energy engineering

Heat Transfer Problems in Heating Systems

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This text was published as a part of the project 'Excellence programming – PK XXI 2.0 Cracow University of Technology development program for the years 2018–22'.
Funding from EU: 18,048,774.96 PLN

eISBN 978-83-66531-21-5

Online edition
4,5 publisher's sheets

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SYMBOLS AND UNITS

A	area, m^2
c_p	specific heat, J/(kgK)
d	diameter, m
f	correction factor, –
H	heat loss coefficient, W/K
L	length, m
$\dot{m}$	mass flow rate, kg/s, kg/h
P	perimeter; pipe spacing in radiation heating, m
q	heat flux density, W/m^2
R	heat transfer resistance, $(m^2K)/W$
s	thickness, m
t	temperature, °C
U	heat transfer coefficient, $W/(m^2K)$
$\dot{V}$	volume flow rate, m^3/h, m^3/s

Greek symbols

α	surface heat transfer coefficient, $W/(m^2K)$
λ	thermal conductivity coefficient, W/(mK)
θ	air temperature; heating surface temperature, °C
ρ	density, kg/m^3
σ	heating medium temperature drop in radiation heating, K
Δp	pressure loss, Pa
Φ	design heat loss; heat flux, W

Subscripts

e	external, exterior
ex	exhaust
equiv	equivalent
F	heating surface
g	ground, radiator
HL	heat loss
i	heated space
inf	infiltration
k	building element
l	linear thermal bridge
m	annual mean
M	mixing
P	return from installation
RH	reheat
su	supply
śr	mean
T	transmission
u	unheated space
V	ventilation
Z	installation feed

1. INTRODUCTION

The book *Heat Transfer Problems in Heating Systems* is dedicated to selected design and calculation issues related to heating installations. Particular attention is paid to the topicality of the referenced standards and ordinances. Given the continuous process of updating, modifying and amending these documents, it is important to provide timely information in the education process. This is why there are numerous references in the book to standards for heating installations, which have the status of current, or current discretionary standards.

In the first part (chapter two), issues related to the design heat load calculations of heated spaces are discussed, and an algorithm for such calculations is presented. Based on the results of these calculations, convection heaters are selected and/or radiation heating is designed. The rules for selecting convection heaters are specified in chapter three. Radiation heating, with particular consideration of floor heating, modern water heating, and air temperature control systems in rooms, and hydraulic control, are discussed in chapter four. A methodology for calculating the thermal efficiency of a radiator is also given.

The book is intended for college/university students of *Energy and Environmental Engineering*. Beyond this, it will be helpful for designers, builders, and operators of heating installations.

2. DESIGN HEAT LOAD CALCULATIONS FOR HEATED SPACES

Heating is the process of providing energy to maintain or raise the temperature of bodies or rooms. Heat for heating rooms is usually supplied to cover the heat losses via partitions, and to heat the ventilation air.

The need to heat rooms is imposed by climatic conditions, while the obligation to use heating systems or equipment results directly from Polish building law. Regulation [2], § 132. 1 provides that: A *building that requires heating due to its intended use should be fitted with a heating system or other heating devices, other than stoves, kitchen stoves or fireplaces.* This regulation provides for two types of heating system. The first is a water heating system. In accordance with § 133. 1. A *water heating system is a system of connected ducts together with fittings, circulation pumps, radiators, and other devices, located downstream of the valves isolating from a heat source, such as a boiler room, individual or group heat centre, solar collectors or a heat pump.* The other is an air heating system. In accordance with § 133. 2. *An air heating system is a system of connected air ducts and conduits together with diffusers and extract grilles, as well as airflow control elements, located between the heat source heating the air and the heated rooms. The air heating function may also be fulfilled by a mechanical ventilation system.*

The design heat load of heated spaces should be calculated in accordance with the standard [1]. The obligation to apply it results from the Regulation of the Minister of Infrastructure of April 12, 2002 [2], as currently amended.

Regulation [2], § 134. 1 provides that: *Building heating systems and devices should have the peak heat output determined in accordance with the Polish Standards for calculating the heat demand of rooms, as well as calculating the thermal resistance and heat transfer coefficient of building partitions.*

Annex No. 1 to the Regulation [2] lists the Polish standards referred to in this Regulation. One of them is PN-EN 12831: 2006 [1], which has been in force since 01.01.2009. This standard, published in Polish, is binding in its entirety.

The methodology for calculating the design heat load of a heated space, part of a building, or a building is valid for buildings with a limited height of rooms (up to 5 m) assuming that they are heated to a steady-state. The results of the calculations for each space of the heated part of the building or the building (total design heat load) are used to dimension the heat supply device, i.e. a heat exchanger or heat source. Also annexed to [1] are instructions for design heat loss calculation in a high room (large volume) and a building with a significant difference between the air temperature and the mean radiation temperature. These special cases, as well as the simplified calculation method of the design heat load, will not be discussed in this study.

The design heat load $\Phi_{HL,i}$ of a heated space (i) consists of the total design heat loss and surplus heat output to compensate for the heated zone's reheat [1]. This load allows the correct selection of radiators for heated spaces, and is calculated from the following formula:

$$\Phi_{HL,i} = \Phi_i + \Phi_{RH,i}. \tag{2.1}$$

The total design heat loss Φ_i is the sum of the design heat loss due to transfer and the ventilation heat loss:

$$\Phi_i = \Phi_{T,i} + \Phi_{V,i}. \tag{2.2}$$

In the above formulas:

$\Phi_{T,i}$ – design heat loss of heated space (i) due to penetration, W,

$\Phi_{V,i}$ – ventilation heat loss of heated space (i), W,

$\Phi_{RH,i}$ – surplus heat output for reheat of the heated zone, W (should be agreed with the customer). This surplus allows the required indoor temperature to be reaching within a certain time after a heating reduction period.

2.1. DESIGN HEAT LOSS DUE TO TRANSFER

The total design heat loss component that accounts for the heat loss due to transfer is determined by the formula:

$$\Phi_{T,i} = \left(H_{T,ie} + H_{T,iue} + H_{T,ig} + H_{T,ij}\right) \cdot \left(\theta_i - \theta_e\right). \tag{2.3}$$

In the above formula:

$H_{T,ie}$ – heat loss coefficient due to transfer from heated space (i) to the exterior (e) via building envelope, W/K,

$H_{T,iue}$ – heat loss coefficient due to transfer from heated space (i) to the exterior (e) via unheated space (u), W/K,

$H_{T,ig}$ – heat loss coefficient due to transfer from heated space (i) to ground (g) under steady conditions, W/K,

$H_{T,ij}$ – heat loss coefficient due to transfer from heated space (i) to adjacent space (j) heated to a significantly different temperature, i.e. adjacent heated space in the same part of the building or in an adjacent part of the building, W/K.

The design internal temperature of heated space θ_i in formula (2.3) can be adopted based on Tab. 2.1. The design external temperature θ_e and mean annual external temperature $\theta_{m,e}$ result from the division of Poland into climate zones (Fig. 2.1) and are specified for each zone in Tab. 2.2.

Fig. 2.1. The Polish territory division into climate zones [1, 3]

Tab. 2.1

Design internal temperature [1, 2]

The purpose or use of rooms	**Room examples**	θ_I °C
– not intended for the permanent stay of people – industrial – with standby heating (if allowed by process considerations)	– unattended warehouses, individual garages, parking halls (without repairs), battery rooms, machine rooms, and passenger elevator shafts	5

– where there is no heat gain, and the one-time stay of people in motion and outdoor clothing does not exceed 1 hour – where there are heat gains from process equipment, lighting, etc., over 25 W per 1m³ of room volume	– staircases in residential buildings – compressor halls, pumping stations, forges, hardening plants, heat treatment departments	8
– where there are no heat gains, intended for the permanent stay of people, in outdoor clothing or performing physical work with energy expenditure above 300 W – where there are heat gains from process equipment, lighting, etc., of 10 to 25 W per 1m³ of room volume	– permanently attended warehouses and storage rooms, entrance halls, waiting rooms at auditoriums without cloakrooms, churches – halls for physical work with energy expenditure over 300 W, moulding halls, cold store machine rooms, battery charging rooms, market halls, fish and butcher shops	12
– where there are no heat gains, intended for the stay of people: – in outdoor clothing, sitting or standing – without outdoor clothing, in motion or performing physical work with energy expenditure up to 300 W – where there are heat gains from process equipment, lighting, etc. of 10 W per 1 m³ of room volume or less	– performance halls without cloakrooms, public EC, outdoor clothing cloakrooms, production halls, gyms – individual kitchens with coal ranges	16
– intended for the permanent stay of people without outdoor clothing, not performing continuous physical work – boiler rooms and heat centres	– living rooms, hallways, individual kitchens equipped with gas or electric ranges, office rooms, meeting rooms, museums and art galleries with cloakrooms, auditoriums	20
– intended for undressing – not intended for the stay of people without clothes	– bathrooms, dressing rooms/cloakrooms, washrooms, shower rooms, swimming pool halls, – doctor's offices where patients undress, baby rooms and children's rooms in nurseries, operating rooms	24

Tab. 2.2

Design external temperature and mean annual external temperature [1]

Climate zone	θ_e, °C	$\theta_{m,e}$, °C
I	–16	7.7
II	–18	7.9
III	–20	7.6
IV	–22	6.9
V	–24	5.5

For calculation of the heat loss due to transfer, the external dimensions, i.e. measured on the outside of the building, should be adopted. The standard [1] does not, however, specify how internal partitions should be dimensioned. In the opinion of the guide [3] authors, the internal partition dimensions should be determined based on the dimensions in the bounding partition axes. When determining horizontal dimensions, half of the thickness of the bounding inner wall and the entire thickness of the bounding outer wall are accounted for. The wall height is measured between floor surfaces (Fig. 2.2).

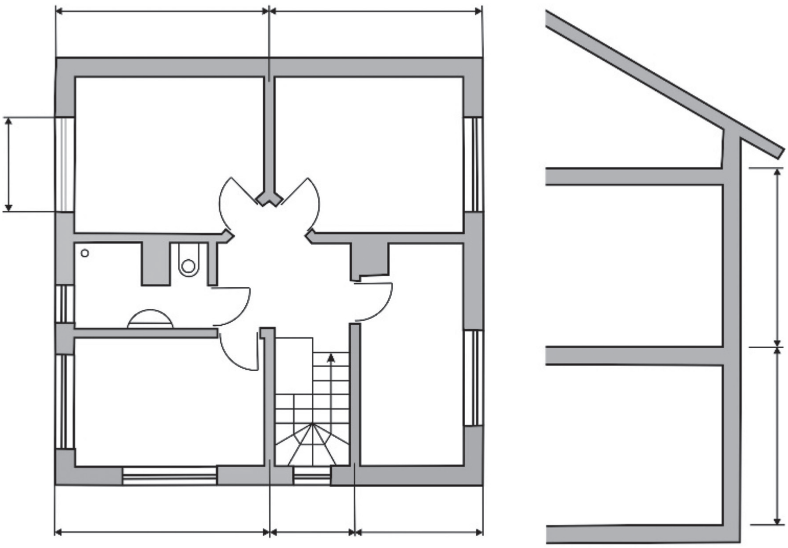


Fig. 2.2. Examples of horizontal and vertical dimensions [3]

2.1.1. HEAT LOSS DIRECTLY TO EXTERIOR

The heat losses directly to the exterior are determined by the formula:

$$H_{T,ie} = \sum_k A_k \cdot U_k \cdot e_k + \sum_l \Psi_l \cdot L_l \cdot e_l \,, \qquad (2.4)$$

where:

A_k – building element area (k), m^2,

U_k – coefficient of heat transfer via partition (k), W/(m^{2}K),

L_l – length of the linear thermal bridge (l) between internal and external spaces, m,

ψ_l – heat transfer coefficient of the linear thermal bridge (l), W/(mK),

e_k, e_l – correction factors due to the orientation, with consideration of the climate impact (various insulation, moisture absorption by building elements, wind speed, and air temperature), where these impacts have not already been taken into consideration in the valuation of coefficient U;

e_k and e_l – should be determined based on national data; in their absence, the approximate value of coefficients e_k and e_l equal 1.

Heat transfer coefficient U_k should be calculated according to standards [4, 5] or based on the recommendations in European technical approvals.

The heat transfer coefficient of linear thermal bridge ψ_l can be determined based on standard [6] in the case of detailed (numerical) calculations, or standard [7] for simplified calculations, using tabulated values. Non-linear thermal bridges are not included in the calculations. For linear heat losses due to transfer by permeation, a simplified method can be used. It consists in adopting the adjusted heat transfer coefficient:

$$U_{kc} = U_k + \Delta U_{tb}, \tag{2.5}$$

where:

U_{kc} – adjusted heat transfer coefficient of the building element (k) including linear thermal bridges, W/(m²K),

ΔU_{tb} – correction factor depending on the building element type, W/(m²K). Indicative values for openings and horizontal and vertical building elements are specified in Tab. 2.3–2.5.

Tab. 2.3

Adjustment factor ΔU_{tb} for openings [1]

Building element area	**ΔU_{tb}, W/(m²K)**
0–2 m²	0.50
> 2–4 m²	0.40
> 4–9 m²	0.30
> 9–20 m²	0.20
> 20 m²	0.10

Tab. 2.4

Adjustment factor ΔU_{tb} for horizontal building elements [1]

Building element			**ΔU_{tb}, W/(m²K)**
Light floor (wood, metal, etc.)			0
Heavy floor (concrete, etc.)	Number of sides in contact with the exterior	1	0.05
		2	0.10
		3	0.15
		4	0.20

Tab. 2.5

Adjustment factor ΔU_{tb} for vertical building elements [1]

Number of insulation intersecting floors	**Number of intersected walls**	**ΔU_{tb}, W/(m²K)**	
		space volume ≤ 100 m³	**space volume > 100 m³**
0	0	0.05	0
	1	0.10	0
	2	0.15	0.05
1	0	0.20	0.10
	1	0.25	0.15
	2	0.30	0.20
2	0	0.25	0.15
	1	0.30	0.20
	2	0.35	0.25

The wall intersecting principle is illustrated in Fig. 2.3.

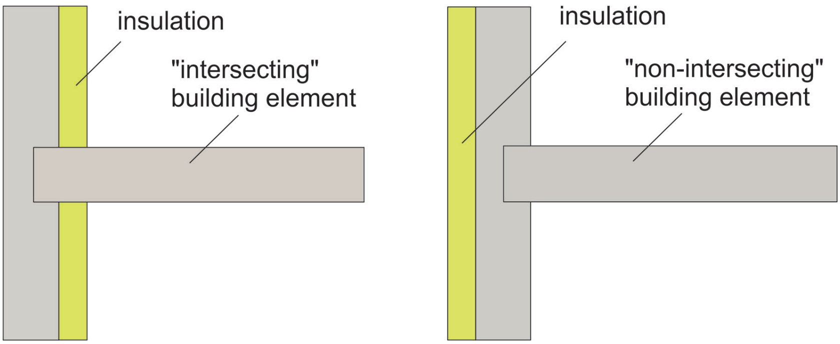


Fig. 2.3. Insulation "intersecting" and "non-intersecting" building elements [1]

2.1.2. HEAT LOSS TO EXTERIOR VIA UNHEATED SPACE

Heat losses to the exterior via unheated space are calculated as follows:

$$H_{T,iue} = \sum_k A_k \cdot U_k \cdot b_u + \sum_l \Psi_l \cdot L_l \cdot b_u \, . \qquad (2.6)$$

In the above formula, b_u is a temperature reduction factor accounting for the difference between the unheated space temperature and the design external temperature. This factor can be determined in a few ways:

a) if the unheated space temperature θ_u, under design conditions is determined or calculated:

$$b_u = \frac{\theta_i - \theta_u}{\theta_i - \theta_e}, \tag{2.7}$$

b) if θ_u is unknown:

$$b_u = \frac{H_{ue}}{H_{iu} + H_{ue}}, \tag{2.8}$$

where:

H_{iu} – coefficient of heat loss from heated space (*i*) to unheated space (*u*), including heat transfer losses and ventilation heat losses, W/K,

H_{ue} – coefficient of heat loss from unheated space (*u*) to the exterior (*e*), taking into account heat losses due to the transfer to exterior and ground, and ventilation heat losses between the unheated space and the exterior, W/K.

c) with reference to the national annex to standard [1] in which values of b_u are specified for most cases. If any value is missing in the annex, Tab. 2.6 can be used.

Tab. 2.6

Temperature reduction factor bu [1]

Unheated space	b_u
Room	
with 1 external wall only	0.4
with at least 2 external walls with no external door	0.5
with at least 2 external walls with external doors (e.g. halls, garages)	0.6
with three external walls (e.g. external staircase)	0.8
Basement (a room can be considered to be a basement if more than 70% of the external walls are in contact with the ground)	
with no external windows/doors	0.5
with external windows/doors	0.8
Attic	
heavily ventilated attic space (e.g. roof covering of tiles or other materials forming a discontinuous covering) without planking covered with felt paper or roofing felt or boards joined by edges	1.0
other uninsulated roofs	0.9
insulated roof	0.7
Internal circulation spaces	
(with no external walls, air exchange rate less than 0.5 h^{-1})	0

Freely ventilated circulation spaces (openings area/space volume > 0.005 m²/m³)	1.0
Underfloor space (floor above intransitive space)	0.8
Unheated passages or through gates closed on both sides	0.9

In computer calculations, it is recommended to determine the unheated space temperature θ_u through heat balance and substitute the resulting value into equation (2.7). In approximate calculations, the tabulated factor for b_u may be adopted.

2.1.3. HEAT LOSS TO GROUND

Another heat loss to consider is the heat loss to the ground. These losses can be calculated in detail according to standard [8], or in a simplified manner, presented in the standard [1]. In the simplified manner, the heat losses to the ground are calculated from formula (2.9):

$$H_{T,ig} = f_{g1} \cdot f_{g2} \cdot \left(\sum_k A_k \cdot U_{\text{equiv},k} \right) \cdot G_w . \tag{2.9}$$

In the above formula:

f_{g1} – correction factor accounting for the effect of annual external temperature fluctuations. It should be determined based on national data, if no data are available: $f_{g1} = 1.45$

f_{g2} – temperature reduction factor accounting for the difference between the mean annual and design external temperatures:

$$f_{g2} = \frac{\theta_i - \theta_{m,e}}{\theta_i - \theta_e}, \tag{2.10}$$

where:

$\theta_{m,e}$ – mean annual external temperature, as per Tab. 2.2,

G_w – correction factor accounting for the groundwater impact. If the distance between the assumed groundwater level and the basement floor level exceeds 1 m, this effect should not be accounted for (G_w = 1.00). Otherwise G_w = 1.15 (these are indicative values when no national values are available),

$U_{\text{equiv},k}$ – equivalent heat transfer coefficient of building element (k) determined according to the floor design (Fig. 2.4–2.7), W/(m²K).

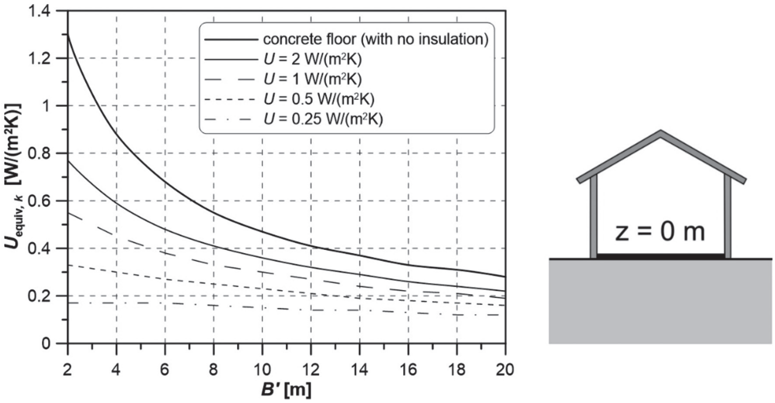


Fig. 2.4. Equivalent heat transfer coefficient of floor at ground level — based on [1]

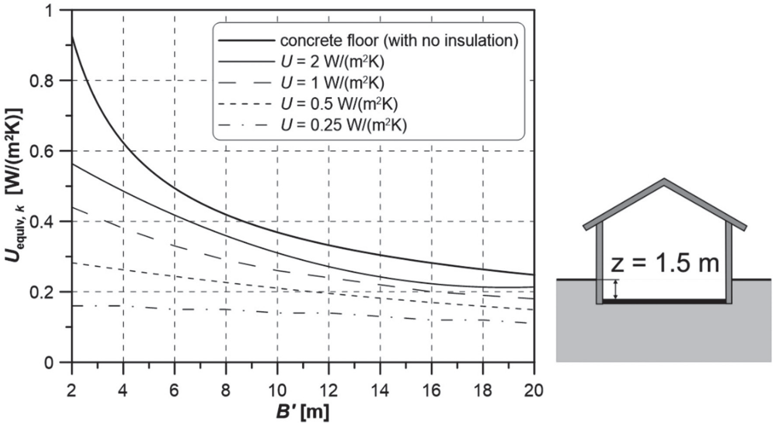


Fig. 2.5. Equivalent heat transfer coefficient of the heated underground floor with the floor slab located 1.5 m below the ground level — based on [1]

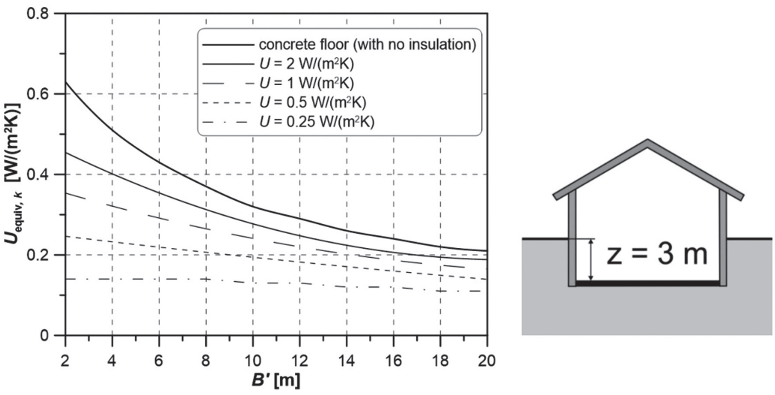


Fig. 2.6. Equivalent heat transfer coefficient of the heated underground floor with the floor slab located 3.0 m below the ground level — based on [1]

In Fig. 2.4–2.6, the $U_{\mathrm{equiv},k}$ values depend on heat transfer coefficient U of the building element and characteristic parameter B'. The values of $U_{\mathrm{equiv},k}$ are given with the assumption that the coefficient of heat transfer to ground λ_g = 2 W/(mK). The effect of side insulation is not considered.

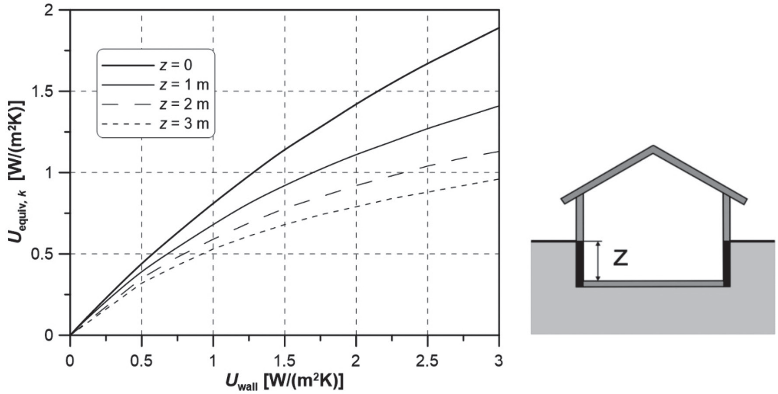


Fig. 2.7. Equivalent heat transfer coefficient of a heated basement wall — based on [1]

Characteristic dimension B' is determined from the relationship (Fig. 2.8):

$$B' = \frac{A_g}{0.5 \cdot P}, \tag{2.11}$$

where:

A_g – floor slab area, m². As regards an entire building, A_g is the total ground floor area. As regards part of a building, i.e. a single terraced building, A_g is the respective ground floor area,

P – floor slab perimeter, m. In terraced houses, perimeter P represents only the length of the external walls separating the heated space from the exterior.

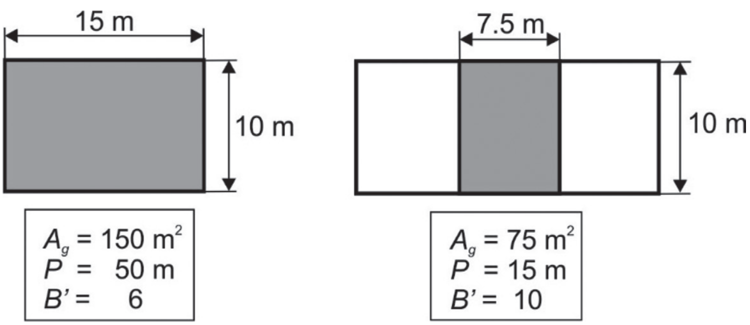


Fig. 2.8. Determination of characteristic dimension B' [1]

According to standard [1], dimension *B'* for individual rooms should be determined as follows:

a) for rooms without external walls, the *B'* value calculated for the entire building is adopted,
b) for all rooms with a well-insulated floor, $U_{floor} < 0.5$ W/(m^{2}K), the *B'* value for the entire building is also adopted,
c) for the other rooms, *B'* should be calculated separately for each room (room-by-room method).

2.1.4. HEAT LOSS BETWEEN SPACES HEATED TO VARIOUS TEMPERATURES

In this section, a heat flux transferred from heated space (*i*) into an adjacent space (*j*), heated to a significantly different temperature, is considered. These losses are included in the calculation of the heat load of individual rooms for the selection of radiators. However, they are not accounted for when determining the heat load of a part of a building or an entire building for the selection of a heat source.

The adjacent space can be an adjacent room in the same apartment (e.g. bathroom), a room in another part of the building (e.g. another apartment), or a room in an adjacent building, which may be unheated [3].

The heat loss coefficient is calculated as:

$$H_{T,ij} = \sum_k f_{ij} \cdot A_k \cdot U_k \, , \tag{2.12}$$

where:

f_{ij} – temperature reduction factor accounting for the difference between the adjacent space temperature (Tab. 2.7) and the design external temperature:

$$f_{ij} = \frac{\theta_i - \theta_{\text{adjacent space}}}{\theta_i - \theta_e} \, . \tag{2.13}$$

Tab. 2.7

Indicative temperatures of adjacent heated spaces [1]

Heat transferred from heated space (*i*) into	$\theta_{\text{adjacent space}}$, **°C**
adjacent room in the same building unit (e.g. the same apartment)	$\theta_{\text{adjacent space}}$ should be determined according to the room's intended use, e.g. for a bathroom: 24°C
adjacent room in another building unit (e.g. another apartment)	$\dfrac{\theta_i + \theta_{m,e}}{2}$
adjacent room in another building (heated or unheated)	$\theta_{m,e}$

2.2. DESIGN VENTILATION HEAT LOSS

The design ventilation heat loss of a heated space is calculated as follows [1]:

$$\Phi_{V,i} = H_{V,i} \cdot (\theta_i - \theta_e), \tag{2.14}$$

where the design ventilation heat loss coefficient:

$$H_{V,i} = \dot{V}_i \cdot \rho \cdot c_p . \tag{2.15}$$

In formula (2.15):

$\dot{V}_i$ – ventilation air volume flow rate in heated space, m³/s,

ρ – air density at temperature θ_i, kg/m³,

c_p – specific heat of air at temperature θ_i, J/(kgK).

The above relationship can also be written (assuming that ρ = 1.2 kg/m³, and c_p = 1005 J/(kgK)) as:

$$H_{V,i} = 0{,}34 \cdot \dot{V}_i , \tag{2.16}$$

while $\dot{V}_i$ is then denominated in [m³/h].

The calculation procedure for determining the appropriate airflow depends on the case under consideration, i.e. whether the room has a ventilation system or not.

2.2.1. ROOMS WITH NO VENTILATION SYSTEM

In this case, it is assumed that the supplied air is characterised by the thermal parameters of the external air. Heat losses are therefore proportional to the difference between the design internal temperature and the external air temperature.

The ventilation airflow rate should be the greater of the two values [1]:

$$\dot{V}_i = \max\left(\dot{V}_{\inf,i}, \dot{V}_{\min,i}\right), \tag{2.17}$$

where:

$\dot{V}_{\text{inf},i}$ – the volume flow rate of air infiltrating into the heated space, m³/s,

$\dot{V}_{\min,i}$ – minimum ventilation airflow rate required for hygiene reasons, m³/h.

The volume flow rate of air infiltrating into heated space (i), caused by the wind and chimney effect on the building, is calculated according to [1] as follows:

$$\dot{V}_{\text{inf},i} = 2 \cdot V_i \cdot n_{50} \cdot e_i \cdot \varepsilon_i . \tag{2.18}$$

In the above formula:

V_i – heated space volume (based on internal dimensions), m^3,

n_{50} – external air exchange rate due to a pressure difference of 50 Pa between the building interior and exterior, including the impact of the air diffuser, h^{-1} (Tab. 2.8),

e_i – cover factor (Tab. 2.9),

ε_i – correction factor accounting for the increase in wind speed depending on the heated space altitude above ground level (Tab. 2.10).

A factor of 2 was introduced because of the n_{50} value for the entire building. The calculation should take into account the worst case in which all infiltrating air enters the building on one side.

Tab. 2.8

Air exchange rate for entire building [1]

Structure	n_{50}, h^{-1}		
	Building envelope tightness (window gasket quality)		
	high (high-quality gaskets in windows and doors)	medium (double glazed windows, standard gaskets)	low (single glazed windows, with no gaskets)
single-family houses	< 4	4–10	< 10
other apartments or buildings	< 2	2–5	< 5

Tab. 2.9

Cover factor [1]

Cover classes	e		
	Number of exposed openings in heated space (windows and doors)		
	0	1	> 1
No cover (building in windy space, tall buildings in city centres)	0	0.03	0.05
Medium cover (buildings in the countryside with trees or other buildings nearby, suburbs)	0	0.02	0.03
Well covered (medium-tall buildings in city centres, buildings in forests)	0	0.01	0.02

Tab. 2.10

Correction factor for altitude [1]

Heated space height above ground level (room centre height above ground level)	ε
0–10 m	1.0
> 10–30 m	1.2
> 30 m	1.5

The minimum flow rate of ventilation air required for hygiene reasons flowing into the heated space (i) is determined as follows:

$$\dot{V}_{\min,i} = n_{\min} \cdot V_i \,, \tag{2.19}$$

where:

$n_{\min}$ – minimum external air exchange rate, h^{-1} (Tab. 2.11).

Tab. 2.11

Minimum external air exchange rate [1]

Room type	$n_{\min}$, h^{-1}
Living room (indicative)	0.5
Kitchen or bathroom with window	1.5
Office room	1.0
Conference room, classroom	2.0

2.2.2. ROOMS WITH VENTILATION SYSTEM

In this case, the supplied air is not always characterised by the thermal parameters of the outside air, e.g. when using heat recovery, central heating of external air, or air supply from adjacent spaces.

If there is a ventilation system, the volume flow rate of ventilation air in a heated space is determined by the formula:

$$\dot{V}_i = \dot{V}_{\inf,i} + \dot{V}_{su,i} \cdot f_{V,i} + \dot{V}_{mech,\inf,i} \,, \tag{2.20}$$

where:

su, – the volume flow rate of the air mechanically supplied to the heated space, m^3/h,

$\theta_{su,i}$ – the temperature of the air supplied to the heated space, °C (calculated based on the heat recovery efficiency, if heat is recovered),

$\dot{V}_{mech,\inf,i}$ – the surplus volume flow rate of the air exhausted from the heated space, m^3/h.

The volume flow rate $\dot{V}_{\mathrm{inf},\,i}$ of the air infiltrating into the heated space is calculated from formula (2.18).

Temperature reduction factor:

$$f_{V,i} = \frac{\theta_i - \theta_{\mathrm{su},\,i}}{\theta_i - \theta_e} \,. \tag{2.21}$$

The temperature of the air supplied to the heated space in the above formula can be calculated as (Fig. 2.9):

$$\theta_{\mathrm{su},\,i} = \theta_e + \eta_v \left(\theta_i - \theta_e\right) \tag{2.22}$$

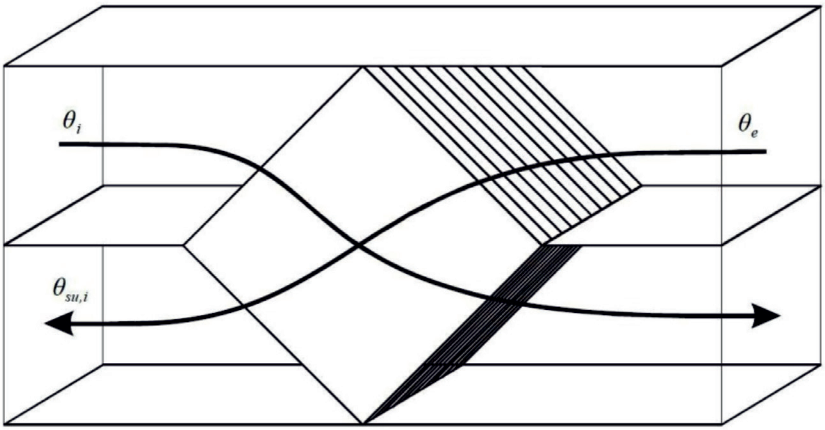


Fig. 2.9. Heat recovery by a plate heat exchanger [3]

Formula (2.22) is valid if heat is recovered (with known recovery efficiency η_v) and the flow rates of supply and exhaust air are equal.

If the flow rate of air exhausted from a room is higher than that of the supplied air, the resulting difference is compensated by the external airflow penetrating the building envelope.

For the entire building, the value of the ventilation airflow rate should be the greater of the two values:

$$\dot{V}_{\mathrm{mech,\ inf}} = \max\left(\dot{V}_{\mathrm{ex}} - \dot{V}_{\mathrm{su}}, 0\right), \tag{2.23}$$

where:

$\dot{V}_{\mathrm{ex}}$ – the volume flow rate of the air mechanically exhausted from the heated space, m^3/h,

$\dot{V}_{\mathrm{su}}$ – the volume flow rate of the air mechanically supplied — for the entire building, m^3/h (in residential buildings, often taken to be zero).

The external airflow distribution to the individual spaces in the building is determined by the formula:

$$\dot{V}_{\text{mech, inf},i} = \dot{V}_{\text{mech, inf}} \cdot \frac{V_i}{\sum V_i}. \tag{2.24}$$

If the ventilation system has not been identified, the ventilation heat losses are calculated as for a building with no ventilation system. When the ventilation system has been identified, the flow rate $\dot{V}_{\text{su},\,i}$ of air supplied to the heated space is calculated based on the system design.

2.3. HEATED SPACES WITH BREAKS OR REDUCTION

A space heated with breaks or reduction requires a thermal power surplus to reach the required design internal temperature after the reduction period [1]. The surplus thermal power should be agreed with the client (employer).

The surplus thermal power depends on the following factors:
- heat capacity of building elements,
- heating time,
- assumed temperature decrease in heating reduction periods,
- system control characteristics.

The thermal power surplus may not be needed, e.g. if:
- control system deactivates the reduction program when external temperatures are low,
- heat loss (ventilation) may be limited in heating reduction periods.

The PN-EN 12831:2006 standard provides a simplified method for calculating the surplus thermal power. This method may be applied to:
- residential buildings:
  - reduction periods (night reduction) 8 hours or less,
  - building structure is not light (such as a timber-framed building).
- non-residential buildings:
  - reduction periods 48 hours or less (weekend reduction),
  - usage time on working days longer than 8 hours a day,
  - design internal temperature from 20°C to 22°C.

The surplus thermal power required to compensate for the heating reduction effects is determined as follows:

$$\Phi_{RH,i} = A_i \cdot f_{RH}, \tag{2.25}$$

where:

A_i – heated space floor area, m²,

f_{RH} – heating factor – depends on the assumed temperature decrease in the heating reduction period and the heating time to reach the required internal temperature (Tab. 2.12 and 2.13).

Tab. 2.12

Heating factor in residential buildings, night reduction for max. 8 hours [1]

Heating time, hrs	**Heating factor f_{RH}, W/m²**		
	Assumed internal temperature decrease by heating reduction [a)]		
	1 K	**2 K**	**3 K**
	large building mass	**large building mass**	**large building mass**
1	11	22	45
2	6	11	22
3	4	9	16
4	2	7	13

[a)] In a well-insulated airtight building, it is unlikely that heating reduction will decrease the internal temperature by more than 2 K. It depends on the climatic conditions and the thermal mass of the building.

Tab. 2.13

Heating factor in non-residential buildings, night reduction for max. 12 hours [1]

Heating time, hrs	**Heating factor f_{RH}, W/m²**								
	Assumed internal temperature decrease by heating reduction [a)]								
	2 K			**3 K**			**4 K**		
	building mass			**building mass**			**building mass**		
	small	**medium**	**large**	**small**	**medium**	**large**	**small**	**medium**	**large**
1	18	23	25	27	30	27	36	27	31
2	9	16	22	18	20	23	22	24	25
3	6	13	18	11	16	18	18	18	18
4	4	11	16	6	13	16	11	16	16

[a)] In a well-insulated airtight building, it is unlikely that heating reduction will decrease the internal temperature by more than 2 K. It depends on the climatic conditions and the thermal mass of the building.

These f_{RH} factors refer to the internal floor area dimensions, and may be adopted for rooms with an average height of up to 3.5 m.

2.4. DESIGN HEAT LOAD

The design heat load may be calculated for a heated space, part of a building, and an entire building, to determine the heat load for the selection of radiators, heat exchanger, heat source, etc.

2.4.1. DESIGN HEAT LOAD OF HEATED SPACE

Substituting (2.2) into (2.1) produces the relationship to calculate the design heat load of a heated space (i):

$$\Phi_{HL,i} = \Phi_{T,i} + \Phi_{V,i} + \Phi_{RH,i} \tag{2.26}$$

Based on the results of the calculations, obtained with formula (2.26) for each heated space (i), radiators for these spaces are selected.

2.4.2. DESIGN HEAT LOAD OF PART OF BUILDING OR ENTIRE BUILDING

The calculations of the design heat load of part of a building or an entire building do not account for the heat exchanged by transfer and ventilation inside the heated part of the building.

The design heat load of part of a building or an entire building, Φ_{HL}, based on which the heating device capacity is selected, should be calculated as follows:

$$\Phi_{HL} = \sum \Phi_{T,i} + \sum \Phi_{V,i} + \sum \Phi_{RH,i} \,. \tag{2.27}$$

In this formula:

$\sum \Phi_{T,i}$ – the sum of heat losses by transfer of all heated spaces, excluding the heat exchanged inside the building, W,

$\sum \Phi_{V,i}$ – heat losses by ventilation of all heated spaces, excluding the heat exchanged inside the building, W,

$\sum \Phi_{RH,i}$ – the sum of surplus thermal powers in all heated spaces to compensate for the heating interruption (reduction) effects, W.

To calculate $\sum \Phi_{V,i}$, the entire airflow in the building is adopted. If the airflow for each heated space has been determined in the most unfavourable conditions assumed in each of them, the air flows of all spaces should not be summed, because the most unfavourable conditions occur simultaneously only in a portion of these spaces [1].

The airflow for a building is calculated as follows:

– with no ventilation system:

$$\sum \dot{V}_i = \sum \max\left(0.5 \cdot \dot{V}_{\inf,i}, \dot{V}_{\min,i}\right) \tag{2.28}$$

In heat calculations for an entire building, there is no need to double $\dot{V}_{\inf,i}$ (as per 2.18), because the unfavourable situation will not occur in rooms on both sides of the building at the same time. Therefore, in relation (2.28), 0.5 appears,

– with a ventilation system:

$$\sum \dot{V}_i = 0.5 \cdot \sum \dot{V}_{\inf,i} + \left(1 - \eta_V\right) \cdot \sum \dot{V}_{\mathrm{su},i} + \sum \dot{V}_{\mathrm{mech,\,inf},i} \ , \qquad (2.29)$$

where:

η_v – heat recovery from exhaust air efficiency, if no heat is recovered, $\eta_v = 0$.

To dimension a heat source, a 24-hour average is adopted. If the supplied air is heated by an adjacent system, this should be taken into account in the required heat load calculation.

Based on the design heat load calculation results, radiators are then selected, and hydraulic calculations performed to, inter alia, select the conduit diameters, the system's hydraulic balance, and circulation pumps [9].

3. RULES OF CONVECTION HEATER SELECTION

The basis for the selection of radiators for individual heated spaces is the design heat load of these spaces (formula 2.26). The required radiator surface area is then calculated from the following formula:

$$A_g \geq \frac{\Phi_g}{U \cdot \Delta t_g \cdot \varepsilon}, \tag{3.1}$$

where:

Φ_g – required radiator heat output, W,
U – coefficient of heat transfer via radiator wall, W/(m^{2}K),
Δt_g – mean temperature difference, °C,
ε – correction factor.

The required radiator heat output depends not only on the design heat load of the heated space $\Phi_{HL,i}$, but also on several other parameters, and is designated as:

$$\Phi_g = \Phi_{HL,i} \cdot \beta_T \cdot \beta_P \cdot \beta_U \cdot \beta_O \cdot \beta_S. \tag{3.2}$$

In the above formula:

β_T – factor that increases the required radiator output when using a thermostatic valve ($\beta_T = 1.15$ is recommended). This factor increases the area of the radiator surface or the mass flow rate of its feed water. It's meant to compensate for the effects of the system's thermal and hydraulic misalignment due to, for example, reduction at night or intensive room ventilation,

β_P – factor accounting for the way the radiator is connected to the system if it does not match that for which the thermal characteristics have been developed. For panel radiators $\beta_P = 1$. For tubular radiators, depending on the number of elements, $\beta_P = 1.05 – 1.33$ [10],

β_U – factor accounting the impact of the radiator location on heat exchange conditions. Computer programs for hydraulic calculations of heating systems offer several options to choose from (Tab. 3.1),

β_O – factor accounting for the impact of the radiator cover on the heat exchange conditions. Exemplary values of this coefficient, available e.g. from computer programs for hydraulic calculations of heating systems, are listed in Tab. 3.2.,

β_S – factor accounting for the effect of water cooling in uninsulated risers on the thermal efficiency of radiators in a two-tube heating system. Values of this factor are listed in Tab. 3.3 [10].

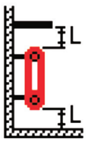
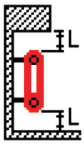
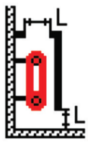
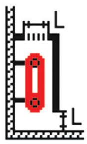
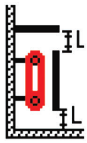
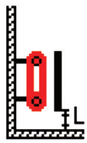
Tab. 3.1

Factors β_U for various radiator locations

Radiator location	**Factor β_U**
Radiator on an external wall just above the floor or near the external walls, balcony doors, and windows	1.00
Radiator on an internal wall just above the floor away from the external walls, balcony doors, and windows	1.10
Radiator mounted under the room ceiling	1.10

Tab. 3.2

Factors β_o for various radiator covers

Radiator cover						
L [mm]						
50			1.30	1.40	1.35	1.05
70	1.05	1.10	1.25	1.35	1.30	1.05
100	1.05	1.05	1.20	1.25	1.20	1.00
≥ 150	1.00	1.00	1.10	1.10	1.10	1.00

The heat transfer coefficient for a radiator is calculated as follows

$$U = C \cdot \Delta t_g^m \cdot \dot{m}^a , \qquad (3.3)$$

where:

C, m, a – fixed parameters for a radiator type and its connection to the duct network (determined experimentally at the manufacturer's test bench).

Mean difference between the heating medium and heated room temperatures for water-fed radiators:

$$\Delta t_g = \frac{t_Z + t_P}{2} - \theta_i , \qquad (3.4)$$

where:

t_Z – heating medium inlet temperature, °C,

t_p – heating medium return temperature, °C.

Tab. 3.3

Factor accounting for the effect of water cooling in uninsulated risers on the thermal efficiency of radiators

No. of floors	Building floor counted from the feed level											
	1	2	3	4	5	6	7	8	9	10	11	12
1	1.00											
2	1.00	1.05										
3	1.00	1.03	1.08									
4	1.00	1.02	1.04	1.09								
5	1.00	1.01	1.03	1.05	1.10							
6	1.00	1.01	1.02	1.04	1.06	1.11						
7	1.00	1.01	1.02	1.03	1.05	1.07	1.12					
8	1.00	1.01	1.02	1.03	1.04	1.05	1.08	1.13				
9	1.00	1.01	1.01	1.02	1.03	1.04	1.06	1.09	1.14			
10	1.00	1.01	1.01	1.02	1.03	1.04	1.05	1.07	1.09	1.14		
11	1.00	1.01	1.01	1.02	1.02	1.03	1.04	1.05	1.07	1.10	1.15	
12	1.00	1.00	1.01	1.02	1.02	1.03	1.04	1.05	1.06	1.08	1.10	1.15
13	1.00	1.00	1.01	1.01	1.02	1.03	1.03	1.04	1.05	1.06	1.08	1.11
14	1.00	1.00	1.01	1.01	1.02	1.02	1.03	1.04	1.04	1.05	1.07	1.08
15	1.00	1.00	1.01	1.01	1.02	1.02	1.03	1.03	1.04	1.05	1.06	1.07
16	1.00	1.00	1.01	1.01	1.02	1.02	1.03	1.03	1.04	1.04	1.05	1.06

Correction factor ε accounts for the variability of the radiator's heat transfer coefficient [10]:

$$\varepsilon = \frac{m \cdot (1 - X)}{\left(\frac{1}{X^m} - 1\right) \cdot \left(\frac{X+1}{2}\right)^{1+m}}, \tag{3.5}$$

where:

$$X = \frac{\Delta t_2}{\Delta t_1} = \frac{t_P - \theta_i}{t_Z - \theta_i}. \tag{3.6}$$

3.1. SELECTION OF RADIATORS AND OPERATING PARAMETERS DIFFERENT FROM CATALOGUE FOR TWO-TUBE HEATING SYSTEMS

Radiators are selected based on their manufacturers' detailed tables of heat outputs for various values of t_Z, t_P, and θ_i. Often, the quoted values are the so-called normative tabular radiator heat outputs determined for normative conditions, which usually are:

$t_Z/t_P/\theta_i = 75/65/20°C$ ($\Delta t_{g\,tab} = 50°C$). For radiators in systems with different parameters of $t_Z/t_P/\theta_i$, correction factors should be applied. Radiators are then selected based on values from the respective tables, based on the adjusted radiator heat output, which is:

$$\Phi'_g = \Phi_g \cdot f\,. \tag{3.7}$$

In this formula:

Φ'_g – adjusted (for normative conditions) radiator heat output, W,

f – correction factor (accounting for the different from normative radiator operating parameters) – it results from the equation of the radiator characteristics, determined by experimental tests.

This factor is most often calculated from the formula:

$$f = \left(\frac{\Delta t_{g\,tab}}{\Delta t_{g\,dsg}}\right)^{1+m}, \tag{3.8}$$

where:

$\Delta t_{g\,tab}$, $\Delta t_{g\,dsg}$ – calculated temperature difference between the heating medium and air in the heated space, for the normative values listed in the manufacturer's tables and for the selected (designed) radiator, respectively.

The simplest form of the radiator thermal characteristics equation is obtained by substituting equation (3.3) into equation (3.1) without correction factor ε:

$$\Phi_g = C \cdot A_g \cdot \Delta t_g^{1+m} \cdot \dot{m}^a\,. \tag{3.9}$$

For panel and panel-convector radiators, the characteristic exponents are: $0.25 < m < 0.35$ (typically $m = 0.3$) and $a = 0.0$ [10].

Examples of the correction factors for "Convector" brand radiators are listed in Tab. 3.4. In this case, listed in the table are the f values calculated as:

$$f = \left(\frac{\Delta t_{g\,dsg}}{\Delta t_{g\,tab}}\right)^{1+m}. \tag{3.10}$$

Tab. 3.4

Selected thermal output correction factors for side-fed and bottom-fed water radiators, and temperatures different from 75/65/20°C [11]

Feed temp.	Return temp.	Ambient temp. θ_i						
		4	**8**	**12**	**16**	**20**	**24**	**30**
90	**85**	1.95	1.83	1.71	1.59	1.48	1.37	1.20
	80	1.87	1.75	1.64	1.52	1.41	1.30	1.13
	75	1.79	1.68	1.56	1.44	1.33	1.22	1.06
	70	1.71	1.60	1.48	1.37	1.26	1.15	0.99
85	**80**	1.80	1.68	1.57	1.45	1.34	1.23	1.07
	75	1.72	1.61	1.49	1.38	1.27	1.16	1.00
	70	1.65	1.53	1.42	1.30	1.19	1.09	0.93
	65	1.57	1.45	1.34	1.23	1.12	1.01	0.86
80	**75**	1.65	1.54	1.42	1.31	1.20	1.10	0.94
	70	1.58	1.46	1.35	1.24	1.13	1.03	0.87
	65	1.50	1.39	1.28	1.17	1.06	0.96	0.80
	60	1.42	1.31	1.20	1.09	0.99	0.88	0.73
75	**70**	1.51	1.40	1.28	1.18	1.07	0.96	0.81
	65	1.43	1.32	1.21	1.11	1.00	0.90	0.75
	60	1.36	1.25	1.14	1.03	0.93	0.83	0.68
	55	1.28	1.17	1.07	0.96	0.86	0.76	0.61

Radiator manufacturers, in their proprietary materials, set out the radiator thermal characteristics determined at test benches. An example equation of this characteristic for a "Convector" brand GP 2/10 radiator (Fig. 3.1) is described by equation (3.11) [11]:

$$\Phi_g = 1.146 \cdot \Delta t_g^{1.3305} \cdot \dot{m}^{0.2614}. \tag{3.11}$$

Fig. 3.1. "Convector GP" side-fed water radiator [11]

Selected calculation results for this radiator (for a single feed temperature) are listed in Tab. 3.5.

Tab. 3.5

Selected calculation results for the GP 2/10 radiator (for temperatures different from 75/65/20°C) [11]

Feed temp.	**Return temp.**	**Ambient temp. θ_i**													
		4		**8**		**12**		**16**		**20**		**24**		**30**	
75	70	152		140		129		118		107		96		81	
		389	885	331	817	279	750	233	685	191	621	155	559	109	469
	65	72		66		61		55		50		45		37	
		88	841	74	773	62	707	51	643	42	580	33	519	23	430
	60	46		42		38		34		31		27		22	
		35	796	29	729	24	664	20	600	16	538	13	478	8	390
	55	32		29		27		24		21		19		15	
		17	750	15	684	12	620	10	557	8	495	6	435	4	349
	50	24		22		20		18		15		13		11	
		10	703	8	638	7	574	5	512	4	451	3	392	2	306

Each cell in Tab. 3.5 contains details of the mass flow rate of water flowing through the radiator, its heat output, and pressure loss. An example reading for a feed temperature of 75°C, with a return temperature of 65°C and an ambient temperature θ_i of 20°C is shown in Fig. 3.2.

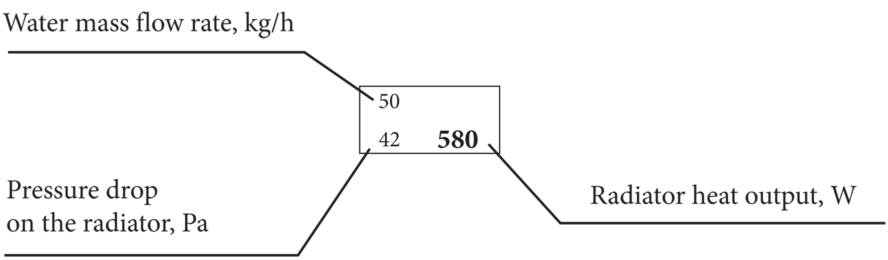


Fig. 3.2. Data from cells of Tab. 3.5

The pressure drop on the radiator (its hydraulic characteristics) is described as follows [11]:

$$\Delta p_g = 0.0168 \cdot \dot{m}^2 . \qquad (3.12)$$

The calculated pressure drop on the radiator (as a function of the water mass flow rate) should be accounted for in the pressure loss calculation in a heating circuit. This will allow for the proper hydraulic balance of the entire system [9].

3.2. SELECTION OF RADIATORS FOR SINGLE-TUBE HEATING SYSTEMS

When selecting radiators for single-tube heating, the tables for two-tube heating can be used. The adjusted (for two-tube heating) radiator heat output is determined as follows:

$$\Phi'_g = \Phi_g \cdot \varphi \,. \tag{3.13}$$

Correction factors φ can be read in the function of the so-called relative cooling of the circulation water (X_n). Examples of φ are listed in Tab. 3.6.

Tab. 3.6

Correction factors φ for radiator selection in single-tube heating systems [10]

X_n	0.00	0.05	0.10	0.15	0.20	0.25	0.30	0.35	0.40	0.45	0.50	0.55	0.60
90/70	0.82	0.83	0.85	0.87	0.88	0.90	0.92	0.94	0.96	0.98	1.00	1.02	1.04
70/55	0.81	0.83	0.84	0.86	0.88	0.90	0.92	0.94	0.96	0.98	1.00	1.02	1.05
X_n	0.65	0.70	0.75	0.80	0.85	0.90	0.95	1.00	1.05	1.10	1.15	1.20	
90/70	1.07	1.09	1.12	1.15	1.18	1.20	1.24	1.27	1.30	1.34	1.37	1.41	
70/55	1.07	1.10	1.13	1.16	1.19	1.22	1.25	1.29	1.32	1.36	1.40	1.45	

Relative cooling of the water flowing through the n-th radiator is determined as follows [10]:

$$X_n = \frac{\sum_{i=1}^{i=n-1} \Phi_i}{\Phi_c} + \frac{\Phi_n}{2 \cdot \beta \cdot \Phi_c} \,, \tag{3.14}$$

where:

$\sum_{i=1}^{i=n-1} \Phi_i$ – the sum of heat outputs of the radiators upstream of the radiator in question, W,

Φ_n – required heat output of the n-th radiator, W,

Φ_c – total radiator heat output in the single-tube heating circuit, W,

β – flow ratio – i.e. ratio of the mass flow of water flowing through the radiator to the mass flow of water flowing at the floor. This ratio is typically $\beta = 0.33–0.5$.

4. RADIANT HEATING

There are many ways to heat rooms and to supply the heat. One of these is radiant heating. In radiant heating, heat is supplied to heated rooms through a warm surface of partitions in the form of the floors, walls, and/or ceilings. By using building partitions, surface heating is characterised by a relatively large heat exchange surface area compared to traditional heating systems. The large heat exchange surface area means a larger radiation share in the heat transfer compared to convection in traditional radiator heating. In radiant heating, the entire partition surface area is involved in heat emission, which means that the heat is evenly distributed, and the temperature stratification is reduced. In in-floor radiant heating, there is minimal natural convection, no drafts, and the resulting temperature profile is similar to the theoretical ideal profile (Fig. 4.1).

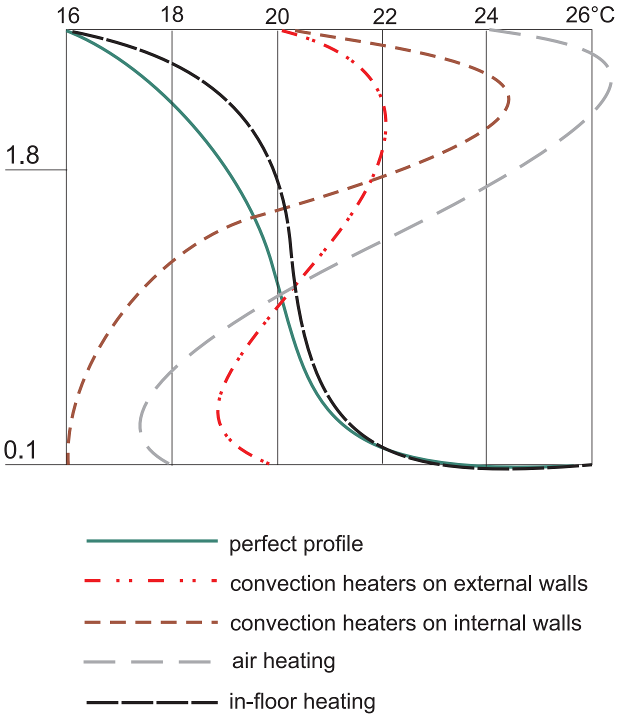


Fig. 4.1. Temperature profile for various heating types [12]

The correct temperature of the surrounding partitions and the air temperature in the heated room are also important for the thermal comfort. According to the König chart (Fig. 4.2), the area of air temperature and surrounding partitions where comfort can be experienced is relatively small. It can be enlarged with radiant heating.

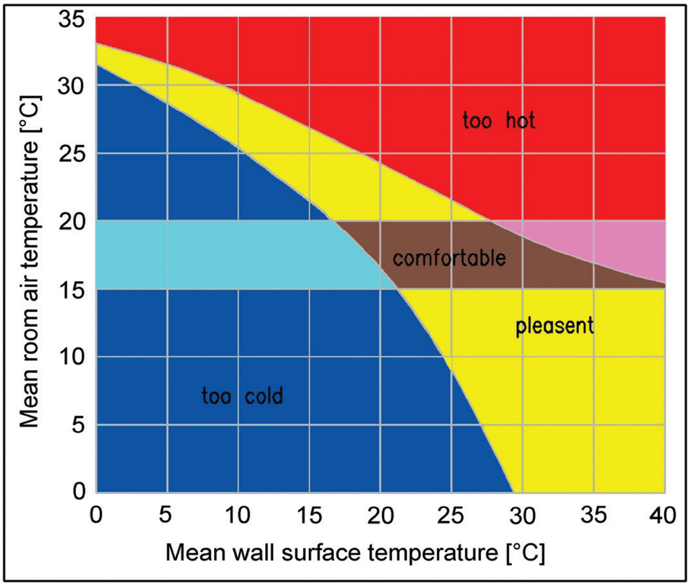


Fig. 4.2. The König chart [12]

Currently, water or electric radiant heating is most often used. In ancient Greece from ca. the 4th century BC, and in ancient Rome from ca. the 1st century BC, so-called hypocausts [13] were used for heating rooms. This was a system that now would be classified as in-floor or wall/floor air heating, where warm air was heated by a furnace in the lower part of the building and circulated through ducts in the floor or walls, heating the rooms. A similar air system, closer to our time, can be observed at the Teutonic castle in Malbork, where warm air circulated through under-floor ducts, heating the rooms.

Radiant heating systems, such as in-floor, wall, and ceiling heating, are becoming commonplace due to their comfort of use and high energy efficiency. The comfort of use is due to the absence of radiators, as the heating surfaces are the existing floors, walls, and/or ceilings. Ongoing maintenance, due to the utility of building partitions, is sufficient to ensure the high energy efficiency of a large surface area heater.

Also important is that with a large heating surface area, its temperature, and therefore also the heating medium temperature, can by naturally lowered. A lower heating medium temperature allows alternative heat sources to be used, such as solar collectors, in the transition period, or the energy efficiency of heat pumps and condensing boilers to be increased.

A large heating surface with a constant temperature generates a homogeneous and intense heat radiation field, and therefore to ensure thermal comfort, the mean room air temperature can be lowered. Each air temperature reduction by 1°C adds up to 5% of heat consumption savings.

Radiant heating advantages:

- comfort and convenience of use,
- large proportion of heat transferred by radiation,
- vertical temperature profile close to ideal,
- low air circulation,
- no spread of dust and mites,
- no wall dirt,
- no special maintenance of heating surfaces needed,
- less heat loss in high rooms,
- good sound insulation of rooms,
- wide selection of heat sources.

Radiant heating disadvantages and limitations:

- risk of damage in construction works,
- need to interfere with partitions in existing facilities,
- heating surface shielding limitations (additional insulation),
- limited unit heat output (W/m^2),
- high inertia,
- no expansion possible without construction works.

4.1. RADIANT HEATER HEAT OUTPUT

The heat output of a radiant heater depends on many factors, such as the heated room temperature, heat exchange surface area, finish type, heater design, diameter, type and spacing of heater tubes, heating medium temperature, and flow rate. The radiant heater heat output can be controlled by adjusting all or any of these factors. In practice, some of these factors are hardly, if at all, adjustable. The temperature in a heated room depends on its intended use, and Technical Conditions [2] strictly determine the value of it, in accordance with §134.2. Of course, at the investor's request, these temperatures may be slightly higher. The heat exchange surface of a radiant heaters is limited by the surface area of the partitions in the room that we want or can use for heating purposes. For in-floor heating, it is the floor surface not covered by furniture or appliances. The finish type is related to the room function and is imposed by the architect and/or investor. The type and diameter of the radiant heating tubes do not significantly impact its heat output. The diameter has a significant effect on the hydraulic resistance to the heating medium flow. However, once the desired flow rate has been accomplished, the radiant heating tube diameter is of secondary importance. This is due to the fact that the basic thermal resistance to heat transfer occurs between the heated room air and the heater's heating surface. The radiant heater output largely depends on the finish type, tube spacing and temperature, and heating medium flow rate. In installation practice, the finish type is imposed, and the medium flow rate is selected so that the medium is cooled by 8–12°C. Therefore, radiant heater output is determined by the tube spacing and the heating medium temperature. In practice, a heating medium with the same temperature feeds several heaters, and their outputs are controlled by tube spacing adjustment. Various radiant heating parameter calculation methodologies are reported in the literature. The most well-known are the trapezoidal method developed by prof. Wasilewski [14], and the methodology proposed by Polish Standard [16].

In-floor heating performance depends to a large extent on meeting several basic criteria from this standard. It provides, inter alia, a classification of in-floor heating systems and information on general requirements. This standard is divided into 5 parts [15–19].

The following calculation algorithm, and description of the criteria and individual steps, applies to water floor heating in apartments, offices, etc. It applies to systems in which other heating media are used (instead of water). The standard provides for four floor heater design solutions, i.e. types A, B, C, and the so-called type D flat systems. The calculating methodology for in-floor heating types A and C is described below (Fig. 4.3).

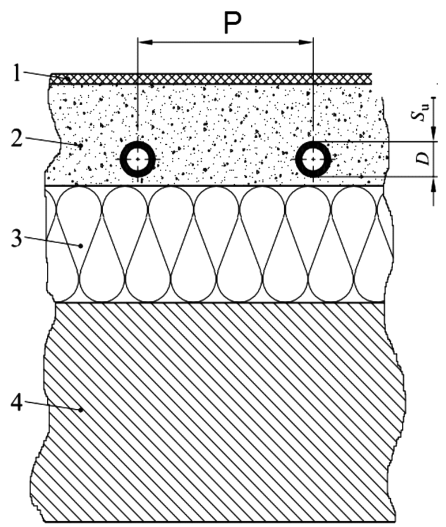


Fig. 4.3. System with in-screed ducting (types A and C) [16];
1 – floor covering ($R_{\lambda,B}$), 2 – screed (λ_E), 3 – insulation layer, 4 – structural base

The design of a type B in-floor heater is shown in Fig. 4.4. It is characterised by ducts laid below the screed, in the insulation layer.

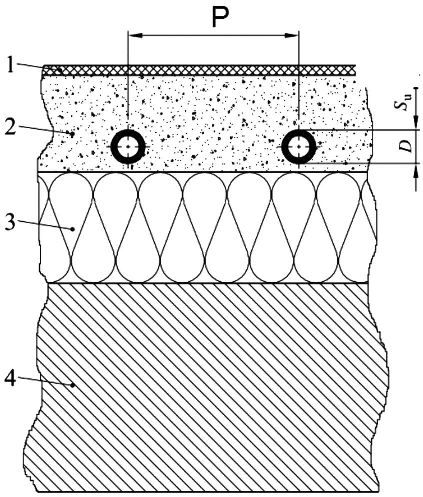


Fig. 4.4. System with under-screed ducting (type B) [16];
1 – floor covering ($R_{\lambda,B}$), 2 – screed (λ_E), 3 – heat conducting element, e.g. steel sheet, 4 – insulation layer, 5 – structural base

In standard [16], it is assumed that, to determine the in-floor heating output, each heating surface with a given mean surface temperature provides the same unit heat flow to each heated room with the same internal temperature θ_i. With this assumption, the surface heat transfer characteristics can be determined, regardless of the heating system. This relation can be applied to all in-floor heating surfaces, including the edge zones with an increased heat output. This relation is illustrated in Fig. 4.5.

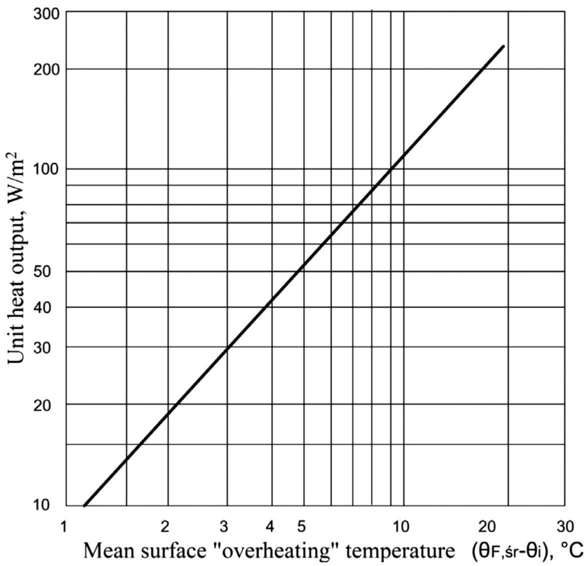


Fig. 4.5. Surface heat transfer characteristics [16]

The density of the heat flux transferred through a heating surface is described by the formula:

$$q = 8.92(\theta_{F,śr} - \theta_i)^{1.1} \tag{4.1}$$

where:
$\theta_{F,śr}$ – mean floor surface temperature, °C,
θ_i – nominal room temperature, °C.

For each in-floor heating system, there is a maximum heat flux density limited by q_G. It is determined by room temperature θ_i = 20°C and maximum heating surface temperature $\theta_{F,\max}$ = 29°C for σ = 0 K, where σ is the difference between the heating medium inlet and return temperatures. For the edge zone, the maximum heat flux density can be assumed for heating surface temperature $\theta_{F,\max}$ = 35°C at σ = 0 K. For heating surface area calculation, regardless of the heating type, the heating surface maximum temperature should be adopted as the reference point $\theta_{F,\max}$, e.g. some finishing materials may limit it. The mean heating surface temperature $\theta_{F,śr}$, which determines the heat flux density according to Fig. 4.5, must be less than the maximum. Floor surface temperature $\theta_{F,\max}$ depends on the heating operating conditions, such as the heating medium flow temperature, return temperature, finish thermal resistance, and unit heat flow downwards.

The basic assumptions used to determine the heat flux density are as follows:
– heat transfer from floor surface to room is consistent with the heat transfer characteristics,
– heating medium temperature drop σ = 0; in the area where a heating medium temperature decrease influences the heat transfer characteristics (Fig. 4.5), this decrease was accounted for by considering heating medium temperature logarithmic difference θ_H,
– turbulent flow, i.e. $\dot{m}_H / d_w > 4000\ \mathrm{kg/(s \cdot m)}$,
– no lateral heat transfer,
– in-floor heat conducting layer is separated from the building structure by an insulating layer.

Heat flux density q on the floor surface is dependent on:
– distance P between floor heater tubes,
– thickness s_u and thermal conductivity λ_E of the layer above the tubes,
– heat conduction resistance $R_{\lambda,B}$ of the finish layer,
– outer tube diameter d_z (for non-insulated tube), or d_M (for insulated tube) where d_M is outer tube diameter with insulation, and on the thermal conductivity of the tube and/or lag material λ_R and, λ_M, respectively. For non-circular ducts, an equivalent diameter should be adopted. The thickness and heat conduction resistance of adjacent layers up to 0.3 mm thick should be neglected.

Heat flux density is proportional to $(\Delta\theta_H)^n$, where the heating medium temperature difference is:

$$\Delta\theta_H = \frac{t_Z - t_P}{\ln \dfrac{t_Z - \theta_i}{t_P - \theta_i}} \tag{4.2}$$

Exponent n, by experimental tests and theoretical considerations, is valued at:

$$1.0 < n < 1.05 \tag{4.3}$$

For further considerations $n = 1$ is adopted. With these assumptions, the heat flux density is formulated as:

$$q = B\Pi_i\left(a_i^{m_i}\right) \cdot \Delta\theta_H \cdot \tag{4.4}$$

In the above formula:

B – factor dependant on the floor heating system, W/(m²·K),

$\Pi_i\left(a_i^{m_i}\right)$ – relationship that combines the floor element parameters such as the cover, distance between tubes, shielding, and outer conduit diameter factors.

Systems should be differentiated between those with tubes in the screed, under the screed, and with flat elements. Equation (4.4) applies directly to typical designs. For systems with additional elements for heat distribution, hollow profiles filled with air, and other elements affecting heat distribution, equation (4.4) should be extended with additional components.

For systems with in-screed ducts (type A and type C), the characteristic curves are calculated from the equation:

$$q = B \cdot a_B \cdot a_P^{m_P} \cdot a_U^{m_U} \cdot a_{dz}^{m_{dz}} \cdot \Delta\theta_H \tag{4.5}$$

where:

$B = 6.7$ W/(m²·K) – factor dependent on the floor heating system with tube material thermal conductivity $\lambda_R = \lambda_{R,0} = 0.35$ W/(m·K), and tube wall thickness $s_R = s_{R,0} =$ $= (d_z - d_w)/2 = 0.002$ m,

a_B – floor shielding factor calculated from the equation:

$$a_B = \frac{\dfrac{1}{a} + \dfrac{s_{u,0}}{\lambda_{u,0}}}{\dfrac{1}{a} + \dfrac{s_{u,0}}{\lambda_E} + R_{\lambda,B}} \tag{4.6}$$

where:

a_P – tube spacing factor, $a_P = f(R_{\lambda,B})$, as per Tab. A.1 [16],
a_U – shielding factor, $a_U = f(P, R_{\lambda,B})$, as per Tab. A.2 [16],
a_{dz} – outer conduit diameter factor, $a_{dz} = f(P, R_{\lambda,B})$, as per Tab. A.3 [16].

$$m_P = 1 - P/0.075 \text{ for } 0.05 \text{ m} \le P \le 0.375 \text{ m} \tag{4.7}$$

$$m_U = 100 \cdot (0.045 - s_u) \text{ for } s_u s_u \ge 0.010 \text{ m} \tag{4.8}$$

$$m_{dz} = 250 \cdot (d_z - 0.020) \text{ for } 0.008 \text{ m} \le d_z \le 0.030 \text{ m} \tag{4.9}$$

In the above formulas:

P – tube spacing, m,
d_z – outer tube diameter, m,
s_u – over-tube layer thickness, m,
$\alpha = 10.8$ W/(m²·K) – heat transfer from floor surface to air factor, as per [16],
$\lambda_{u,0} = 1$ W/(m·K) – screed heat transfer, as per [16],
$s_{u,0} = 0.045$ m – partition thickness, as per [16].

Standard [16] refers to the notion of a heat flux density boundary curve (Fig. 4.6). This determines the relationship between the heat flux density and temperature difference between the heating medium and the room air, at the maximum allowable heating surface "overheating" (9K or 15K). The boundary curves are calculated from the formula:

$$q_G = \varphi \cdot B_G \cdot \left[\frac{\Delta\theta_H}{\varphi}\right]^{n_G} \tag{4.10}$$

where:

B_G – factor from Tab. A.4a (if $s_u/\lambda_E \le 0.0792$)), or Tab. A.4b (if $s_u/\lambda_E > 0.0792$) for A and C type systems — table numbering as per [16],
n_G – exponent from Tab. A.5a (if $s_u/\lambda_E \le 0.0792$), or Tab. A.5b (if $s_u/\lambda_E > 0.0792$) for A and C type systems — table numbering as per [16],
φ – factor dependent on $\theta_{F,\max}\theta_{F,\max}$ and θ_i:

$$\varphi = \left[\frac{\theta_{F,\max} - \theta_i}{\Delta\theta_0}\right]^{1.1}; \ \Delta\theta_0 = 9K \tag{4.11}$$

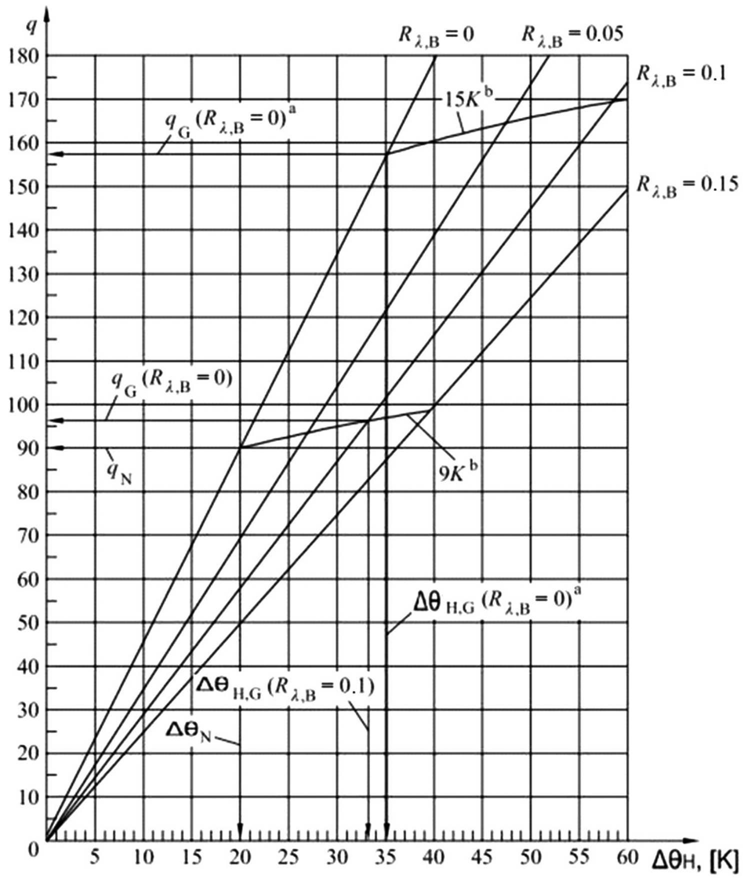


Fig. 4.6. In-floor heating performance characteristics, with marked boundary curves [16]; superscripts a and b indicate the edge zone and the boundary curve, respectively

To determine the maximum heat flux density q_G for temperature difference $\Delta\theta_H$ between the heating medium and the heated room, for a finish with thermal resistance $R_{\lambda,B}$ subject to the maximum allowable heating surface "overheating", the intersection point of the heat transfer characteristics with the limit curve is identified according to the formula:

$$\Delta\theta_{H,G} = \varphi \cdot \left[\frac{B_G}{B \cdot \Pi_{a_i}^{m_i}} \right]^{\frac{1}{1-n_g}} \tag{4.12}$$

Temperature difference $\Delta\theta_H$ (heating medium/room) is calculated from equation (4.2). This is how the heating medium temperature drop effect is accounted for. The in-floor heating performance characteristics describe the relationship between heat flux density q of the system and the required logarithmic difference of the heating medium temperature. For simplicity, it is assumed that the heat flux density is directly proportional to the heating medium temperature difference:

$$q = K_H \cdot \Delta\theta_H \tag{4.13}$$

where:

$$K_H = B \cdot \Pi\left(a_i^{m_i}\right). \tag{4.14}$$

In-floor heating characteristics with specified conduit spacing P should contain at least the performance characteristics for floor shielding thermal resistance $R_{\lambda,B} = 0$; $R_{\lambda,B} = 0.05$; $R_{\lambda,B} = 0.1$; and $R_{\lambda,B} = 0.15$ (m^{2}K)/W, as per Fig. 6.6.

The boundary curves in the system characteristics scope describe the relationship between the heating medium temperature difference and the unit heat flux at physiologically acceptable floor surface temperature $\theta_{F,\max}$ (29°C for comfort zones and 35°C for edge zones). The boundary curve for ($\theta_{F,\max}$ θ_i) = 9 K may also be applied to bathrooms, where θ_i = 24°C. When designing, they are used to determine the heat flux density, and are associated with the temperature difference, where:

$$0K < \sigma \leq 5K \tag{4.15}$$

The boundary curves are also used to determine the maximum feed temperature (also with regard to Fig. 6.8).

In order to limit the heat flux towards the rooms below, thermal resistance of the insulating layer is required $R_{\lambda,ins}$ (as per Tab. 1 available in [18]). This resistance is calculated from the equation:

$$R_{\lambda,ins} = \frac{s_{ins}}{\lambda_{ins}} \tag{4.16}$$

In the above formula:

s_{ins} – insulating layer thickness, m,
λ_{ins} – insulating layer thermal conductivity, W/(m·K).

For traditional in-floor heating systems with flat (non-profiled) insulation, the insulating layer thickness is adopted as s_{ins} (Fig. 4.7).

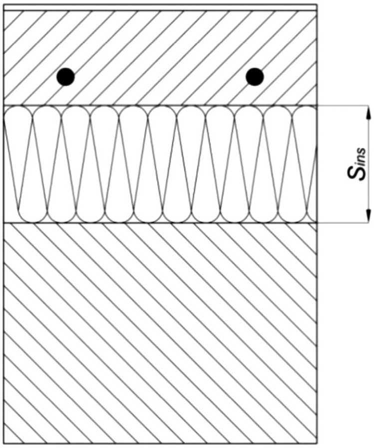


Fig. 4.7. Cross-section through a traditional floor heater with non-profiled thermal insulation [17]

4.2. IN-FLOOR HEATING SYSTEM DESIGNING

First, the unit heat flux q_{des} of the in-floor heating system in the room should be determined as:

$$q_{des} = \frac{\Phi_{N,f}}{A_F} \tag{4.17}$$

In the above formula:

$\Phi_{N,f}$ – nominal heat loss in the heated room, W,
A_F – heated floor area, m^2.

Usually, in-floor system heat output Φ_F should equal nominal heat loss $\Phi_{N,f}$. Otherwise, additional heating surfaces will be needed. The heat output of an entire floor surface provided with heating tubes is proportional to the conduit length L_R:

$$\Phi_F = q \cdot P \cdot L_R \tag{4.18}$$

The calculated conduit length is determined from the equation:

$$L_R = \frac{A_F}{P} \tag{4.19}$$

The conduit elbow length is omitted in formula (4.19). If there is an edge zone, the heat flux density should be calculated subject to division into the edge zone A_R and the comfort zone A_A:

$$q = \frac{A_R}{A_F} q_R + \frac{A_A}{A_F} \cdot q_A \tag{4.20}$$

The in-floor heating feed temperature is determined by the feed temperature of the room with the highest required unit heat flux density q_{max}, excluding bathrooms. A uniform thermal resistance of the floor's finish layer in heated rooms is assumed. Uniform shielding with resistance $R_{\lambda,B} = 0.1$ (m^2 · K)/W is assumed for the comfortable heating of rooms. If resistance $R_{\lambda,B}$ is higher, it should be accounted for. For bathrooms, thermal resistance $R_{\lambda,B} = 0$ is assumed. For heated rooms, temperature drop $\sigma \leq 5$ K is assumed. The need to divide the room heating into heating circuits should be reckoned with. In such a case, the unit heat flux (heat flux density) limit will correspond to the maximum in-floor heating output (Fig. 4.8).

For a room with q_{max}, the in-floor heating conduit spacing should be so selected that q_{max} remains less than or equal to heat flux density limit q_G defined by the boundary curve ($q_{max} \leq q_G$, Fig. 4.8). A small tube spacing is recommended for these cases. If $q_{max} < q_G$, a design temperature difference between the heating medium feed and the in-floor heated room is allowed, according to the formula:

$$\Delta\theta_{V,des} \leq \Delta\theta_{H,G} + 2.5\text{K} \tag{4.21}$$

The maximum allowable temperature difference between the medium and room air is then:

$$\Delta\theta_{V,des} = \Delta\theta_{H,des} + \frac{\sigma}{2} \tag{4.22}$$

where:

$$\Delta\theta_{H,des} \leq \Delta\theta_{H,G} \tag{4.23}$$

This equation is valid if $\sigma/\Delta\theta_H \leq 0.5$, where temperature drop $\sigma = 5$ K means that $\Delta\theta_H \geq 10$ K.

If ratio $(\sigma/\Delta\theta_H) > 0.5$, the following formula is applied:

$$\Delta\theta_{V,des} = \Delta\theta_{H,des} + \frac{\sigma}{2} + \frac{\sigma^2}{12 \cdot \Delta\theta_{H,des}} \tag{4.24}$$

In the above formulas:

$\Delta\theta_{V,des}$ – design temperature difference between the heating medium feed and the in-floor heated room, determined by the room, where q_{max}, °C,

$\Delta\theta_{H,des}$ – design temperature difference between the heating medium and the in-floor heated room, °C,

$\Delta\theta_H$ – temperature difference between the heating medium and the in-floor heated room, calculated from equation (4.2), °C,

$\Delta\theta_{H,G}$ – limit temperature difference between the heating medium and the in-floor heated room, °C.

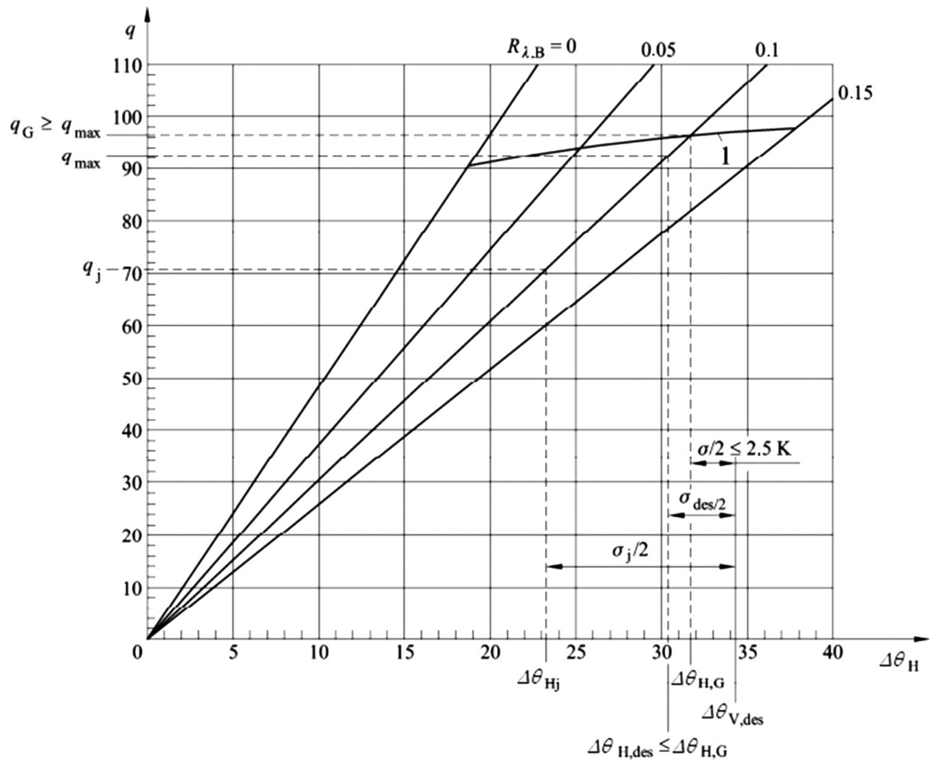


Fig. 4.8. Determination of design temperature difference $\Delta\theta_{H,des}$ and temperature drop σ_j for the j-th rooms with different loads [17]

For all other j-th rooms fed with a heating medium of the same temperature $\theta_{V,des}$ and subject to dependence $\sigma_j/\Delta\theta_{H,j} < 0.5$, the respective heating medium temperature drops σ_j should be determined based on Fig. 4.8 or calculated based on the relationship (using the temperature difference $\Delta\theta_{H,j}$ for various unit heat flux densities q_j according to Fig. 4.8):

$$\frac{\sigma_j}{2} = \Delta\theta_{V,des} - \Delta\theta_{H,j} \tag{4.25}$$

where:

$\Delta\theta_{H,j}$ – temperature difference between the heating medium and the air temperature in the j-th room (for a room with a unit load less than q_{max}, °C).

For $\sigma j/\Delta\theta_{H,j} > 0.5$, temperature drop σ_j is calculated from the formula:

$$\sigma_j = 3 \cdot \Delta\theta_{H,j} \left[\left(1 + \frac{4 \cdot \left(\Delta\theta_{V,des} - \Delta\theta_{H,j} \right)}{3 \cdot \Delta\theta_{H,j}} \right)^{\frac{1}{2}} - 1 \right] \tag{4.26}$$

If q_{des} cannot be achieved under the above conditions at any conduit spacing in a designed floor heating, it is recommended to use an edge zone and/or to provide additional heating surfaces. The additional heating surfaces should be selected so as to enable outputting an additional heat stream. The additional heat output is determined by the following equation:

$$\Phi_{out} = \Phi_{N,f} - \Phi_F \tag{4.27}$$

Design heating medium mass flow $\dot{m}_H$ in a heating circuit should be calculated as follows:

$$\dot{m}_H = \frac{A_{F \cdot q}}{\sigma \cdot c_p}\left(1 + \frac{R_o}{R_u} + \frac{\theta_i - \theta_u}{q \cdot R_u}\right) \tag{4.28}$$

In the above formula:

R_o – partial resistance to heat transfer upwards, $(m^2K)/W$,
R_u – partial resistance to heat transfer downwards, $(m^2K)/W$,
θ_u – nominal temperature of the room below, °C.

4.3. HEATING MEDIUM PARAMETERS – CONTROL SYSTEMS

For radiant heating, traditional heat sources such as gas, oil, solid fuel boilers, heat exchangers, and low-temperature heat sources (condensing boilers, heat pumps, solar systems) can be used. Whatever the choice, however, temperature control systems must be used that reduce the heating medium feed temperature at the inlet to the radiant heaters. If a heat source feed temperature is adequate to obtain the required radiant heater output, no control system is needed to reduce it. However, this is a special and rare case.

The temperature of the heating medium feeding a radiant heater depends on the demand for heat flux, the tube spacing, and the thermal resistance of heat transfer from the heating medium to the air in the heated room. The heating medium temperature usually ranges from 35 to 50°C for typical radiant heaters. A lower heating medium temperature often suffices for feeding radiant heaters with good thermal conductivity. For in-floor heating, these are heaters with a ceramic finish. Good thermal conductivity is characteristic for wall and ceiling heaters, as a rule with no low thermal conductivity coating. For dry or wet wall heating, the walls may be covered with a thin gypsum finishing coat, wallpaper or plaster. A higher heating medium temperature range is required when the radiant heater is finished (shielded) with a high thermal resistance material, e.g. wooden floor, parquet, panels or carpet. The above heating medium feed temperature range is not obligatory and may be slightly lower or higher. For wall heating, the heating medium temperature may be lower than 35°C. Similarly, for in-floor heaters shielded with a finish with a high thermal insulation, the heating medium temperature may be higher than 50°C. However, these are rather rare cases. In special cases, possible problems, e.g. drying out of wood, increased odour emission from carpets, etc. should be reckoned with.

The heating medium parameters and the heater tube spacing must be selected so as to ensure thermal comfort in accordance with the König chart and respective in-floor heating standards [15–19].

4.3.1. THERMOSTATIC TEMPERATURE CONTROL SYSTEM WITH A STRAIGHT-WAY THERMOSTATIC VALVE

The most popular solution for the reduction of radiant heater feeding medium temperature t_G is the temperature control system with a straight-way thermostatic valve (Fig. 4.9). It is a thermostatic system because outlet temperature t_M it is constant and depends on the thermostatic head setting (1):

$$t_G = t_M \tag{4.29}$$

where:

t_G – radiant heater feeding medium temperature,

t_M – heating medium temperature in the mixing node.

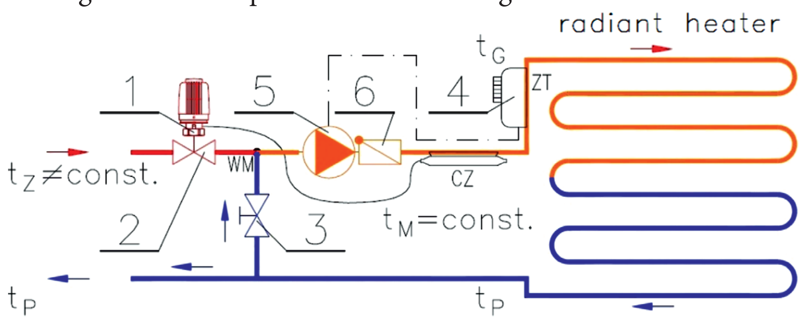


Fig. 4.9. Thermostatic temperature control system with a straight-way valve [20]:
1 – thermostatic head with contact sensor, 2 – straight-way thermostatic valve, 3 – manual control valve, 4 – safety switch, 5 – circulation pump, 6 – check valve

The operating principle of the heating medium feed temperature reduction system involves the phenomenon of mixing two medium streams with different temperatures t_Z and t_P, resulting in a heating medium with intermediate temperature t_M, such that:

$$t_P \leq t_M < t_Z \tag{4.30}$$

High temperature heating medium t_Z flows through thermostatic valve (2) – Fig. 4.9. The medium flow throttle rate depends on the setting of thermostatic head (1) and the temperature at the point of application of the thermostatic head of sensor **CZ**.

In mixing node **WM**, high temperature heating medium t_Z is mixed with low temperature cooled medium t_p returning from the radiant heater. Then, the reduced temperature heating medium flows through circulation pump (5), check valve (6) and the pipe to which contact sensor **CZ** of thermostatic head (1) is attached. When the heating medium temperature matches the temperature set on the thermostatic head knob, the opening of thermostatic valve (2) does not change. If the medium temperature at point **CZ** is higher than the temperature set on the knob of thermostatic head (1), the thermostatic head closes valve (2) until the temperature at point **CZ** matches the temperature set on the thermostatic head knob. If the medium temperature at point **CZ** is lower than the temperature set on the knob of thermostatic head (1), the thermostatic head opens valve (2) until the temperature at point **CZ** matches the temperature set on the thermostatic head knob.

The temperature of the heating medium feeding a radiant heater depends on the mixing ratio of the streams: the greater the share of the medium with low temperature t_P returning from the heater, the lower the resultant temperature (after mixing the streams). In the boundary case, the heating medium temperature at point **WM** is equal to the low temperature t_P of the medium returning from the heater. This happens when thermostatic valve (2) is completely closed and the entire stream of the heating medium returning from the heater is returned by circulation pump (5) to the supply. Thermostatic valve (2) can be completely closed when source temperature t_Z is significantly higher than the temperature set on the thermostatic head (this is a protection of the heater against overheating). Another extreme case is when the thermostatic valve is fully open; then the temperature of the medium after mixing t_M is close to feed temperature t_Z. This may happen when the heating medium feed temperature is too low compared to the temperature set on the knob of thermostatic head (1). However, it can never happen that the medium temperature after mixing t_M equals medium feed temperature t_Z. Even with thermostatic valve (2) completely open, part of heating medium t_P returning from the radiant heater will be sucked by the circulation pump into mixing node **WM** and will reduce temperature t_M, so that $t_M < t_Z$. In an existing system, the media mixing ratio depends on the opening rate of manual control valve (3): the higher the valve (3) opening rate, the greater the share of the medium returning from the radiant heater, which reduces temperature t_M. The more valve (3) is closed, the smaller the share of the return stream in the mix, and consequently the higher temperature t_M. Manual control valve (3) cannot be closed because it would disturb the basic control function of the temperature reduction system. By adjusting control valve (3), maximum temperature t_M (for maximum temperature t_Z) may be set, even with thermostatic valve (2) fully open. This is a protection of the radiant heater against overheating, e.g. in case of damage to the capillary tube of thermostatic head (1).

In practice, the control valve (3) opening should be set by trial and error on an operating system at the heat source maximum temperature, with temperature sensor **CZ** removed (temporarily) from the "measuring" tube and at the maximum temperature set on thermostatic head (1). The maximum setting and sensor **CZ** removal is to minimise the hydraulic resistance of thermostatic valve (2). With the above settings, manual control valve (3) should be slowly opened from the closed position, while watching the temperature rise at the mixing point. Once the maximum radiant heater feed temperature (+ 5°C) has been reached, the valve (3) setting should be locked or the valve knob removed so that the setting cannot be accidentally changed. After the control valve (3) adjustment, sensor **CZ** should be properly attached on the "measuring" tube, and thermostatic head (1) should be set appropriately to the radiant heater feed temperature. With all control operations completed, the radiant heating temperature thermostatic control system with a straight-way thermostatic valve will allow reliable operation of the heaters.

An additional protection is thermal switch (4) that turns circulation pump (5) off when the radiant heater feeding medium temperature at point **ZT** exceeds the thermal switch knob (4) set point. For this reason, the thermal switch (4) setpoint should be approx. 5°C higher than the thermostatic head (1) setpoint. The temperature at point **ZT** may exceed the setpoint if, for example, a pollutant from the system has locked the thermostatic valve head open or when the system pressure has overcome the thermostatic head pressure force.

Check valve (6) is installed in the system to ensure the right medium flow direction.

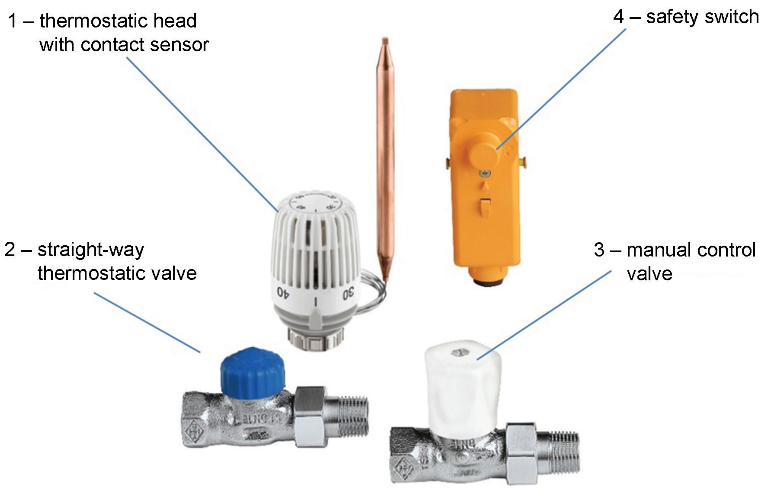


Fig. 4.10. Elements of a thermostatic temperature control system with a straight-way valve [21]

With typical thermostatic fittings available on the market (Fig. 4.10), the radiant heater feed temperature can be thermostatically controlled in a temperature range of 20–50°C. Due to the limited capacity of the thermostatic fittings and limited pressure force of the typical thermostatic heads, the maximum total surface area of radiant heaters fed from one mixing system should not exceed 160 m^2.

The thermostatic control system simplicity determines the reliability and universality of the above solution.

4.3.2. THERMOSTATIC TEMPERATURE CONTROL SYSTEM WITH A THREE-WAY THERMOSTATIC VALVE AT RETURN

The radiant heater feeding medium temperature control system based on a straight-way thermostatic valve and a thermostatic head with a capillary is cheap, simple and reliable. However, it has a certain limitation that reduces its applicability with low-temperature heat sources and heat sources with a wide heating medium temperature range. As mentioned earlier, heating medium temperature after mixing t_M is always lower than heating medium temperature t_Z, according to the formula:

$$t_M < t_Z \tag{4.31}$$

According to (4.29) and (4.31):

$$t_G < t_Z \tag{4.32}$$

For a high temperature heat source, this is the right relationship at all times. With low-temperature heat sources for which heating medium temperature t_Z changes temporarily, there may be such periods that the need to meet condition (4.32) will make t_Z "artificially" overrated. This may happen in a central heating system with a condensing boiler also feeding panel or convector radiators, or fan coil heaters. For a condensing boiler, with weather-compensated control of heating medium temperature t_Z, the temperature changes depending on the external air temperature. For external conditions consistent with the external temperatures calculated according to [22], the heating medium temperature will be above temperature t_G. If the external air temperature is higher than the calculated external conditions, e.g. at the beginning or end of the heating season, heat source feed temperature t_Z can match t_G or be lower, according to the relationship:

$$t_Z \leq t_G \tag{4.33}$$

This is due to the simple fact that, in this case, t_G is constant, while t_Z is variable. Another similar case is a system fed by a heat source with a large range of heating medium temperature changes, such as a periodically operated solid-fuel boiler with a buffer or a system with solar collectors. As regards a solid-fuel boiler, the large range of the heating medium temperature changes is due to its periodic operation, and the heating medium accumulation (heat storage) in the heat buffer. As regards a system with solar collectors, the large range of the heating medium temperature changes is due to the solar radiation variability over time. In either case, a system with radiant heaters will perform effectively if condition (4.32) is met. The system's performance will decrease for condition (4.33).

In automatic heating medium temperature control mode, it may turn out that due to condition (4.32), it is necessary to artificially maintain a higher temperature t_Z than that resulting from the weather-compensated control curve. For a system with low-temperature heat sources or sources with a wide feed temperature range, it is desirable to use a radiant heater feeding medium temperature control system, for which the following condition will be met:

$$t_Z \geq t_G \tag{4.34}$$

This condition is met by a temperature control system with a three-way thermostatic valve at return (Fig. 4.11). It is a thermostatic system because outlet temperature t_M is constant, and depends on thermostatic head setting (1):

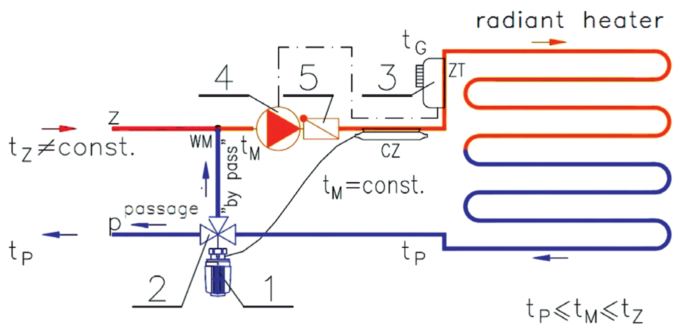


Fig. 4.11. Thermostatic temperature control system with a three-way valve at return [23]: 1 – thermostatic head with contact sensor, 2 – three-way isolation valve, 3 – safety switch, 4 – circulation pump, 5 – check valve

The operating principle of radiant heating feed temperature reduction involves the phenomenon of mixing two heating medium streams with different temperatures t_Z and t_P, resulting in a heating medium with intermediate temperature t_M.

Similarly, as before, high temperature heating medium t_Z (coming out of the heat source) flows to mixing node **WM** and is mixed with the cooled medium with temperature t_P, returning from the surface heater. Temperature t_M of heating medium exiting point **WM**, after the mixing of the two mass streams with different temperatures t_P and t_Z, depends on the proportion of these streams. Then, reduced temperature heating medium t_M flows through circulation pump (4), check valve (5), and the pipe to which contact sensor **CZ** of thermostatic head (1) is attached. When the heating medium temperature matches the temperature set on thermostatic head knob (1), then the opening of three-way valve (2) does not change. If the medium temperature at point **CZ** is higher than the temperature set on the knob of thermostatic head (1), the thermostatic head closes valve (2) until the temperature at point **CZ** matches the temperature set on the thermostatic head knob. Closing the three-way valve reduces the medium flow through the so-called passage directing its surplus towards the bypass. This increases the share of the lower temperature medium flowing to mixing node **WM**. A larger share of the lower temperature medium at point **WM** reduces temperature t_M. If the medium temperature at point **CZ** is lower than the temperature set on the thermostatic head knob, the thermostatic head opens valve (2) until the temperature at point **CZ** matches the temperature set on the thermostatic head knob. Opening the three-way valve increases the medium flow out of the system (point **P**), which increases the medium flow from the heat source (point **Z**). A larger share of higher temperature medium t_Z flowing to point **WM** increases temperature t_M of the heating medium flowing from node **WM** to the radiant heater. The temperature of the heating medium feeding a radiant heater depends on the mixing ratio of the streams. The larger the share of low temperature heating medium t_P returning from the heater, the lower the resultant temperature (after mixing the streams). In the boundary case, the medium temperature at point **WM** is equal to the low temperature t_P of the medium returning from the radiant heater. This happens when three-way valve (2) is completely closed and the entire stream of the heating medium returning from the radiant heater is returned by circulation pump (4) to the radiant heater feed. Three-way valve (2) can be completely closed if source temperature t_Z is significantly higher than the thermostatic head setpoint, or if the thermostatic head (1) setpoint is close to the temperature of the room in which the radiant heater is installed. This is a protection against overheating the heater. A three-way isolation valve view is shown in Fig. 4.12.

The other case is when three-way valve (2) is fully open. Then, medium temperature t_M downstream of mixing node **WM** is equal to feed temperature t_Z. This may happen if feed temperature t_Z is equal to or lower than the thermostatic head knob setpoint. The above case represents a fundamental difference in operation of the temperature control system with a three-way thermostatic valve, compared to the mixing system with a straight-way thermostatic valve, described above.

In the mixing system with a straight-way thermostatic valve, temperature t_M downstream of the mixing node is always lower than feed temperature t_Z.

The availability, downstream of mixing node **WM,** of the medium with temperature t_M equal to heat source temperature t_Z ($t_M = t_Z$) is desirable if the heat is fed from a low-temperature heat source, such as a heat pump or condensing boiler, or if the system is fed via a buffer heat (solid fuel boilers, solar systems). As regards

Fig. 4.12. Three-way isolation valve [21]

low-temperature heat sources, to increase the system efficiency index, the maximum feed temperature reduction is pursued. However, it is possible that a low-temperature source's output temperature is temporarily increased (despite worsening its performance) to obtain the appropriate domestic hot water temperature or a higher conventional radiator feed temperature, or to heat the room faster. In this case, it is necessary to use a radiant heater feed water temperature control system to protect the heater against overheating, and to make temperature t_M downstream of the mixing node equal to heat source temperature t_Z during a low-temperature operation period.

The other case is when the system is fed via a heat buffer. The feed temperature differences are then significant. The feed temperature is high in the final phase of "loading" the buffer through a solid-fuel boiler or solar system, while it is low in the starting phase of "loading" the buffer. With the buffer's ability to operate with significant temperature differences, its capacity can be decreased, because in a borderline case with the radiant heater feed water control system, the heating medium can be transferred from the heat source to the radiant heater without lowering the feed temperature. Similarly to the previous point, the additional radiant heater protection is thermal switch (3), which turns circulation pump (4) off when the medium temperature at point **ZT** exceeds the thermal switch knob setpoint. For this reason, the thermal switch setpoint should be approx. 5°C higher than the thermostatic head (1) setpoint. When the **ZT** thermal switch setpoint is equal to or slightly higher than the temperature set on thermostatic head (1), the system operation may be disturbed by switching off the pump. With the **ZT** thermal switch setpoint lower than the temperature set on thermostatic head (1), the circulation pump will switch off cyclically. The pump's mean operating time will depend on the difference between temperatures t_Z and t_P, and the on/off switching frequency will depend on the system's thermal inertia. The temperature at point **ZT** may exceed the setpoint if, for example, a pollutant from the system has locked the three-way valve head (2) open, or when the system pressure has overcome the thermostatic head pressure force, or when the capillary becomes damaged. Simple temperature control systems with direct-action controllers are the best guarantee of the entire system's reliability.

4.3.3. THERMOSTATIC TEMPERATURE CONTROL SYSTEM WITH THERMOSTATIC THREE-WAY VALVE AT FEED

Described above are thermostatic radiant heater feed water temperature control systems that also provide heater protection against overheating. For this purpose, straight-way thermostatic valves and three-way thermostatic isolation valves are installed in the return circuit.

This part of the textbook is devoted to a similar system of heater temperature control and protection against overheating with a three-way mixing valve, but installed in the feed circuit (Fig. 4.13). It is a thermostatic system because outlet temperature t_M is constant and depends on the thermostatic head setting (1).

The operating principle of radiant heating feed temperature reduction involves the phenomenon of mixing two heating medium streams with different temperatures t_Z and t_P, resulting in a heating medium with intermediate temperature t_M.

The control element is a thermostatic head with a contact sensor and a capillary.

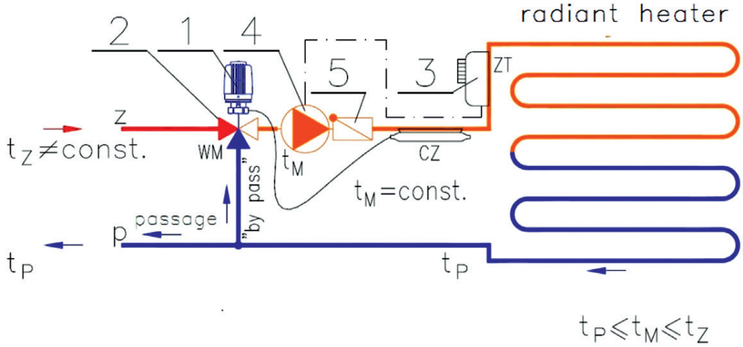


Fig. 4.13. Thermostatic temperature control system with a three-way valve at feed [24]: 1 – thermostatic head with contact sensor, 2 – three-way mixing valve, 3 – safety switch, 4 – circulation pump, 5 – check valve

Similarly as before, high temperature heating medium t_Z (coming out of the heat source) flows to three-way valve (2) and is mixed with the cooled medium with temperature t_P, returning from the radiant heater. In this case, the three-way mixing valve is also the **WM** mixing node. The temperature of the heating medium exiting three-way mixing valve (2), after the mixing of the two water streams with different temperatures (t_P and t_Z), depends on the proportion of these streams. Reduced temperature heating medium t_M exiting the three-way valve flows through circulation pump (4), check valve (5) and the tube to which contact sensor **CZ** of thermostatic head (1) is attached. Contact sensor **CZ** is connected to thermostatic head (1) by a flexible capillary, which transfers information about the tube temperature in the place where sensor **CZ** is attached. Depending on the temperature measured by sensor **CZ**, the thermostatic head closes or opens the thermostatic valve. When the heating medium temperature matches the temperature set on the thermostatic head knob, the opening of three-way valve (2) does not change. If the medium temperature at point **CZ** is higher than the temperature set on the knob of the thermostatic head, the thermostatic head closes valve (2) until the temperature at point **CZ** matches the temperature set on the thermostatic head knob. Closing the three-way valve reduces the medium flow with temperature t_Z (through the so-called passage) to mixing node **WM** (mixing valve (2)), while increasing the flow of medium at temperature t_P to mixing node **WM** (through the bypass). A larger share of the lower temperature medium at point **WM** reduces temperature t_M. If the medium temperature at point **CZ** is lower than the temperature set on the thermostatic head knob, the thermostatic head opens valve (2) until the temperature at point **CZ** matches the temperature set on the thermostatic head knob. Opening the three-way

Fig. 4.14. Three-way mixing valve [25]

valve increases the flow of medium at temperature t_Z to point **WM**, while reducing the flow of medium with lower temperature t_P through the bypass. A larger share of higher temperature medium t_Z flowing to point **WM** increases temperature t_M of the medium flowing from node **WM** to the radiant heater. The temperature of the heating medium feeding a radiant heater depends on the mixing ratio of the streams. The larger the share of the low temperature heating medium t_P returning from the heater, the lower the resultant temperature (after mixing the streams). In the boundary case, the medium temperature at point **WM** is equal to low temperature t_P of the medium returning from the radiant heater. This happens when three-way valve (2) is completely closed and the entire stream of the heating medium returning from the heater is returned by circulation pump (4) through the bypass to the radiant heater feed. Three-way valve (2) can be completely closed if source temperature t_Z is significantly higher than the thermostatic head setpoint or if the thermostatic head (1) setpoint is close to the temperature of the room in which the radiant heater is installed. This is a protection against overheating the heater. The other case is when three-way valve (2) is fully open. In this case, medium temperature t_M downstream of mixing node **WM** is equal to feed temperature t_Z. This may happen if feed temperature t_Z is equal to or lower than the thermostatic head knob setpoint. The above case represents a fundamental difference in operation of the temperature control system with a three-way thermostatic valve, compared to the mixing system with a straight-way thermostatic valve. A view of a three-way thermostatic mixing valve is shown in Fig. 4.14.

In the mixing system with a straight-way thermostatic valve, temperature t_M downstream of the mixing node s always lower than feed temperature t_Z ($t_M < t_Z$). This is the same with a three-way valve with incomplete bypass closing, e.g. valves in single-pipe systems. Thermostatic systems for radiant heater feed temperature control and overheating protection with thermostatic three-way mixing valves are similar to the systems with thermostatic three-way isolation valves. The main difference is the three-way valve location. An important practical aspect in favour of three-way mixing valves installed on the feed is their larger product range, in particular in the scope of large k_{VS} coefficients and the feasibility of their operation at higher pressure differences.

4.4. ROOM AIR TEMPERATURE – ADJUSTMENT

The previous sections describe issues related to thermostatic control of the radiant heater feed water temperature, and heater protection against overheating. For this purpose, thermostatic straight-way valves and three-way isolation or mixing valves are used. In all of these systems, the control element is a thermostatic head with a contact sensor and a capillary. A thermostatic valve with a thermostatic head and capillary is a proportional direct-acting controller (the system requires no external energy for proper operation). Thermostatic systems for radiant heater feed water temperature control are necessary for the correct operation of any radiant heating system, but are not sufficient. Their task in the feeding system is primarily to protect radiant heaters against overheating.

For full comfort and energy savings in radiant heated rooms, it is important to control the temperature not so much of the heating plane, but above all the air temperature in the occupied zone. This section is devoted to air temperature control systems in rooms heated with radiant heaters.

4.4.1. HEATED ROOM AIR TEMPERATURE CONTROL BY A HEAD WITH A CAPILLARY

The simplest heated room air temperature control system is also equipped with a thermostatic valve and a head with a capillary (Fig. 4.15).

The heater is protected against overheating by a system with a three-way mixing thermostatic valve, as described in the previous section.

When designing any type of heating, and thus also radiant heating, the starting point for the selection of equipment or system components is the heat balance of the rooms. The heat balance of rooms is made based on applicable standards and regulations, as well as technical expertise. Depending on the room's intended use, the thermal comfort conditions are set, the heat losses through partitions are calculated, and the demand for the heat flux to heat the ventilation air is calculated. The heat balance of rooms is made, for which design assumptions are adopted with some reserve. Therefore, the resultant heating system is slightly oversized.

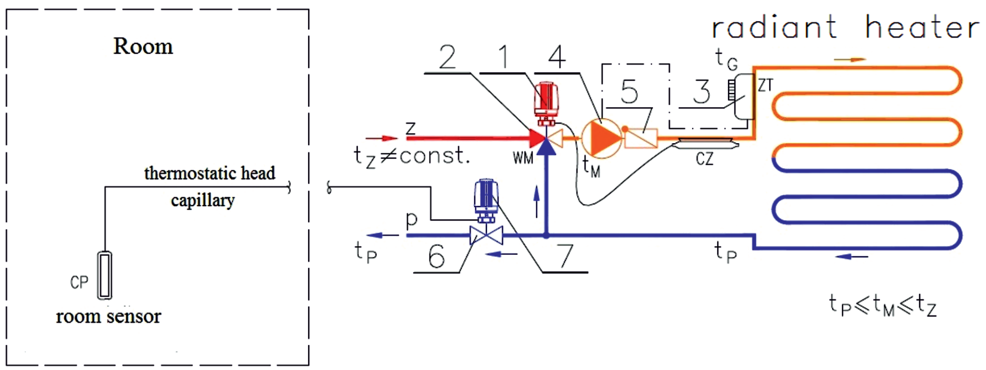


Fig. 4.15. Room air temperature control system with thermostatic head and capillary [26]: 1 – thermostatic head with contact sensor, 2 – three-way mixing valve, 3 – safety switch, 4 – circulation pump, 5 – check valve, 6 – straight-way thermostatic valve, 7 – thermostatic head with room sensor

This is why an option should be provided to adjust its performance depending on the conditions prevailing inside the heated room. Such a performance adjustment is possible with a feeding system with an additional thermostatic valve (6) and thermostatic head (7) with a capillary tube terminated with room sensor **CP** (Fig. 4.15). Any temperature surplus in the heated room over the thermostatic head (7) setpoint throttles the heating medium flow to the system, which in turn reduces radiant heater temperature t_G. This negative feedback, in terms of control, is caused by room sensor **CP** of thermostatic head (7) in the heated room. Information about the temperature in this room is transmitted from room sensor **CP** of thermostatic head (7) via the capillary tube to the head's actuator. The thermostatic head actuator can be a bellows or a piston set in the cylinder. The thermostatic head bellows or piston presses the thermostatic valve (6) stem, and the stem transfers the pressure to the valve head, causing the head to stick to the valve seat and reducing the heating medium flow. The heating medium streams mixing system, consisting of thermostatic head (1), three-way thermostatic valve (2), safety switch (3), circulation pump (4), and check valve (5), operates similarly to the earlier described system. The throttling of the system feeding the heating medium by thermostatic valve (6) adjusted to the heated room temperature, reduces the stream of high temperature heating medium t_Z flowing to mixing node **WM**, while pump (5) sucks more water returning from the radiant heater at low temperature t_P. Such additional throttling of high temperature heating medium t_Z reduces temperature t_M of the water leaving the mixing node.

The operation of the described system (Fig. 4.15) differs from that of the system without thermostatic valve (6) and thermostatic head (7), shown in Fig. 4.13, in the first phase of throttling the medium flow through valve (6). Then, temperature t_M downstream of mixing node **WM** will be lower than the thermostatic head (1) setpoint. In this phase, thermostatic head (1) attempts to compensate for the decrease in temperature t_M downstream of mixing node **WM** by opening thermostatic valve (2). Ultimately, thermostatic valve (2) will be fully open. Further closing of thermostatic valve (6), and thus throttling the heating medium flow into the system, will lower temperature t_M of the heating medium directly feeding the radiant heater, thus reducing its efficiency to the value equal to the actual demand for the heat flux resulting from losses at partitions and ventilation air heating.

Fig. 4.16. Head with remote sensor [21]

Thermostatic head (7) is also called a head with a remote sensor (Fig. 4.16), which means that the temperature to be maintained in the room is set directly on the head knob. Information about the temperature in the heated room is obtained by means of a room sensor and transferred by a capillary to the actuator in the form of a bellows or piston.

In the solution of room temperature control by means of a thermostatic head with a remote sensor, the temperature can be set in the location of the control unit installation (Fig. 4.15). The installation location of such a set is usually a closed installation cabinet in generally inaccessible rooms. Such a solution is particularly suited for temperature control in public areas, where the control system is systemically protected against third party access, e.g. in schools, offices, and public utility rooms. It should be noted that there must be a sensor in a room where the air temperature is controlled. The sensor is usually installed in a special ventilated protective casing or outside the direct zone of people's stay. The adjustment head (7) can also be a thermostatic head with a remote setter (room unit) (Fig. 4.17). The difference between a thermostatic head with a remote sensor (Fig. 4.16) and a head with a remote setter (Fig. 4.17) is that in a room where the temperature is controlled, an on-wall room temperature setter is installed.

Fig. 4.17. Head with remote setter, on-wall type [25]

There is an integrated room sensor in the setter knob that retrieves information about the heated room temperature. The on-wall head with a remote setter (and room sensor) also has a capillary, at the end of which there is an actuator in the form of, e.g. a bellows actuator or a piston to be mounted on thermostatic valve (6) – Fig. 4.15. In this case, the capillary transfers pressure to the actuator drive. The solution with a thermostatic head with a remote setter is characterised by the fact that in a room where the temperature is regulated, there is a setter and an air temperature sensor. This solution is applied in single-family houses and in apartments where the capability to change the settings by household members according to individual needs is desirable.

Temperature control in a room with one radiant heating loop and thermostatic heads with capillaries and remote sensors or remote setters has its limitations. The basic limitation is the finite length of the capillaries offered. Heads with 2 to 10 m long capillaries are available on the market. Where the control system is more distant, other solutions are necessary, which will be discussed below.

4.4.2. HEATED ROOM AIR TEMPERATURE CONTROL BY AN ELECTRICAL CONTROLLER

The previous part discussed thermostatic temperature control systems with a thermostatic head with a remote sensor or a thermostatic head with a remote setter (and sensor). To protect heaters against overheating, thermostatic straight-way valves and three-way isolation or mixing valves can be used. In all of these systems, the control element is a thermostatic head with a contact sensor and a capillary.

With a thermostatic head with a remote sensor or a thermostatic head with a remote setter, information about the heated room temperature can be transferred to an actuator built-in on the thermostatic valve. The information is transferred via a capillary. In either case, there is an air temperature sensor in the heated room, and an actuator built-in on the thermostatic valve. The difference between a system with a head with a remote sensor, and a system with a head with a remote setter is only the different signal carried by the capillary. With a head with a remote sensor, information about the prevailing room temperature is transmitted through the capillary, while the setter and actuator are installed on the thermostatic valve. With a head with a remote setter, there is a setter with a built-in sensor in the room. The capillary carries the pressure signal that drives the bellows of the actuator mounted on the thermostatic valve.

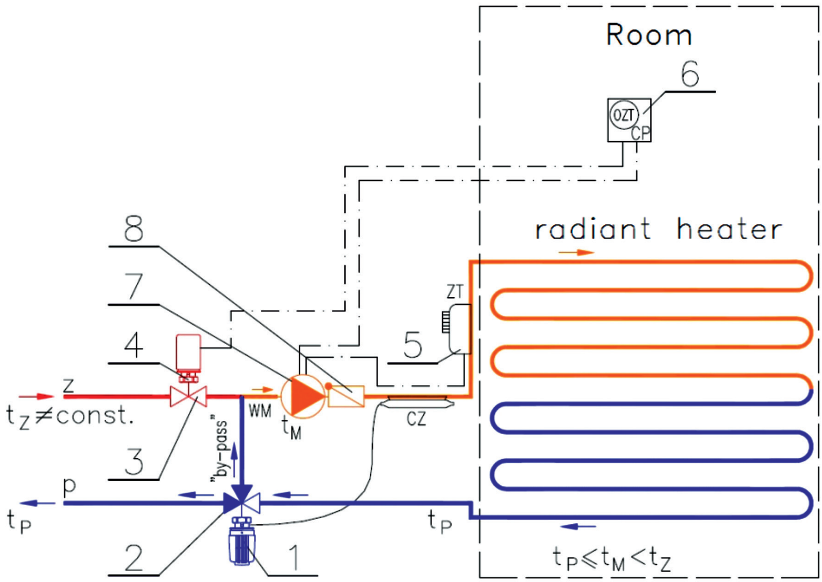


Fig. 4.18. Room air temperature control system with electrical controller [27]:
1 – thermostatic head with contact sensor, 2 – three-way isolation valve with minimal bypass, 3 – straight-way thermostatic valve, 4 – thermal actuator, 5 – safety switch, 6 – room controller, 7 – circulation pump, 8 – check valve

Both of the above solutions have many advantages, such as simplicity of operation, reliability, no need for power supply and sufficient control accuracy. They are also not without certain limitations due to the process characteristics and simplicity of operation, such as:
– finite capillary length of up to 10 m,
– high inertia of operation,
– limited heat output,
– if the capillary is damaged, the head must be replaced.

To eliminate the above limitations of the direct-acting system, a room air temperature control system can be equipped with an electromechanical or electronic controller and an actuator in the form of a thermal actuator (Figs 4.18 and 4.19).

The radiant heater is protected against overheating by a system with a three-way isolation thermostatic valve with a minimal bypass (2) and thermostatic head with a capillary and a contact sensor (1). With a three-way valve with a minimal bypass 100% opened, ca. 20–30% of the heating medium returning from the floor heater flows through the valve bypass, i.e. to mixing node **WM**. This solution is designed to protect the pump against flow loss in the event that thermostatic valve (3) closes completely. Three-way isolation valves with minimal bypass are also used in single-pipe installations.

Thermostatic temperature control in a radiant heated room operates the same way as a system with a thermostatic head with a capillary and remote setter built into the room or with a thermostatic head with a capillary and a remote sensor built into the room. They differ by different actuators (4) and controllers (6). The controller in the later solution is electrically powered. The actuator is thermal actuator (4) mounted on thermostatic valve (3). A voltage signal applied to thermal actuator (4) triggers an electric current in the resistive element integrated with the actuator and it heating.

Fig. 4.19. Thermal actuator [21]

Fig. 4.20. Electrical controller [25]

Heating of the working medium in the actuator causes its thermal expansion and pressure on the thermostatic valve stem, which results in throttling the flow. Removing the thermal actuator powering voltage causes the working medium to cool in the actuator element, its shrinkage, and return of the thermostatic valve stem to its original state, i.e. its opening. The controller (6) is typically a simple electronic or electromechanical room temperature controller (6), a so-called on/off two-position controller with a changeover contact. Integrated room sensor **CP** and temperature setter **OZT** with a rotary knob are contained in the controller. The setter's setting range depends on the controller type, and is usually within the typical indoor temperature range of 5–30°C. In an electronic controller, the changeover contact position is determined by the electronic system based on the room temperature information from temperature sensor **CP** built into the controller and depending on the **OZT** setter setpoint. In an electromechanical controller, the changeover contact position is determined by a simple thermostatic-mechanical system depending on the room temperature (**CP**) and the **OZT** setter setpoint. In either case, when the measured room temperature is lower than the **OZT** setpoint, the basic contact is closed, and when the measured temperature is higher than the setpoint, the basic contact is open. There is a certain hysteresis of the controller operations, in the range of 0.2 to 0.5°C. The operation of the control system consisting of room temperature controller (6) and thermal actuator (4) boils down to supplying voltage to the thermal actuator, i.e. closing the thermostatic valve when the measured temperature is higher than the setpoint, and cutting off the voltage supplying the thermal actuator (opening the thermostatic valve) when the measured temperature is lower than the controller setpoint. The above described actuator is of the NO type, i.e. normally open. Actuators with the inverse operation, i.e. NC (normally closed), are also available on the market. There are also thermal actuators, the operation variant of which is selected at the assembly.

The advantages of systems with simple electric controllers (Fig. 4.20) and thermal actuators are:

– distance between setter and thermal actuator can be increased,
– simplicity of operation,
– reliability in operation,
– temperature control accuracy (small systems),
– simple repair if control cable is damaged,
– actuator power supply circuit is typical electrical wiring,

– heating system can be switching off centrally (master switch),
– typical system components,
– several actuators can be connected to one controller.

However, such a solution has limitations due to its simplicity, control method, and applied actuators, such as:
– two simple on/off states separated by control hysteresis,
– no programming possible (operating time, temperature reductions, control hysteresis, etc.),
– temperature control at the controller installation location,
– high control inertia,
– temperature fluctuations in large rooms.

With heating systems of high inertia, simple on/off control with a simple electronic or electromechanical controller causes undesirable temperature fluctuations and changes in the heated room. In particular, this applies to large radiant heaters, i.e. radiators with a large control time constant. The control mechanism of a simple on/off type controller consists in applying a control voltage to thermal actuator (4) – Fig. 4.18, and slow opening of thermostatic valve (3), caused by the slow heating of the working medium in the thermal actuator. Due to its high thermal inertia, a radiant heater heats up slowly, especially at a low feed temperature (weak forcing). As the radiant heater heats up, heat is transferred from it to the heated room, so the room temperature rises. It will keep rising until it exceeds the controller setpoint by the hysteresis value. Once the temperature has exceeded the setpoint plus the hysteresis, the thermal actuator (4) powering voltage switches off, the actuator slowly cools down and thermostatic valve (3) slowly closes. After closing thermostatic valve (3), the radiant heater slowly cools down, and the room temperature slowly decreases. The control voltage is applied to thermal actuator (4) only when the room temperature has decreased in relation to the controller setpoint by the hysteresis value. Applying the control voltage makes thermal actuator (4) slowly warm up, thermostatic valve (3) open, the radiant heater heat up, heat transfer to the heated room, and the air temperature increase. The air temperature in the heated room will rise until the setter setpoint is exceeded by the hysteresis value, etc. This control method, due to the heater and thermal actuator (4) inertia, and the controller hysteresis, causes temperature fluctuations in the heated room. To mitigate this phenomenon, more advanced control devices are used.

4.4.3. HEATED ROOM AIR TEMPERATURE CONTROL BY AN ELECTRONIC CONTROLLER

The on/off control of heating causes temperature fluctuations in the heated room due to the heater's and thermal actuator's inertia, and the simple controller's hysteresis. To mitigate this phenomenon, an electronic pulse controller is used to adjust the room temperature. The radiant heater can be protected against overheating by a system with a three-way isolation thermostatic valve with minimal bypass (2) and a thermostatic head with a capillary and a contact sensor (1) – Fig. 4.21.

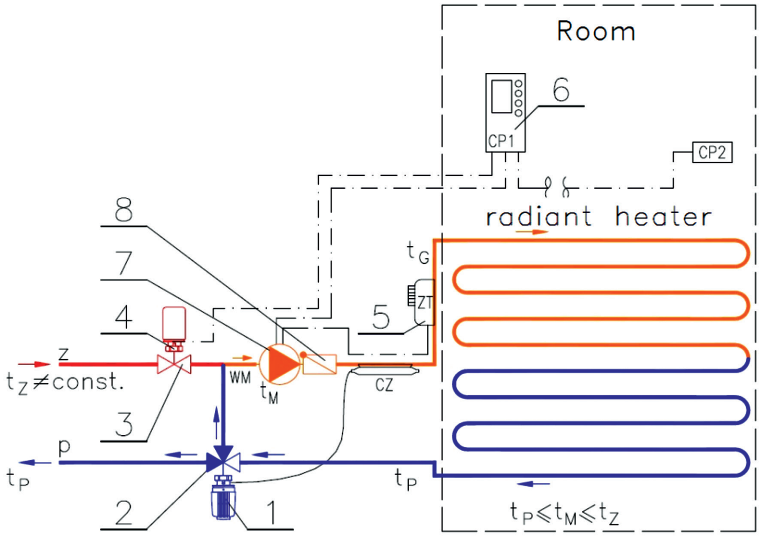


Fig. 4.21. Room temperature control system with electronic controller [28]: 1 – thermostatic head with contact sensor, 2 – three-way isolation valve with minimal bypass, 3 – straight-way thermostatic valve, 4 – thermal actuator, 5 – safety switch, 6 – room controller, 7 – circulation pump, 8 – check valve

The control by an electronic regulator is also of the on/off type, i.e. control voltage is applied to the thermal actuator or there is no voltage. However, the control dynamics and sequence of on to off states change, which is why the controller is called a pulse and smart. Pulse controller (6), e.g. of PI type, will in a sense predict the heating system response and will adapt to changing conditions. In the same situation, a simple electric or electromechanical controller will withhold its response until the heated room temperature exceeds the setpoint. Electronic pulse controller (6), based on information about the difference between the controller setpoint and the temperature measured in the room, will adjust its response to the actual situation. Of course, the response is always the same, i.e. “apply voltage” to the thermal actuator (on) or “do not apply” voltage to the thermal actuator (off). However, not only the state itself is important, but also the duration of each state. With the thermal actuator performance, which is characterised by significant inertia, applying the voltage to it (on) makes it slowly open, and removing the voltage (off) makes it slowly close. With a quick sequence of on and off states, while maintaining an appropriate ratio of the on and off durations, the position of the actuator (and thus also the valve) remains unchanged in the intermediate position between the “open” (on) and “closed” (off) states. And the intermediate position depends on the ratio of the pulse duration (on) to the interval between pulses (off). Partial opening of thermostatic valve (3) results in a quantitative adjustment of the radiant heater output depending on the actual demand for the heating medium flux. The measure of the demand is the error, i.e. the difference between the controller setpoint and the temperature measured in the room. In simple terms, it can be said that a large temperature difference means a large demand for heat flux, while a small difference means a state of near equilibrium or a small demand for heat flux. With a large temperature difference, thermostatic valve (3) opens considerably by maintaining the on state by controller (6) on actuator (4) much longer than the off state. With a small temperature difference, thermostatic valve (3) opens slightly by maintaining the on state by controller (6) on actuator (4) much shorter than the off state.

Fig. 4.22. Electronic controller [29]

Electronic pulse controllers (Fig. 4.22) usually have extensive features related to programming the operating time, control type selection, setting control constants, and additional functionality. A standard feature is programming the operating time in weekly or annual mode with an automatic clock switching to summer or winter time. In weekly mode, three temperature values: reduced, normal, and comfort temperature, can be selected. An important feature is the ability to choose the control type, e.g. PI or P type, along with the option of setting constant adjustments and, e.g. hysteresis values. Controllers with an internal **CP1** temperature sensor are standard. With some controllers, an external **CP2** sensor can be used. With a controller with an external sensor, the room air temperature can be adjusted in the sensor's place of installation, e.g. in a public location, generally accessible, while the controller itself with the setter can be located in a place inaccessible to third parties. This is the same solution as the control by a thermostatic head with a remote sensor.

With a "smart" pulse controller, simple on/off control is as effective as continuous control, while the apparent disadvantage of the thermal actuator, i.e. its inertia, is used to precisely control the heated room temperature.

4.5. CONTROL OF RADIANT HEATING WITH MULTIPLE HEATING LOOPS

The previous sections discuss temperature adjustment of radiant heating. and the protection of heaters against overheating in systems with one heating loop. This section is devoted to temperature adjustment and hydraulic control in systems with multiple radiant heating loops (Fig. 4.23). Systems with one radiant heating loop have an inherited limitation in that they can be used to heat or support heating one room only. This limitation is due to several reasons. For underfloor heating, there will be room walls in small rooms or expansion joints in large rooms. No wall can be erected on a floor heater, because heaters must be dilated from fixed partitions. Of course, heating two adjacent rooms with a loop consisting of two parts is theoretically possible. The loops must then be connected in series. In this case, however, one of the rooms must be dominant, where the room temperature control system adjusts the performance of the entire loop. In practice, this means that the temperature sensor of the air temperature control device is installed in the dominant room. Temperature in the other room is then resultant, and the desired temperature is obtained e.g. from an additional heat source controlled from an additional control system. The radiant heating output in the other room is considered the so-called heat gains. Another limitation of radiant heating systems with one loop is their limited length and, consequently, their limited heating capacity. In practice, the loop length of e.g. in-floor heating should not exceed 120 m, which means that such a system can be used for small and medium-sized rooms.

If a large room or several rooms need to be heated, a system with multiple heating loops should be used (Fig. 4.23).

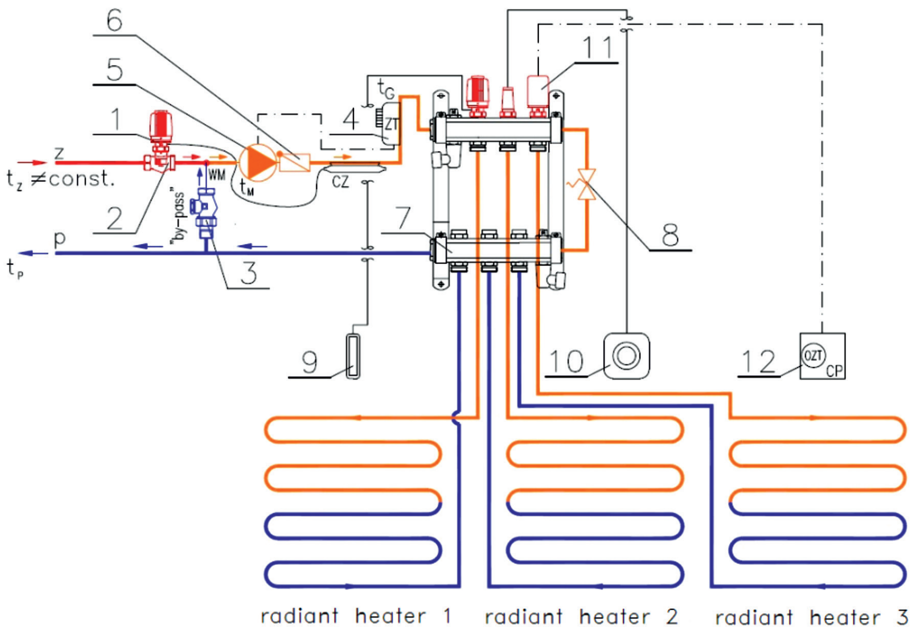


Fig. 4.23. Temperature control system of radiant heating with multiple heating circuits [30]: 1 – thermostatic head with contact sensor, 2 – straight-way thermostatic valve, 3 – bypass control valve, 4 – safety switch, 5 – circulation pump, 6 – check valve, 7 – floor heating manifold, 8 – relief valve, 9 – thermostatic head with remote sensor, 10 – thermostatic head with remote setter, 11 – thermal actuator, 12 – electronic (or electric) controller

A system with many heating loops is made up of two sub-systems: radiant heater protection against overheating and individual room air temperature control components (1)–(6) in Fig. 4.23 protect radiant heaters and the loops' heating medium drive module. The operating principle of the radiant heating feed temperature reduction involves the phenomenon of mixing two heating medium streams with different temperatures t_Z and t_P, resulting in a heating medium with intermediate temperature t_M.

This operating principle is described in detail in the previous sections. The other components (7)–(12) make up the sub-system of temperature control in the individual rooms heated by the radiant heaters, and the pump's "dry running" protection. In a system with several radiant heating loops, radiant heating distributor and collector (7) must be used. This is a special fitting with appropriate equipment. For distributor beam (7), these are thermostatic inserts on which thermostatic heads with a remote sensor (9) or heads with a remote setter (10) and thermal drives (11) can be installed. For collector beam (7), these are adjustment or shut-off inserts (Fig. 4.24).

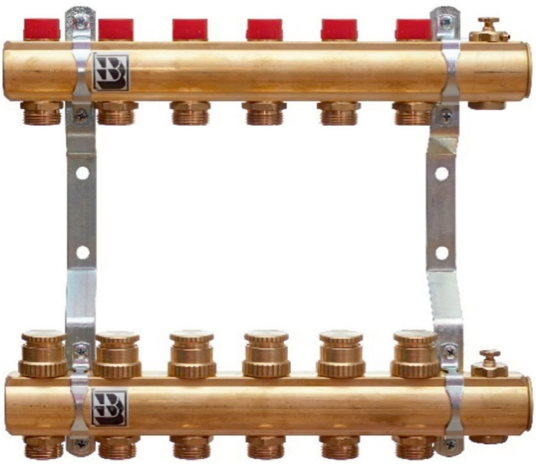


Fig. 4.24. Radiant heating distributor and collector with thermostatic and shut-off inserts [31]

The manifold is provided with the inserts just because it is the only place in a radiant heating circuit where thermostatic and control elements can be installed. The remaining parts of the circuits are heating loops routed in floor layers for an in-floor heater or in the heater itself. Besides this, radiant heating manifold (7) almost always has air vents and drain valves. The task of thermostatic head with remote sensor (9), head with remote setter (10), and the thermal actuator is to adjust the heating medium flow in each loop to obtain the set temperature in the heated rooms. The heads' special design (capillary) is meant to enable the transfer of information about the temperatures prevailing in the heated rooms to the actuator built-in on the thermostatic inserts of distributor beam (7). To thermal actuator (11), information and control are transmitted electrically from the controller (12) located in the heated room.

Fig. 4.25. Pressure relief valve [21]

The purpose of relief valve (8) is to protect the pump against "dry running" when all thermostatic inserts in distributor beam (7) close. When the pressure difference due to the closing of the thermostatic inserts increases, relief valve (8) opens and ensures minimal flow through circulation pump (5). A pressure relief valve is shown in Fig. 4.25.

The overheating protection system designed for many radiant heaters limits the liberty in designing the heaters themselves. The feed temperature set on the thermostatic head with contact sensor (1) must be adjusted to the radiant heater with the largest resistance to heat transfer or/and the heater in the room with the largest heat loss per surface unit. In practice, this means that radiant heaters are fed with the medium with the maximum allowable temperature. The thermostatic head (1) setpoint must not exceed the maximum feed temperature of any of the radiant heaters. If there is one radiant heater, its required output can be obtained at various heating medium feed temperatures (wide temperature range). The proper performance is obtained by appropriately adjusting the tube spacing in the radiant heater. If there are several radiant heaters, the feed temperature must be matched to the heater, which requires the maximum feed temperature, so the surplus heat output of the other heaters should be adjusted by increasing the tube spacing. This situation requires experience in the design of radiant heaters. The most difficult cases are those whereby radiant heaters with significantly different thermal resistances are fed from one mixing circuit. In-floor heaters can be provided with a ceramic or wood-like finish. In most similar cases, the heating medium temperature is set for wood-like finished heaters, and the surplus output of heaters with a ceramic finish is compensated by increased tube spacing. In this case, in-floor heaters with a wood-like finish have long loops with high hydraulic resistance. Heaters with a ceramic finish have short loops with low hydraulic resistance. This significantly hinders the hydraulic control at the stage of radiant heating installation balancing. Moreover, the thermostatic inserts built into distributor beams have the same capacity (k_{VS}), which even more differentiates the hydraulic resistances of individual loops. For this reason, the recommended authority (0.3–0.7) of thermostatic inserts in bathrooms and other small spaces is rarely achieved. This problem may be solved by increasing the number of heating loops with a wood-like finish. However, this requires an additional expansion joint that worsens the finish aesthetics.

Systems with multiple radiant heaters are used in large rooms or where several rooms are heated. In a large room, more than one radiant heater should be deployed, to:
- protect against rupture of a too-large heater due to its thermal expansion,
- reduce heating medium flow resistance,
- enable division into heating zones,

– enable the use of typical tubes, actuators, thermostatic heads and inserts in the manifold.

Where several rooms are heated, there is at least one radiant heater in each of them. To enable individual temperature control in each heated room, the loop of each heater is individually connected to the input of a distributor with a thermostatic insert and the output of a collector with a control or shut-off insert. Thermostatic control of the heated room air temperature is performed by heads with remote sensors (9), with a remote setter and sensor (10) or thermal actuators (11) with controller (12) – Fig. 4.23. Head sensor (9), head setter (10) or/and controller (12) are deployed in the heated room. The drives of heads (9) and (10) and the actuator(s) are installed on the thermostatic inserts of manifold (7). The task of the thermostatic head with remote sensor (9), head with remote setter (10) and the thermal actuator is to adjust the heating medium flow in each loop to obtain the set temperature in the heated rooms. In a heated room with more than one radiant heater, one room controller (12) and several thermal actuators (11) should be used. One controller controls several thermal actuators. The maximum number of actuators that can be connected to one controller is determined by the maximum current load of the signal output. The electrical power of most thermal actuators is ca. 3W, which is why manufacturers in most cases specify the maximum number of thermal actuators that can load the controller.

Each radiant heater should have individual thermal actuator (11) to enable flow control in its loops. Thermal actuators with electric or electronic controllers are used when the capillary length is insufficient (above 10 m) or control with night or weekend reduction should be provided.

For a full description of a control system, an analysis of the heating medium flow on the heating loop side is necessary. Where the tube lengths in individual radiant heating loops are similar, individual heaters' demands for heat flux are comparable, and the heating medium temperature drops are the same (e.g. 10°C), the system "balances itself" in hydraulic terms. This may happen when the demands for heat flux and the heater designs are the same in each room. A self-balancing system can also occur when the heated rooms' demand for heat flux is a multiple of the radiant heater output or in one large room heaters with the same output are deployed. These hydraulically self-balancing systems can occur when there is no individual thermostatic control of the radiant heaters. Any additional heat loss or gain in a heated room triggers the thermostatic control system's response, changing the heating medium flow in the radiant heater loop in the room. In a room with more than one radiant heater, the flows in other heaters may also change. A larger opening of the thermostatic insert in the manifold (7) supply beam that feeds a radiant heater will increase the heating medium flow in the respective loop and decrease the pressure difference between the supply and return beams. A decrease in the pressure difference (the so-called available pump pressure) is caused by the shift to the right of the pump (5) operating point due to the increase in its efficiency. A decrease in the pump (5) available pressure will simultaneously reduce the heating medium flow

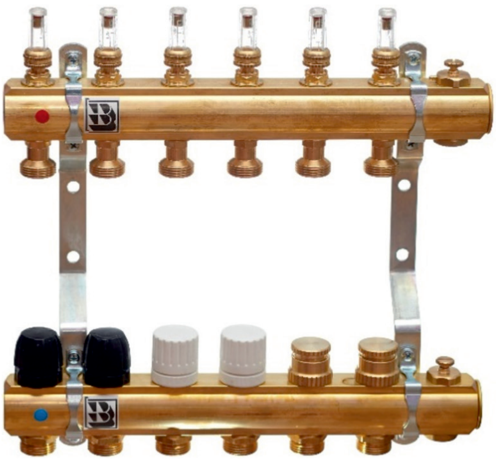


Fig. 4.26. Radiant heating manifold with thermostatic inserts and rotameters [31]

through other heaters. Activation of the thermostatic control systems of each radiant heater changes the heating medium flows in the other heaters, which changes their heat outputs. The rule of communicating vessels applies here. After the stabilisation of disturbances, on the heat gains and losses side, the whole system acquires equilibrium after some time. The time after which a system will reach equilibrium depends on many factors, but the basic view is the system span and thermal inertia of the radiant heaters. To shorten this, an electronic pump with constant available pressure may be applied as circulation pump (5). Now, it is a formal requirement of applicable regulations. This solution warrants the radiant heater's stable operation. Even significant changes in the flow caused by thermostatic control of one heater's output should not interfere with the others' performance.

Fig. 4.27. Radiant heating rotameter [31]

The aforementioned hydraulically self-balancing systems are rare. Most often used are radiant heaters with various loop lengths, outputs, and cooling. Radiant heater designers usually assume the same heating medium temperature drops, but in practice, the actual drops in various loops can vary significantly. In most heating systems, to balance the flows in individual radiant heating loops, a distributor-collector integrated manifold (Fig. 4.26) is used, with control and shut-off inserts, coupled with rotameters (Fig. 4.27).

Self-balancing is performed in the same way as in the heater system, by static throttling the surplus available pressure. Design flows determined by the designer based on the balance should be set with control elements, so-called rotameters. The popular term "rotameter" does not fully describe the distributor beam's equipment and features. A rotameter is a measuring instrument used to measure the flow of a flowing medium. With a rotameter built-in on the distributor beam, the flow can also be throttled by turning its head with simultaneous reading of the actual flow. If the feed beam of integrated manifold (7 in Fig. 4.23) is equipped with control and shut-off inserts (Fig. 4.28) only, then the actual heating medium flow cannot be read out. Adjustment by trial and error is burdensome due to the large inertia of radiant heaters.

Each new setting made during the adjustment process can give the desired effect only after a few hours, while the other radiators lose their balance. Experienced installers know that a traditional radiator system (with panel radiators) can be accurately adjusted only after a few iterations within a few hours. The same applies to radiant heaters, with the difference that the radiant heating adjustment in an building with many radiant heaters can take up to several days. Therefore, integrated distributor-collector manifolds with thermostatic inserts coupled with rotameters are often used to enable precise flow adjustment and real-time control of actual flows. This is particularly important in a building heated by many radiant heaters with varying heating loop lengths. It should also be noted that while trying to adjust one heating circuit, the others that have operated properly are misaligned. With the ability to read flow changes in the circuits that are not adjustable, their settings can be changed on an ongoing basis, without having to wait for the misalign effect. Another advantage of control elements with rotameters is the flowing heating medium's visibility, which allows its condition (colour, uniformity, impurities) to be assessed. This is very important when starting up a system that has already been vented. With such a static control system, the adjustment time of a radiant heating system can be significantly shortened, its operation controlled, and appropriate performance warranted.

Fig. 4.28. Control and shut-off insert [25]

4.6. WALL HEATING

This section is devoted to the wall heating system implemented in dry technology (Fig. 4.29). The term "dry" refers to the implementation technology. The wall heater is made without semi-liquid plaster with modifiers. In the traditional wet system, the heating pipe is attached to a prepared wall and covered with a layer of plaster in such a way that after the completion of construction works it is an integral part of the building partition. The dry system is based on prefabricated heating panels fixed to the wall, similar to the plasterboard system. These are fundamental differences between the systems in terms of technology, but these are not the only differences. The dry wall heating system is based on 15 mm gypsum-fibre boards with built-in heating tubes. These are multilayer plastic tubes with an aluminium divider, of the PE-RT/AL/PE-HD system. The heating tube's outer diameter is 10 mm, its wall is 1.3 mm thick, and the aluminium divider is 0.2 mm thick. The aluminium divider in a heating tube wall is butt-connected by laser welding. The gypsum fibre boards are reinforced with cellulose fibres. This makes them homogeneous and of high-density. The gypsum fibre boards are fireproof (F 30) and moisture resistant. Multilayer tubes are factory-pressed into grooves milled in the boards. Heating plates, also called heating panels, are intended for direct mounting on a supporting structure on a wall, ceiling or floor. The available panels dimensions are 2000 × × 625, 2000 × 310, and 1000 × 625. 15 mm gypsum fibre panels with built-in 10 × × 1.3 mm oxygen-tight multilayer tubes are shown in Fig. 4.29. The maximum heating medium temperature should not exceed 45°C.

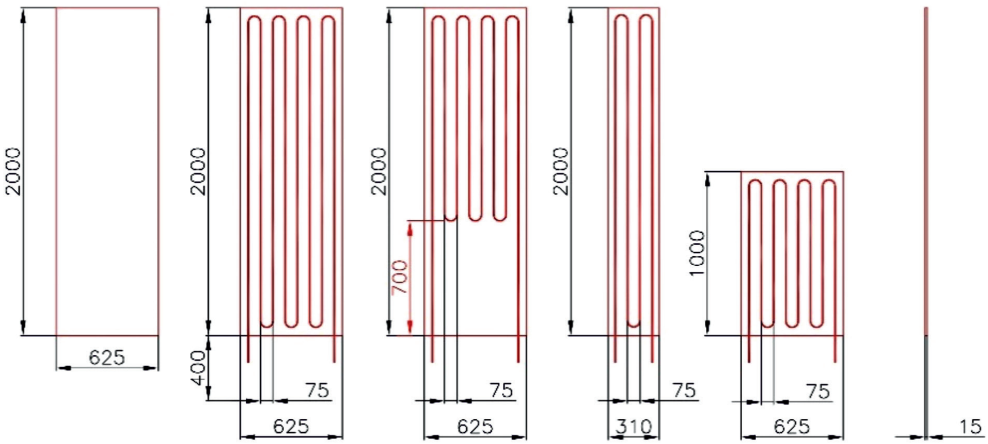


Fig. 4.29. Dry system heating panels [29]

Wall heating panels should be mounted on a support structure suitable for dry installation inside the building (Fig. 4.30). The support structure is made of metal slats (similar to the plasterboard assembly system) or wood vertically attached to the walls. The distance between the mounting strips should be half the board width, i.e. 31.2 cm. The space between the fastening strips should be filled with insulating material – pressed mineral wool is suggested. Heating plates are attached to the structure and glued to each other. Every few boards, a 3–5 mm expansion gap should be provided. The expansion joint is filled with a special flexible mass. The maximum length of a set of panels without expansion joints cannot exceed 8 m. The expansion joint filling is applied from a cartouche, and its surplus is removed after drying. Moisture contained in the air is used to properly harden the expansion joints. The filler material absorbs moisture, which makes it solidify, retaining flexibility to allow minimal thermal displacement of the boards.

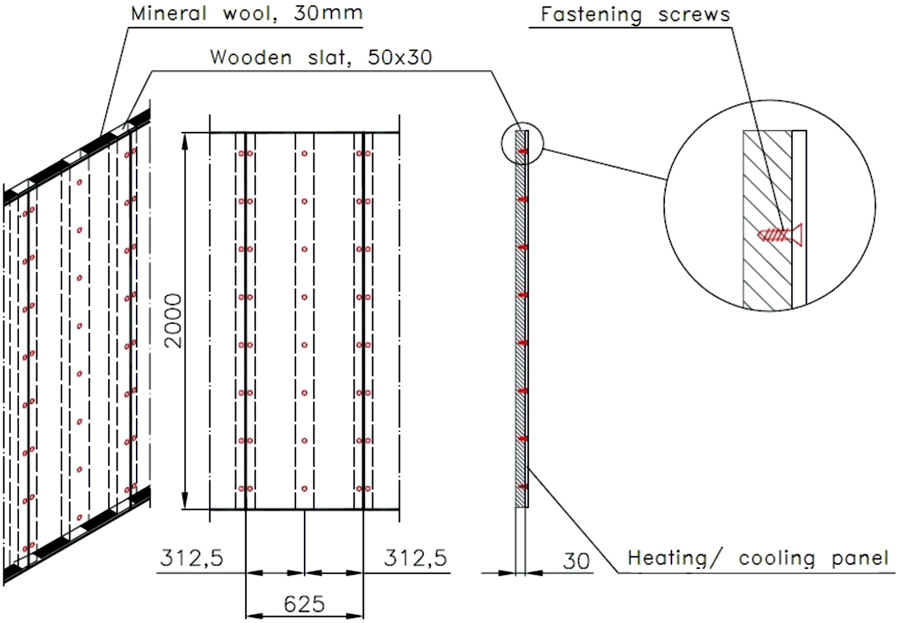


Fig. 4.30. Heating panel mount to wall [29]

The panels are pressed under pressure and have sharp edges. They may be cut with a hand-held circular saw. The boards are mounted with the smooth side towards the room. Fast-mount 30 or 45 mm long screws are used for fixing. 30 mm long screws are used for steel strips, 45mm long screws are used for wooden slats. The screws must have appropriate geometry, shape and thread pitch to ensure adequate mount strength. To avoid accidental drilling through a tube in a panel with a screw when fitting, recesses are routed on the panel. Unused recesses and fixing points should be levelled after assembly work is completed. The recesses are deep enough that a screw after panel fixing is recessed to depth of ca. 2 mm. After trowelling the joints and mounting holes, the heating walls can be painted, wallpapered, tiled, or covered with a thin layer of plaster. Gypsum fibre boards should be laid in temperatures above +5°C. The order of assembly is as follows: laying the panels, filling the expansion joints and holes, drying the room.

Wall heating panels are connected in series (two or three system panels) – Fig. 4.31 – so that the length of one heating loop does not exceed 55 m. Each wall heating loop is connected directly to the manifold in an installation cabinet, similarly to floor heating loops. Tichelmann system connections are recommended. The panels are connected in series with press fittings and connected directly to a 20 mm distribution or collection pipe.

The wall heating output depends on the heating medium temperature and its drop, as well as the room temperature. For example, with an average temperature difference between the heating medium and room of 15°C, feed temperature of 40°C, return temperature of 30°C, and room temperature of 20°C, the single plate output is 92 W.

Wall heating system panels can be adapted for in-floor or ceiling heating. For this purpose, the dry screed from joining two 10 mm thick Fermacell boards is used (Fig. 4.32). Screed boards are laid in the same way as for floor heating on wooden slats laid on the floor, spaced by 31.2 cm. The space between the slats is filled with insulating material. Wall heating system panels are glued and screwed to the top of composite screed panels. The screed panels support the heating panels and strengthen them mechanically. The floor cladding (finish) is fitted directly to the heating panels. The cladding must be suitable for in-floor heating.

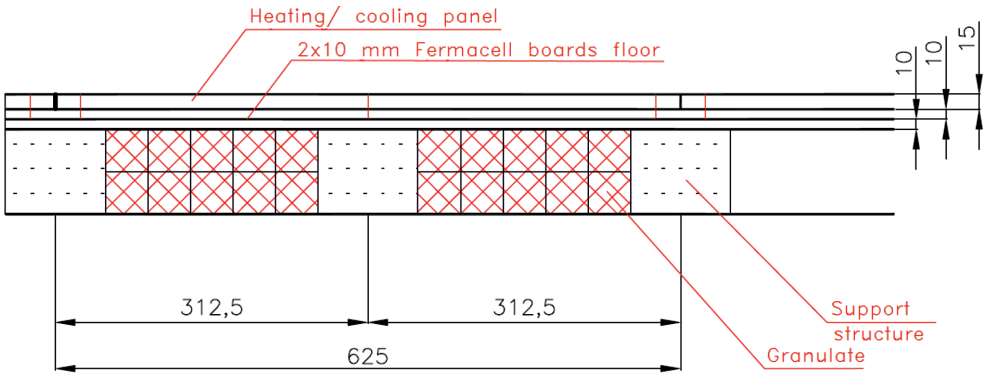


Fig. 4.32. Dry system in-floor heating [29]

Wall heating panels can also be used for ceiling heating, with the difference that they are attached from below to a supporting structure attached to the ceiling of the heated room (Fig. 4.33). The support structure can be the same as for suspended ceiling installation. The heating panels in the ceiling void thus formed should be covered with 50 to 100 mm thick insulation mats.

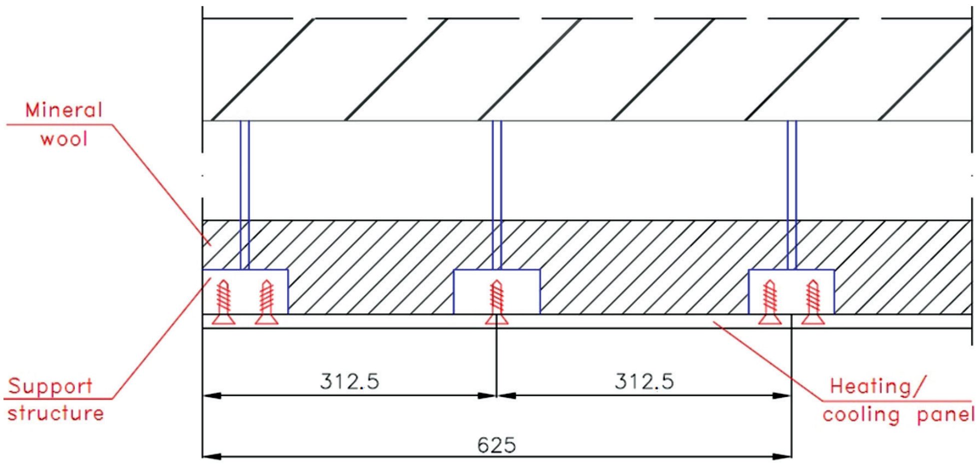


Fig. 4.33. Dry system ceiling heating [29]

The wall heating system can also be used for wall cooling. The dry wall heating system can compete to some extent with the wet wall heating system. These systems, due to their specificity related to the assembly technology and features, complement each other.

Dry system boards for wall heating and cooling are no different. Their different operation is due only to different parameters of the feeding medium. When a dry system is fed with a heating medium, it heats the room. When a dry system is fed with a cooling medium, it cools the room.

The driving forces behind the development of new technologies in this heating technology segment are undoubtedly the growing demand of building users and increasing energy prices. The basic feature of radiant systems is the use of building partition surfaces to transfer the cooling flux to cover room heat gains. This solution allows for the reduction of the interior design elements intended only for cooling or air-conditioning, such as fan coils or air-cooling systems. By completely covering the heat gains from people, partitions and room equipment, the cooling ventilation airflow can be significantly reduced. If heat gains are absorbed by a wall, ceiling or floor cooling system, then the inflow (ventilation) airstream will be reduced to hygienic needs coverage only. The inflow will only be fresh air, without recirculated air.

The reduction in the number of cooling devices results in more space available that can be arranged freely without constraints due to air conditioning system components. In office or commercial buildings, rooms are designed as high-volume, so-called "open space" with separate sanitary facilities. Such large rooms are often divided into smaller ones after their sale or rental. Smaller rooms are separated with plasterboard partition walls. With the dry technology, the heating and cooling system can be integrated within the partition wall system, in which standard plasterboard panels are replaced with heating or cooling panels. A characteristic feature of the wall cooling system is the significant "spread" of the cooling surface compared to traditional solutions with fan coils or air conditioners. A large cooling surface area means a relatively high temperature of the surface itself while maintaining the required overall capacity. This issue should be considered in two aspects: economic and thermal comfort. The economic aspect means a reduction in the demand for cold, resulting from a higher temperature in cooled or air-conditioned rooms, and more effective use of traditional or non-standard sources of cold. With a large cooling surface area, the contribution of low-temperature partitions (less radiation) to thermal comfort and heat absorption is greater than with traditional cooling. Thermal comfort is, therefore, felt at higher air temperatures. The higher temperature in cooled or air-conditioned rooms means less demand for "cold" flux. This is so because with a smaller average temperature difference on both sides of a partition, there are smaller heat gains resulting from heat transfer through it. This applies to partitions between cooled rooms and rooms without cooling or external partitions of cooled rooms. Lower demand for the cold flux is also due to the lower demand for cold to cool the supply air. The supply air doesn't need so much cooling to provide the same effect. When radiant cooling operates at a higher coolant temperature, cold sources such as chillers or heat pumps can work more efficiently. The relatively high feed temperature means that a portion of the "cold" from unconventional sources, e.g. tap water, can be practically utilised and later used for domestic hot water. Another unconventional cold source can be the lower source of a heat pump with a ground collector during regeneration, so-called passive cooling. In a heat pump operated in reverse mode, whereby the desired effect is cold, in certain conditions, closing the average temperatures of the lower and upper heat sources to each other by 1°C can improve the device efficiency by up to 10%. An interesting issue when developing a room cooling concept is the aspect of the thermal comfort of the people staying there. It is often the case that designers limit the thermal comfort only to ensuring the design temperature of 24°C in the summertime, in accordance with the standard specifying the conditions of thermal comfort for air conditioning [32]. The appropriate mean temperature in cooled rooms is one of the elements of thermal comfort. The uniformity of the temperature and radiation field in cooled rooms is also important. All of these parameters depend on the building partitions, amount of glazing, heat source locations, i.e. factors that the designer of sanitary installations does not have any influence on. Therefore, systems that allow adverse conditions in cooled rooms to be reduced should be used. Examples of such a solution are wall, floor or ceiling cooling systems. Radiant cooling, due to the cooling surface spread over a large area (large area cooling), allows a significant reduction in the temperature difference in cooled rooms, and makes the radiation field homogeneous throughout the entire area. With radiant cooling, cooling by air systems can be reduced, which is of great importance for improving the comfort of staying in cooled or air-conditioned rooms without the draft effect. However, radiant cooling systems have limitations too. The basic one is the unit efficiency and the inability to intensify it. Due to the partition temperature lowering, the summer comfort temperature can be assumed as 28°C. With a coolant feed temperature of 17°C and heating it by 3°C to 20°C, a unit cold output of ca. 40 W/m^2 can be obtained. The above limitation results from the coolant feed temperature of 17°C. Such a medium temperature may cause condensation of moisture from the air on the partition wall. With a room temperature of 28°C and a cooling medium temperature of 17°C, the maximum air humidity should not exceed 48% while maintaining a 1°C difference between the feed temperature and the condensation temperature of the moisture in the air cooled in the room. This limitation reduces the cooling system's performance if the room air humidity exceeds 48%. An indispensable requirement is the use of dew point sensors to protect the partitions against uncontrolled condensation, and automatic systems capable of partition surface humidity control. Another limitation is the low cooling circuit capacity. This is due to the cooling loop lengths being limited to 55 m, and the small temperature difference between the cooling medium feed and return, which is only 3°C. In traditional heating systems, the feed and return temperature differences are between 15 and 25°C, while in traditional cooling systems, these differences are between 5 and 7°C. With a low medium temperature drop, the flows have large volumes, which in turn causes high hydraulic resistance, large tube diameters, and high capital expenditures. The applicability of dry cooling systems is also limited by the range of available cooling panel shapes, and their suitability for construction of flat or angular partitions only. Dry system panels can form neither circular, nor arching and elliptical partitions. The low power output limits the application of only dry surface cooling systems to buildings that are well-insulated or have low heat gains. Due to its specificity, the system always needs to be designed and implemented by highly qualified professionals.

When combining radiant heating and cooling (Fig. 4.34), the system sizing should be based on the required cooling demand. Heating capacity will almost always be less demanding.

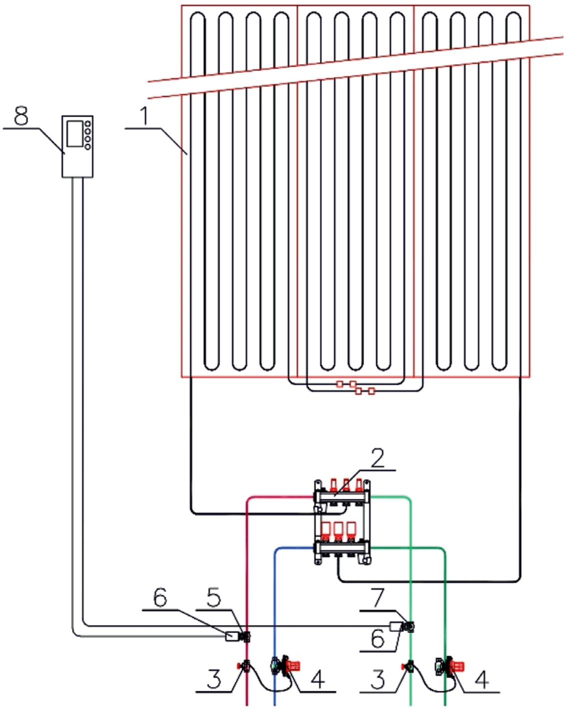


Fig. 4.34. Example of heating and cooling combined into one system [29]; 1 – wall panel, 2 – manifold, 3 – balancing valve, 4 – differential pressure regulator, 5 and 7 – thermostatic valves, 6 – thermal actuator, 8 – controller

The system described in this section, used alone, despite its many advantages, also has limitations. They can be minimised if the dry wall cooling system is used in configuration with other systems. In combination with air conditioning or cooling ventilation, it is a perfect complement to the air system. A well-designed radiant heating system operated with heat pumps in the wintertime, in the summertime can cool rooms without additional costs by reversing the heat pump operation or, more interestingly, by passive cooling with ground heat exchanger regeneration. Therefore, it is always worth considering the possibility of using a dry radiant cooling system when designing a dry radiant heating system.

4.7. HEAT SOURCES IN ENERGY-EFFICIENT BUILDINGS WITH RADIANT HEATING AND COOLING

There are two aspects to consider in the issues related to saving heat energy. The first aspect concerns solutions of passive protection of buildings against heat losses, such as effective thermal insulation of partitions, protection against unwanted infiltration of external air and minimisation of the number of thermal bridges. The second aspect refers to active solutions, which involve systems with which the need for thermal energy can be minimised. An example of such solutions is ventilation systems with heat recovery, operating periodically depending on the needs. Other solutions can be energy-saving heating systems and heat sources with high energy efficiency. Examples of energy-saving heating systems are radiant systems, i.e. in-floor, wall, and ceiling heating. Minimising the energy consumption by radiant systems can be achieved in several ways. The basic premise is the ability to lower the mean temperature in the heated rooms while maintaining the thermal comfort of the people staying in them. This is possible because the thermal comfort of a person depends on the air temperature in the room and the mean temperature of the surrounding partitions. The partitions' temperature and area determine the amount of heat radiation emitted. Heating a room through the floor and walls increases the amount of energy supplied by radiation, which can reduce the air temperature in this room. The overall heat loss for heat transfer through partitions and the heat demand for ventilation air heating are largely determined by the air temperature. Reducing the air temperature reduces the heat loss, i.e. saves heat. In radiant heating and cooling systems, typical combined regulation and control stations are applied for individual temperature control in heated rooms in terms of the value and time. Individual temperature control in heated rooms allows to the thermal comfort conditions to be adjusted, and periods of temperature reduction in them to be programmed. It is assumed that lowering the temperature by 1°C from ca. 20°C can save up to 5% of thermal energy. Radiant heating primarily heats the zone of people's stay, ensuring almost perfect temperature distribution.

Radiant heating systems can be fed by low-temperature and energy-saving heat sources, such as heat pumps, solar collectors (Fig. 4.35) and heat recovery systems from refrigeration circuits. At our latitude, solar systems can provide auxiliary heat only in the heating season, however, their output can be a significant contribution in the transition periods, when they may account for half of the heating season.

Since it is an environment-friendly and free source of heat, it is worth considering when choosing a heating system. Heat pumps are used as environment-friendly, energy-saving and year-round sources of heat and cold. Most often they feed the central heating and domestic hot water systems. Available heat pumps can cooperate with the lower heat sources in the form of ground collectors, vertical probes and water from wells or underground and surface watercourses. The cooperation of heat pumps with surface heating systems results in the pumps' high energy efficiency ratios and COP coefficients over 5. It is important for energy-saving heating systems with heat pumps that the upper heat source (heating system)

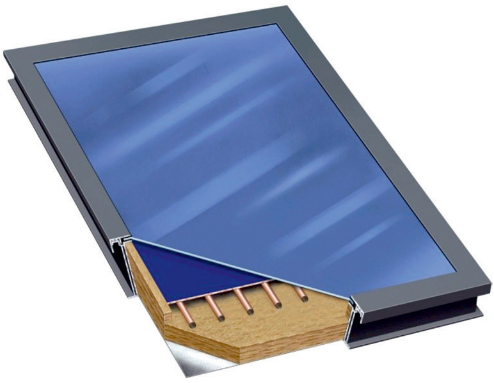


Fig. 4.35. Flat solar collector [29]

Fig. 4.36. Vatra heat pump [33]

operates at the lowest possible temperature. In terms of W10/W35 heat pump operating parameters, reducing the feed temperature by 1°C increases the system efficiency (COP) by about 5%. Therefore, coupling heat pumps and panel systems is very beneficial. With glycol/water heat pumps and a radiant heating system, the heat pump operation can be so reversed that the heating system cools the rooms in the summertime without the need for its special expansion. Comprehensive heating and cooling systems integrated with heat and cold sources provide such opportunities.

For the sake of low costs of heating (and/or cooling), systems with radiant heating (and/or cooling) should be fed by heat pumps with high energy efficiency. An example of such a heat and cold source is the Vatra CT 8B-200 family of heat pumps, with a heating capacity of 8.3 kW. In systems with radiant heating, water/water or glycol/water heat pumps are most often used. For example, at B2/W35 normative parameters, the COP efficiency coefficient of the Vatra CT 8B-200 glycol/water heat pump may be as high as 4.81. To obtain high energy ratios, reduce operating costs and obtain high comfort of use, comprehensive solutions of energy-saving heating and cooling systems with modern and ecological sources of heat and cold are proposed.

4.8. WARM FLOOR COMFORT

The most common heating system now is central heating with panel, fin, or ladder radiators. These are used in small buildings, such as single-family houses, and large blocks of flats, hotels, guest houses, public facilities. In residential facilities, some rooms require the comfort of a warm floor. These include bathrooms, where standing on a warm floor is desirable, especially after leaving a bathtub or shower cubicle.

However, in-floor heaters require individual feeding systems and temperature control. This applies to:

– radiant heater protection against overheating by thermostatic temperature reduction systems,
– special manifolds with thermostatic inserts and rotameters,
– thermostatic heads with capillaries or thermal actuators with electric or electronic controllers.

This particularly applies to facilities with traditional radiators (e.g. panel radiators) and radiant heaters alike. Due to the mutual temperature and hydraulic mismatch of these systems, they are separated by independent heating circuits fed from one heat source. There are two reasons for the temperature mismatch. Firstly, traditional radiators are fed at a higher temperature than radiant heaters, secondly, traditional

Fig. 4.37. RTL return limiter [21]

radiators normally operate with heating medium temperature drop 15–25°C, and radiant heaters with 8–12°C. The hydraulic mismatch results from the different hydraulic resistance of circuits with panel radiators and radiant heaters. Circuits with radiant heaters feature higher hydraulic resistance compared to circuits with typical radiators (due to the considerable length of tubes in a radiant heating loop). If the area heated with radiant heating is comparable with the area heated with traditional heating, the systems are separated by independent heating circuits, adapted to the heaters' specificity. A certain problem arises if a significant portion of the rooms in the facility is heated by panel radiators, while one or two rooms are heated by in-floor heaters, e.g. kitchen, bathroom. The question arises whether it is worth using for one or two small loops the special mixing systems and systems to protect the radiant heater from overheating, or whether special manifolds with thermostatic inserts and rotameters should be used.

There is a compromise solution, which, however, rather than in-floor heating should be called the warm floor comfort. Its applicability is limited by many conditions. Such a solution is the so-called RTL return temperature limiter, which warrants comfort at moderate costs and simplicity of installation (Fig. 4.37 and 4.38).

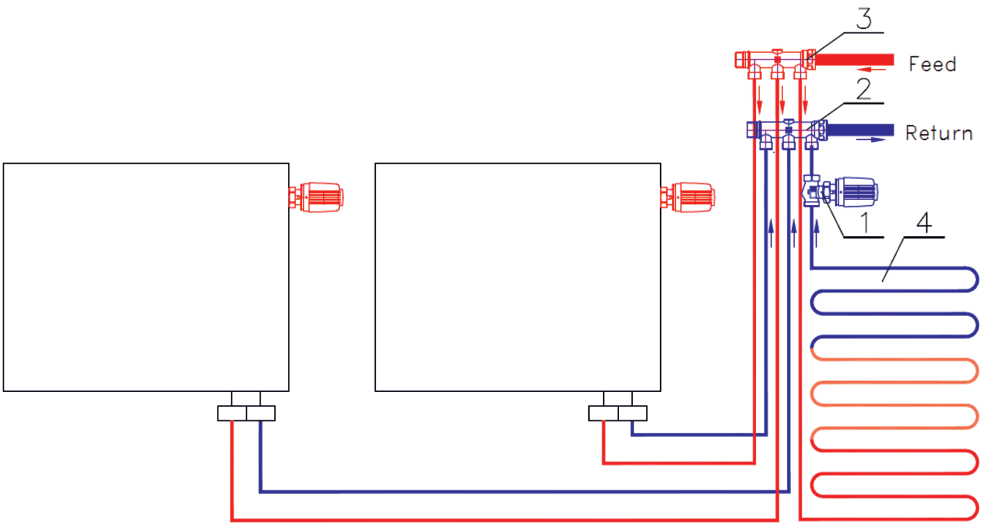


Fig. 4.38. Floor comfort system with panel radiators with incorrect RTL connection [34]; 1 – RTL thermostatic return temperature limiter, 2 – working medium collector beam, 3 – working medium distributor beam, 4 – floor heater

This solution can be applied to small rooms in single-family houses or buildings with a single-story heating system fed from a low-temperature heat source. The heating medium feeds a typical low-temperature heater system via the distributor beam (3) — Fig. 4.38 — in-floor water heater.

The heating medium flowing at a low speed through the water loop of floor heater (4) gives off heat to the heated floor. At the end of the in-floor heating loop, RTL return temperature limiter (1) is installed to control the temperature of the medium returning from the in-floor heater's heating loop. When the flowing medium temperature is lower than the setpoint, the limiter automatically increases its opening, and when the temperature is too high, it throttles the flow. The cooled medium returns to collector (2). Under steady-state conditions, the system always reaches equilibrium, which makes the floor temperature stabilise. The mean floor temperature depends on the RTL return temperature limiter (1) setpoint. This solution is appropriate for low-temperature systems and when the heated floor area is not too large and does not exceed a few square meters.

A common mistake when using RTL is to fit the valve part of the limiter directly to the collector beam. This solution makes the control element respond to the mean collector temperature, and not to the temperature of the medium returning from the floor heater loop. The collector (2) temperature is close to the average temperature of the medium returning from the radiator system, which is significantly higher than the temperature of return from the floor heater circuit. The excessive collector temperature closes the return temperature limiter and interrupts the heating medium flow through the floor heater, which leaves floor cold. Shifting the RTL adjustment range does not solve the problem, because when the RTL setpoint is significantly higher than the expected floor temperature, then control over the floor temperature is lost. To avoid the effect of the collector beam temperature on the RTL return limiter performance, a distance of a few centimetres should be kept between the RTL and the collector.

The warm floor system with an RTL return limiter does not provide air temperature control in the heated room, which is important for thermal comfort. To respond to market needs, fittings manufacturers have implemented a special radiant heating control system for installation concealed under plaster and with double temperature adjustment (Fig. 4.39).

An example control system consists of the Floor Fix control unit (Fig. 4.39) and a thermostatic head, with remote sensor and setter. Optionally, the thermostatic head with remote sensor and setter may be replaced with an electronic controller and a thermal actuator. The idea of a temperature control system with the Floor Fix control unit is to combine all of the essential features of a radiant heating system in a single device so that it can be used in a traditional radiator system. Floor Fix has two thermostatic elements — a return temperature limiter and a traditional thermostatic insert (as in thermostatic valves). The return temperature limiter's operating range is adjustable within 30–60°C. The thermostatic insert's operating range depends on the thermostatic head type, and is usually 6–28°C. The radiant heater and the Floor Fix unit are connected with the central heating system the same way as in systems with an RTL (or RTB).

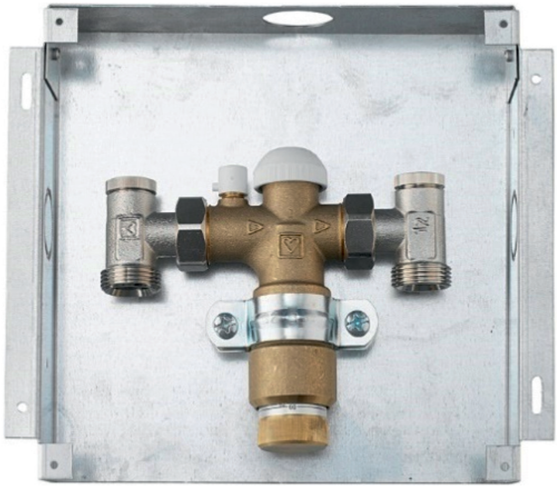


Fig. 4.39. Floor Fix floor heating control unit [29]

a) b)

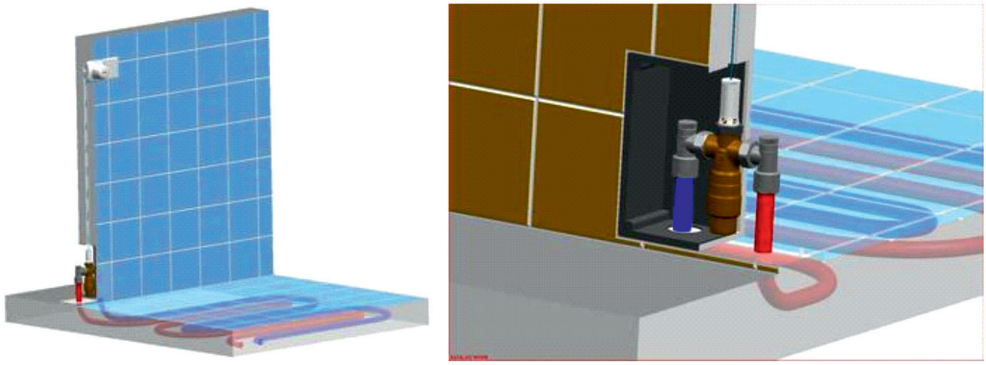


Fig. 4.40. In-floor heating control system with double temperature adjustment [29]; a) control system components, b) Floor Fix unit connection detail

The feed part of the radiant heater is connected directly to the central heating system's distributor. The return part of the radiant heater is connected to the Floor Fix feed part, the Floor Fix return part is connected to the central heating system's collector directly or via a second radiant heating loop. The Floor Fix control unit is capable of limiting the temperature of the heating medium flowing into the second radiant heating loop through the built-in return limiter. This is full protection against overheating of the second radiant heating loop and partial protection of the first loop. Due to the lack of a mixing system, the temperature of the first heater loop's "first meters" is similar to that of the central heating system feed, therefore it is recommended to use low-temperature heat sources. With the second thermostatic insert and the thermostatic head in Floor Fix, the heated room air temperature can be controlled. The temperature measurement and adjustment location is determined by the thermostatic head knob location. It is noteworthy that Floor Fix itself does not have to be deployed in the heated room, which is significant if the radiant heating loop is in a high standard bathroom. Moreover, the air temperature measurement and control are easily accessible, e.g. at ca. 1.2 m above the floor. The Floor Fix box is 55 mm deep and can be installed in a plasterboard partition. With the special solution in the double temperature adjustment device, a professional floor heating system can be installed in a small room without expanding the system with traditional radiators.

The current level of installation technology in Poland, and the high requirements for heating comfort set a high bar for system designers and installers. A designer must thoroughly identify the building's specifics, and the future user's requirements at the concept stage. When designing, they must optimise solutions in terms of energy consumption, and installation and operation costs. During project implementation, the installer must pay attention to compliance with the design and functionality, the system's installation diligence, and operational stability. These conditions are necessary, but not sufficient, to fully achieve the assumed goal. Very important user satisfaction factors are the comfort of use and aesthetics of implementation. For this reason, the connection unit to ladder radiators and an in-floor heating loop deserves attention, as it enables the joint feed of a traditional ladder radiator and a floor heater (Fig. 4.41).

This unit is particularly recommended for bathrooms and showers, where, in addition to a traditional ladder or panel radiator, the warm floor effect is desirable, while maintaining high aesthetics of the installation. The connection unit is a comprehensive solution in this respect. To a radiator system with traditional manifolds, a small in-floor heating loop and a ladder radiator can be connected. They can be connected to a single radiator outlet in the manifold without the need for a complicated and expensive system to reduce

Fig. 4.41. Connection unit to ladder radiator and floor heater [29]

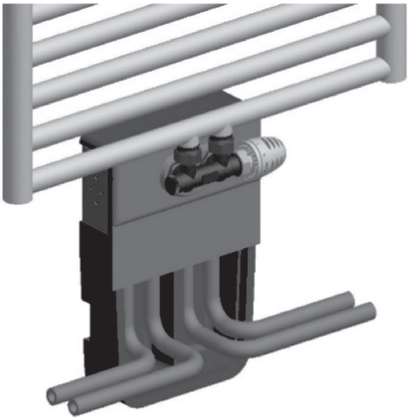


Fig. 4.42. Connection unit with ladder radiator [29]

the heating medium temperature (mixing system). The basic heat source in this system is the radiator, the output of which is adjusted by the thermostatic head.

The ladder radiator is connected to the connection unit through a typical radiator connection valve with a thermostatic insert (Fig. 4.42 and 4.43), on which the thermostatic head is installed. This solution is not limited to ladder radiators only. In the same way, a regular panel radiator without a thermostatic insert can be connected. The thermostatic insert for heated room temperature adjustment is built-in on the radiator connection valve.

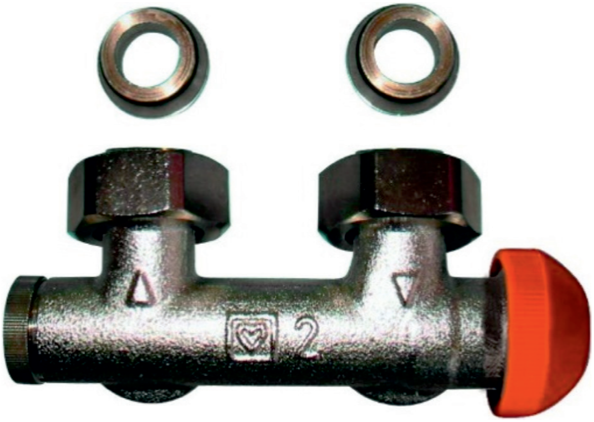


Fig. 4.43. Radiator connection valve with thermostatic insert [29]

The floor heater is connected to the connection unit (Fig. 4.44) by a typical thermostatic valve with visible TS 98V presetting on the return and return valve RL1 on the feed. This connection method is related to the necessity to install a return temperature limiter on the thermostatic valve on the return from the floor heater loop.

A floor heater is designed first of all to provide a feeling of comfort associated with a warm floor in rooms where we stay in clothing or to eliminate the feeling of discomfort in rooms where we touch the floor with a bare, often wet feet. A floor heater also supports the primary heat source. The traditional radiator output is adjusted to the room temperature and the temperature set on the thermostatic head knob. The in-floor heater is adjusted to the temperature of the medium returning from the in-floor heating loop and to the value set on the knob of the return temperature limiter installed on the thermostatic insert of the thermostatic valve integrated with

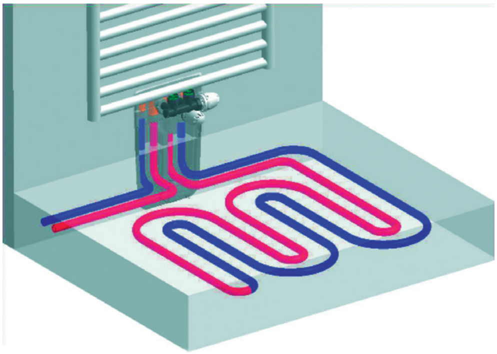


Fig. 4.44. Connection unit with floor heater [29]

the connection unit. In either case, hydraulic regulation is performed by means of pre-settings of the thermostatic valve inserts. The in-floor heater and traditional radiator work in parallel in this system. A similar solution is known to the installation engineering that consists of a thermostatic valve and a thermostatic head on a ladder radiator feed, and a temperature limiter on the ladder radiator return – the so-called series connection (to protect the in-floor heater from overheating). However, there is a fundamental difference in the operation of these systems. The series connection of all system components significantly increases the circuit's hydraulic resistance, and mutual interference of the thermostatic valve and return temperature limiter. With an oversized ladder radiator, the low temperature of the returning medium is insufficient to obtain the warm floor effect. With an undersized ladder radiator, the high return temperature closes the return limiter, thus reducing the flow, and the room's thermal comfort. If the two heaters are connected in parallel, there are no disruptions to their operation – the heaters are complementary. If the in-floor radiator overheats the heated room, then the ladder radiator output is automatically reduced by activating the thermostatic head in response to the room air temperature. When the main share of heating is provided by the ladder radiator, the in-floor heater is not determined by maintaining a high return temperature, and thus a too-high floor surface temperature. This system's only limitation is related to the heat source. Namely, the connection sets should not be used for solid fuel boilers or sources with high feed temperatures (because of the in-floor radiator).

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eISBN 978-83-66531-21-5

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of Technology