) is given in Formula (6):

$$q_l = \frac{\theta_{si} - \theta_{se}}{R_l} \tag{6}$$

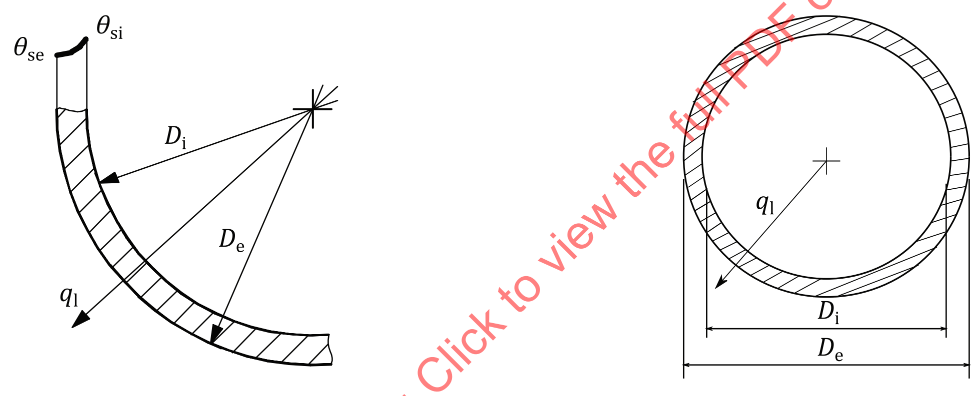
where R_l is the linear thermal resistance of a single-layer hollow cylinder [m·K/W], as given in Formula (7):

$$R_l = \frac{\ln \frac{De}{D_i}}{2 \cdot \pi \cdot \lambda_D} \tag{7}$$

where

D_e is the outer diameter of the layer, expressed in m;

D_i is the inner diameter of the layer, expressed in m.



a) **Temperature distribution in a single layer-hollow cylinder**

b) **Front view of a hollow cylinder**

Figure 3 — Temperature distribution in a single-layer hollow cylinder

For a multi-layer hollow cylinder (see Figure 4), the linear density of heat flow rate, q_l, is given in Formula (6), where R_l is given by Formula (8)

$$R_l = \frac{1}{2 \cdot \pi} \sum_{j=1}^{n} \left(\frac{1}{\lambda_{D_j}} \ln \frac{D_{e,j}}{D_{i,j}} \right) \tag{8}$$

where

$D_{i,1} = D_i$

$D_{e,n} = D_e$

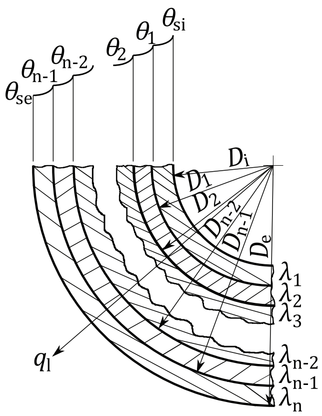


Figure 4 — Temperature distribution in a multi-layer hollow cylinder

For curved surfaces with a diameter larger than 1 200 mm, it is recommended to use formulae for a plane wall.

The heat flow rate of a sphere, Φ_{sph}, of a single-layer hollow sphere (see Figure 5) is given by Formula (9):

$$\Phi_{sph} = \frac{\theta_{si} - \theta_{se}}{R_{sph}} \tag{9}$$

where R_{sph} is the thermal resistance of a single-layer hollow sphere [K/W], as given in Formula (10):

$$R_{sph} = \frac{1}{2 \cdot \pi \cdot \lambda_D} \left(\frac{1}{D_i} - \frac{1}{D_e} \right) \tag{10}$$

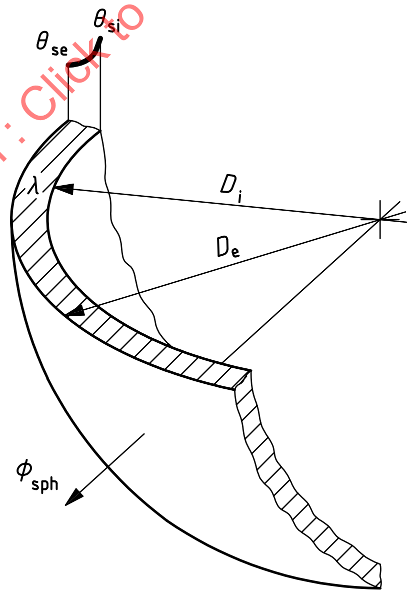


Figure 5 — Temperature distribution in a single-layer hollow sphere

For a multi-layer hollow sphere (see Figure 6), the heat flow rate of a sphere, Φ_{sph}, is given in Formula (9), where R_{sph} is given by Formula (11):

$$R_{sph} = \frac{1}{2 \cdot \pi} \cdot \sum_{j=1}^{n} \frac{1}{\lambda_{D_j}} \cdot \left(\frac{1}{D_{j-1}} - \frac{1}{D_j} \right) \tag{11}$$

where

$D_0 = D_i$

$D_n = D_e$

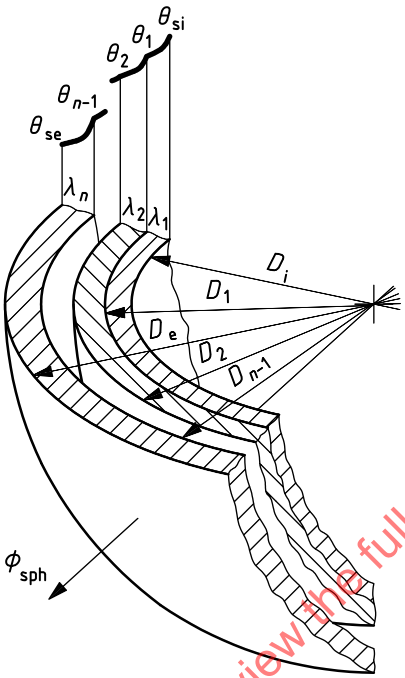


Figure 6 — Temperature distribution in a multi-layer hollow sphere

The linear density of heat flow rate, q_l, through the wall of a duct with rectangular cross-section (see Figure 7) is given by Formula (12):

$$q_l = \frac{\theta_{si} - \theta_{se}}{R_l} \tag{12}$$

The linear thermal resistance of a duct, R_l [m·K/W], of the wall of such a duct can be approximately calculated by Formula (13):

$$R_l = \frac{2 \cdot d}{\lambda_D \cdot (P_e + P_i)} \tag{13}$$

where

d is the thickness of the insulating layer, expressed in m;

P_i is the inner perimeter of the duct, expressed in m;

P_e is the external perimeter of the duct, expressed in m, as given in Formula (14):

$$P_e = P_i + (8 \cdot d) \tag{14}$$

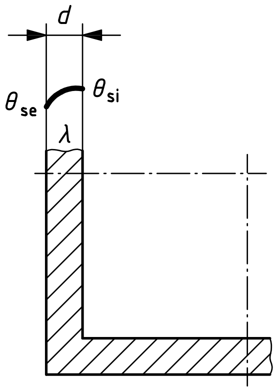


Figure 7 — Temperature distribution in a wall of a duct with rectangular cross-section with temperature-dependent thermal conductivity

4.1.3 Surface coefficient of heat transfer

In general, the radiative and convective heat transfer at the surface area given by Formulae (15) and (16) occurs at the surface:

$$q_r = h_r \cdot (\theta_1 - \theta_{MRT}) \tag{15}$$

$$q_{cv} = h_{cv} \cdot (\theta_1 - \theta_a) \tag{16}$$

where

q_r	is the density of radiative heat flow, expressed in W/m²;
q_{cv}	is the density of convective heat flow, expressed in W/m²;
h_r	is the radiative part of the surface coefficient of heat transfer, expressed in W/(m²·K);
h_{cv}	is the convective part of the surface coefficient of heat transfer, expressed in W/(m²·K);
θ_1	is the surface temperature of surface 1, expressed in °C;
θ_{MRT}	is the mean radiant temperature of the surrounding, expressed in °C;
θ_a	is the ambient air temperature, expressed in °C.

NOTE 1 h_r is dependent on the temperature and the emissivity of the surface. Emissivity is defined as the ratio between the radiation coefficient of the surface and the black body radiation constant (see ISO 9288).

NOTE 2 h_{cv} is, in general, dependent on a variety of factors, such as air movement, temperature, the relative orientation of the surface, the material of the surface and other factors.

The combined surface heat transfer can be given by Formula (17):

$$q = q_r + q_{cv} = h_r \cdot (\theta_1 - \theta_{MRT}) + h_{cv} \cdot (\theta_1 - \theta_a) \tag{17}$$

When the mean radiant temperature is almost equal to the ambient air temperature, the combined heat transfer at the surface is given by Formula (18):

$$q = h_r \cdot (\theta_1 - \theta_a) + h_{cv} \cdot (\theta_1 - \theta_a) = (h_r + h_{cv}) \cdot (\theta_1 - \theta_a) = h_{se} \cdot (\theta_{se} - \theta_a) \tag{18}$$

where

h_{se} is the external surface coefficient of heat transfer, expressed in W/(m²·K);

θ_{se} is the external surface temperature, expressed in °C.

In 4.1.4 and 4.1.5, the coefficient $h_{se} = h_r + h_{cv}$ is used to calculate the external surface resistance, R_{se} and thermal transmittance, U, hence the approximation, mean radiant temperature equals the ambient temperature, is considered.

NOTE 3 When a surface receives the solar radiation (e.g outdoor pipes, tank roofs), the total heat flow due to radiant and convective heat transfer is calculated using the following Formula (19).

$$q = q_r + q_{cv} = h_{se} \cdot \left(\theta_{se} - \theta_a - \frac{A_s \cdot J_s}{h_{se}} \right) \quad (19)$$

where

J_s is the solar radiation, expressed in (W/m²);

A_s is the absorption coefficient of solar radiation.

4.1.3.1 Radiative part of surface coefficient, h_r

The radiative part of surface coefficient between two surfaces at different temperatures, h_r, is given by Formula (20):

$$h_r = a_r \cdot C_r \quad (20)$$

where

a_r is the temperature factor, expressed in K³;

C_r is the radiation coefficient, expressed in W/(m²·K⁴), as given by Formula (23).

The temperature factor, a_r, is given by Formula (21):

$$a_r = \frac{T_1^4 - T_2^4}{T_1 - T_2} \quad (21)$$

where

T_1 is the absolute temperature of surface 1, expressed in K;

T_2 is the absolute temperature of surface 2, expressed in K.

Formula (21) can be approximated as follows.

$$a_r \approx 4 \cdot \left(\frac{T_1 + T_2}{2} \right)^3 = 4 \cdot T_{av}^3 \quad (22)$$

where T_{av} is the arithmetic mean of the temperatures T_1 and T_2, expressed in (K).

NOTE This approximation is only valid up to 200 K temperature difference between the component (surface 1) and the surroundings (surface 2).

When a component is surrounded by different surfaces at different temperatures, the temperature T_2 should be the mean radiant temperature of the surroundings.

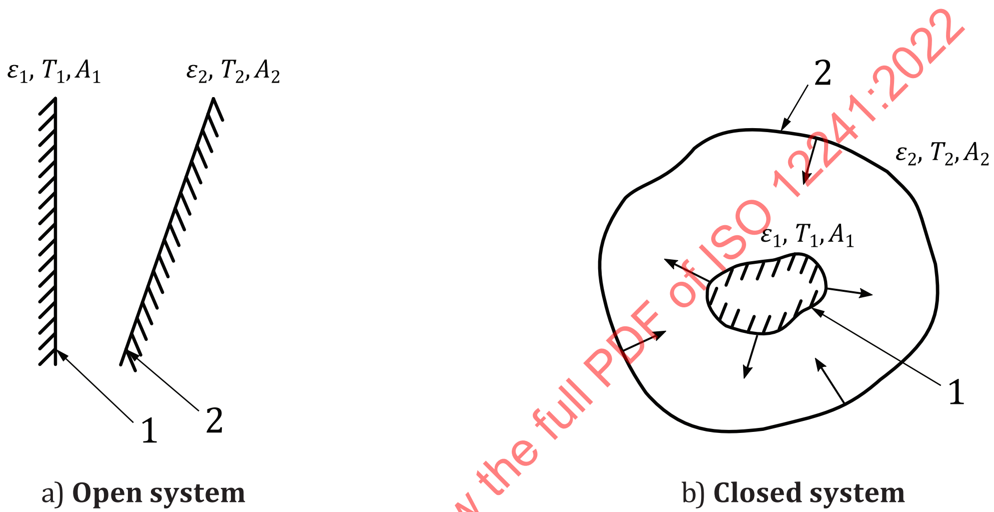
The radiation coefficient, C_r, is given by Formula (23):

$$C_r = \varepsilon \cdot \sigma \quad (23)$$

where

σ is the Stefan-Boltzmann constant [5,67×10^{-8} W/(m^2·K^4)];

ε is the effective emissivity consisted of emissivity ε_1, ε_2 and configuration factor as shown in Figure 8.



Key

1 is surface 1;

2 is surface 2.

Figure 8 — Radiation exchange between two surfaces [8]

When the mean radiant temperature is considered, the surface 2 is assumed as black ($\varepsilon_2 = 1$), and $\varepsilon = \varepsilon_1$.

Usually, the surrounding surface consists of several surfaces, each of them has generally different temperature, emissivity, and configuration factor. Here, we assume (approximate) that the surrounding surface has hypothetical uniform temperature T_2 and emissivity ε_2.

Table 3 gives some general values of emissivity for different surfaces. The emissivity value varies significantly depending on external agents, e.g. dust, corrosion, surface finish.

Table 3 — Emissivity values

Surface	ε
Aluminium, bright rolled	0,05
Aluminium, oxidized	0,13
Galvanized sheet metal, blank	0,26
Galvanized sheet metal, dusty	0,44
Austenitic steel	0,15
Aluminium-zinc sheet, lightly oxidized	0,18
Non-metallic surfaces	0,94

For more detailed information about surfaces emissivity refer to the VDI 2055 [8].

4.1.3.2 Convective part of surface coefficient, h_{cv}

4.1.3.2.1 General

Formulae for the convective part have been taken from[8]. B.1 shows examples of the following calculations.

Convection is a heat transport mechanism that occurs in liquids and gases which involves heat flows in form of internal energy flow by a mass movement. The concept of free or natural convection is used if the movement is caused by buoyancy due to temperature or concentration difference, while forced convection is caused by external forces like the wind or a fan.

For convection, it is necessary to make a distinction between the internal, h_{si}, and external, h_{se}, surface coefficients. h_{si} is defined from the point of view of the confined medium (e.g. inside pipes, vessels, boilers) and h_{se} is defined from the surrounding medium.

NOTE 1 In most cases, h_{si} can be very large and so the inner surface temperature nearly equals the temperature of the medium.

In order to describe the convection, experimental methods and their corresponding dimensionless numbers are used. The dimensionless numbers involve some fluid properties, such as thermal conductivity, density, viscosity, and heat capacity, which are determined at the mean temperature of the interface boundary layer by Formula (24):

$$\overline{\theta_f} = 0{,}5 \cdot (\theta_{se} + \theta_a) \quad (24)$$

The Nusselt number, *Nu*, which describes the relation between the convective heat transfer of a fluid layer and the conductive part within the fluid, is given by Formula (25):

$$Nu = \frac{h_{cv} \cdot l}{\lambda_f} \quad (25)$$

where

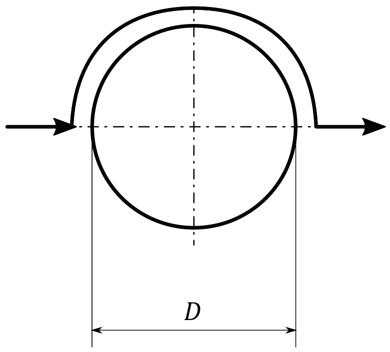
h_{cv} is the convective part of the surface coefficient, expressed in W/(m²·K);

l is the characteristic length, expressed in m;

λ_f is the thermal conductivity of the air or other fluid, expressed in W/(m·K).

NOTE 2 The characteristic length l corresponds to a body dimension, which varies according to the specific application case. It can be the diameter of a pipe or the length from the leading edge in the direction of the flow on a wall.

EXAMPLE For the determination of the Nusselt number for crossflow over cylinders, the characteristic length is the streamwise length, as shown in Figure 9:



$$l = 0{,}5 \cdot \pi \cdot D$$

Figure 9 — Characteristic length for the determination of Nusselt number for cross-flow over a cylinder [16]

Consider the proper characteristic length in each case, see Table 4, characteristic length, *l*. For any other cases the characteristic length can be approximated by Formula (26):

$$l = \frac{A}{P} \tag{26}$$

where

A is the area, expressed in m^2;

P is the perimeter, expressed in m.

The Grashof number, *Gr*, is the ratio of the buoyancy force to the viscosity force within the fluid. It must be noted that the *Gr* is the most important dimensionless number to describe the free convection, and is given by Formula (27):

$$Gr = \frac{g \cdot l^3 \cdot |\Delta\theta|}{{\nu_f}^2 \cdot T_f} \tag{27}$$

$$T_f = \overline{\theta_f} + 273{,}15 \tag{28}$$

where

g is the acceleration of gravity, expressed in m/s^2;

$|\Delta\theta|$ is the absolute value of temperature difference between surface and ambient air or fluid, expressed in °C;

ν_f is the kinematic viscosity of the air or other fluid, expressed in m^2/s;

T_f is the temperature of the air or other fluid, expressed in K.

The Prandtl number, *Pr*, is the ratio of the momentum diffusivity to thermal diffusivity, i.e. the *Pr* describes the relation between the flow field and the temperature field. It is given by Formula (29):

$$Pr = \frac{\rho_f \cdot \nu_f \cdot c_{p_f}}{\lambda_f} \tag{29}$$

where

ρ_f is the density of the air or other fluid, expressed in kg/m^3;

c_{pf} is the specific heat capacity of the air or other fluid, expressed in J/(kg·K);

λ_f is the thermal conductivity of the air or other fluid, expressed in W/(m·K).

NOTE 3 The Prandtl number depends only on fluid properties according to their temperature and pressure.

The Reynolds number, *Re*, specifies the relation between the inertia force and the friction force within the fluid, and is given by Formula (30).

$$Re = \frac{w_f \cdot l}{\nu_f} \tag{30}$$

where, w_f is the velocity of the air or other fluid, expressed in (m/s).

The numerical value of *Re* is the crucial criterium to decide whether a flow remains in a stable laminar mode, or it may undergo a transition to turbulent flow: For the fluid flow inside a circular pipe, the critical Reynolds number is Re_{cr} = 2 300. For $Re < Re_{cr}$ the flow is laminar, for $Re > Re_{cr}$ it can become turbulent. The characteristic length *l* in this case is usually taken as the inner diameter of the tube.

For parallel flow over a flat plate, the characteristic length *l* is the length in flow direction, measured from the leading edge. The critical Reynolds number for this flow is about $Re_{cr} = 5{\cdot}10^5$.

In case of dry air at standard pressure, Formulae (31), (32), and (33) shall be applied.

These formulae are only valid for air as a fluid and for the given range of temperature.

Thermal conductivity of the air in W/(m·K), is given by Formula (31):

$$\lambda_f = 0{,}024\ 3 + 7{,}842\ 1 \cdot 10^{-5} \cdot \bar{\theta}_f - 2{,}075\ 5 \cdot 10^{-8} \cdot \bar{\theta}_f^{\ 2} \tag{31}$$

where $\bar{\theta}_f$ is between −170 °C and 1 000 °C.

Kinematic viscosity of the air in m^2/s, is given by Formula (32):

$$\nu_f = \frac{4{,}211\ 3 \cdot 10^{-9} \cdot T_f^{\ 2{,}5}}{112 + T_f} \tag{32}$$

where $\bar{\theta}_f$ is between −50 °C and 100 °C.

Density of the air in kg/m^3, is given by Formula (33):

$$\rho_f = \frac{348{,}35}{T_f} \tag{33}$$

The specific heat capacity and the Prandtl number of the air in a temperature range of −50 °C to 100 °C can be expressed as an average value according to Formulae (34) and (35):

$$c_p = 1\ 007\ \text{J/(kg.K)} \tag{34}$$

$$Pr = 0{,}709 \tag{35}$$

4.1.3.2.2 Formulae to determine the convective part of the surface coefficient h_{se}, and h_{si}

The Nusselt number shall be determined to calculate the convective part of the surface coefficient, solving the Formula (25), as follows:

$$h_{cv} = \frac{Nu \cdot \lambda_f}{l} \tag{36}$$

When calculating the Nusselt number, a distinction shall be made between the case of external and internal convection. For the external convection the medium is flowing around the object in question, see Table 4, and for internal convection a confined medium is considered, see Table 5.

In case of a directional mixed convection there is a superposition of free and forced convection. The heat transfer coefficient can be calculated using Formulae (37) and (38):

For unidirectional mixed convection:

$$Nu = \sqrt[3]{Nu_{forced}^3 + Nu_{free}^3} \tag{37}$$

For mixed convection in the opposite direction:

$$Nu = \sqrt[3]{\left| Nu_{forced}^3 - Nu_{free}^3 \right|} \tag{38}$$

Table 4 — Formulae for the determination of Nusselt number — External convection case

Free convection				
Case		**Nusselt Number**	**Range of validity**	**Characteristic length, *l***
Vertical	**Wall**	$Nu_{free} = (0,825 + 0,306\,3 \cdot Gr^{1/6})^2$	$0,14 < Gr < 1,4 \cdot 10^{12}$	$l = H$
Vertical	**Pipe**	$Nu_{free} = (0,825 + 0,306\,3 \cdot Gr^{1/6})^2 + 0,87 \cdot (l / D_e)$	$0,14 < Gr < 1,4 \cdot 10^{12}$	$l = H$
Horizontal	**Pipe**	$Nu_{free} = (0,752 + 0,303 \cdot Gr^{1/6})^2$	$0,14 < Gr < 1,4 \cdot 10^{12}$	$l = \frac{\pi \cdot D_e}{2}$
Horizontal	**Wall**: heat dissipation on the top (or cooling on the bottom)	Laminar $Nu_{free,lam} = 0,593 \cdot Gr^{1/5}$	$Gr \leq 2,4 \cdot 10^5$	$l = \frac{a \cdot b}{2 \cdot (a+b)}$
		Turbulent $Nu_{free,tur} = 0,098\,8 \cdot Gr^{1/3}$	$Gr > 2,4 \cdot 10^5$	
	Wall: heat dissipation on the bottom (or cooling on the top)	$Nu_{free} = 0,453 \cdot Gr^{1/5}$	$4 \cdot 10^3 < Gr < 4 \cdot 10^{10}$	
Forced convection				
Laminar $Nu_{lam} = 0,592 \cdot Re^{1/2}$				
Turbulent $Nu_{tur} = \frac{0,026\,2 \cdot Re^{0,8}}{1 - \frac{0,5}{Re^{0,1}}}$				

Table 4 *(continued)*

Free convection			
Case	**Nusselt Number**	**Range of validity**	**Characteristic length, *l***
Wall	$Nu_{\text{forced}} = \sqrt{Nu_{\text{lam}}^2 + Nu_{\text{tur}}^2}$	$10 < Re < 10^7$	*l* longitude of the wall in the direction of the flow
Pipe	$Nu_{\text{forced}} = 0{,}3 + \sqrt{Nu_{\text{lam}}^2 + Nu_{\text{tur}}^2}$		$l = \dfrac{\pi \cdot D_{\text{e}}}{2}$

Table 5 — Formulae for the determination of Nusselt number — Internal convection case

Case	Nusselt number	Range of validity	Characteristic length, *l*
Flow through circular cross section	$Nu_{\text{forced}} = 0{,}021\ 4 \cdot (Re^{0{,}80} - 100) \cdot Pr^{0{,}4}$	$2\ 300 < Re$ and $0{,}5 < Pr < 1{,}5$	$l = D_{\text{i, pipe}}$
	$Nu_{\text{forced}} = 0{,}012 \cdot (Re^{0{,}87} - 280) \cdot Pr^{0{,}4}$	$2\ 300 < Re$ and $1{,}5 < Pr < 500$	
Flow through non-circular cross section	$Nu_{\text{forced}} = \dfrac{(\xi / 8) \cdot Re \cdot Pr}{1 + 12{,}7 \cdot \sqrt{\xi / 8} \cdot (Pr^{2/3} - 1)}$ $\xi = (1{,}8 \cdot \log(Re) - 1{,}5)^{-2}$	$10^4 \leq Re \leq 10^6$ and $0{,}1 < Pr \leq 1\ 000$	$l = \dfrac{4 \cdot A}{P}$

4.1.4 External surface resistance

The reciprocal of the outer surface coefficient, h_{se}, is the external surface resistance.

For plane walls, the surface resistance, R_{se}, is given by Formula (39):

$$R_{\text{se}} = \frac{1}{h_{\text{se}}} \tag{39}$$

For pipe insulation, the linear surface resistance, $R_{\text{l,se}}$, is given by Formula (40):

$$R_{\text{l,se}} = \frac{1}{h_{\text{se}} \cdot \pi \cdot D_{\text{e}}} \tag{40}$$

For hollow spheres, the surface resistance, $R_{\text{sph,se}}$, is given by Formula (41):

$$R_{\text{sph,se}} = \frac{1}{h_{\text{se}} \cdot \pi \cdot D_{\text{e}}^2} \tag{41}$$

4.1.5 Thermal transmittance

The thermal transmittance, U [W/m²K], is defined by Formula (42):

$$U = \frac{q}{\theta_{\text{i}} - \theta_{\text{a}}} \tag{42}$$

where

θ_{a} is the ambient air (or fluid) temperature, expressed in °C;

θ_{i} is the internal air temperature for plane walls or the temperature of the medium inside pipes, ducts and vessels, expressed in °C.

For plane walls, the thermal transmittance, U, can be calculated by Formula (43):

$$\frac{1}{U}=\frac{1}{h_{\mathrm{si}}}+R+\frac{1}{h_{\mathrm{se}}}=R_{\mathrm{si}}+R+R_{\mathrm{se}}=R_{\mathrm{T}} \tag{43}$$

For pipe insulation, the linear thermal transmittance, U_{l} [W/m·K], can be calculated by Formula (44):

$$\frac{1}{U_{\mathrm{l}}}=\frac{1}{h_{\mathrm{si}}\cdot\pi\cdot D_{\mathrm{i}}}+R_{\mathrm{l}}+\frac{1}{h_{\mathrm{se}}\cdot\pi\cdot D_{\mathrm{e}}}=R_{\mathrm{l,si}}+R_{\mathrm{l}}+R_{\mathrm{l,se}}=R_{\mathrm{l,T}} \tag{44}$$

For rectangular ducts, the linear thermal transmittance, U_{d} [W/m·K], can be calculated by Formula (45):

$$\frac{1}{U_{\mathrm{d}}}=\frac{1}{h_{\mathrm{si}}\ P_{\mathrm{i}}}+R_{\mathrm{d}}+\frac{1}{h_{\mathrm{se}}\ P_{\mathrm{e}}}=R_{\mathrm{d,si}}+R_{\mathrm{d}}+R_{\mathrm{d,se}}=R_{\mathrm{d,T}} \tag{45}$$

For hollow spheres, the thermal transmittance, U_{sph} [W/K], is given by Formula (46):

$$\frac{1}{U_{\mathrm{sph}}}=\frac{1}{h_{\mathrm{si}}\cdot\pi\cdot D_{\mathrm{i}}^{2}}+R_{\mathrm{sph}}+\frac{1}{h_{\mathrm{se}}\cdot\pi\cdot D_{\mathrm{e}}^{2}}=R_{\mathrm{sph,si}}+R_{\mathrm{sph}}+R_{\mathrm{sph,se}}=R_{\mathrm{sph,T}} \tag{46}$$

In Formulae (5), (8), (11), and (13), R, R_{l}, R_{sph}, and R_{d} are surface-to-surface thermal resistances. They shall be calculated using the design values of thermal conductivity which represent the conditions of the expected application. According to ISO 23993, the design thermal conductivity, λ_D, takes into consideration two factors. The first one is overall conversion factor, F, that takes into consideration effects like ageing or open joint, among others. The second one represents the influence of elements placed into the insulation material as spacers, $\Delta\lambda$.

Components which are regularly placed in the insulating layer such as sub-constructions, modify the thermal performance of the installed insulation material, adding an extra conductivity, $\Delta\lambda$, to the declared thermal conductivity of the insulation, $F\lambda_d$, i.e. the design thermal conductivity can be larger than the declared.

The design thermal conductivity can be calculated by Formula (47)

$$\lambda_D=F\cdot\lambda_{\mathrm{d}}+\Delta\lambda \tag{47}$$

where

- λ_D is the design thermal conductivity, expressed in W/(m·K);
- λ_{d} is the declared thermal conductivity, expressed in W/(m·K);
- F is overall conversion factor for thermal conductivity;
- $\Delta\lambda$ is the extra conductivity due to regularly placed components in the insulation system, expressed in W/(m·K).

NOTE Estimated values for the extra conductivity due to regularly placed components which are part of the insulation system are given in Annex A.

In most cases, h_{si} can be very large which leads to a very small surface resistance of the flowing media inside a pipe $R_{\mathrm{l,si}}$, and therefore it can be neglected. For the external surface coefficient, h_{se}, formulae from Table 4 (4.1.3.2.2) apply. For ducts, it is necessary to include the internal surface coefficient. It can be calculated using the appropriate formulae in Table 5 (4.1.3.2.2) taking into consideration the velocity of the medium in the duct.

The reciprocal of thermal transmittance, U, is the total thermal resistance, R_{T}, for plane walls, the total linear thermal resistance, $R_{\mathrm{l,T}}$, for pipes and $R_{\mathrm{sph,T}}$ for hollow sphere.

4.1.6 Heat flow rate

The heat flow rate is expressed in W.

The heat flow rate of a plane wall is given by Formula (48):

$$\Phi = U \cdot A \cdot (\theta_i - \theta_a) \quad (48)$$

The heat flow rate of a pipe is given by Formula (49):

$$\Phi_l = U_l \cdot L \cdot (\theta_i - \theta_a) \quad (49)$$

The heat flow rate of a sphere is given by Formula (50):

$$\Phi_{sph} = U_{sph} \cdot (\theta_i - \theta_a) \quad (50)$$

4.1.7 Temperatures of the layer boundaries

The general formula for the density of the heat flow rate in a multi-layer wall is written in the general form given by Formulae (51) and (52) (see also Figure 10):

$$q = \frac{\theta_i - \theta_a}{R_T} \quad (51)$$

$$R_T = R_{si} + R_1 + R_2 + ... + R_n + R_{se} \quad (52)$$

where

$R_1 ... R_n$ are the thermal resistances of the individual layers, expressed in ($m^2 \cdot K/W$);

R_{si} and R_{se} are the surface resistances of the internal and external surfaces, respectively, expressed in ($m^2 \cdot K/W$).

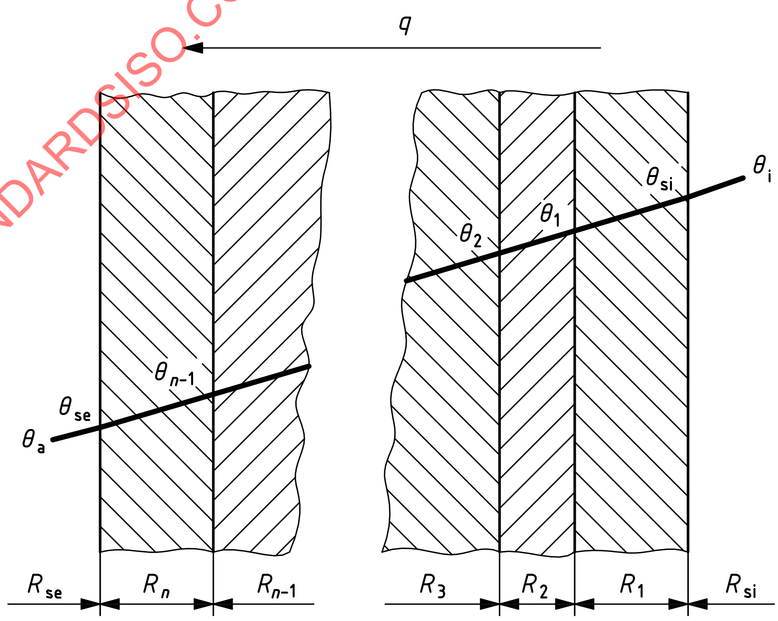


Figure 10 — Temperature distribution for a multi-layer plane wall in relation to the surface resistance and the thermal resistances of layers

The ratio between the resistance of each layer or the surface resistance with respect to the total resistance gives a measure of the temperature change across the particular layer or surface, expressed in °C, as given in Formulae (53) to (56):

$$\theta_i - \theta_{si} = \frac{R_{si}}{R_T} \cdot (\theta_i - \theta_a) \quad (53)$$

$$\theta_{si} - \theta_1 = \frac{R_1}{R_T} \cdot (\theta_i - \theta_a) \quad (54)$$

$$\theta_1 - \theta_2 = \frac{R_2}{R_T} \cdot (\theta_i - \theta_a) \quad (55)$$

$$\theta_{se} - \theta_a = \frac{R_{se}}{R_T} \cdot (\theta_i - \theta_a) \quad (56)$$

R_T is calculated for plane walls according to Formula (43), for cylindrical pipes according to Formula (44), for rectangular ducts according to Formula (45) and for spherical insulation according to Formula (46).

4.2 Determination of the influence of thermal bridges

4.2.1 General

In industrial installations there are two types of thermal bridges; thermal bridges linked to the insulation system itself and thermal bridges linked to the installation. In both cases, they represent an additional heat transfer. Their influence is explained in 4.2.2 and 4.2.3.

4.2.2 Insulation system related thermal bridges

In heat transfer calculation, the influence of insulation related thermal bridges, such as spacers or support construction, is taken into consideration using the design thermal conductivity according to 4.1.5.

4.2.3 Installation related thermal bridges

Every installation related thermal bridge, such as hangers, valves, or other fittings, represents an extra heat flow. Their influence on the heat transfer calculation shall be determined individually for each thermal bridge using the thermal bridge coefficient K according to Formula (57):

$$\Phi_{tb} = K \cdot (\theta_i - \theta_a) \quad (57)$$

where

Φ_{tb} is the additional heat flow rate due to an installation related thermal bridge, expressed in W;

K is the thermal bridge coefficient, expressed in W/K.

The thermal bridge coefficient can be also expressed as an equivalent area, ΔA, or length, ΔL, of the component where the thermal bridge is placed.

For plane walls, it can be expressed as Formula (58):

$$K = \Delta A \cdot U \quad (58)$$

where

ΔA is the equivalent area, expressed in m²;

U is the thermal transmittance of the component where the installation related thermal bridge is placed, expressed in W/(m²·K).

For pipes, it can be expressed as the Formula (59):

$$K_l = \Delta L \cdot U_l \tag{59}$$

where

ΔL is the equivalent length, expressed in m;

U_l is the linear thermal transmittance of the pipe where the installation related thermal bridge is placed, expressed in W/(m·K).

A.2 (informative) provides a calculation methodology of the thermal bridge coefficient, *K*, for valves, flanges, pumps, and insulation boxes.

4.3 Determination of total heat flow rate for plane walls, pipes and spheres

The total heat flow rate is influenced by additional transfer due to installation related thermal bridges. It is expressed in [W].

The total heat flow rate of a plane wall is given by Formula (60):

$$\Phi_T = U \cdot A \cdot (\theta_i - \theta_a) + \sum_{j=1}^{n} \Phi_{tb,j} \tag{60}$$

where, $\Phi_{tb,j}$ is the heat flow rate due to an installation related thermal bridge *j*, expressed in (W).

The total heat flow rate of a pipe is given by Formula (61):

$$\Phi_{l,T} = U_l \cdot L \cdot (\theta_i - \theta_a) + \sum_{j=1}^{n} \Phi_{tb,j} \tag{61}$$

The total heat flow rate of a sphere is given by Formula (62):

$$\Phi_{sph,T} = U_{sph} \cdot (\theta_i - \theta_a) + \sum_{i=1}^{n} \Phi_{tb,j} \tag{62}$$

4.4 Surface temperature

Usually, industrial processes involve extreme temperatures, either very high or low. If the related components are not properly insulated the extreme temperatures can produce contact accidents or adjacent processes can be affected. For these operational and/or safety reasons, a certain surface temperature is often specified.

The surface temperature is not only dependent on the design thermal conductivity but also on other operating and ambient conditions. It can be calculated using Formula (56). The operating and ambient conditions which determine the surface temperature include:

— the process medium and its temperature, pressure and mass flow,

— the insulation system (which includes the cladding),

— the ambient temperature,

— the air movement around the surface,

— the effect of adjacent radiating bodies (including irradiation from the sun),

— Installation and insulation related thermal bridges.

Outdoor environmental conditions for the calculation are measured only in a limited moment and place since wind, air temperature, solar radiation, and rain always vary. Also, the operating conditions can vary according to the specific requirements. Every deviation between the used data for the calculation and the measured conditions leads to a different surface temperature. Therefore, the surface temperature is not a suitable measure to evaluate the quality of the thermal insulation.

4.5 Prevention of surface condensation

Surface condensation depends not only on the parameters affecting the surface temperature but also on the relative humidity of the surrounding air, which very often cannot be stated accurately by the customer. The higher the relative humidity, the more the fluctuations of humidity or of surface temperatures, the risk of surface condensation increases. Using Formula (65) or (66), the necessary insulation thickness to prevent dew formation can be obtained.

The vapor pressure of the ambient air p_a in Pa at temperature θ_a is given as in Formula (63):

$$p_a = p_{sat}(\theta_a) \cdot \frac{\varphi_a}{100} \tag{63}$$

where

$p_{sat}(\theta)$ is the saturated vapor pressure at temperature θ in °C, expressed in Pa;

φ_a is the relative humidity of the ambient air, expressed in %.

If this vapor pressure is lower than the saturated vapor pressure at the surface on the ambient air side, $p_{sat}(\theta_{se})$, the condensation will not occur. Therefore, by using Formulae (56) and (52), Formula (64) is obtained.

$$p_{sat}(\theta_a) \cdot \frac{\varphi_a}{100} \le p_{sat}(\theta_{se}) = p_{sat} \cdot \left(\theta_a - \frac{R_{se}}{R_T} \cdot (\theta_a - \theta_i) \right) = p_{sat} \cdot \left(\theta_a - \frac{R_{se}}{R_{se} + R + R_{si}} \cdot (\theta_a - \theta_i) \right) \tag{64}$$

When φ_a is low and/or the resistance of insulation R is large, the possibility of condensation occurrence is low.

The necessary insulation thickness (necessary thermal resistance R) is given by Formula (65) or Formula (66),

$$R \geq \frac{R_{se} \cdot (\theta_a - \theta_i)}{\theta_a - \dfrac{237{,}3 \cdot \ln \dfrac{p_a}{610{,}5}}{17{,}269 - \ln \dfrac{p_a}{610{,}5}}} - R_{se} - R_{si} \qquad (\text{for } \theta_{se} \geq 0\ °\text{C}) \tag{65}$$

or

$$R \geq \frac{R_{se} \cdot (\theta_a - \theta_i)}{\theta_a - \dfrac{265{,}5 \cdot \ln \dfrac{p_a}{610{,}5}}{21{,}875 - \ln \dfrac{p_a}{610{,}5}}} - R_{se} - R_{si} \qquad (\text{for } \theta_{se} < 0\ °\text{C}) \tag{66}$$

where $p_{sat}(\theta)$, is approximated by an exponential function (Tetens formula), given by Formula (67) or Formula (68):

$$p_{sat} = 610{,}5 \cdot e^{\frac{17{,}269 \cdot \theta_a}{237{,}3 + \theta_a}} \qquad (\text{for } \theta_a \geq 0\ °\text{C}) \tag{67}$$

or

$$p_{sat} = 610{,}5 \cdot e^{\frac{21{,}875 \cdot \theta_a}{265{,}5 + \theta_a}} \qquad (\text{for } \theta_a < 0\ °\text{C}) \tag{68}$$

The insulation thickness that prevents surface condensation [Formulae (65) and (66)] does not warrant the prevention of interstitial condensation in the insulation material. To prevent the interstitial condensation, the design of the insulation system shall follow ISO 13788.

NOTE An example calculation is given in B.6.

5 Calculation of the temperature change in pipes, vessels, and containers

5.1 General

Temperature change and cooling time are dependent on total heat flow rate. However, for simplification purposes, 5.2 and 5.3 only consider the heat flow rate without the influence of installed thermal bridges. The calculated outputs in theses clauses can have considerable deviations from reality due to the significant influence of the installation related thermal bridges.

Accurate temperature changes or cooling time shallbe calculated with special programs or methods such as the Finite Element Method.

5.2 Longitudinal temperature change in a pipe

To obtain an accurate value of the longitudinal temperature change in a pipe with a flowing medium, i.e. liquid or gas, Formulae (69) and (70) apply:

$$|\theta_{ex} - \theta_a| = |\theta_{en} - \theta_a| \cdot e^{-\alpha \cdot L} \quad (69)$$

where

$$\alpha = \frac{U_l}{\dot{m} \cdot c_p} \quad (70)$$

and

θ_{ex} is the exit temperature of the medium, expressed in °C;

θ_{en} is the entrance temperature of the medium, expressed in °C;

θ_a is the ambient temperature, expressed in °C;

c_p is the specific heat capacity at constant pressure of the flowing medium, expressed in J/(kg·K);

$\dot{m}$ is the mass flow rate of the flowing medium, expressed in kg/s;

L is the length of the pipe, expressed in m;

U_l is the linear thermal transmittance, expressed in W/(m·K).

Formulae (69) and (70) can also be used for ducts with rectangular cross-section if U_l is replaced by U_d [Formula (45)].

Since, in practice, the allowed temperature change is often small, Formula (71) may be used for an approximate calculation:

$$\Delta\theta = \frac{\Phi_l}{\dot{m} \cdot c_p} \quad (71)$$

where

Φ_l is the heat flow rate in a pipe, expressed in W;

$\Delta\theta$ is the longitudinal temperature change, expressed in K.

Formula (71) yields results of sufficient accuracy only for relatively short pipes and a relatively small temperature change [$\Delta\theta \leq 0{,}06 \cdot (\theta_{en}-\theta_a)$].

NOTE An example calculation is given in B.2.

5.3 Temperature change and cooling times in pipes, vessels, and containers

The cooling time, t_v, for a given temperature drop, expressed in seconds, is calculated by Formula (72):

$$t_v = \frac{(\theta_{in} - \theta_a)\, m \cdot c_p \cdot \ln\left(\frac{(\theta_{in} - \theta_a)}{(\theta_{fi} - \theta_a)}\right)}{\Phi} \quad (72)$$

where

Φ is given by Formula (48) for plane walls, by Formula (49) for pipes and by Formula (50) for spheres, expressed in W;

θ_{in} is the initial medium temperature, expressed in °C;

θ_{fi} is the final medium temperature, expressed in °C;

θ_a is the ambient temperature, expressed in °C;

m is the mass of medium, expressed in kg;

c_p is the specific heat capacity of the medium, expressed in J/(kg·K).

The approximate time-dependent temperature drop can be calculated by Formula (73):

$$\Delta\theta = \frac{\Phi}{m \cdot c_p} \cdot t \tag{73}$$

NOTE 1 Calculating the cooling time, it is assumed that no heat is absorbed by the media during cooling. The cooling time obtained on this basis is the shortest, which means there is a safety factor built into the calculation (for design purposes). For small containers, the heat capacity of the container itself can be taken into account by including in Formula (72) a term analogous to that in Formula (74).

NOTE 2 An example calculation is given in B.3.

6 Calculation of cooling and freezing times of stationary liquids

NOTE An example calculation is given in B.4.

6.1 Calculation of the cooling time to prevent the freezing of water in a pipe

It is not possible to prevent the freezing of a liquid in a pipe, even though insulated, over an arbitrarily long period of time if the ambient temperature is below the freezing point of the liquid.

As soon as the liquid (normally water) in the pipe stops moving, the process of cooling starts. The heat flow rate in a pipe, Φ_l, of a stationary liquid is determined by the temperature difference, the properties of the thermal insulation as well as the pipe geometry. In addition, the energy stored in the liquid, $m_w \cdot c_{pw}$, and in the pipe material, $m_p \cdot c_{pp}$, as well as by the freezing enthalpy required to transform water to ice, shall be taken into account. If $m_p \cdot c_{pp} << m_w \cdot c_{pw}$, then $m_p \cdot c_{pp}$ may be neglected.

The time until freezing starts, t_{wp} expressed in seconds, is calculated using Formula (74):

$$t_{wp} = \frac{(\theta_{in} - \theta_a) \cdot (m_w \cdot c_{pw} + m_p \cdot c_{pp}) \cdot \ln\left(\frac{\theta_{in} - \theta_a}{\theta_{fi} - \theta_a}\right)}{\Phi_l} \tag{74}$$

where

$$\Phi_l = U_l \cdot (\theta_{in} - \theta_a) \cdot L \tag{75}$$

and

U_l is the linear thermal transmittance given by Formula (44), expressed in W/m·K;

Φ_l is the heat flow rate, expressed in W;

θ_{in} is the initial medium temperature, expressed in °C;

θ_{fi} is the final medium temperature, expressed in °C;

θ_a is the ambient temperature, expressed in °C;

c_{pw} is the specific heat capacity of water, expressed in J/(kg·K);

c_{pp} is the specific heat capacity of the pipe, expressed in J/(kg·K);

m_w is the mass of water, expressed in kg;

m_p is the mass of the pipe, expressed in kg.

In practice, for the calculation of Φ_l, both internal and external surface resistances may be neglected for insulated pipes.

If a comparison is made between uninsulated and insulated pipes, neglecting thermal bridges, the influence of the surface coefficient of the uninsulated pipe shall be taken into consideration. The heat flow rate of the uninsulated pipe is given by Formula (76):

$$\Phi_l = h_{se} \cdot (\theta_{in} - \theta_a) \cdot \pi \cdot D_e \cdot L \tag{76}$$

The approximation neglecting the internal thermal resistance is generally not allowed in case of stagnant water.

The time until freezing starts is calculated by using the procedure above with θ_{fi} equal to the freezing point of the liquid.

6.2 Calculation of the freezing time of water in a pipe

The freezing time in seconds, t_{fr}, is dependent on the heat flow rate and the diameter of the pipe when neglecting thermal bridges. It is given by Formula (77):

$$t_{fr} = \frac{f}{100} \cdot \frac{\rho_{ice} \cdot \pi \cdot D_{i,p}^2 \cdot \Delta h_{fr}}{\Phi_{l,fr} \cdot 4} \tag{77}$$

where

f is the mass fraction of water that is frozen, expressed in %;

$D_{i,p}$ is the interior pipe diameter, expressed in m;

Δh_{fr} is the latent heat of ice formation, equal to 334 000 J/kg;

ρ_{ice} is the density of ice at 0 °C, equal to 920 kg/m^3.

The heat flow rate of freezing, $\Phi_{l,fr}$, can be calculated from Formula (78):

$$\Phi_{l,fr} = \frac{\pi \cdot (0 - \theta_a)}{\frac{1}{2 \cdot \lambda_D} \cdot \ln \frac{D_e}{D_i}} \cdot L \tag{78}$$

The percentage, f, of water that is frozen shall be chosen according to a requirement, i.e. 25 % (f = 25). [8]

To allow for the effect of the reduced cross-section of slides, taps and fittings, it is recommended that the cooling and freezing times, t_{wp} and t_{fr}, given in 6.1 and this subclause, respectively, be reduced by 25 %.

7 Calculation of heat loss for underground pipelines

7.1 General

Pipelines are laid in the ground with or without thermal insulation, either in channels or directly in the soil.

NOTE An example calculation is given in B.5.

7.2 Single line without channels

The heat flow rate per metre, $q_{l,E}$, for a single underground pipe is calculated by Formula (79):

$$q_{l,E} = \frac{\theta_i - \theta_{s,E}}{R_l + R_E} \tag{79}$$

where

θ_i is the medium temperature, expressed in °C;

$\theta_{s,E}$ is the surface temperature of the soil, expressed in °C;

R_l is the linear thermal resistance of the pipe, expressed in (m·K)/W;

R_E is the linear thermal resistance of the ground for a pipe laid in homogeneous soil, expressed in (m·K)/W.

7.2.1 Uninsulated pipe

The linear thermal resistance of the ground for an uninsulated pipe, as shown in Figure 11, is given by Formula (80):

$$R_E = \frac{1}{2 \cdot \pi \cdot \lambda_E} \cdot \text{arcosh} \frac{2 \cdot H_E}{D_i} \tag{80}$$

where

λ_E is the design thermal conductivity of the ambient soil, expressed in W/(m·K);

H_E is the distance between the centre of the pipe and the ground surface, expressed in m.

For $H_E/D_i > 2{,}5$, it may be simplified to Formula (81):

$$R_E = \frac{1}{2 \cdot \pi \cdot \lambda_E} \ln \frac{4 \cdot H_E}{D_i} \tag{81}$$

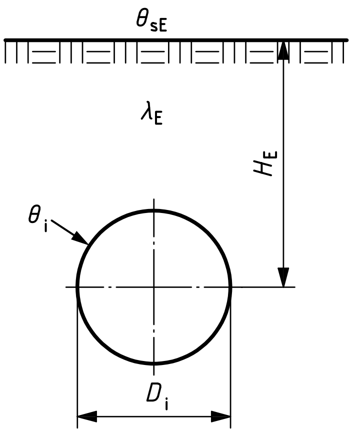
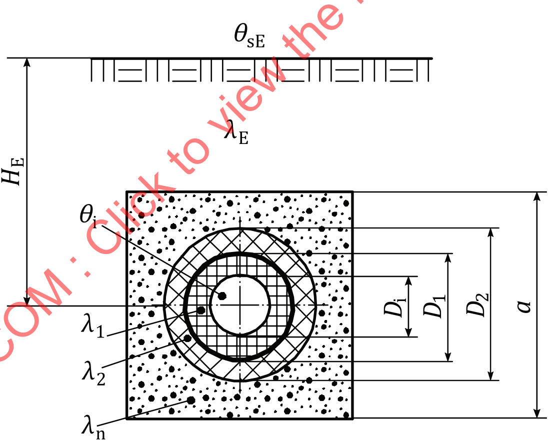


Figure 11 — Underground pipe without insulation

7.2.2 Insulated pipe

For underground pipes with insulating layers, as shown in Figure 12, the thermal resistance is calculated by Formula (82):

$$R_l = \frac{1}{2\cdot\pi}\sum_{j=1}^{n}\left(\frac{1}{\lambda_j}\cdot\ln\frac{D_{e,j}}{D_{i,j}}\right) \quad (82)$$



NOTE The concentric layers can consist of, for example, insulating material and sheathing (e.g. jacket pipe) embedded in a bottoming (e.g. sand) with a square cross-section.

Figure 12 — Underground pipe comprising several concentric layers

The square cross-section of the outer layer with side length, a, is taken into consideration with an equivalent diameter, D_n, as given by Formula (83):

$$D_n = 1{,}073 \cdot a \tag{83}$$

The internal diameter, D_i, is identical to D_0 (where j = 1). The linear thermal resistance of the ground, R_E, becomes, in this case, as given by Formula (84):

$$R_E = \frac{1}{2 \cdot \pi \cdot \lambda_E} \operatorname{arcosh} \frac{2 \cdot H_E}{D_n} \tag{84}$$

For $H_E/D_n > 2{,}5$, it may be simplified to Formula (85):

$$R_E = \frac{1}{2 \cdot \pi \cdot \lambda_E} \ln \frac{4 \cdot H_E}{D_n} \tag{85}$$

7.3 Other cases

Calculation methods are available for the determination of the heat flow rate and temperature field in the ground for several adjacent pipes, i.e. double lines or laid systems; see References [11] and [12].

In the case of commonly used jacket pipes that are laid adjacent to each other, if $\lambda_1 << \lambda_E$ (λ_1 is the thermal conductivity of the insulation layer) calculation as a single pipe is generally sufficient as an initial approach, as heat interchange between the pipes can be disregarded.

A simplified calculation is not permissible for pipes embedded in insulating masses without additional insulation.

Annex A
(informative)

Thermal bridges

A.1 Insulation related thermal bridges

A.1.1 General

Insulation related thermal bridges are components regularly placed in the insulation layer, like spacers or fasteners, which modify the thermal performance of installed insulation material by adding an extra conductivity, $\Delta\lambda$. This additional thermal conductivity influences the design thermal conductivity which shall be used in order to calculate thermal transmittances.

NOTE ISO 23993 provides calculation procedures for special insulation systems.

A.1.2 Spacers for sheet metal pipeline jackets

The additional thermal conductivity depends on several variables, like material, dimensions and frequency. The following values are approximations and can be applied to common insulation systems in thicknesses from 100 mm to 300 mm.

EXAMPLE Additions to thermal conductivity for:

steel spacers	$\Delta\lambda$ = 0,010 W/(m·K)
austenitic steel spacers	$\Delta\lambda$ = 0,004 W/(m·K)
ceramic spacers	$\Delta\lambda$ = 0,003 W/(m·K)

A.1.3 Spacers for sheet metal jackets for walls

Steel spacers in the form of a flat bar increase the thermal conductivity depending on the number of spacers per square meter and their dimensions:

$$\Delta\lambda = N \cdot \Delta\lambda_{sq} \quad (A.1)$$

where

N is the number of spacers per square metre, expressed in $1/m^2$;

$\Delta\lambda_{sq}$ is the additional thermal conductivity due to spacers per square meter, expressed in W/(m·K)/($1/m^2$).

Spacers are defined by the two dimensions of the flat bar cross-section.

EXAMPLE

Type 1, cross section 30 mm by 3 mm	$\Delta\lambda_{sq}$ = 0,003 5 W/(m·K)/($1/m^2$)
Type 2, cross section 40 mm by 4 mm	$\Delta\lambda_{sq}$ = 0,006 0 W/(m·K))/($1/m^2$)
Type 3, cross section 50 mm by 5 mm	$\Delta\lambda_{sq}$ = 0,008 5 W/(m·K))/($1/m^2$)

A.1.4 Mechanical fasteners penetrating an insulation layer

Additional $\Delta\lambda$ to thermal conductivity for fasteners depends on the number of fasteners per square metre ($1/m^2$) and on their geometry. The total addition is calculated by:

$$\Delta\lambda = N \cdot \Delta\lambda_{fas} \tag{A.2}$$

where $\Delta\lambda_{fas}$ is the additional conductivity due to fastener, expressed in W/(m·K)/($1/m^2$)

EXAMPLE

For steel fasteners with diameter 4 mm, $\Delta\lambda_{fas}$ = 0,000 7 W/(m·K)/($1/m^2$).

For austenitic steel fasteners with diameter 4 mm, $\Delta\lambda_{fas}$ = 0,000 5 W/(m·K)/($1/m^2$).

A.2 Installation-related thermal bridges in pipe insulation

A.2.1 General

Formulae for installation-related thermal bridges have been taken from VDI 4610-2:2018 [14].

A.2.2 Calculation of thermal bridge coefficient for uninsulated flanges

The thermal bridge coefficient for flange pairs K_{fl} is calculated as:

$$K_{fl} = f_{fl} \cdot h_{se} \cdot A_{fl} \tag{A.3}$$

where

f_{fl} is the correction factor for flange;

h_{se} is the external heat transfer coefficient, expressed in W/(m^2·K);

A_{fl} is the surface area for pair of flanges, expressed in m^2, as per Formula (A.5) in connection with Table A.1.

For the heat transfer coefficient h_{se} of uninsulated components, the formulae for turbulent, free convection in case of horizontal and vertical pipes as well as for radiation were adapted:

$$h_{se} = 1{,}56 \cdot \sqrt[3]{|\theta_i - \theta_a|} + 4 \cdot \varepsilon \cdot \sigma \cdot \left[\frac{\theta_i + \theta_a}{2} + 273{,}15\right]^3 \tag{A.4}$$

where

σ is the Stefan-Boltzmann constant ($\sigma = 5{,}67 \cdot 10^{-8}$ W/(m^2·K^4)); ε is

the effective emissivity of surface, see Table 3;

θ_i is the fluid temperature, expressed in °C;

θ_a is the ambient temperature, expressed in °C.

The formula is valid for fluid temperatures between −60 °C and 100 °C. Outside this temperature range, the relations given in this document in 4.1.3 shall be used.

The heat-releasing area A_{fl}, in m², results from the pipe outside diameter $D_{p,e}$, in m (uninsulated pipe diameter), as:

$$A_{fl} = a_0 + a_1 \cdot D_{p,e} + a_2 \cdot D_{p,e}{}^2 + a_3 \cdot D_{p,e}{}^3 \quad (A.5)$$

The coefficients for calculating the heat-releasing area A_{fl} are listed in Table A.1. They are valid for pipe diameters from 10 mm to 1 200 mm.

Table A.1 — Coefficients for calculating the uninsulated surface areas of flange pairs

PN	a0 m²	a1 m	a2	a3 m⁻¹
2,5	0,011	0,881	1,559	−0,753
6	0,006	1,200	0,455	0,182
10	−0,003	1,613	−0,390	1,094
16	−0,017	1,743	0,296	0,639
25	−0,017	1,794	1,268	0,471
40	0,000	1,193	4,087	0,000
63	0,002	2,068	1,136	5,393
100	−0,015	3,042	−5,236	25,646
160	0,002	2,092	4,923	3,662
320	−0,007	3,388	4,784	30,713

The heat loss was calculated by means of FEM simulation using the external heat transfer coefficients as per Formula (A.4) and real geometries. From the heat losses obtained by FEM simulation, the correction factor f_{fl} was calculated using Formula (A.6).

$$f_{fl} = \frac{\Phi_{fl,FEM}}{(\theta_i - \theta_a)} \quad (A.6)$$

Flanges with different internal heat transfer coefficients were investigated and are represented in the correction factor with two reference values: 20 W/(m²·K) and 1 000 W/(m²·K). Table A.2 lists the factors for the uninsulated flanges.

Table A.2 — Correction factor f_{fl} for calculating the heat loss from uninsulated flanges

Internal heat transfer coefficient h_{si} W/(m²K)	Factor f_{fl}
1 000	1,09 to $5{,}21 \cdot 10^{-4} \cdot \theta_i$
20	0,644 to $1{,}5 \cdot 10^{-3} \cdot \theta_i$ to $1{,}32 \cdot 10^{-6} \cdot \theta_i^2$

If the end disc from the piping insulation system has contact with the pipe, an allowance of 15 % shall be added to the factor f_{fl}.

NOTE Pipes transporting superheated steam or heat transfer oils normally have an internal heat transfer coefficient from 750 W/(m²·K) to 3 000 W/(m²·K), for flowing air, the heat transfer coefficient is approximately 20 W/(m²·K).

A.2.3 Calculation of thermal bridge coefficient for uninsulated valves

Different types of valves have been grouped in Table A.3, which provides heat loss coefficients, correction factors as a function of the fluid temperature and the formulae for the heat-releasing surface of various valves.

According to Formula (A.3) the heat losses from the thermal bridges can thus be determined based on thermal bridges coefficient K or K_A for valves. The formulae in Table A.3 contain the heat losses from the protruding elements with hand wheels.

Table A.3 — Determination of thermal bridge coefficients for uninsulated valves

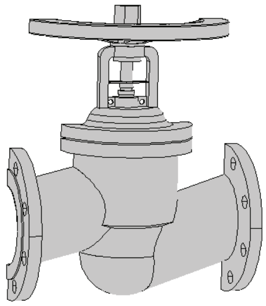
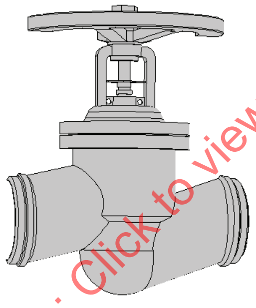
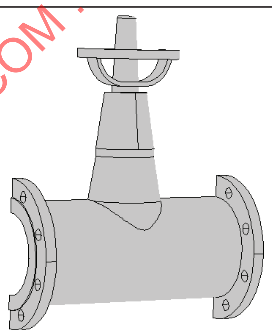
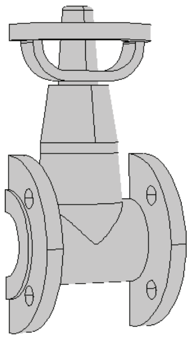
N	Type of valve	Geometry	**Calculation formulae** **Range of validity** (DN, nominal diameter in mm, PN, nominal pressure in bar) f_A, correction factor; A_A, heat-releasing area, m^2; K_A, thermal bridge coefficient, W/K.
1	**Manual blocking valve for industrial installations flanged connection**		DN 15–DN 200; PN = (16...25) $f_A = -\frac{0,43}{1\,000} \cdot \theta_i + 0,708\,6$ $A_A = 23,2 \cdot D_{p,e}^2 + 1,37 \cdot D_{p,e} + 0,071\,8$ $K_A = f_A \cdot h_{se} \cdot A_A + K_{fl}$
2	**Manual blocking valve for industrial installations, welded connection**		DN 15–DN 200; PN = (16...25) $f_A = -\frac{0,31}{1\,000} \cdot \theta_i + 0,737\,4$ $A_A = 24,6 \cdot D_{p,e}^2 + 0,437 \cdot D_{p,e} + 0,073\,1$ $K_A = f_A \cdot h_{se} \cdot A_A$
3	**Shut-off device with hand wheel for heating installations, flanged connection, elongated design**		DN 15–DN 200; PN = 16 $f_A = -\frac{0,45}{1\,000} \cdot \theta_i + 0,813\,3$ $A_A = 8,78 \cdot D_{p,e}^2 + 1,48 \cdot D_{p,e} + 0,040\,2$ $K_A = f_A \cdot h_{se} \cdot A_A + K_{fl}$
4	**Shut-off device with hand wheel for heating installations, flanged connection, short design**		DN 15–DN 200; PN = (6...16) $f_A = -\frac{0,46}{1\,000} \cdot \theta_i + 0,748$ $A_A = 3,21 \cdot D_{p,e}^2 + 1,62 \cdot D_{p,e} + 0,032\,2$ $K_A = f_A \cdot h_{se} \cdot A_A + K_{fl}$

Table A.3 *(continued)*

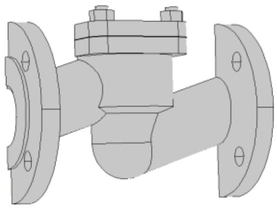
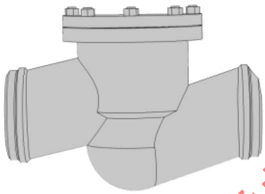
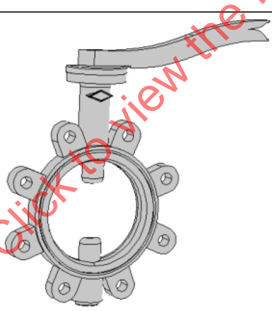
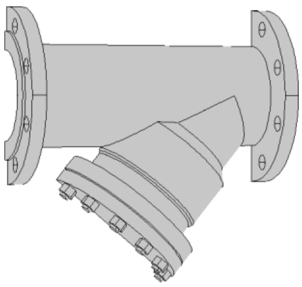
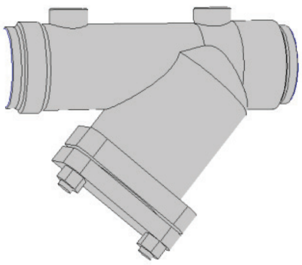
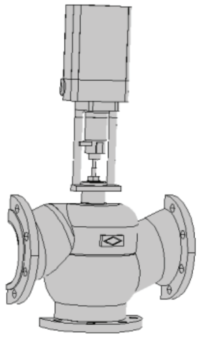
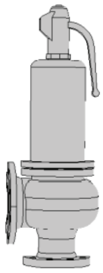
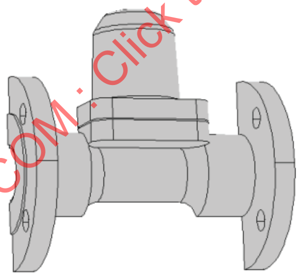
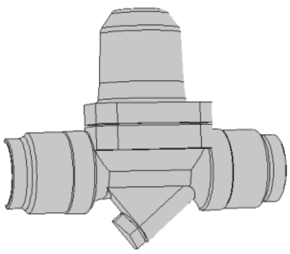
N	Type of valve	Geometry	**Calculation formulae** **Range of validity** (DN, nominal diameter in mm, PN, nominal pressure in bar) f_A, correction factor; A_A, heat-releasing area, m^2; K_A, thermal bridge coefficient, W/K.
5	**Check valve, flanged connection**		DN 15–DN 200; PN = (16...25) $f_A = -\frac{0,34}{1\ 000} \cdot \theta_i + 0,956$ $A_A = 16,8 \cdot D_{p,e}^2 + 0,31 \cdot D_{p,e} + 0,078$ $K_A = f_A \cdot h_{se} \cdot A_A + K_{fl}$
6	**Check valve, welded connection**		DN 15–DN 200; PN = (6...16) $f_A = -\frac{0,29}{1\ 000} \cdot \theta_i + 0,966$ $A_A = 15,8 \cdot D_{p,e}^2 + 0,51 \cdot D_{p,e} + 0,023$ $K_A = f_A \cdot h_{se} \cdot A_A$
7	**Shut-off flap**		DN 15–DN 200; PN = (6...16) $f_A = -\frac{0,49}{1\ 000} \cdot \theta_i + 0,714$ $A_A = 1,07 \cdot D_{p,e}^2 + 0,629 \cdot D_{p,e} + 0,013\ 5$ $K_A = f_A \cdot h_{se} \cdot A_A + K_{fl}$
8	**Dirt arrester, flanged**		DN 15–DN 200; PN = (6...16) $f_A = -\frac{0,37}{1\ 000} \cdot \theta_i + 0,938$ $A_A = 18,4 \cdot D_{p,e}^2 + 0,969 \cdot D_{p,e} + 0,034\ 6$ $K_A = f_A \cdot h_{se} \cdot A_A + K_{fl}$
9	**Dirt arrester, welded**		DN 15–DN 200; PN = (6...16) $f_A = -\frac{0,32}{1\ 000} \cdot \theta_i + 0,951$ $A_A = 15,1 \cdot D_{p,e}^2 + 0,754 \cdot D_{p,e} + 0,028$ $K_A = f_A \cdot h_{se} \cdot A_A$

Table A.3 *(continued)*

N	Type of valve	Geometry	**Calculation formulae** **Range of validity** (DN, nominal diameter in mm, PN, nominal pressure in bar) f_A, correction factor; A_A, heat-releasing area, m²; K_A, thermal bridge coefficient, W/K.
10	**Adjustment valve**		DN 15–DN 200; PN = (6...16) $f_A = -\frac{0{,}30}{1\ 000} \cdot \theta_i + 0{,}600$ $A_A = 14{,}8 \cdot D_{p,e}^2 + 2{,}8 \cdot D_{p,e} + 0{,}39$ $K_A = f_A \cdot h_{se} \cdot A_A + K_{fl}$
11	**Safety valve**		DN 15–DN 200; PN = (6...16) $f_A = -\frac{0{,}50}{1\ 000} \cdot \theta_i + 0{,}709$ $A_A = 14{,}0 \cdot D_{p,e}^2 + 1{,}7 \cdot D_{p,e} + 0{,}092$ $K_A = f_A \cdot h_{se} \cdot A_A + K_{fl}$
12	**Condensate drain, flanged**		DN 15–DN 50; PN = 40 $f_A = -\frac{0{,}52}{1\ 000} \cdot \theta_i + 0{,}875$ $A_A = 1{,}4 \cdot D_{p,e} + 0{,}069$ $K_A = f_A \cdot h_{se} \cdot A_A + K_{fl}$
13	**Condensate drain, welded**		DN 15–DN 50; PN = 40 $f_A = -\frac{0{,}41}{1\ 000} \cdot \theta_i + 0{,}934$ $A_A = 1{,}6 \cdot D_{p,e} + 0{,}020$ $K_A = f_A \cdot h_{se} \cdot A_A$

Each FEM calculation was carried out assuming six different temperatures between 50 °C and 500 °C. The internal heat transfer coefficient was 1 000 W/(m²·K), the external heat transfer coefficient of the component was considered according to Formula (A.4). Different pressure stages and special designs were not considered.

When the valve to be investigated is not listed in Table A.3, the procedure shall be as follows:

Valves with flange connection:

An additional flange pair shall be added in this case:

$$K_A = f_A \cdot h_{se} \cdot A_A + K_{fl} \tag{A.7}$$

The proper correction factor is determined according to:

$$f_A = -\frac{0,33}{1\ 000} \cdot \theta_i + 0,629 \tag{A.8}$$

The valve surface area shall be obtained from the manufacturer or shall be calculated.

Valves with welded connection:

In this case:

$$K_A = f_A \cdot h_{se} \cdot A_A \tag{A.9}$$

The proper correction factor is determined according to:

$$f_A = -\frac{0,21}{1\ 000} \cdot \theta_i + 0,638 \tag{A.10}$$

The valve surface area shall be obtained from the manufacturer or shall be calculated.

A.2.4 Calculation of thermal bridge coefficient for uninsulated pumps

The formulae have been developed from FEM simulations for heating-water pumps (≤ DN 150) with direct flange-mounted drive. Formula (A.11) represents the thermal bridge coefficient for an uninsulated pump:

$$K_P = (14 \cdot D_{p,e} - 0,09) \cdot \left(\frac{4 \cdot \theta_i}{1\ 000} + 0,83 \right) \tag{A.11}$$

where $D_{p,e}$ pipe outside diameter (uninsulated pipe), expressed in m.

A.2.5 Calculation of heat losses for insulation boxes

Insulation boxes for flanges, valves, and other components are calculated using the pipe model, see Figure A.1.

The linear density of heat flow rate for insulation boxes, $q_{l,Ka}$ [W/m], is calculated using Formula (A.12):

$$q_{l,Ka} = \frac{\Phi_{tb}}{l_i} = \frac{\theta_i - \theta_a}{R_{si} + R_{se} + R_{Ka}} \tag{A.12}$$

with the internal heat transfer resistance R_{si}

$$R_{si} = \frac{1}{h_{si} \cdot D_{p,i} \cdot \pi} \tag{A.13}$$

and the external heat transfer resistance R_{se}

$$R_{se} = \frac{1}{h_{se} \cdot (D_{p,e} + 2 \cdot d_R + 2 \cdot d_{Ka}) \cdot \pi} \tag{A.14}$$

The thermal resistance of the insulation box, R_{Ka}, is determined using Formula (A.15)

$$R_{Ka} = \frac{1}{f_{Ka}} \cdot \frac{\ln\left(\dfrac{D_{p,e} + 2 \cdot d_R + 2 \cdot d_{Ka}}{D_{p,e} + 2 \cdot d_R}\right)}{2 \cdot \pi \cdot \lambda_{Ka}} \tag{A.15}$$

where

l_i is the insulation box inside length, expressed in m;

θ_i is the fluid temperature, expressed in °C;

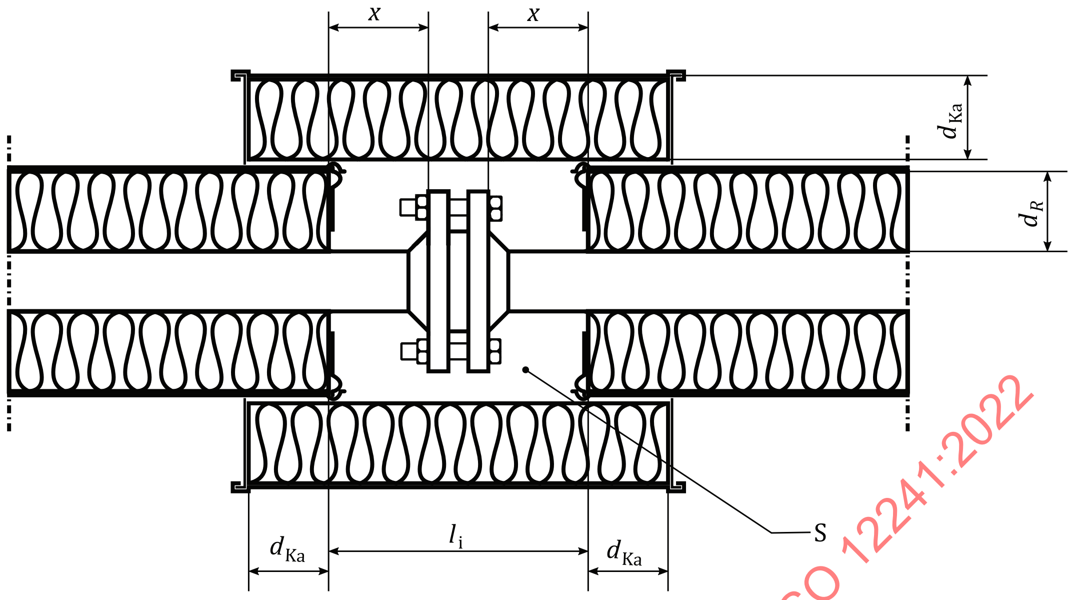
θ_a is the ambient temperature, expressed in °C;

$D_{p,e}$ is the pipe outside diameter (uninsulated pipe), expressed in m;

d_R is the insulation layer thickness of the pipe, expressed in m;

d_{Ka} is the insulation layer thickness of the insulation box, expressed in m;

λ_{Ka} is the thermal conductivity of insulation box material at mean temperature of insulation box, expressed in W/(m·K).



Key

l_i insulation box inside length

d_R insulation layer thickness of the pipe

d_{Ka} insulation layer thickness of the insulation box

x bolt length + 20 mm

S space inside the insulation box

Figure A.1 — Geometry of an insulation box

NOTE The space inside the insulation box (S) can be filled with or without insulation.

The dimensionless insulation box factor f_{Ka} depends on the pipe diameter, the temperature, the cladding material, the type of end disc, and the insulation box fill.

Guide values for insulation box factors are listed in Table A.4 and Table A.5.

Table A.4 — Mean values for insulation box factors f_{Ka} (void without fill)

Cladding	Without end disc	End disc in contact	Thermally separated end disc
Steel sheet	1,58	2,73	2,5
Stainless steel	1,44	1,88	1,84
Aluminium	2,65	--	9,53
Plastic	1,4	1,4	1,4

Table A.5 — Mean values for insulation box factors f_{Ka} (void filled with insulation material)

Cladding	Without end disc	End disc in contact	Thermally separated end disc
Steel sheet	0,68	1,97	1,39
Stainless steel	0,65	1,25	1,06
Aluminium	1,23	--	3,65
Plastic	0,94	0,94	0,94

The thermal bridge coefficient for an insulation box can be calculated using Formula (A.16):

$$K_{Ka} = \frac{l_i}{R_{si} + R_{se} + R_{Ka}} \tag{A.16}$$

The insulation box inside lengths for valve boxes can be approximated using Formula (A.17) and Table A.6

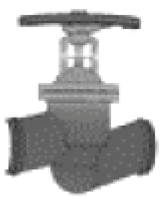
$$l_i = c_1 + c_0 \cdot D_{p,e} \tag{A.17}$$

For flange boxes, the insulation box inside length including counter flange and 20 mm can be calculated from the outside pipe diameter $D_{p,e}$ and the nominal pressure PN using the approximation Formula (A.18):

$$l_i = 0{,}228 + 0{,}044 \cdot D_{p,e} \cdot PN^{0{,}722} \tag{A.18}$$

Elements protruding from insulation boxes shall be calculated as specific thermal bridges.

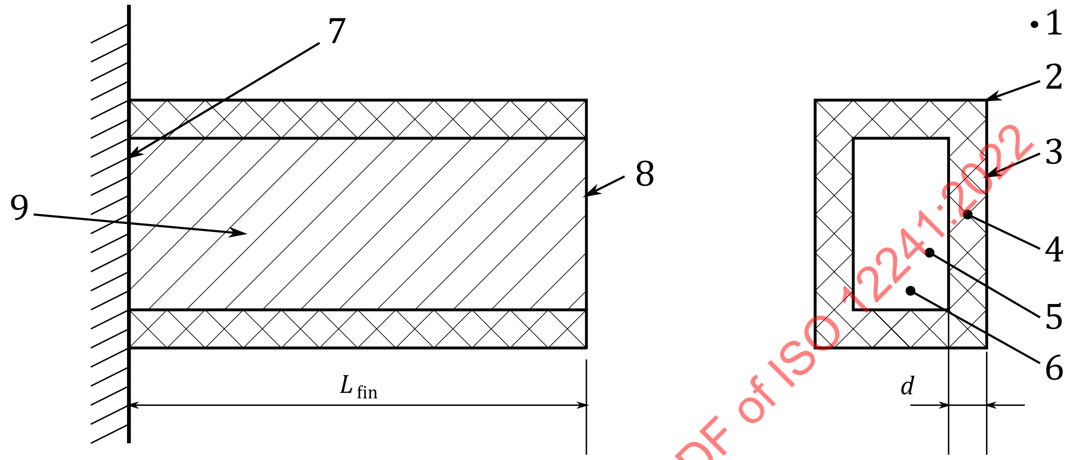
Table A.6 — Calculation parameters for valve lengths up to PN 40

Type	c_o	c_1
Flanged valve 	2,660 6	0,275 1
Flanged short valve 	0,928 1	0,298 5
Intermediate flange valve 	0,456 1	0,221 3
Welded valve 	2,336 8	0,087 6

A.3 Projecting thermal bridges of roughly constant cross-section

A.3.1 General

A metal fin of roughly constant cross-section, which can be surrounded by an insulating layer (see Figure A.2), has the temperature, θ_b, at the fin base and releases heat to the environment at temperature θ_a.



Key

1 point of measurement of the ambient temperature, θ_a, expressed in °C

2 point of measurement for calculating the surface coefficient of heat transfer, h_{se}, expressed in W/(m²·K)

3 point of measurement of the perimeter, P_{fin}, expressed in m

4 point of measurement for calculating the design thermal conductivity of the insulation, λ_D, expressed in W/(m·K)

5 cross-sectional area of the thermal bridge, $A_{cs,fin}$, expressed in m²

6 point of measurement for calculating the design thermal conductivity of the thermal bridge material, λ_{fin}, expressed in W/(m·K)

7 point of measurement of the temperature at the fin base, θ_b , expressed in °C

8 point of measurement for calculating the thermal transmittance on the frontal of the thermal bridge, U_{fa}, expressed in W/(m²·K)

9 heat flow rate, Φ_{fin}, expressed in W

L_{fin} length of the thermal bridge, expressed in m

d thickness of the insulation, expressed in m

Figure A.2 — Mounting with insulation

The thermal bridge coefficient, K_{fin}, is given by Formula (A.19) and expressed in W:

$$K_{fin} = \frac{A_{cs,fin} \cdot \lambda_{fin} \cdot k}{L_{fin}}$$

$$K_A = f_A \cdot h_{se} \cdot A_A \quad (A.19)$$

The dimensionless factor, k, can be calculated from Formula (A.20) or determined by using Figure A.3 after having calculated the dimensionless parameters B and B_{fa} using Formulae (A.21), (A.22), (A.23), and (A.24):

$$k = B \cdot \frac{B \cdot \sinh B + B_{fa} \cdot \cosh B}{B \cdot \cosh B + B_{fa} \cdot \sinh B} \quad (A.20)$$

where

$$\cosh B = \frac{e^B + e^{-B}}{2} \quad (A.21)$$

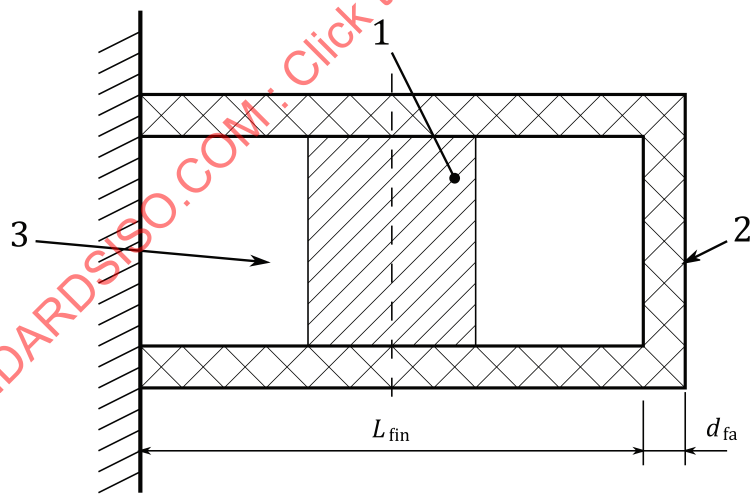
$$\sinh B = \frac{e^B - e^{-B}}{2} \quad (A.22)$$

$$B = L_{fin} \cdot \sqrt{\frac{P_{fin}}{\lambda_{fin} \cdot A_{cs,fin} \cdot \left(\frac{1}{h_{se}} + \frac{d}{\lambda}\right)}} \quad (A.23)$$

$$B_{fa} = \frac{U_{fa} \cdot L_{fin}}{\lambda} \quad (A.24)$$

The external surface coefficient, h_{se}, can be calculated in accordance with 4.1.3, while the thermal transmittance, U_{fa}, on the frontal area can be assessed for each case in A.3.2 to A.3.4.

A.3.2 Insulated or free frontal area



Key

1 area of the cross-section of the thermal bridge, $A_{cs,fin}$, expressed in m^2

2 point of measurement for calculating the heat transfer coefficient, h_{se}, expressed in $W/(m^2 \cdot K)$

3 heat flow, Φ_{fin}, expressed in W

d_{fa} thickness of the insulation on the frontal area, expressed in m

L_{fin} length of the thermal bridge, expressed in m

Figure A.3 — Mounting with insulated or free frontal area

The thermal transmittance in case of the insulated frontal area can be calculated from Formula (A.25):

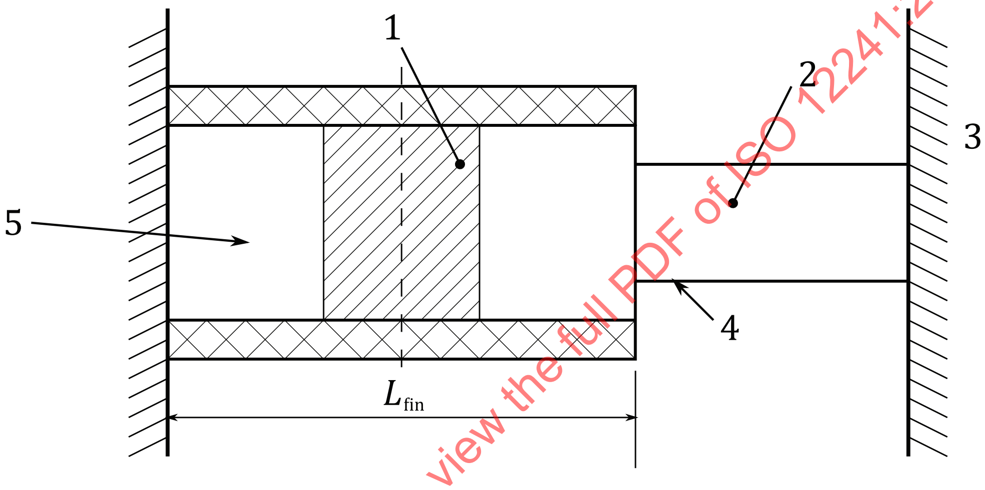
$$\frac{1}{U_{\text{fa}}} = \frac{d_{\text{fa}}}{\lambda} + \frac{1}{h_{\text{se}}} \quad \text{(A.25)}$$

The thermal transmittance in case of the free frontal area can be calculated from Formula (A.26):

$$U_{\text{fa}} = h_{\text{se}} \quad \text{(A.26)}$$

A.3.3 Anchoring structure

An anchoring structure of any desired shape, made from material that has good conducting properties and is based in the ground or in concrete, is attached to the frontal area (see Figure A.4).



Key

1 area of the cross-section of the thermal bridge, $A_{\text{cs,fin}}$, expressed in m²

2 area of the surface of the anchorage, A_{anc}, expressed in m²

3 concrete or ground

4 point of measurement for calculating the heat transfer coefficient, h_{se}, expressed in W/(m²·K)

5 heat flow, Φ, expressed in W

Figure A.4 — Mounting with anchorage, in ground

If A_{anc} is the total surface of the anchoring structure, the maximum value for U_{fa} can be assessed by Formula (A.27):

$$U_{\text{fa}} = \frac{A_{\text{anc}} \cdot h_{\text{se}}}{A_{\text{cs,fin}}} \quad \text{(A.27)}$$

Where the shaping of the connecting structure is permitted, the latter can itself be considered in turn as a fin, and the heat flow on the interface between the two fins and the resulting value of U_{fa} can be calculated in accordance with Formula (A.28).

A.3.4 Fin frontal area is in good heat contact with free metal supports

The rough estimate given by Formula (A.28) applies:

$$\frac{1}{U_{fa}} = 0 \qquad (A.28)$$

For geometries with widely varying cross-section, a calculation using numerical methods is recommended.