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Building environment design — Design, dimensioning, installation and control of embedded radiant heating and cooling systems —

Part 1: Definition, symbols, and comfort criteria

Conception de l'environnement des bâtiments — Conception, construction et fonctionnement des systèmes de chauffage et de refroidissement par rayonnement —

Partie 1: Définition, symboles et critères de confort

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Foreword

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ISO 11855-1 was prepared by Technical Committee ISO/TC 205, *Building environment design*.

ISO 11855 consists of the following parts, under the general title *Building environment design — Design, dimensioning, installation and control of embedded radiant heating and cooling systems*:

— Part 1: *Definition, symbols, and comfort criteria*

— Part 2: *Determination of the design and heating and cooling capacity*

— Part 3: *Design and dimensioning*

— Part 4: *Dimensioning and calculation of the dynamic heating and cooling capacity of Thermo Active Building Systems (TABS)*

— Part 5: *Installation*

— Part 6: *Control*

Part 1 specifies the comfort criteria which should be considered in designing embedded radiant heating and cooling systems, since the main objective of the radiant heating and cooling system is to satisfy thermal comfort of the occupants. Part 2 provides steady-state calculation methods for determination of the heating and cooling capacity. Part 3 specifies design and dimensioning methods of radiant heating and cooling systems to ensure the heating and cooling capacity. Part 4 provides a dimensioning and calculation method to design Thermo Active Building Systems (TABS) for energy-saving purposes, since radiant heating and cooling systems can reduce energy consumption and heat source size by using renewable energy. Part 5 addresses the installation process for the system to operate as intended. Part 6 shows a proper control method of the radiant heating and cooling systems to ensure the maximum performance which was intended in the design stage when the system is actually being operated in a building.

Introduction

The radiant heating and cooling system consists of heat emitting/absorbing, heat supply, distribution, and control systems. The ISO 11855 series deals with the embedded surface heating and cooling system that directly controls heat exchange within the space. It does not include the system equipment itself, such as heat source, distribution system and controller.

The ISO 11855 series addresses an embedded system that is integrated with the building structure. Therefore, the panel system with open air gap, which is not integrated with the building structure, is not covered by this series.

The ISO 11855 series shall be applied to systems using not only water but also other fluids or electricity as a heating or cooling medium.

The object of the ISO 11855 series is to provide criteria to effectively design embedded systems. To do this, it presents comfort criteria for the space served by embedded systems, heat output calculation, dimensioning, dynamic analysis, installation, operation, and control method of embedded systems.

Building environment design — Design, dimensioning, installation and control of embedded radiant heating and cooling systems —

Part 1: Definition, symbols, and comfort criteria

1 Scope

This part of ISO 11855 specifies the basic definitions, symbols, and a comfort criteria for radiant heating and cooling systems.

The ISO 11855 series is applicable to water based embedded surface heating and cooling systems in residential, commercial and industrial buildings. The methods apply to systems integrated into the wall, floor or ceiling construction without any open air gaps. It does not apply to panel systems with open air gaps which are not integrated into the building structure.

The ISO 11855 series also applies, as appropriate, to the use of fluids other than water as a heating or cooling medium. The ISO 11855 series is not applicable for testing of systems. The methods do not apply to heated or chilled ceiling panels or beams.

2 Normative references

ISO 7726:1998, *Ergonomics of the thermal environment — Instruments for measuring physical quantities*

ISO 7730:2005, *Ergonomics of the thermal environment — Analytical determination and interpretation of thermal comfort using calculation of the PMV and PPD indices and local thermal comfort criteria*

ISO 13731:2003, *Ergonomics of the thermal environment — Vocabulary and symbols*

3 Terms and definitions

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

2.1
additional thermal resistance
thermal resistance representing layers added to the building structure and acting mostly as thermal resistances because of their own low thermal inertia

EXAMPLE Carpets, moquette, and suspended ceilings.

2.2
average specific thermal capacity of the internal walls
thermal capacity related to one square metre of the internal walls

NOTE Since internal walls are shared with other rooms, then just half of the total specific thermal capacity of the wall must be taken into account, since the second half is influenced by the opposite rooms that are considered to be at the same thermal conditions as the one under consideration.

2.3
average surface temperature
$\theta_{s,m}$
average value of all surface temperatures in the occupied or peripheral area

2.4
basic characteristic curve
curve or formula reflecting the relationship between the heat flux and the mean surface temperature difference

NOTE This depends on heating/cooling and surface (floor/wall/ceiling) but not on the type of embedded system.

2.5
calculation time step
length of time considered for the calculation of the temperatures and heat flows in the room and slab

NOTE This is typically assumed to equal 3 600 s.

2.6
circuit
section of system connected to a distributor which can be independently switched and controlled

2.7
circuit total thermal resistance
thermal resistance representing the circuit as a whole, determining a straight connection between the water inlet temperature and the mean temperature at the pipe level

NOTE It includes the water flow thermal resistance, the convection thermal resistance at the pipe inner side, the pipe thickness thermal resistance, and the pipe level thermal resistance.

2.8
clothing insulation
basic clothing insulation that is the resistance of a uniform layer of insulation covering the entire body that has the same effect on sensible heat flow as the actual clothing under standardized (static, wind-still) conditions

NOTE The definition of clothing insulation also includes the uncovered parts of the body, e.g. the head. It is described as the intrinsic insulation from the skin to the clothing surface, not including the resistance provided by the air layer around the clothed body, and is expressed in the clo unit or in m^2K/W; 1 clo = 0,155 m^2K/W.

2.9
conductive region of the slab
region of the slab that includes the pipes with thermal conductivities of the layers higher than 0,8 W/(m·K)

NOTE Due to the subdivision of the slab into an upper slab and a lower slab, the conductive region is also subdivided into an upper conductive region and a lower conductive region.

2.10
convection thermal resistance at the pipe inner side
thermal resistance associated to the convection heat transfer taking place between the water flowing in the pipe and the pipe inner side, thus connecting the mean water temperature along the circuit with the mean temperature of the pipe inner side

2.11
convective heating and cooling system
system that directly conditions the air in the room for the purpose of heating and cooling

2.12
convective peak load
maximum cooling load to be extracted by a virtual convective system used to keep comfort conditions in the room

2.13
daily average temperature of the conductive region of the slab
average temperature of the conductive region of the slab during the day

2.14
design cooling capacity
$Q_{H,c}$
thermal output by a cooling surface at design conditions

2.15
design cooling load
$Q_{N,c}$
required thermal output necessary to achieve the specified design conditions in outside summer design conditions

2.16
design sensible cooling load
required sensible thermal output necessary to achieve the specified design conditions in outside summer design conditions

2.17
design dew point
$\theta_{Dp,des}$
dew point determined for the design

2.18
design supply temperature of heating/cooling medium
$\theta_{V,des}$
value of flow water temperature with the thermal resistance of the chosen floor covering, at maximum value of heat flux q_{max}

NOTE The flow and the supply temperature are the same throughout the EN 1264 series.

2.19
design heat flux
q_{des}
heat flow divided by the heating or cooling surface, taking into account the surface temperature required to reach the design thermal capacity of a surface heated or cooled space, Q_H, reduced by the thermal capacity of any supplementary heating or cooling equipment, if applicable

2.20
design heating capacity
$Q_{H,h}$
thermal output from a heating surface at design conditions

2.21
design heating load
$Q_{N,h}$
required thermal output necessary to achieve the specified design conditions in outside winter design conditions

NOTE When calculating the value of the design heat load, the heat flow from embedded heating systems into neighbouring rooms is not taken into account.

2.22
design heating/cooling medium differential temperature
$\Delta\theta_{H,des}$
temperature difference at design heat flux

2.23
design heating medium differential supply temperature
$\Delta\theta_{V,des}$
temperature difference between the design supply medium temperature and indoor temperature at design heat flux

2.24
design heating/cooling medium flow rate
m_H
mass flow rate in a circuit which is needed to achieve the design heat flux

2.25
design indoor temperature
θ_i
operative temperature at the centre of the conditioned space used for calculation of the design load and capacity

NOTE The operative temperature is considered relevant for thermal comfort assessment and heat loss calculations. This value of internal temperature is used for the calculation method.

2.26
distributor
common connection point for several circuits

2.27
draught
unwanted local cooling of a body caused by movement of air and related to temperature

2.28
electric floor (wall, ceiling) heating system
several panel systems that convert electrical energy to heat, raising the temperature of conditioned indoor surfaces and the indoor air

2.29
embedded surface heating and cooling system
system consisting of circuits of pipes embedded in floor, wall or ceiling construction, distributors and control equipments

2.30
equivalent heat transmission coefficient
K_H
coefficient describing the relationship between the heat flux from the surface and the heating/cooling medium differential temperature

2.31
family of characteristic curves
curves denoting the system-specific relationship between the heat flux, q, and the required heating medium differential temperature $\Delta\theta_H$ for conduction resistance of various floor coverings

2.32
heat flux
q
heat flow between the space and surface divided by the heated/cooled surface

NOTE For heating it is a positive value and for cooling it is a negative value.

2.33
heat transfer coefficient
h_t
combined convective and radiative heat transfer coefficient between the heated or cooled surface and the space operative temperature (design indoor temperature)

2.34
heating or cooling surface
surface (floor, wall, ceiling) covered by the embedded surface heating/cooling system between the pipes at the outer edges of the system with the addition of a strip at each edge of width equal to half the pipe spacing, but not exceeding 0,15 m

2.35
heating or cooling surface area
A_F
area of surface (floor, wall, ceiling) covered by the embedded surface heating/cooling system between the pipes at the outer edges of the system with the addition of a strip at each edge of width equal to half the pipe spacing, but not exceeding 0,15 m

2.36
heating/cooling capacity for circuit
Q_{HC}
heat exchange between a pipe circuit and the conditioned room

2.37
heating/cooling medium differential temperature
$\Delta\theta_H$
logarithmically determined average difference between the temperature of the heating/cooling medium and the design indoor temperature

2.38
internal convective heat gains
convective contributions by internal heat gains acting in the room

NOTE Mainly due to people or electrical equipment.

2.39
internal radiant heat gains
radiant contributions by internal heat gains acting in the room

NOTE Mainly due to people or electrical equipment.

2.40
internal thermal resistance of the slab conductive region
total thermal resistance connecting the pipe level with the middle points of the upper conductive region and lower conductive region of the slab

2.41
limit curves
curves in the field of characteristic curves showing the pattern of the limit heat flux depending on the heating medium differential temperature and the floor covering

2.42
limit heat flux
q_G
heat flux at which the maximum or minimum permissible surface temperature is achieved

2.43
limit heating medium temperature difference
$\Delta\theta_{H,G}$
intersection of the system characteristic curve with the limit curve

2.44
maximum cooling power
maximum thermal power of the cooling equipment, referring only to the room under consideration

2.45
maximum permissible surface temperature
q_{max}
required design heat flux in the room in order to design supply medium temperature

2.46
maximum operative temperature allowed for comfort conditions
maximum operative temperature allowed in the room according to comfort requirements in cooling conditions

2.47
maximum operative temperature drift allowed for comfort conditions
maximum drift in operative temperature allowed in the room according to comfort requirements

2.48
maximum permissible surface temperature
$\theta_{S,max}$
maximum temperature permissible for physiological reasons or for the physical building, for calculation of the limit curves, which may occur at a point on the surface (floor, wall, ceiling) in the occupied or peripheral area depending on the particular usage at a temperature drop σ of the heating medium equal to 0

2.49
mean radiant temperature
uniform surface temperature of an imaginary black enclosure in which an occupant would exchange the same amount of radiant heat as in the actual non-uniform enclosure

2.50
mean surface temperature difference
difference between the average surface temperature $\theta_{S,m}$ and the design indoor temperature θ_i

NOTE It determines the heat flux.

2.51
metabolic rate
rate of transformation of chemical energy into heat and mechanical work by aerobic and anaerobic metabolic activities within an organism, usually expressed in terms of unir area of the total body surfaces

NOTE The metabolic rate varies with each activity. It is expressed in the met unit or in W/m^2; 1 met = 58,2 W/m^2. 1 met is the energy produced per unit surface area of a sedentary person at rest. The surface area of an average person can be determined by Dubois Equation, Body Surface Area (m^2) = 0,20 247 × Height (m)^{0,725} × Weight (kg)^{0,425}.

2.52
minimum permissible surface temperature
$\theta_{S,min}$
minimum temperature permissible for physiological reasons or for the physical building, for calculation of the limit curves, which may occur at a point on the surface (floor, wall, ceiling) in the occupied or peripheral area depending on the particular usage at a temperature drop σ of the heating medium equal to 0

2.53
nominal heat flux
q_N
limit heat flux achieved without surface covering

2.54
nominal heating/cooling medium differential temperature
$\Delta\theta_N$
absolute temperature difference at nominal heat flux q_N

2.55
non-active area
area of the surface not covered by a heating/cooling system

2.56
number of active surfaces
number of surfaces in straight thermal connection with the pipe level, so that it distinguishes whether the slab transfers heat both through the floor side and through the ceiling side or whether the ceiling side is much more active than the floor side

NOTE Two active surfaces when the conductive region extends from the floor to the ceiling, one active surface otherwise.

2.57
number of operation hours of the circuit
length of time during which the system runs in the day

2.58
occupied area
A_A
surface area which is heated or cooled, excluding peripheral area

2.59
occupied zone
part of the conditioned zone in which persons normally reside and where requirements as to the internal environment are satisfied

NOTE Normally, the zone between the floor and 1,8 m above the floor and 1,0 m from outside walls/windows and heating/cooling appliances, 0,5 m from internal surfaces.

2.60
open air gap
air gap in the floor, wall, or ceiling construction, where air exchange with space or the outside may occur

2.61
operative temperature
OT
θ_O
uniform temperature of an imaginary black enclosure in which an occupant exchanges the same amount of heat by radiation and convection as in the actual non-uniform environment

2.62
orientation of the room
orientation of the main windowed external wall: East, South or West

NOTE It is used to determine when the peak load from heat gains happens, since internal heat gains are considered almost constant and the widest variation is expected to happen in solar heat gains.

2.63
outward heat flux
q_U
heat flow which is exchanged through the construction with unconditioned spaces, another building entity, the ground or outdoor air

2.64
peak load
maximum cooling load to be extracted by the system used to keep comfort conditions in the room

2.65
peripheral area
A_R
surface area which is heated or cooled to a higher or lower temperature

NOTE It is generally an area of 1 m maximum in width along exterior walls. It is not an occupied area.

2.66
pipe level
virtual plane where the pipe circuit lies

2.67
pipe level thermal resistance
thermal resistance associated to the 2-D conduction heat transfer taking place between the pipes and the embedding layer, virtually referred to the pipe level, thus connecting the mean temperature of the pipe outer side with the mean temperature at the pipe level

2.68
pipe spacing
spacing or distance between pipes embedded in the surface

2.69
pipe thickness thermal resistance
thermal resistance associated to the conduction heat transfer taking place through the pipe wall, thus connecting the mean temperature of the pipe inner side with the mean temperature of the pipe outer side

2.70
predicted mean vote
PMV
index that predicts the mean value of the thermal sensation votes of a large group of persons on a 7-point thermal sensation scale

2.71
predicted percentage of dissatisfied
PPD
index that establishes a quantitative prediction of the percentage of thermally dissatisfied people who are either too warm of too cool

2.72
primary air convective heat gains
heat gains acting in the room due to the infiltration or primary air inflow

2.73
radiant surface heating and cooling system
heating and cooling system that controls the temperature of indoor surfaces on the floor, walls, or ceiling

2.74
radiant temperature asymmetry
difference between the plane radiant temperature of the two opposite sides of a small plane element

2.75
relative air velocity
air velocity relative to the occupant, including body movements

2.76
regional dew point
$\theta_{\mathrm{Dp,R}}$
dew point specified depending on the climatic conditions of the region

2.77
running mode
running mode of the circuit that defines whether the system is currently switched on or off

2.78
slab
horizontal building structure separating two rooms placed one below the other, hence being the ceiling for one and the floor for the other

2.79
solar heat gains
solar heat gains acting in the room due to high-frequency radiation transmission through windows

2.80
specific daily energy gains
total energy to be extracted during the day in order to avoid a net increase in internal energy in the room and maintain comfort conditions

2.81
supplementary heating equipment
additional heating facility with the additional heat output Q_{out}

EXAMPLE Convector, radiators.

NOTE It may have its own control equipment.

2.82
surface heating and cooling components
insulating layer (for thermal and/or impact noise insulation), protection layer (to protect the insulating layer), the pipes or plane sections, the load and thermal distribution layer where pipes are embedded, covering and other items

NOTE 1 Other items include conducting devices, peripheral strips, attachment items, etc.

NOTE 2 Components may differ depending on the system.

2.83
system insulation
insulation with the thermal resistance $R_{\lambda,\text{ins}}$ according to ISO 11855-5:2012, Table 2, to limit the heat loss of heating and cooling systems

NOTE For floor heating and cooling systems, as a rule the thermal resistance $R_{\lambda,\text{ins}}$ is provided by the insulation layers which are integral parts of the system. National rules shall be consulted for this subject. For wall and ceiling heating and cooling systems, the thermal resistance $R_{\lambda,\text{ins}}$ may be determined taking into account the effective thermal resistance of the building structure.

2.84
Thermally Active Building System
TABS
water based heating and cooling system where the pipes are embedded in the central concrete core of a building construction

2.85
temperature drop
σ
difference between the supply and return temperature of the heating/cooling medium in a circuit

2.86
temperature of the heating/cooling medium
θ_m
average temperature between the supply and the return temperature defined as $\theta_\text{m} = \theta_\text{i} + \Delta\theta_\text{H}$

2.87
thermal node
node summarizing the thermal behavior of a material or air volume as regards heat transfer calculations

2.88
thermal output of surface system
Q_S
sum of the products of the heating or cooled surfaces of a space with the associated design heat fluxes

NOTE For heating it is a positive value. For cooling it is a negative value.

2.89
total convective heat gains
sum of all convective contributions from heat gains acting in the room, hence it is the sum of internal convective heat gains, primary air convective heat gains and a fraction of transmission heat gains

2.90
total radiant heat gains
sum of all radiant contributions from heat gains acting in the room (internal radiant heat gains, solar heat gains and a fraction of transmission heat gains)

2.91
transmission heat gains
heat gains acting in the room due to conductive heat transmission through the external walls and windows

2.92
vertical air temperature difference
difference in air temperature measured at 1,1 m and 0,1 m above the floor

NOTE The distances 1,1 m and 0,1 m are theoretical average values for head and ankle height of a sedentary person.

2.93
wall surface thermal resistance
thermal resistance representing the connection between the core of the internal walls and their surface on the room side

NOTE It usually corresponds to the layer of plaster covering the internal side of the walls.

2.94
water based floor (wall, ceiling) heating and cooling system
floor (wall, ceiling) system where pipes carrying water with or without additives as a medium are laid in the floor (wall, ceiling)

2.95
water flow thermal resistance
thermal resistance that expresses the variation in temperature of the water flowing in the pipe along the circuit, so it connects the water inlet temperature with the mean water temperature along the circuit

4 Symbols and abbreviations

For the purposes of this part of ISO 11855, the symbols and abbreviations in Table 1 apply.

Table 1 — Symbols and abbreviations

Symbol	Unit	Quantity
A_A	m^2	Area of the occupied surface
A_F	m^2	Area of the heating/cooling surface
A_R	m^2	Area of the peripheral surface
A_W	m^2	Total area of internal vertical walls (i.e. vertical walls, external façades excluded)
a_i	-	Parameter factors for calculation of characteristic curves
B, B_G, B_0	$W/(m^2 \cdot K)$	Coefficients depending on the system
b_u	-	Calculation factor depending on the pipe spacing
C	$J/(m^2 \cdot K)$	Specific thermal capacity of the thermal node under consideration
C_W	$J/(m^2 \cdot K)$	Average specific thermal capacity of the internal walls
c_j	$J/(kg \cdot K)$	Specific heat of the material constituting the j-th layer of the slab
c_W	$J/(kg \cdot K)$	Specific heat of water
D	m	External diameter of the pipe, including sheathing where used
d_a	m	External diameter of the pipe
d_i	m	Internal diameter of the pipe
d_M	m	External diameter of sheathing
E_Day	kWh/m^2	Specific daily energy gains
$F_\mathrm{v\ F\text{-}C}$	-	View factor between the floor and the ceiling
$F_\mathrm{v\ F\text{-}EW}$	-	View factor between the floor and the external walls
$F_\mathrm{v\ F\text{-}W}$	-	View factor between the floor and the internal walls
f_s	-	Design safety factor
f_{rm}^h	-	Running mode (1 when the system is running; 0 when the system is switched off) in the h-th hour

Table 1 *(continued)*

Symbol	Unit	Quantity
H_A	W/K	Heat transfer coefficient between the thermal node under consideration and the air thermal node ("A")
H_C	W/K	Heat transfer coefficient between the thermal node under consideration and the ceiling surface thermal node ("C")
$H_{Circuit}$	W/K	Heat transfer coefficient between the thermal node under consideration and the circuit
$H_{CondDown}$	W/K	Heat transfer coefficient between the thermal node under consideration and the next one
H_{CondUp}	W/K	Heat transfer coefficient between the thermal node under consideration and the previous one
H_{conv}	-	Fraction of internal convective heat gains acting on the thermal node under consideration
H_F	W/K	Heat transfer coefficient between the thermal node under consideration and the floor surface thermal node ("F")
$H_{Inertia}$	W/K	Coefficient connected to the inertia contribution at the thermal node under consideration
H_{IWS}	W/K	Heat transfer coefficient between the thermal node under consideration and the internal wall surface thermal node ("IWS")
H_{Rad}	-	Fraction of total radiant heat gains impinging on the thermal node under consideration
$h_{A\text{-}C}$	W/(m^2·K)	Convective heat transfer coefficient between the air and the ceiling
$h_{A\text{-}F}$	W/(m^2·K)	Convective heat transfer coefficient between the air and the floor
$h_{A\text{-}W}$	W/(m^2·K)	Convective heat transfer coefficient between the air and the internal walls
h_c	W/(m^2·K)	Convective heat transfer coefficient
$h_{F\text{-}C}$	W/(m^2·K)	Radiant heat transfer coefficient between the floor and the ceiling
$h_{F\text{-}W}$	W/(m^2·K)	Radiant heat transfer coefficient between the floor and the internal walls
h_r	W/(m^2·K)	Radiant heat transfer coefficient
h_t	W/(m^2·K)	Total heat transfer coefficient (convection + radiation) between surface and space
J	-	Number of layers constituting the slab as a whole
J_1	-	Number of layers constituting the upper part of the slab
J_2	-	Number of layers constituting the lower part of the slab
K_H	W/(m^2·K)	Equivalent heat transmission coefficient
K_{WL}	-	Parameter for heat conducting devices
k_{CL}	-	Parameter for heat conducting layer
k_{fin}	-	Parameter for heat conducting devices
L_{fin}	m	Width of fin (horizontal part of heat conducting device seen as a heating fin)
L_R	m	Length of installed pipes
L_{WL}	m	Width of heat conducting devices
m	-	Exponents for determination of characteristic curves
m_H	kg/s	Design heating/cooling medium flow rate
$\dot{m}_{H,sp}$	kg/(m^2·s)	Specific water flow in the circuit, calculated on the area covered by the circuit
m_j		Number of partitions of the *j*-th layer of the slab
n	-	Actual number of iteration in iterative calculations
n, n_G	-	Exponents
n_h	h	Number of operation hours of the circuit
n^{Max}	-	Maximum number of iterations allowed in iterative calculations
	W	Maximum cooling power reserved to the circuit under consideration in the h-th hour
$P^{Max}_{Circuit,Spec}$	W	Maximum specific cooling power (per floor square metre)
PB	-	Polybutylene

Table 1 *(continued)*

Symbol	Unit	Quantity
PE-MDX	-	Cross-linked polyethylene, medium density
PE-RT-Systems	-	Polyethylene of raised temperature resistance
PE-X	-	Cross-linked polyethylene
PP	-	Polypropylene
PVC-C	-	Chlorinated polyvinyl chloride
Q_C^h	W	Heat flow impinging on the ceiling surface (“C”) in the h-th hour
$Q_{Circuit}^h$	W	Heat flow extracted by the circuit in the h-th hour
Q_{Conv}^h	W	Total convective heat gains in the h-th hour
Q_{des}	W	Design capacity
Q_F^h	W	Heat flow impinging on the floor surface (“F”) in the h-th hour
$Q_{IntConv}^h$	W	Internal convective heat gains in the h-th hour
Q_{IntRad}^h	W	Internal radiant heat gains in the h-th hour
Q_{IWS}^h	W	Heat flow impinging on the internal wall surface (“IWS”) in the h-th hour
Q_{N}	W	Design load
$Q_{\mathrm{N,c}}$	W	Design cooling load
$Q_{\mathrm{N,h}}$	W	Design heating load
$Q_{\mathrm{N,l}}$	W	Design latent cooling load
$Q_{\mathrm{N,s}}$	W	Design sensible cooling load
Q_{out}	W	Heat output of supplementary heating equipments
$Q_{PrimAir}^h$	W	Primary air convective heat gains in the h-th hour
Q_{Rad}^h	W	Total radiant heat gains in the h-th hour
Q_{s}	W	Thermal output of surface heating-cooling
Q_{Sun}^h	W	Solar heat gains in the room in the h-th hour
Q_{Transm}^h	W	Transmission heat gains in the h-th hour
Q_{W}	W/m^2	Average specific cooling power
q	W/m^2	Heat flow density at the surface
q_{A}	W/m^2	Heat flow density in the occupied area
q_{des}	W/m^2	Design heat flow density
$q_{\mathrm{des,A}}$	W/m^2	Design heat flow density of occupied area
$q_{\mathrm{des,R}}$	W/m^2	Design heat flow density of peripheral area
q_{G}	W/m^2	Limit heat flow density
q_{i}	W/m^2	Inward specific heat flow
q_{max}	W/m^2	Maximum design heat flow density
q_{N}	W/m^2	Nominal heat flow density
q_{R}	W/m^2	Heat flow density in the peripheral area
q_{U}	W/m^2	Outward heat flow density

Table 1 *(continued)*

Symbol	Unit	Quantity
$R_{\text{Add C}}$	$(m^2 \cdot K)/W$	Additional thermal resistance covering the lower side of the slab
$R_{\text{Add F}}$	$(m^2 \cdot K)/W$	Additional thermal resistance covering the upper side of the slab
RCAC	K/W	Convection thermal resistance connecting the air thermal node (“A”) with the ceiling surface thermal node (“C”)
RCAF	K/W	Convection thermal resistance connecting the air thermal node (“A”) with the floor surface thermal node (“F”)
RCAW	K/W	Convection thermal resistance connecting the air thermal node (“A”) with the internal wall surface thermal node (“IWS”)
R_{int}	$(m^2 \cdot K)/W$	Internal thermal resistance of the slab conductive region
$R_{\text{L},p}$	$(m^2 \cdot K)/W$	Conduction thermal resistance connecting the p-th thermal node with the boundary of the (p+1)-th thermal node
R	$(m^2 \cdot K)/W$	Generic thermal resistance
R_{o}	$(m^2 \cdot K)/W$	Partial inwards heat transmission resistance of surface structure
R_{r}	$(m^2 \cdot K)/W$	Pipe thickness thermal resistance
R_{t}	$(m^2 \cdot K)/W$	Circuit total thermal resistance
R_{u}	$(m^2 \cdot K)/W$	Partial outwards heat transmission resistance of surface structure
$R_{\text{U},p}$	$(m^2 \cdot K)/W$	Conduction thermal resistance connecting the p-th thermal node with the boundary of the (p-1)-th thermal node
R_{Walls}	$(m^2 \cdot K)/W$	Wall surface thermal resistance
R_{W}	$(m^2 \cdot K)/W$	Water flow thermal resistance
R_{X}	$(m^2 \cdot K)/W$	Pipe level thermal resistance
R_{Z}	$(m^2 \cdot K)/W$	Convection thermal resistance at the pipe inner side
$R_{\lambda,\text{B}}$	$(m_2 \cdot K)/W$	Thermal resistance of surface covering
$R_{\lambda,\text{ins}}$	$(m^2 \cdot K)/W$	Thermal resistance of thermal insulation
S	m	Thickness of the screed (excluding the pipes in type A systems)
s_{h}	m	In Type B systems, thickness of thermal insulation from the outward edge of the insulation to the inward edge of the pipes (see Figure 2)
s_{ins}	m	Thickness of thermal insulation
s_{l}	m	In Type B systems, thickness of thermal insulation from the outward edge of the insulation to the outward edge of the pipes (see Figure 2)
s_{r}	m	Thickness of the pipe wall
s_{u}	m	Thickness of the layer inward from the pipe
s_{WL}	m	Thickness of heat conducting device
s_1	m	Thickness of the upper part of the slab
s_2	m	Thickness of the lower part of the slab
v_{max}	m/s	Maximum air velocity
W	m	Pipe spacing
x	m	Distance to the surface
α	$W/(m^2 \cdot K)$	Heat exchange coefficient
δ_j	m	Thickness of the j-th layer of the slab
η	-	Rate of the extra capacity of the heat source
Δt	s	Calculation time step
$\Delta\theta$	K	Generic temperature difference
$\Delta\theta_{\text{Comfort}}^{\text{Max}}$	°C	Maximum operative temperature drift allowed for comfort conditions
$\Delta\theta_{\text{H}}$	K	Heating/cooling medium differential temperature

Table 1 *(continued)*

Symbol	Unit	Quantity
$\Delta\theta_{H,des}$	K	Design heating/cooling medium differential temperature
$\Delta\theta_{H,G}$	K	Limit of heating/cooling medium differential temperature
$\Delta\theta_{N}$	K	Nominal heating/cooling medium differential temperature
$\Delta\theta_{V}$	K	Heating/cooling medium differential supply temperature
$\Delta\theta_{V,des}$	K	Design heating/cooling medium differential supply temperature
θ_{A}^{h}	°C	Temperature of the air thermal node (“A”) in the h-th hour
$\theta_{Comfort,Ref}$	°C	Maximum operative temperature allowed for comfort conditions in the reference case
θ_{c}^{h}	°C	Temperature of the ceiling surface thermal node (“C”) in the h-th hour
$\theta_{Comfort}^{Max}$	°C	Maximum operative temperature allowed for comfort conditions
θ_{d}	°C	External design temperature
$\theta_{F,max}$	°C	Maximum surface temperature
$\theta_{F,min}$	°C	Minimum surface temperature
θ_{F}^{h}	°C	Temperature of the floor surface thermal node (“F”) in the h-th hour
θ_{IW}^{h}	°C	Temperature of the core of the internal walls thermal node (“IW”) in the h-th hour
θ_{IWS}^{h}	°C	Temperature of the internal wall surface thermal node (“IWS”) in the h-th hour
θ_{i}	°C	Design indoor temperature
θ_{MR}^{h}	°C	Room mean radiant temperature in the h-th hour
θ_{m}	°C	Temperature of the heating/cooling medium
θ_{Op}^{h}	°C	Room operative temperature in the h-th hour
θ_{p}^{h}	°C	Temperature of the p-th thermal node in the h-th hour
θ_{PL}^{h}	°C	Temperature of the pipe level thermal node (“PL”) in the h-th hour
θ_{Slab}^{Av}	°C	Daily average temperature of the conductive region of the slab
$\theta_{i,min}$	°C	Minimum indoor air temperature
θ_{o}	°C	Operative temperature
θ_{r}	°C	Mean radiant temperature
$\theta_{s,m}$	°C	Average surface temperature
$\theta_{s,max}$	°C	Maximum surface temperature
$\theta_{s,min}$	°C	Minimum surface temperature
θ_{R}	°C	Return temperature of heating/cooling medium
θ_{V}	°C	Supply temperature of heating/cooling medium
$\theta_{V,des}$	°C	Design supply temperature of heating/cooling medium
$\theta_{V,des,max}$	°C	Maximum heating water flow temperature
θ_{u}	°C	Indoor temperature in an adjacent space
$\theta_{Water,In}^{h}$	°C	Water inlet actual temperature in the h-th hour

Table 1 *(continued)*

Symbol	Unit	Quantity
$\theta_{Water,In}^{Setp,h}$	°C	Water inlet set-point temperature in the h-th hour
$\theta_{Water,In,Ref}^{Setp}$	°C	Water inlet set-point temperature in the reference case
$\theta_{Water,Out}^{h}$	°C	Water outlet temperature in the h-th hour
λ	W/(m·K)	Thermal conductivity
λ_b	W/(m·K)	Thermal conductivity of the material of the pipe embedded layer
λ_{ins}	W/(m·K)	Thermal conductivity of the thermal insulation layer
λj	W/(m·K)	Thermal conductivity of the material constituting the j-th layer of the slab
λ_r	W/(m·K)	Thermal conductivity of the material constituting the pipe
σ	K	Temperature drop $\theta_V - \theta_R$
ξ	°C	Actual tolerance in iterative calculations
ξ_{max}	°C	Maximum tolerance allowed in iterative calculations
ρ_j	kg/m^3	Density of the material constituting the j-th layer of the slab
ϕ	-	Conversion factor for temperatures
ψ	-	Content by volume of the attachment burrs in the screed
ω	various	Slope of correlation curves

5 Comfort criteria

An occupant's thermal comfort would be the primary objective that any HVAC system pursues. Radiant heating and cooling systems can be used as primary or hybrid systems which are combined with an air system and provide unique and cost-effective approaches to dealing with numerous conditions affecting human thermal comfort. Radiant heating and cooling systems directly transfer heat in order to condition a space to a specific temperature. Meanwhile, radiant heating and cooling systems can be used to directly provide heat to humans as well as to spaces.

As long as the occupants are radiantly heated in a radiant heating system, the same comfort level can be maintained with a lower air temperature in comparison to a convective heating system. For radiant cooling systems, with a higher air temperature in comparison to convective cooling, maintaining the same comfort level is possible. Therefore, compared with conventional heating and cooling systems, it is possible to reduce the energy loss due to ventilation, and infiltration is possible while maintaining the same comfort level.

Thermal comfort can be defined as the psychological condition that expresses satisfaction with the thermal environment. Therefore, thermal comfort would be evaluated by asking all the occupants if they are satisfied with their thermal environment. However, in order to design and control radiant heating and cooling systems, it is necessary to predict the thermal comfort in a room without resorting to a polling result.

To provide an acceptable thermal environment to the occupants, the requirements for general thermal comfort, e.g. predicted mean vote (PMV), operative temperature, and local thermal comfort (surface temperature, vertical air temperature differences, radiant temperature asymmetry, draft, etc.) shall be taken into account.

5.1 General thermal comfort

Operative temperature and PMV can be used as a single index to evaluate general thermal comfort. For sizing and dimensioning of radiant heating and cooling systems, operative temperature can be chosen as general thermal comfort because the systems use radiative heat transfer from the surfaces. In order to design a hybrid system combined with convective systems or to design considering factors related to the occupant such as metabolic rate and clothing, a more comprehensive index, PMV can be used as a general thermal comfort criterion. Meanwhile, when operative temperature and PMV are used in the control as well as the design, it is possible not only to obtain a better comfort condition but also to save energy in buildings.

5.1.1 Operative temperature

In order to provide acceptable thermal conditions, two parameters, air temperature and mean radiant temperature, should be taken into account. The combined influence of these two temperatures is expressed as the operative temperature. In a place where air velocities are low (< 0,2m/s) or the difference between mean radiant temperature and air temperature is small (< 4°C), the operative temperature can be approximated with the simple average of air and mean radiant temperature. This means that air temperature and mean radiant temperature have an equal importance with respect to the level of thermal comfort in a space. Compared with a convective heating and cooling system, a radiant heating system can achieve the same level of operative temperature at a lower air temperature and a radiant cooling system at a higher air temperature.

5.1.2 Definition

Operative temperature is defined as the temperature of a uniform isothermal black enclosure in which the occupant exchanges the same amount of heat by radiation and convection as in the actual non-uniform environment (see ISO 7730). In a physical sense, operative temperature is the temperature that the occupant perceives in his or her surroundings based on convection and radiation. Operative temperature can be said to be the weighted average of the air temperature and the mean radiation temperature (MRT). The weighted average is calculated by the combination of the convective heat transfer coefficient and the linearized radiant heat transfer coefficient.

$$\theta_\mathrm{o} = \frac{h_\mathrm{r} \cdot \theta_\mathrm{r} + h_\mathrm{c} \cdot \theta_\mathrm{i}}{h_\mathrm{r} + h_\mathrm{c}}$$

where

h_r is radiant heat transfer coefficient;

h_c is convective heat transfer coefficient.

5.1.2.1 Relationship to thermal comfort

The air temperature alone is not an appropriate thermal indicator because the room in a building shows a non-uniform radiant field. The air temperature does not account for the heat loss caused by radiant energy exchange with the walls, windows or the radiant heating system. When much heat exchange occurs by radiant energy, operative temperature is a better index for general thermal comfort.

When the values of humidity, air velocities, metabolic rate and clothing insulation are given, comfort zone can be determined. Comfort zone is defined by the range of operative temperature which can provide an acceptable thermal environmental condition or the combination of mean radiant temperature and air temperature which people accept thermally.

The appropriate range of operative temperature that satisfies comfort conditions can be different depending on the occupant’s clothing insulation and metabolic rate. ISO 7730 shows the optimum operative temperature and the permissible temperature range as a function of clothing and activity for each of the three categories. ISO 7726 describes methods and techniques to measure operative temperature.

5.1.3 PMV (predicted mean vote)/PPD (predicted percentage of dissatisfied)

If the humidity and air velocities along with air temperature are taken into account, more precise comfort criteria can be provided. Radiant heating and cooling systems can be hybrid when combined with a convection system. Especially for a hybrid system which combines the radiant heating/cooling and ventilation system, humidity and air velocities along with air and operative temperature can be important factors that determine thermal comfort. Thus, in order to evaluate and control the thermal comfort of the system, an index which takes into account all of these factors is necessary. PMV/PPD is one of the most common indices used for this purpose.

Factors that affect the PMV are metabolic rate, clothing insulation, air temperature, mean radiant temperature, air speed and relative humidity. PPD is an index expressing the thermal comfort level as a percentage of thermally dissatisfied people, and is directly determined from PMV. The PPD index is based on the assumption that people voting ± 2 or ± 3 on the thermal sensation scale are dissatisfied, and the simplification that PPD is symmetric around a neutral PMV (= 0). Both PMV and PPD are based on general (whole body) thermal comfort. Much more details, including calculation methods of PMV and PPD, are described in ISO 7730.

PMV can be used as a comfort indicator to evaluate the whole thermal comfort of the space in which radiant heating and cooling systems are applied. It can also be used as a mathematical model for devices to control comfort in the buildings which apply radiant heating and cooling systems. To get much better comfort control than the general thermostat which only uses the air temperature, PMV can be used as control variable.

5.2 Local thermal comfort

The local thermal discomfort caused by a vertical air temperature difference between the feet and the head, by an asymmetric radiant field, by local convective cooling (draft), or by contact with a hot or cold floor shall be taken into account in determining conditions for acceptable thermal comfort.

5.2.1 Surface temperature limit

In radiant heating and cooling systems, floor, walls and ceilings can be used as the heat transfer surface for heating and cooling. For this reason, special care shall be paid to the surface temperature limit on the floor and wall with which the occupants can have direct contact.

5.2.1.1 Floor heating and cooling

The floor temperature has a direct impact on the comfort of the feet or buttocks. In ISO 7730, a floor temperature range of 19°C to 29°C is recommended in the space with sedentary and/or standing occupants wearing normal shoes. This is a limiting factor when deciding the capacity of floor heating and cooling systems. For heating, the maximum temperature is 29°C and for cooling, the minimum temperature is 19°C. However, this temperature range of 19°C to 29°C might be changed by the factor of whether the occupants wear shoes or not, or whether they usually sit on the floor or stand up in the occupied zone. Thus, the range of the surface temperature can be different depending on lifestyle habits. For this reason, it is recommended to follow the widely accepted standards of each country when deciding on the optimum range of floor surface temperature. See A.1 for recommended ranges for the floor temperature of radiant heating and cooling systems.

A floor temperature range of 19°C to 29°C is based on an average between seated and standing occupants. Seated persons would prefer 1 K higher floor temperatures and standing persons 1 K lower surface temperatures. At a higher metabolic rate, even lower floor temperatures may be acceptable. If it is outside the occupied zone, i.e. within 1 m from outside walls/windows, 35°C is acceptable for the design floor temperature. In spaces where occupants may have bare feet (bathrooms, swimming pools, and dressing rooms), the most appropriate comfort floor temperature depends on the floor covering material.

Especially in spaces where people have bare feet, the range of comfort temperature depends on the material of the floor. More detailed information on the floor temperature can be found in ISO/TS 13732-2. Information on the range of comfortable temperature depending on floor material can be found in A.1.

For an electric heating system, an electrically-heated floor may cause discomfort and even skin burn if occupants have prolonged contact with the floor. This is due to the constant supply of heat from an electrical heating source, whereas, for a water based heating system, the increase in surface temperature is limited by the water temperature. Therefore it is important to control the electrical heating source in order to keep the floor surface temperature under the lower limit of discomfort and skin burn. The relation between floor temperature and skin temperature in an electric heating system is described in A.2. Skin temperatures that cause discomfort and burns are explained in A.3

5.2.1.2 Wall heating and cooling

For wall heating, the maximum recommended surface temperature is in the range of 35°C to 50°C. The maximum temperature depends on factors such as whether occupants may easily have contact with the surface or whether buildings are used for more sensitive persons such as children or the elderly. When a skin temperature is 42°C to 45°C, there is a risk of burns and pain. The losses to the rear walls and its influence on neighbouring spaces should be taken into account.

For wall cooling, the surface temperature should be higher than the dew point temperature to avoid condensation and cold draft caused by the cooled surface.

Annex B shows the equations for calculation of maximum air velocity and minimum air temperature along the floor caused by cold draft from cooled surfaces. By using these equations, it is possible to determine the minimum permissible temperature of the wall surface in order to prevent discomfort caused by low air temperature and high air velocity.

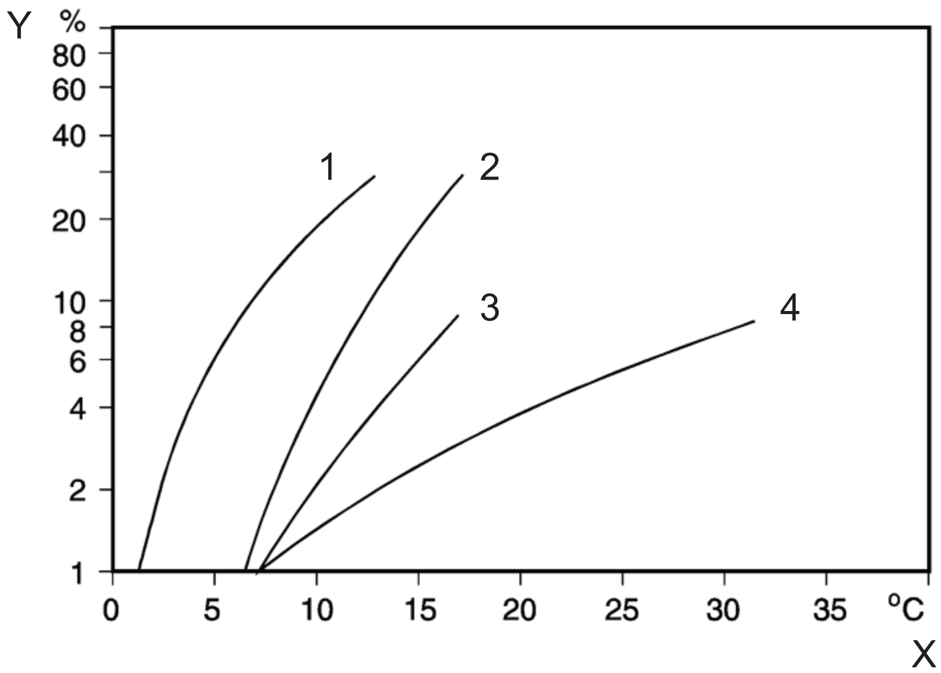
5.2.2 Radiant temperature asymmetry

In all practical thermal environments, a radiation field has an asymmetric feature to some degree. If the asymmetry is sufficiently large, then it can cause discomfort. For example, discomfort might be felt by persons exposed to the asymmetric radiation such as the open door of a furnace, direct sunlight, heated ceilings or cooled windows or walls.

The radiant temperature asymmetry is the difference between the plane radiant temperatures of the two opposite sides of a small plane element. The detailed calculation method of percentage dissatisfied due to the radiant temperature asymmetry is described in ISO 7730.

5.2.2.1 Relationship to thermal comfort

The human body is most sensitive to radiant asymmetry caused by warm ceilings or cool walls/windows. Thus, for making ceilings heated by applying the radiant heating, the radiant temperature asymmetry should be maintained at less than 5°C (in relation to a small horizontal plane 0,6 m above the floor). For making walls/windows cooled by applying the radiant cooling, the radiant temperature asymmetry should be less than 10°C (in relation to a small vertical plane 0,6 m above the floor). Figure 1 shows discomfort level (percentage dissatisfied) due to radiant temperature asymmetry in case of ceiling cooling/heating and walls-windows cooling/heating.



Key

X radiant temperature asymmetry
Y dissatisfied
1 warm ceiling
2 cool wall
3 cool ceiling
4 warm wall

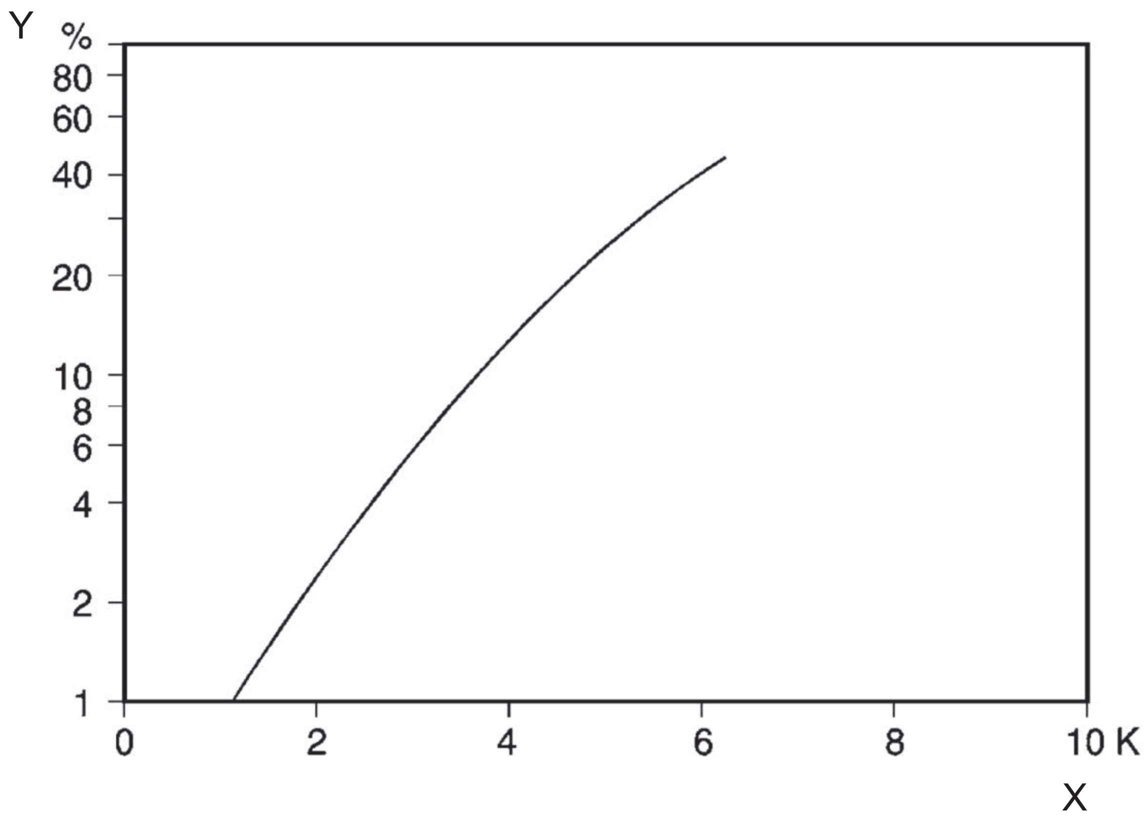
Figure 1 — Local thermal discomfort caused by radiant temperature asymmetry

5.2.3 Vertical air temperature difference

One of the important features of radiant heating and cooling systems is that it is possible to get uniform temperature conditions from floor to ceiling. According to measurements, the application of the floor heating/cooling and the large wall panel heating units under the window has a uniform temperature profile. The more convective systems (baseboard under window, warm air system) or high temperature systems result in 2°C to 3°C gradients between floor and ceiling and even up to 7°C in more severe cases. For systems relying on much more convection, the temperature profile becomes less uniform.

Thermal stratification that results in the air temperature at the head level being warmer than at the ankle level may cause thermal discomfort. Thermal stratification in the opposite direction is rare, but is preferred by occupants. Therefore, it is not addressed in this part of ISO 11855.

The differences in air temperature from the ankle level to the head level are recommended to be within 3°C. Figure 2 can be used in conjunction with the PPD limit for vertical temperature differences to determine the allowable ranges of vertical temperature differences.



Key

X air temperature difference between head and feet

Y dissatisfied

Figure 2 — Local thermal discomfort caused by vertical air temperature difference

5.3 Acoustical comfort

5.3.1 Water velocity and noise

Closed-loop hydronic system piping is generally sized below certain upper limits, such as a velocity limit of 1,2 m/s for a 50 mm pipe and under, and a pressure drop limit of 400 Pa/m for piping over 50 mm in diameter. Pipes with velocities in excess of 1,2 m/s should have much larger size. Although this limitation is based on relatively inconclusive experience with noise in piping, it is generally accepted. Water velocity noise is not caused by water but by free air, sharp pressure drops, turbulence, or a combination of these, which in turn cause cavitation or flashing of water into steam. Therefore, higher velocities may be allowable if proper precautions are taken to eliminate air and turbulence.

When selecting a pipe size for a given flow rate, the resulting maximum flow velocity should be lower than 1,2 m/s based on minimizing noise generated by the flow.

5.3.2 Acoustical comfort in water based heating and cooling systems

In a building equipped with a HVAC system, the noise is usually caused by the aerodynamic air motion from the fan or pump, high airflow velocity along ducts, or hydrodynamic noise in pipes. The excessive airflow rates in all air systems can result in high air velocities and noise in ducts, terminal diffusers, etc. When the water based system cares for sensible cooling and/or heating load and ventilation system only for the air renewal and IAQ, then the sizes of ducts can be scaled down and the noise caused by high air velocity can be decreased.

In the water based heating and cooling systems, balancing valves are used for the temperature control in a multi-zone building. The excessive pressure drop across these balancing valves causes the velocity increase, turbulence, even cavitation and consequently noise. This noise vibrates the pipe and the vibration is delivered to the structure where the pipe is laid, leading to uncomfortable noise. Thus, it is necessary to design and operate the system not to make the excessive pressure drop in the major elements such as balancing valves that constitute the water based heating and cooling systems.

5.3.3 Acoustical comfort in Thermally Active Building Systems (TABS)

For TABS, the use of suspended panels (embedded surface heating and cooling systems) as an acoustic absorption surface will decrease the heat exchange with the occupants and other room surfaces. The decrease in heat exchange will depend on factors like type and position of acoustical panels, type and position of internal loads, ventilation concept, etc. Vertical acoustic plates affect the heat flow to a lesser extent. Acoustic plaster will act as an additional insulation. This effect shall be directly taken into account when calculating the capacity of TABS. The following factors should be considered in acoustical analyses:

— Floor covering;

— Equipment and furniture (position, surface absorptivity);

— Acoustic plates (absorptive surfaces);

— Position of heat-exchange surfaces in the room;

— Lighting system installation.

When the raised floor or the thermal/acoustic insulation in floor is installed, the upper heat flow from the TABS can be significantly decreased. If this structure is not used, then using floor covering materials with the ability of acoustic absorption can obtain good acoustic properties.

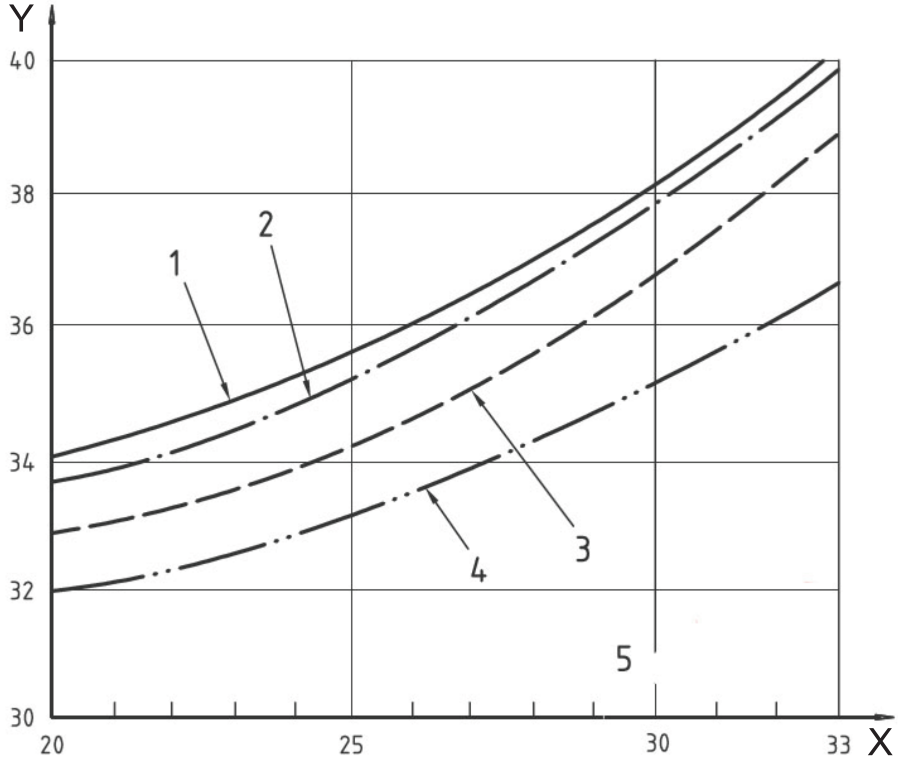
Annex A
(informative)

Floor surface temperature for thermal comfort

A.1 Acceptable range of floor surface temperature

References	Variable	Temperature range	Remarks
ISO 7730		Allowable range of floor surface temperature: 19~29°C	In America and Europe
ISO/TS 13732-2:2001	According to the material of the floor	Textile layer: 20≈28°C Wilton-carpet (velvet): 21≈28°C Sisal-carpet: 22,5≈28°C Pinewood floor: 22,5≈28°C Oakwood floor: 24,5≈28°C Wooden floor: 23≈28°C Vinyl-asbestos tile: 27,5≈29°C PVC-sheet (2 mm): 26,5≈28,5°C Hard linoleum (2,5 mm) on wood: 24≈28°C Hard linoleum (2,5 mm) on concrete: 26≈28,5°C Painted concrete floor: 27,5≈29°C Concrete floor: 26≈28,5°C Marble: 28≈29,5°C For all of these temperature ranges, sedentary people prefer temperatures 1≈2°C higher	In America and Europe, 15 % dissatisfied
ASHRAE Fundamentals handbook, 2009	According to the material of the floor	Textiles (rugs): 21≈28°C Pine floor: 22,5≈28°C Oak floor: 24,5≈28°C Hard linoleum: 24≈28°C Concrete: 26≈28,5°C	In America and Europe
Song, GS et al., 2001	According to the material of the floor	Clay: 24,9≈31,5°C Pine: 18,1≈29,0°C Urethane: 12,7≈23,2°C Veneer: 24,7≈31,0°C	In Korea
Ling Zhang et al.		Allowable range of floor surface temperature: 25≈31°C Comfortable range of floor surface temperature: 26≈30°C	In Japan

A.2 Relation between floor temperature and skin temperature in electric heating system



Key

X floor temperature, °C

Y skin temperature, °C

1 contact time, 90 min

2 contact time, 60 min

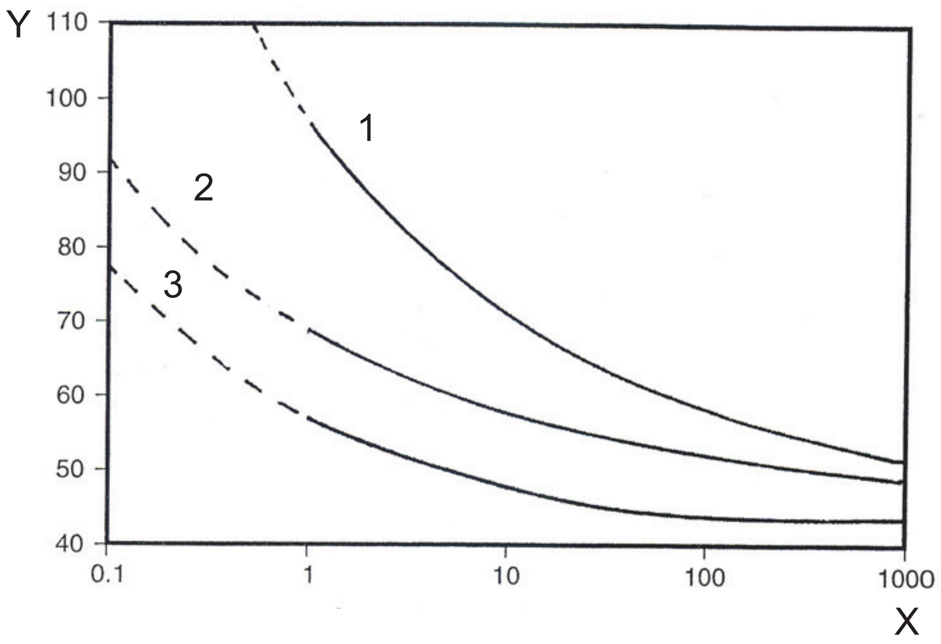
3 contact time, 30 min

4 contact time, 10 min

5 wood floor covering with electric floor heating panel

Figure A.1 — Relation between floor temperature and skin temperature when seated on an electrically heated floor

A.3 Relation between skin temperature and discomfort



Key

X contact time, s

Y skin temperature, °C

1 full skin thickness burn

2 partial skin thickness burn

3 discomfort

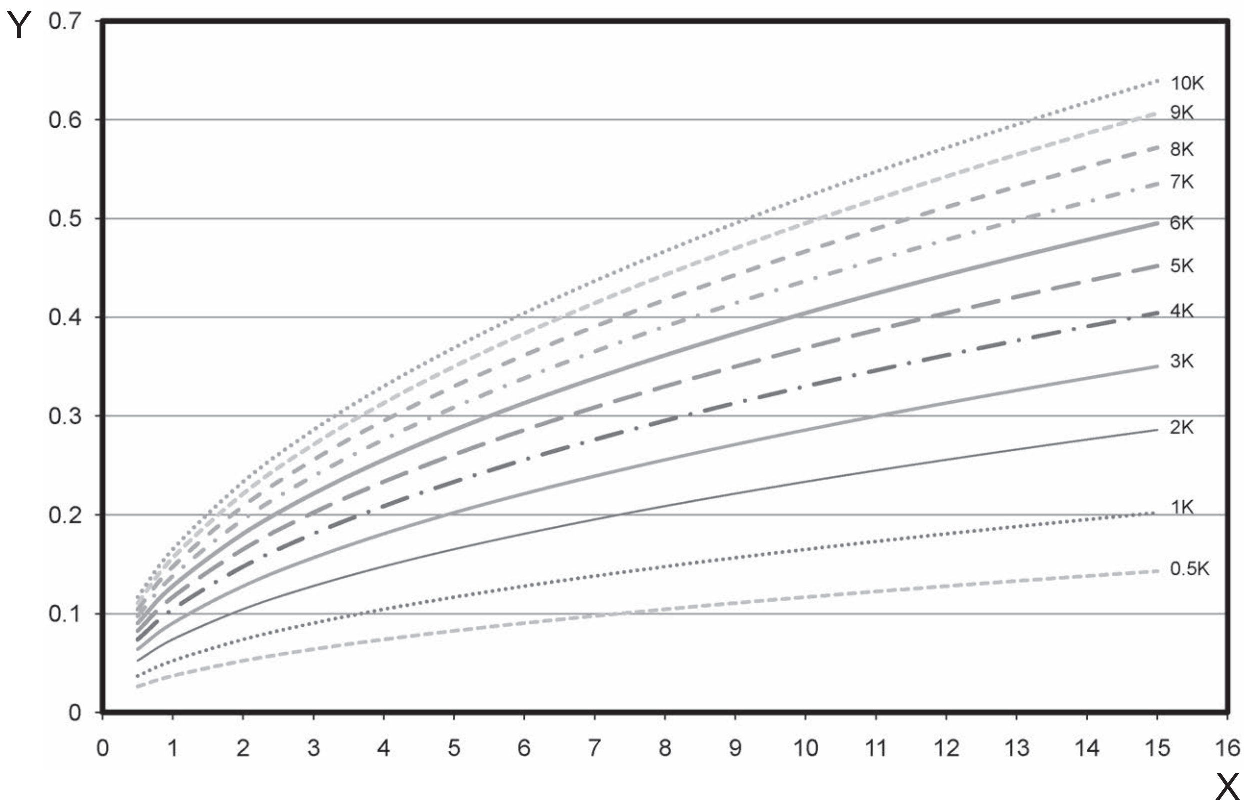
Figure A.2 — Skin temperatures that cause discomfort and burns

Annex B
(informative)

Draught

B.1 Equations for the calculation of maximum air velocity and minimum air temperature along the floor caused by cold draught from cooled surfaces

Parameter	Distance to the cold surface	Equations	Unit
Minimum air temperature, $\theta_{i,min}$, along the floor	x (m)	$\theta_{i,min} = \theta_i - (0{,}30 - 0{,}034 \cdot x)(\theta_i - \theta_{s,m})$	°C
Maximum air velocity, v_{max}, along the floor	$x < 0{,}4$ m	$v_{max} = 0{,}055 \cdot (\lvert\theta_i - \theta_{s,m}\rvert \cdot h)^{0,5}$	m/s
	$0{,}4$ m $< x < 2{,}0$ m	$v_{max} = \dfrac{0{,}095}{x + 1{,}32} \cdot (\lvert\theta_i - \theta_{s,m}\rvert \cdot h)^{0,5}$	m/s
	$2{,}0$ m $< x$	$v_{max} = 0{,}028 \cdot (\lvert\theta_i - \theta_{s,m}\rvert \cdot h)^{0,5}$	m/s



Key

X height of cooled wall, m

Y maximum air velocity along the floor 0,5 m from wall, m/s

Figure B.1 — Maximum air velocity along the floor 0,5 m from a cooled wall as a function of the temperature difference between room and cooled surface (EN 15377-1)

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