# Energy Efficient Heating and Ventilation of Large Halls

REHVA GUIDEBOOK

Karel Kabele (Editor)
Ondřej Hojer
Karel Kabele
Miroslav Kotrbatý
Klaus Sommer
Dušan Petráš

**REHVA**
Federation of European Heating, Ventilation and Air-conditioning Associations

GUIDEBOOK NO 15

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ISBN 978-2-930521-06-0

Printed in Finland, Forssa Print 2011

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## Working Group

This book is developed with a working group consisting of the following experts:

- Ondřej Hojer (Czech Technical University in Prague, Faculty of Mechanical Engineering, Czech Republic)
- Karel Kabele, (Czech Technical University in Prague, Faculty of Civil Engineering, Czech Republic)
- Miroslav Kotrbatý (Kotrbaty Ltd. Czech Republic)
- Klaus Sommer (Cologne University of Applied Sciences, Germany)
- Dušan Petráš (Slovak Technical University in Bratislava, Faculty of Civil Engineering, Slovakia)


## Acknowledgements

The authors wish to thank Richard D.Watson, Vincenc Butala, Daniela Koudelková, Zuzana Veverková (Krtková) for their contributions and valuable comments. The authors also wish to thank Mr. Jarkko Narvanne for the final layout and typesetting of the Guidebook.

The publication was prepared with financial support from the Czech State program to promote energy savings for 2010 - Part A - Program EFEKT.

# Foreword

Heating and ventilation of large halls is domain, where we meet with high performance and large heating systems. In terms of energy consumption are among the most energy consuming and reflecting to the global downward pressure on the energy performance of buildings, we must not forget this category of buildings. In the context with the development of knowledge in the theory of the internal environment of buildings, technological advances in the field of the various elements of the heating system technical solutions, control systems, methods of mathematical simulation and experience with implementations in recent years has been published a series of European standards EN.

This Guidebook was created as a tool for practitioners, planners and educators within the REHVA Task Force - Heating and focuses on the design and implementation of large halls heating systems, mostly industrial. This publication preceded by a series of seminars organized on the subject, which culminated in a workshop at the congress Clima 2010 in Antalya. Guidebook was written by a team of authors from the Czech Republic, Germany and Slovakia.

Chapters 1 and 2 written by Professor Karel Kabele from the Faculty of Civil Engineering Czech Technical University in Prague, are focused on specific issues of indoor environment and thermal comfort in the large-space buildings.

Chapters 3 and 4 are written by Engineer Miroslav Kotrbatý, a leading expert practitioner from the Czech Republic. Chapters address issues of conceptual design principles of large-space buildings and realization of radiant heating strips. In this part there is published completely original zone design method for radiant strips, whose use leads to significant savings on investments together with improvement of the indoor environment quality in such a heated halls.

Chapters 5 and 6 by Dr. Ondřej Hojer from the Faculty of Mechanical Engineering Czech Technical University in Prague, Czech Republic are focused on the design and implementation of gas infrared heaters and warm air heating. In addition to its own design addressing also the issues associated with the das installation and flue gas exhaust.

The Chapter 7 dealing with the issue under floor heating was written by Professor Klaus Sommer, of Cologne University of Applied Sciences / Germany (Chapter 7.1 to 7.4) and Professor Dusan Petráš of Slovak Technical University in Bratislava (Chapter 7.5). This section provides a summary of design methods and calculations under floor heating in accordance with the European standards.

The Guidebook summarizes the latest scientific and practical knowledge in heating large Hals and we believe it will benefit not only for narrowly focused professionals but for all readers interested in issues of heating and reducing energy consumption of buildings.

**Karel Kabele**
Professor, Czech Technical University
Prague, Czech Republic

# 1 INTRODUCTION

The problem of heating large-space buildings (production, storage or workshop) with a clear height of 4-10 m and more is in the design of such systems which can deliver the required output to the places where it is necessary to ensure thermal comfort – mostly in the lower part of the hall in a layer of 2 m where people are present.

The guidebook is focused on modern methods for design, control and operation of energy efficient heating systems in large spaces and industrial halls. The chapters answer questions related to thermal comfort, light and dark gas radiant heaters, panel radiant heating, and industrial air heating systems. Principles are supported with case studies and preview of design and modelling tools.

The traditional system used for heating hall building is a hot-air heating. All heat loss of the building is covered with circulating heated air. As a result of rising warm air up there, however, in applying these systems to the temperature difference between the heights of the room, so to achieve the desired temperature in the working zone, the temperature in the space below the roof substantially higher, thereby increasing the heat losses in this part of the building. The second characteristic is the absence or opposite orientation radiant component of heat transfer between man and environment, which especially in poorly insulated buildings with a low surface temperature leads to the need to increase the air temperature so as to achieve thermal comfort and thus high energy efficiency equipment.

The system, which partly eliminates this problem, is the heating infrared emitters, where, thanks to relatively strong radiant heat transfer component is sufficient to achieve thermal comfort (final temperature) lower air-temperatures, which reduces the heat loss of the building. Rectified infrared heaters emit infrared radiation, which is a small part (about 15%) absorbed by the air (depending on its purity) directly affects both the person moving irradiated area, and on the floor. Due to the large absorption surface of the human body and the floor most of the radiation absorbed, which in humans causes a feeling of heat and temperature increase in floor construction. Irradiated floor therefore has a higher temperature than the other walls and acts as a secondary heating source that heats air in the hall by the convection.

Heating with infrared heaters are particularly suitable for large-hall, workshops, garages, warehouses, depots, waiting rooms and lobbies, sports facilities etc.

Infrared radiant heating systems provide a reasonable and, possibly, energy-efficient alternative to convective heating systems in large enclosed spaces like the industrial halls or gyms. Ideally, the radiant heating system focuses thermal energy on the occupants and only indirectly heats the rest of the heated space.

It is very common to see building designs based on the air temperature as a basic reference parameter. This is not possible if we use infrared radiant heating systems, where we need to know operative temperature distribution in entire hall, which describes simultaneous effect of radiation and convective heat transfer modes. Operative temperature is parameter, which is usually used for evaluation of thermal comfort [Lit.13]. Several tools using for evaluation of radiant heating spaces are based on calculation of intensity of radiation. This simulation tools enables to make safe design, but not optimal. Optimal design could be based on model, which is able to describe the indoor environment quality and evaluate thermal comfort based on operative temperature. The problem is to balance detail of the mathematical model with possible accuracy of the inputs (description of the boundary conditions) and required accuracy of the outputs.

# 2 THERMAL COMFORT OF LARGE-SPACE BUILDINGS

Thermal comfort in the working environment in accordance with applicable health regulations is given to such temperature conditions of the workplace, where thermal equilibrium is reached without a man sweating at the optimal temperature distribution in space and time. Heat balance in the human environment depends mainly on ambient air and surface temperature of the surrounding walls. These two temperatures are in practice replaced by a single value - the final temperature (globeteplotou, spherical bulb temperature). In order to achieve thermal comfort is important and the ratio of heat shared between humans and the surrounding flow (convection - air flow around a man) and radiation (radiation between the surface of human and surrounding walls), which should be around a value of 1, which meets most of the water heating systems. The radiant heating may reach the surface temperature radiators up to 1 000°C and the ratio between the radiant and convective component can be a large imbalance. Therefore, for radiant heating systems are addressed not only thermal equilibrium, but the ratio between the components of heat flow. In practice this means that assessment of the area must not only know the final temperature of the spherical bulb as well as temperature and air velocity.

On the basis of physiological variables, reflecting the activity of man and his clothes, hygiene regulations are set desired range of values of physical, objectively reflecting the outcome of the heated area. Design heating system is therefore based on the requirements to achieve such physical values that most users are located in the area of thermal comfort.

## 2.1 Operative temperature requirements

In the halls without given distribution of the working places is advisable to design the system so that the entire heated area to achieve a uniform final temperature with the fact that, for practical reasons, permitted the operative temperature fluctuations in the surface hall is less than or equal to the requirements for uniformity of the operative temperature in the room which varies depending on the garment in the range of ±0.5 K to ±1.2 K. In other cases, the proposed heating so that the desired final temperature was reached people in places of residence. The value of the desired final temperature implied by the activities of people in the hall, air velocities and their clothing. Values of normal cases are presented in Table 2.1.

## 2.2 Air temperature requirements

To ensure optimum working environment are prescribed threshold temperature difference resulting tan and the air temperature depending on the anticipated activities of people. $T_{g,\,stereo}$ is the final temperature in the rooms with a unilateral source of radiant flux, measured in half-space.

***Table 2.1*** *Minimal and comfort operative temperature for different activities.*

| Human activities, room | Minimal operative temperature $\theta_o$, °C | Comfort operative temperature $\theta_o$, °C |
|---|---|---|
| Man at rest | 19 | 22 to 24 |
| Very light physical work (such as seamstresses, hand typesetters, drawing, crane operators, etc.) | 18 | 20 to 22 |
| Light physical work (such as toolmakers, mechanics, machinists, welders, ironing, kitchen staff, etc.) | 16 | 18 to 20 |
| Moderate physical labor (such as blacksmiths, Rolling Mills, founders, a greater number of machine tools, etc.) | 14 | 16 to 18 |
| Heavy physical work (such as carpenters, diggers, working with a shovel, etc.) | 10 to 12 | 14 to 16 |
| Living rooms, offices | 19 | 22 to 24 |
| Classrooms, dining room | 18 | 20 to 22 |
| Muster room | 16 to 18 | 18 to 22 |
| Gyms, sports halls - playing area<br>auditorium | 10 to 12<br>15 to 16 | 14 to 16<br>18 to 20 |

***Table 2.2*** *Thresholds for calculating the temperature optimum in terms of human thermal stress in the workplace.*

| Activity | $\theta_a - \theta_o$ [°C] | $\theta_a - \theta_{o,stereo}$ [°C] |
|---|---|---|
| Seated calmly in | 0.2 | 1.4 |
| Light work while sitting | 0.5 | 3.0 |
| Hard work | 1.1 | 7.0 |

# 3 CONCEPTUAL DESIGN PRINCIPLES

## 3.1 Introduction

Heating of industrial halls and other large space buildings is significantly different than heating of smaller spaces. The creation of thermal comfort in such spaces must come from complex view of the problem. It is not just demanded microclimate conditions in the occupancy area, but also the influence of their creation on the energy performance of the building. Technical solution should be also influenced by operation of whole system, including the heat source and distribution system. Considerable influence has also durability of the system as well as further service and maintenance demands.

Thanks to the large dimensions of considered spaces and applied physical laws there are created both horizontal and vertical zones over the whole space that desire individual technical solution of the heating system. Current strategies of heating load calculation and placement of heating devices are based on an assumption that the whole space is in general nothing more than one room and so it can be considered the same way as smaller spaces. However this assumption is wrong. Workplaces situated near external walls or places with different demands on design temperature are the main factors influencing the resulting technical solution of whole heating system. Exactly these factors create within the considered space zones with different demands on heat delivery. With radiant heating systems it is possible to deliver into these zones in particular time different amount of energy without separating these zone with solid curtains.

Uniform location of heating devices, especially in case of radiant heating systems, can cause in some parts of the building either over- or insufficient heating. In order to avoid this problem it is reasonable to divide the whole space into zones with different demands on the heat delivery (time and amount).

In the next step each zone should be considered individually, from the evaluation of heating load over the choice of the heating system (when necessary), up to calculating of required power output of a device in the zone. The possibility of supplying certain amount of energy to every zone (possibility to localize the delivery) is the biggest benefit of radiant heating systems. The height of large space buildings allows creating of vertical zones, layout dimensions then creating of horizontal.

## 3.2 Heating delivery principles

For large space heating are in general used two principles, convective and radiant heat delivery. In case of convective heating system the final devices are wall-mounted air-handling units or large air-handling units with complex ducting system with diffusers dampers, filters and silencers. The second principle is radiant heating systems which are represented in Europe mainly by water radiant panels and gas-burning suspend able infrared heaters.

### Convective heating with heat transfer medium - water

The majority of air-handling units is still working on the principle of constant mass air flow over the heat exchanger with control of the primary side of the circuit (either changing temperature in water systems or starting the burner in gas systems). The results of such operation are shown on the Figure 3.1.

As the external temperature decreases, the heat losses increase and also do the water inlet temperature. Hence the outflow air temperature increases as well. The air flow pattern is changing. Two specific cases were chosen for clarity:

a) external (outdoor) temperature range $\theta_e$ = +12ºC ... –3ºC Left side of Figure 3.1

b) external temperature range $\theta_e$ = –3ºC…–15ºC Right side of Figure 3.1

***Operation case a)***

At this case air-handling units are delivering the air of lower temperature – related to the actual need for heat delivery. The flow pattern shows supply into occupancy zone. The airflow is slowly turning upwards and losing its velocity (kinetic energy). The air temperatures are shown in Figure 3.1 both numerically and in color scale.

***Operation case b)***

Increased demand on the heat delivery is compensated by increased water temperature from the heat source. As the water inlet temperature increases the outflow air temperature increases as well. Bigger difference between supplied and ambient air causes immediate change in flow direction and the whole heated amount is rising directly to the roof level. There takes place an absurd situation, the air temperature at roof level increases and so do heat losses, however in occupancy zone the air temperature decreases. Orientation values are shown on the right hand side of the Figure 3.1.

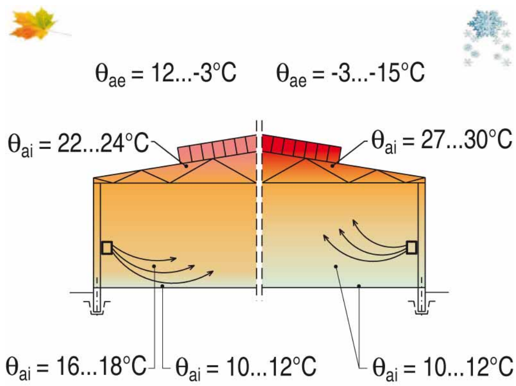



***Figure 3.1*** *Temperature conditions in a hall in case of convective heating; Heat transfer medium: water; spring / autumn on left, winter on right; fluent control.*

### Convective heating with heat transfer medium - gas

Gas-burning air-handling units are mostly controlled just “on - off”. It means that during operation they supply 100% power output until a higher thermostat set point in a control loop is reached. Then units in the control loop are switched off and wait until the air temperature in heated space falls again below lower thermostat set point. Flow pattern and related temperatures are shown in Figure 3.2. On the left hand side the installed unit is off – power output 0%, on the right hand side full operation – power output 100%.

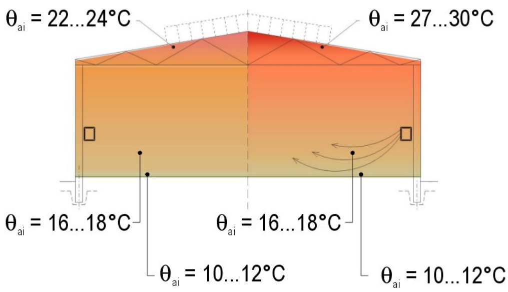



*__Figure 3.2__ Temperature conditions in a hall with convective heating. Heat transfer medium: gas; units operation: 100% power output – right side 0% power output – left side.*

This operation regime principle and air outflow temperatures is a bit better that the case with water-based air-handling units because the burner makes the unit completely separate device which gives the control more possibilities, operate the unit without any relations to the others.

### Radiant heating

Typical devices delivering most of the heat to the ambient by radiation are water/steam radiant strips or gas-burning or electrical infrared heaters (plaque (luminous), tube). It is completely different principle of heat delivery. Incident radiant energy heats up the floor, increases its temperature and just then the heat is transferred to the air by convection. This principle changes temperature conditions in whole heated space. On the Figure 3.3 there are shown both numerically and graphically air temperatures in a cross section of a hall. On the left there are spring / autumn temperature conditions when the outside temperatures varies from +12°C to approx. –3°C, on the right the same for outside temperatures approx. –3°C to –15°C.

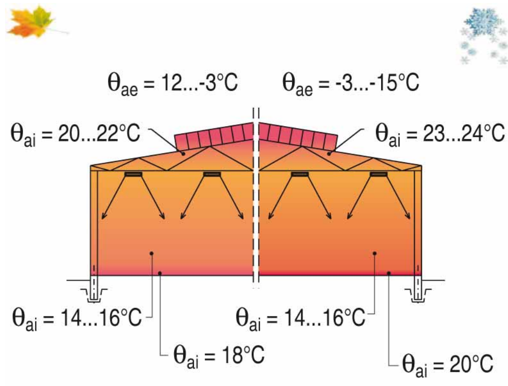



*__Figure 3.3__ Temperature conditions in a hall in case of radiant heating. Heat transfer medium: water; suspend able radiant strips. Heat transfer medium: gas; infrared heaters.*

## 3.3 Zonal approach

In order to explain the influence of external conditions on the thermal comfort within heated space, a manufacture hall was chosen as an example. The hall is 60 m long, made of three bays of 18 m in width. For an analysis and determination of particular zones, radiant heating with suspend-able strips was chosen. This heating system along with infrared heaters belongs to the most economical systems for industrial and large space heating. Besides, they both allow the localization of heat delivery to the thermal zones exactly according to the needs.

### Vertical zoning

When considering a cross section of a large space building (heights higher than 6 m), it can be divided into three vertical zones (Figure 3.4 – A, B, C). Each zone has specific influence on overall energy performance of the building.

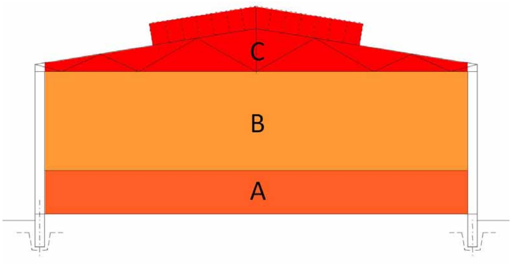



***Figure 3.4** Vertical zoning of a hall building. A – occupancy zone, b - neutral zone, c – energy performance zone.*

***Occupancy zone (Figure 3.4 - A)***

Goal of the design is creation of uniform thermal comfort conditions in occupancy zone. Height of this zone is about 2 m. Influencing factors are air temperatures and mean radiant temperature.

***Neutral zone (Figure 3.4 - B)***

This zone and air temperatures within does not influence directly the thermal comfort in zone A, however vertical temperature gradient influences temperature below the roof where it could increases heat losses via roof and skylights. Vertical temperature gradient varies with used heating system:

$g$ = 0.9…1.1 K/m
convective heating (local units assembled on the walls);

$g$ = 0.3…0.5 K/m
radiant heating

The goal of the design should be maximal utilization of thermal capacity of this zone.

***Energy performance zone (Figure 3.4 - C)***

Very important are areas below roof deck. These directly influence thermal performance (heat losses) of whole building. In accordance with physical principles, in this zone are the highest air temperatures. Therefore it is necessary to put right on this zone the biggest attention. Heating systems creating the lowest air temperatures at roof deck should be preferred. On the other hand, some of the heat from this zone can be reutilized by ventilation for pre-heating of fresh outside air.

### Horizontal zoning

Because of the radiant heat delivery principle, radiant strips are able to deliver heat to the building exactly according to the requirements; the crucial is therefore design stage. Various workplaces are influenced by ambient conditions differently. It is obvious that in a hall of layout dimensions e.g. 100 x 100 meters can poor windows or low insulated walls influence thermal comfort of a person standing in the middle of the hall just marginally. The same can be stated for heat sources. The requirements on a heating system are variable. Significant difference can be observed between the envelope surfaces and inner parts (Figure 3.5).

There are many possible sources causing discomfort. At external walls and windows the ambient air is cooled down and decreases slowly to the floor creating cold flow (1). Increased influence on the flow pattern at the external wall has infiltration (2). These two flows together significantly influence air temperatures in this area. They create so called cold peak (1 + 2 = 4). Significant influence has also "cold" radiation (3), respectively body's heat losses by radiation towards cold external environment and wall.

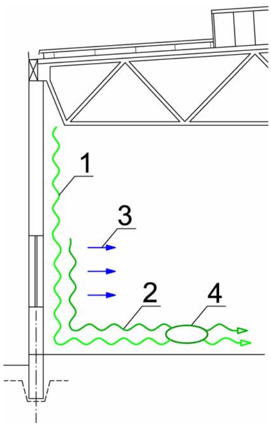



***Figure 3.5*** *Influence of the building envelope on microclimatic conditions in marginal parts of a building. 1 – cold convective flow; 2 – cold infiltration flow; 3 – "cold" radiation; 4 – peak of convective flow.*

### *View factor*

Uniform location of heating surfaces within considered space at uniform temperature does not itself ensure uniform microclimatic conditions over the occupancy zone. Actual radiant heat delivery can be obtained by evaluating of the view factor $\varphi$ (Figure 3.6, 3.7 and 3.8). The approximate value for particular points at height of 1.5 meters above the floor can calculated from following equation:

$$\varphi = \frac{\frac{\omega_{\max}}{360}}{\frac{\omega_{\min}}{360}} = \frac{\omega_{\max}}{\omega_{\min}} \tag{3.1}$$

View factor for the middle point is:

$$\varphi_m = \frac{\omega_{\max}}{360} \tag{3.2}$$

View factor for point at external wall is:

$$\varphi_e = \frac{\omega_{\min}}{360} \tag{3.3}$$

where:

$$tg\left(\frac{\omega_{\min}}{360}\right) = \frac{B}{2(h\text{-}1{,}5)} \tag{3.4}$$

$$tg(\omega_{\min}) = \frac{B}{h\text{-}1{,}5} \tag{3.5}$$

$B$ = width of the building, m;

$h$ = height of radiant strips' suspension, m.

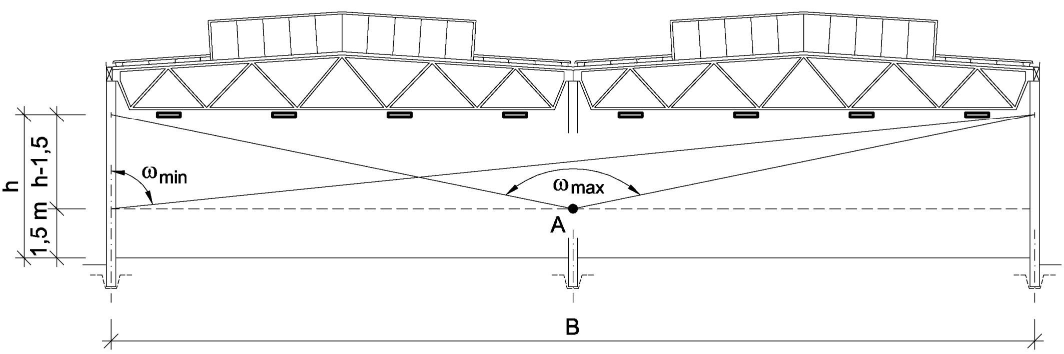



***Figure 3.6*** *Input values for determination of view factor (φ).*

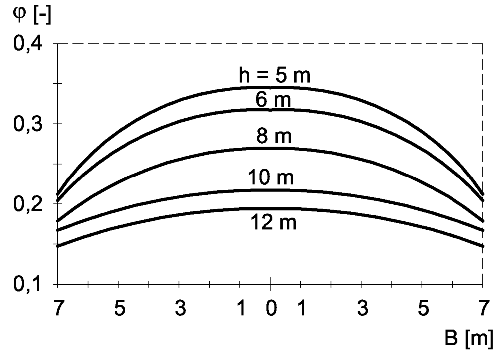



***Figure 3.7** Relation between view factor (φ) and height of suspension h – narrow halls.*

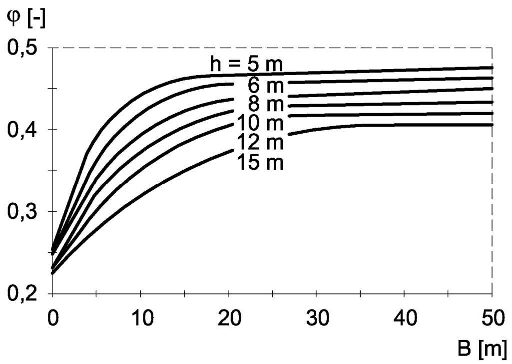



***Figure 3.8** Relation between view factor (φ) and height of suspension h – wide halls.*

## Energy performance zone

The goal of radiant heating system is to provide adequate heat delivery into each thermal zone. First step when solving the project is the differentiation of whole space into thermal zones. Dimensions of zones can be best determined by using view factors.

***a) Width of the zone***

As an example was used three-bay hall with dimensions 60 m x 54 m (Figure 3.13).

First criterion: maximal distance between two radiant strips $l < h$ (applying this rule, 4 strips are resulting for heights of suspension 5.4 m; 6.0 m a 7.2 m. For height of suspension $h$ = 3.5 m, 6 strips are necessary.

Second criterion: even number of radiant strips. This criterion allow on one hand significant reduction of connecting tubes on the other hand providing uniform delivery of heat both in lengthwise and crosswise directions.

From the view factor curve it appears ideal to install in one bay four strips of 57 m long. The inflexion point on the curve shows the differentiation of the bay into two thermal zones (marginal and inner), each with different design heat load. In the marginal zone plays role beside roof (skylights) and floor also external wall and windows including infiltration. In both halves two strips are placed.

The graphs offer possibility to differentiate the space into thermally uniform zones and then design heating surfaces adequately. Uniform comfort conditions can be achieved this way. On the Figure 3.9 view factors in both cross-sections are shown for heights of suspension 3.5; 5.4; 6.0; 7.2 m.

***b) Length of the zone (Figure 3.9)***

With regards to the figure possibilities, just one-bay hall was chosen for the example.

***First criterion**:*
*Inflexion point on the view factor curve*

***Second criterion**:*
*Possibility of changing panel width*

In the given example it is suitable to determine width of marginal area: $l_2$ = 1.5 m (distance between end of radiant strip and external wall) + 2 x 6 m (length of two panels) = 13.5 m. Number of panels included into marginal zone is being picked according to the same rules as in previous example in inflexion point of view factor curve.

Based on the determination of widths of marginal zones, whole building space will be divided into ten individual zones where the power output should be delivered (Table 3.1).

***c) Examples***

On the following figures (3.9…3.14) there are examples of application of zoning approach and determination of energy performance zones on a typical 1 up to 3-bay industrial building with various heights $H$ and hence heights of radiant strips' suspension $h$.

***Table 3.1*** *Ten individual zones where the power output should be delivered.*

| Zone | | |
|---|---|---|
| 1 and 9 | Corner | Influence of lengthwise and crosswise walls, floor, roof, one half of gates, windows and skylight; |
| 2 and 10 | Length-wise | Lengthwise wall, floor, roof, windows and skylight; |
| 3 and 7 | Front | Front wall, one half of gates, floor, roof and windows; |
| 4 and 8 | Length-wise Inner | Lengthwise inner, roof and skylight; |
| 5 | Front inner | Front wall, floor, roof, windows and skylight; |
| 6 | Inner middle | Floor, roof and skylight. |

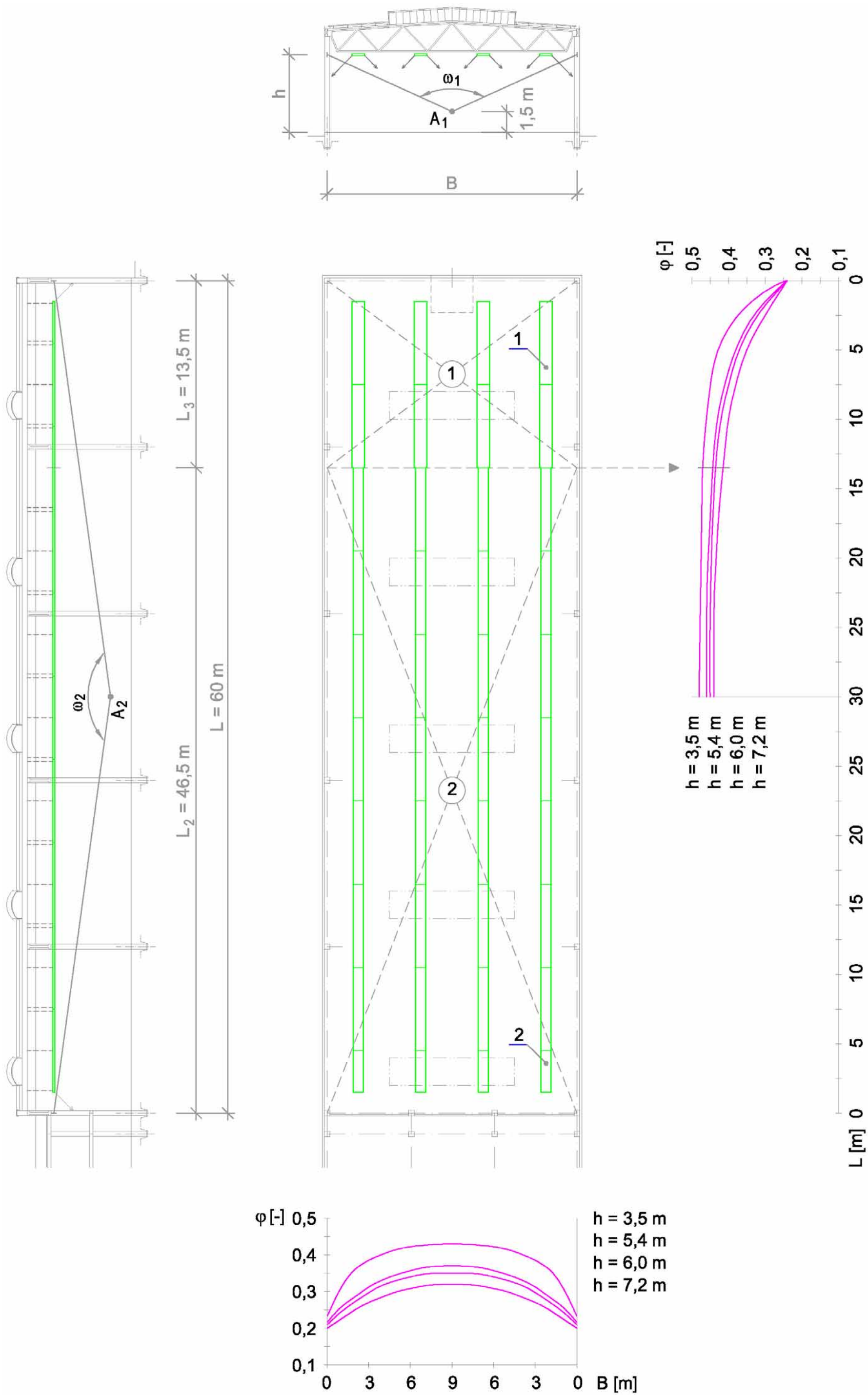



***Figure 3.9*** *Energy performance zones in 1-bay industrial hall 60 m x 18 m with annex; height of strips' suspension h = 5.4 m; 6.0 m; 7.2 m. At height of suspension h = 3.5 m, six strips/bay are necessary.*

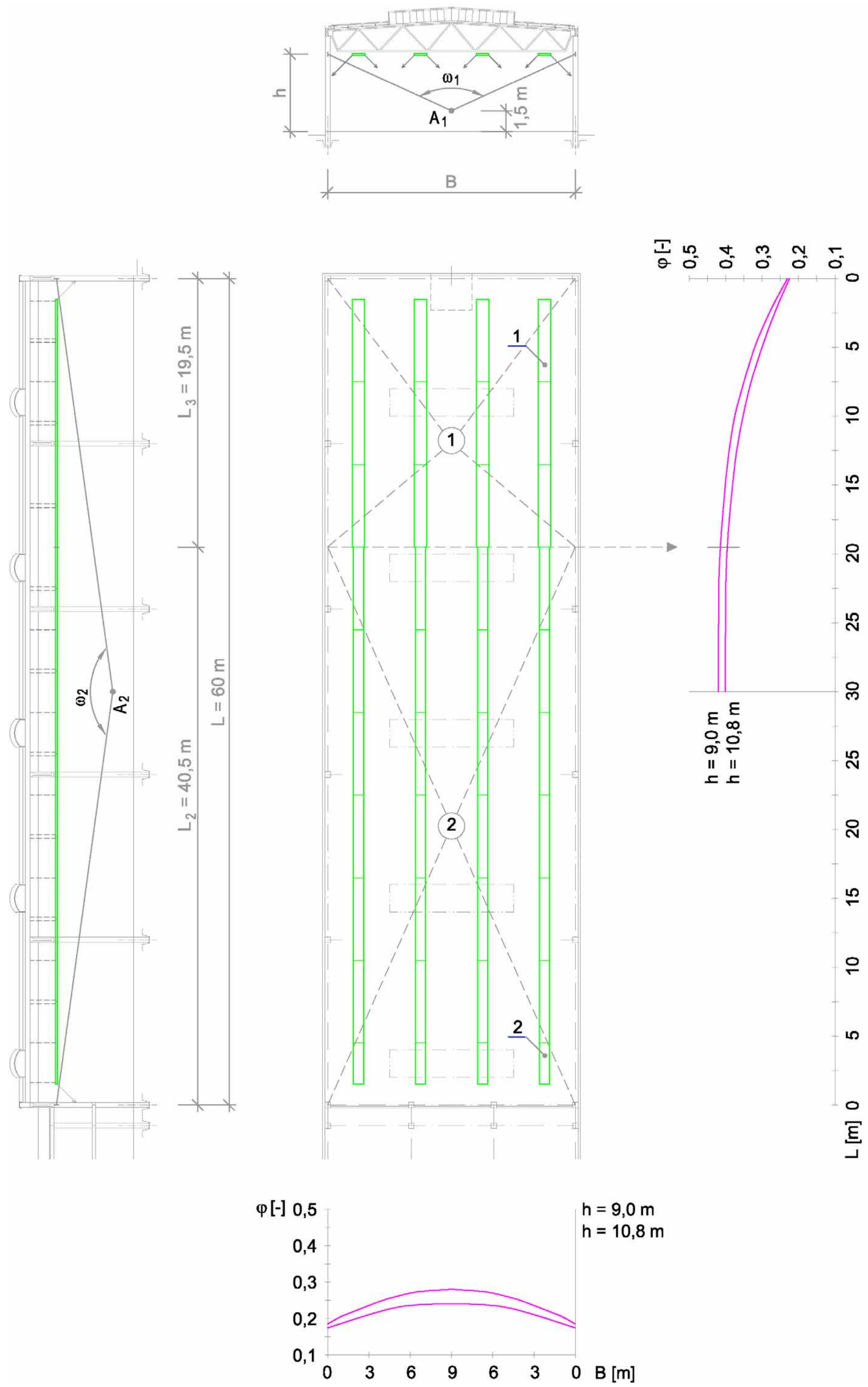



***Figure 3.10*** *Energy performance zones in 1-bay industrial hall 60 m x 18 m with annex; height of strips' suspension h = 9.0 m,resp.10.8 m*

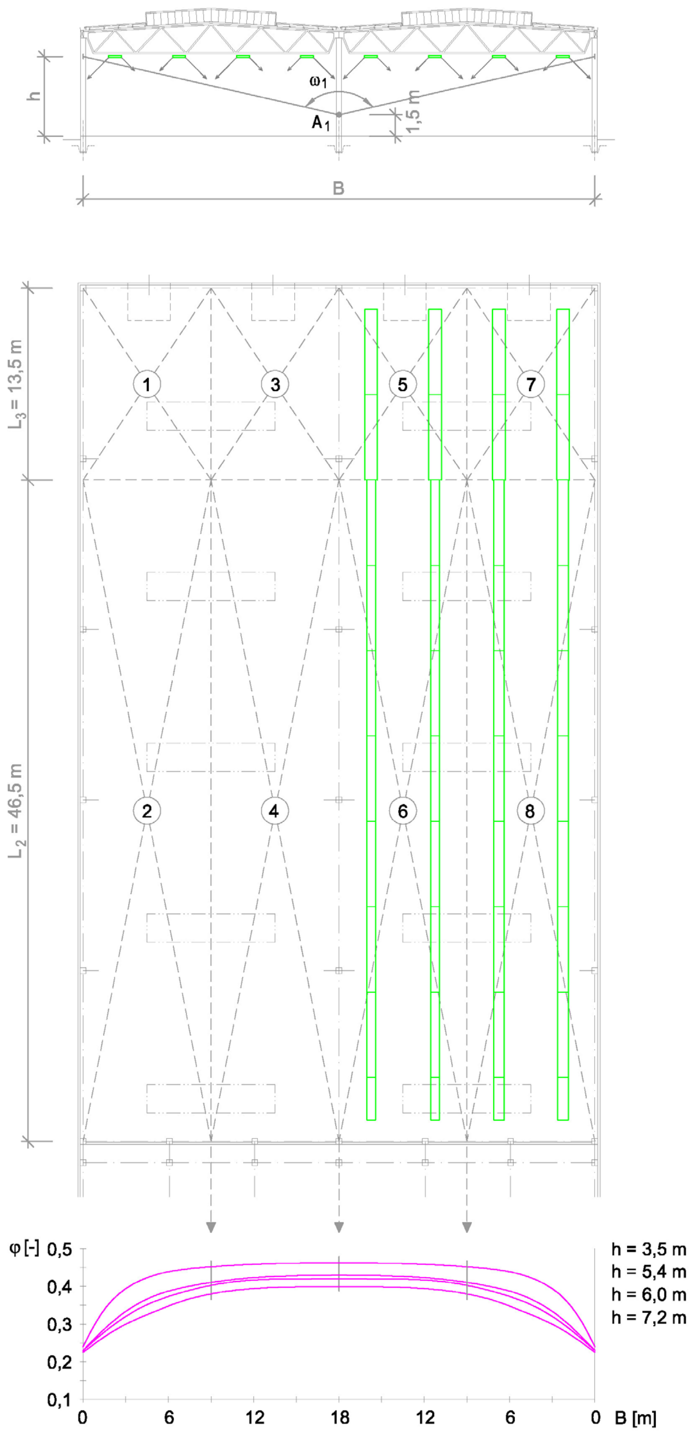



***Figure 3.11*** *Energy performance zones in 2-bay industrial hall 60 m x 36 m with annex; height of strips' suspension h = 5.4 m; 6.0 m; 7.2 m. At height of suspension h = 3.5 m, six strips/bay are necessary.*

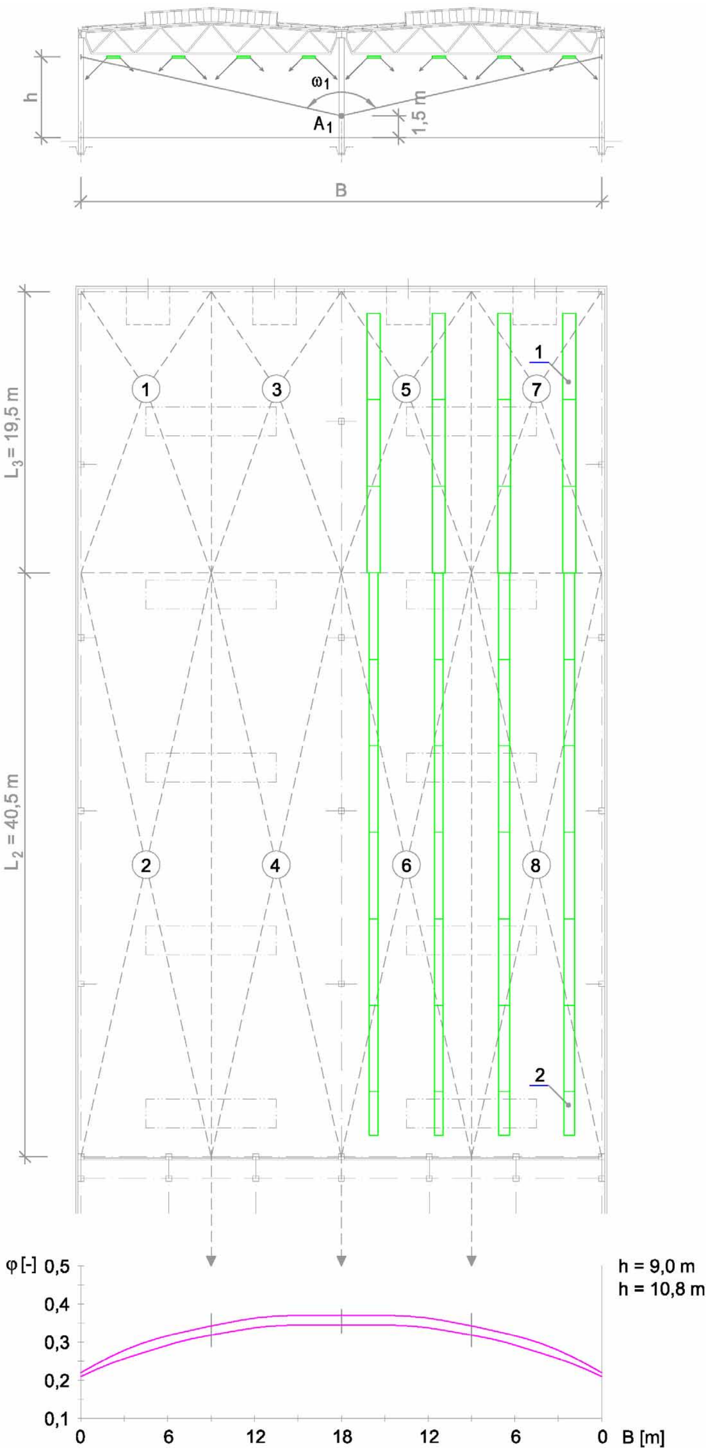



***Figure 3.12*** *Energy performance zones in 2-bay industrial hall 60 m x 36 m with annex; height of strips' suspension h= 9.0 m, resp.10.8 m*

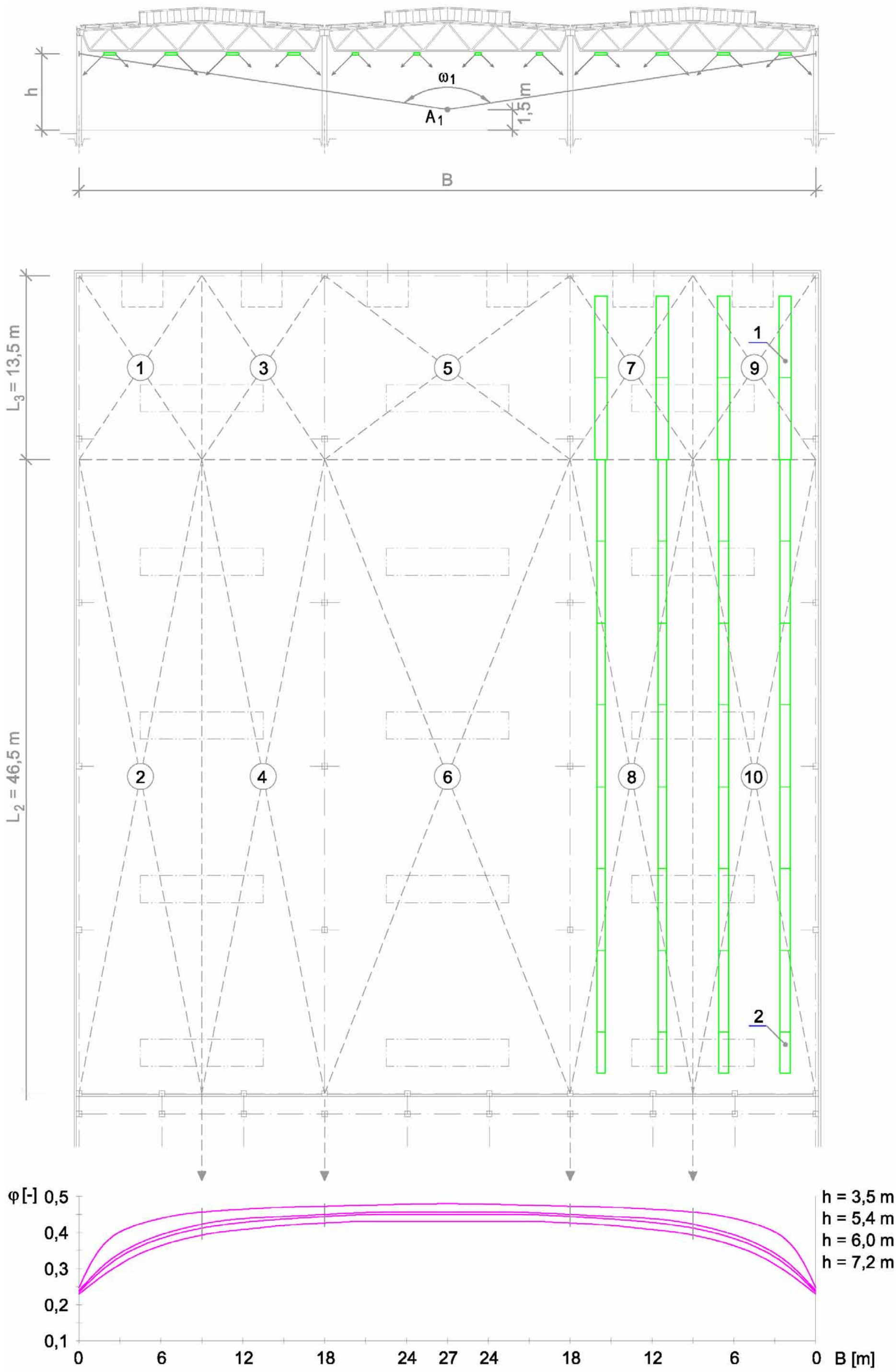



***Figure 3.13** Energy performance zones in 3-bay industrial hall 60 m x 54 m with annex, height of strips' suspension h= 5.4 m; 6.0 m; 7.2 m. At height of suspension h = 3.5 m, six strips / bay are necessary.*

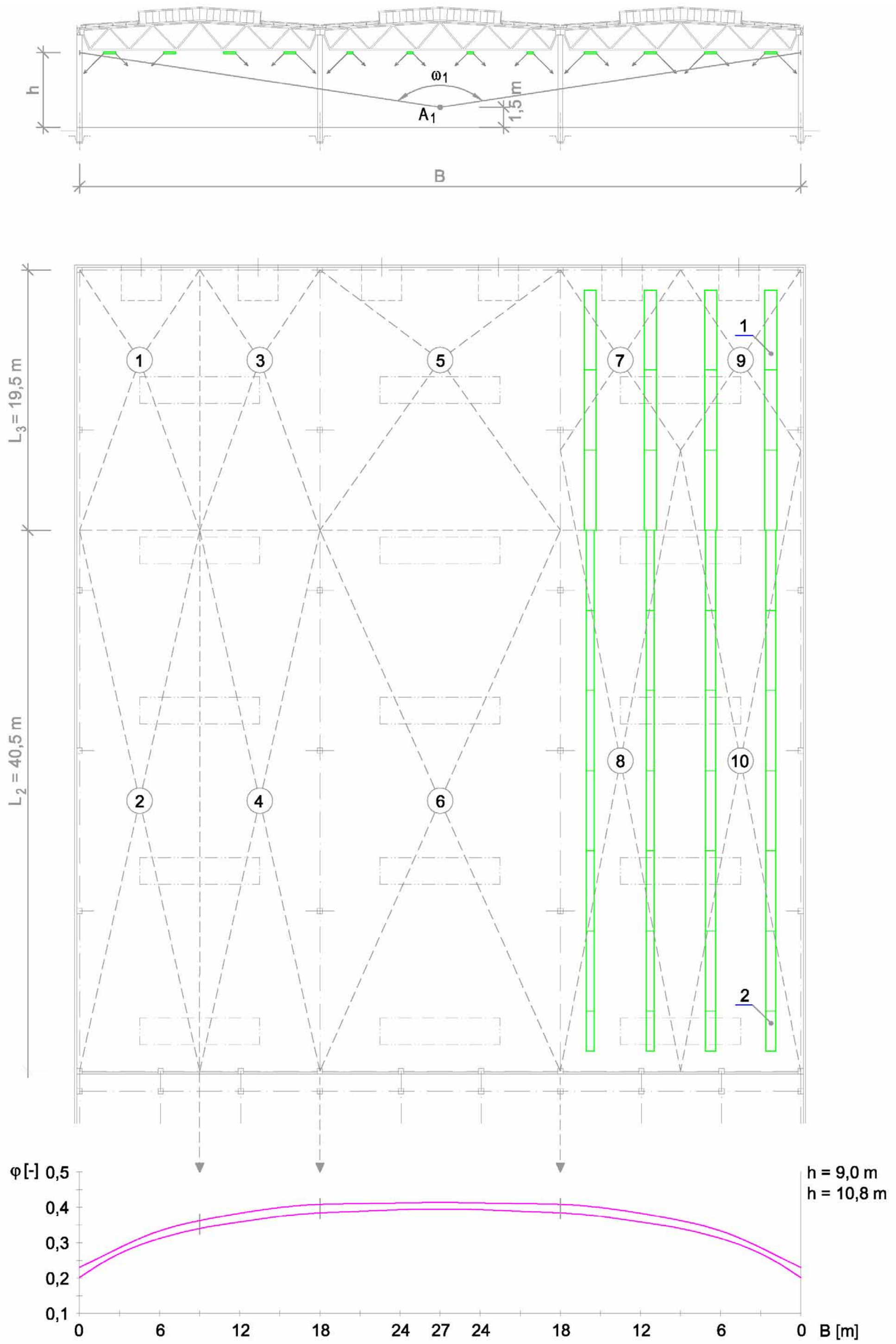



***Figure 3.14** Energy performance zones in 3-bay industrial hall 60 m x 54 m with annex; height of strips' suspension h= 9.0 m, resp.10.8 m.*

# 4 RADIANT STRIPS

Based on the differentiation of the heated space into ten thermal zones (Figure 3.13) with different requirements on heat delivery, it is necessary to divide also heating system into ten separated zones. If whole space heating with uniform requirements is desired, it is possible, with regards to simplicity, to control separately just three zones (along with bays) but design the heating surfaces in all ten zones separately and connect them into heating system according to control loops.

## 4.1 Power output evaluation

First, the initial location of radiant strips should be prepared to be able to determine thermal zones according to previous chapter. Then design heat load in each thermal zone must be calculated and this can be done by any locally used method (recommended EN 12 831). This is initial value before any other step of the procedure can be done.

### Power output correction of a heating system (zone)

When designing the heating with radiant strips it is necessary to take into account various factors that can increase requirements on the system's power output.

**Correction factor** $f_1$ – height of radiant strips' suspension (negative influence – dustiness of an environment)

Dustiness influences significantly radiation heat delivery into occupancy zone. Airborne dust particles within heated space absorb some part of radiation and reducing the heat amount incident on the floor. For standard operation without significant dust sources and height of suspension lower than 6 m, almost clean environment can be considered In other cases correction factor $f_1$ (Table 4.1) have to be used.

***Table 4.1*** *Correction factor $f_1$.*

| Minimal pollutants and dust | | Operation with higher dustiness | |
|---|---|---|---|
| ***h*** **[m]** | $f_1$ **[ - ]** | ***h*** **[m]** | $f_1$ **[ - ]** |
| 6 | 1.00 | 6 | 1.08 |
| 8 | 1.08 | 8 | 1.12 |
| 10 | 1.12 | 10 | 1.18 |
| 12 | 1.18 | 12 | 1.25 |
| 15 | 1.25 | 15 | 1.32 |

**Correction factor** $f_2$ - height of radiant strips' suspension (positive influence – lower height of suspension in higher building)

In case of it is possible (technology, crane installation, …) to suspend the radiant strips lower, required heating surface can be reduced. Correction factor $f_2$ depends on dimensions of heated space and height of suspension (Figure 4.1). Suitable factor can be from Table 4.2.

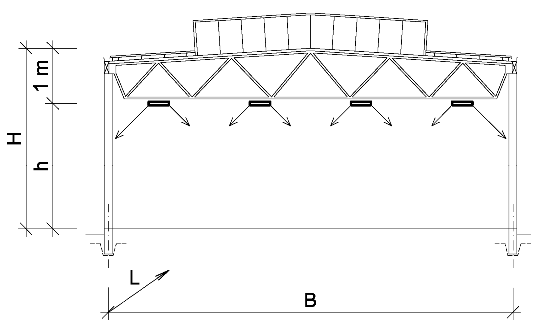



***Figure 4.1*** *Dimensions for evaluation of correction factor $f_2$.*

***Table 4.2*** *Correction factor $f_2$.*

| $\frac{h}{H-1}$ | L / B | | |
|---|---|---|---|
| | **2** | **2...5** | **5** |
| **1.00** | 1.000 | 1.000 | 1.000 |
| **0.95** | 0.967 | 0.981 | 0.989 |
| **0.90** | 0.935 | 0.963 | 0.979 |
| **0.85** | 0.904 | 0.944 | 0.969 |
| **0.80** | 0.874 | 0.927 | 0.959 |
| **0.75** | 0.845 | 0.910 | 0.949 |
| **0.70** | 0.817 | 0.839 | 0.939 |
| **0.65** | 0.790 | 0.877 | 0.930 |
| **0.60** | 0.764 | 0.861 | 0.920 |
| **0.55** | 0.739 | 0.845 | 0.911 |
| **0.50** | 0.715 | 0.830 | 0.902 |
| **0.45** | 0.692 | 0.816 | 0.893 |
| **0.40** | 0.670 | 0.802 | 0.884 |

***Example***

Building dimensions:
$L = 60$ m; $B = 18$ m; $H = 10$ m

Height of radiant strips' suspension
$h = 5$ m

$$\frac{h}{H-1} = \frac{5}{10-1} = 0{,}56 \qquad (4.1)$$

$$\frac{L}{B} = \frac{60}{18} = 3{,}33 \quad \text{i.e. range } 2\ldots5 \qquad (4.2)$$

From the Table 4.2 correction factor would be $f_2$ = 0.845. Therefore installed power output of radiant strips can be lower:

$$\phi_{HL,i} = \phi_{HL} \cdot f_2 = \phi_{HL} \cdot 0{,}845 \text{ W} \qquad (4.3)$$

Where $\phi_{HL}$ is the design heat load (evaluated e.g. according to EN 12 831) [W].

<u>**Correction factor**</u> $f_3$ - tilted suspension of radiant strips

Next negative influence has tilted suspension of radiant strips. With growth of suspension angle, the convective part increases and therefore lower part of power output is delivered to the occupancy zone (Figure 4.2). Therefore correction factor $f_3$ is introduced.

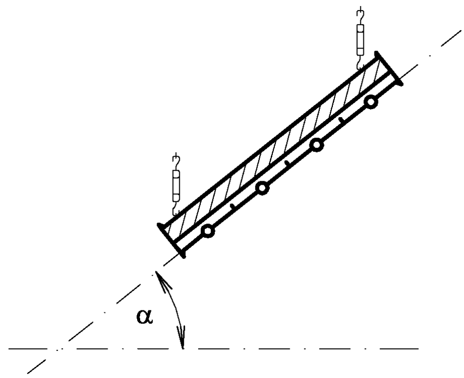



***Figure 4.2*** *Determination of suspension angle $\alpha$ when radiant panels is suspended tilted.*

***Table 4.3*** *Correction factor $f_3$.*

| Suspension angle α | Correction factor $f_3$ [ - ] |
|---|---|
| 30° | 1.10 |
| 45° | 1.15 |

***Evaluation of hygienic conditions***

<u>**Irradiation intensity**</u> $I_s$

To avoid unpleasant overheating, it is recommended from the praxis that the irradiation intensity at the top of the head shouldn't exceed 200 W/m². The value can be calculated theoretically from following equation:

$$I_s = \frac{\Phi_{HL,i} \cdot \eta_s}{A_1} \leq I_{s,norm} \qquad (4.4)$$

where:
$\Phi_{HL,i}$ = installed power output of radiant strips, W;
$\eta_s$ = radiant efficiency, - (Figure 4.7);
$A_1$ = floor surface area, m².

**Example**

$\Phi_{HL,i}$ = 190 000 W; $A_1$ = 1 080 m²;

$\theta_m = \frac{130+70}{2} = 100\,°C$; $\eta_s = 0.72$

$$I_s = \frac{\Phi_p \cdot \eta_s}{A_1} = \frac{190\,000 \cdot 0.72}{1080} =$$

126.7 W/m² < 200 W/m² → **Succeed.**

***Height of suspension check***

Following figure (Figure 4.3) can serve as a guide to find an optimum between height of suspension given by space possibilities (crane, technology, etc.) and by economical heat delivery. There is shown relation between mean water temperature $\theta_m$ and so called cover rate of floor surface with radiant strips:

$$A_0 = \frac{A_p}{A_1} \qquad (4.4a)$$

where:
$A_p$ = radiant strips surface, m²
$A_1$ = floor surface area, m².

**Example**

$A_p$ = 246 m²; $A_1$ = 1080 m²; $h$ = 6 m;

$$\theta_m = \frac{130+70}{2} = 100°C;$$

$$A_0 = \frac{A_p}{A_1} = \frac{246}{1080} = 0.227.$$

From Figure 4.3 for $A_0$ = 0.227 the maximal acceptable temperature can be $\theta_{m,max}$ = 105°C. → **Succeed.**

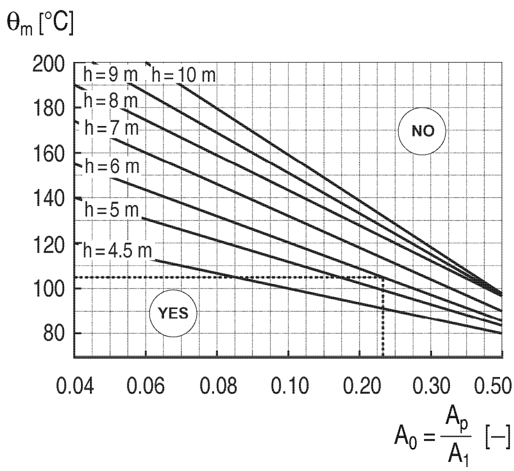



***Figure 4.3*** *Allowable suspension heights of radiant strips.*

## 4.2 Creation of heating surfaces

In order to obtain uniform heating all over the considered space, it is necessary to design suitable sizes of radiant panel into all zones according to requirements and simultaneously treat the heating surface as one unit. From the Figure 4.4 it is obvious that the whole building has in different places (thermal zones) different requirements on heat delivery. However also in such situation it is necessary to keep whole heating system as simple as possible at least for the control and maintenance. Hence, the system was divided in three sections along with bays within the hall. Each bay can be controlled separately.

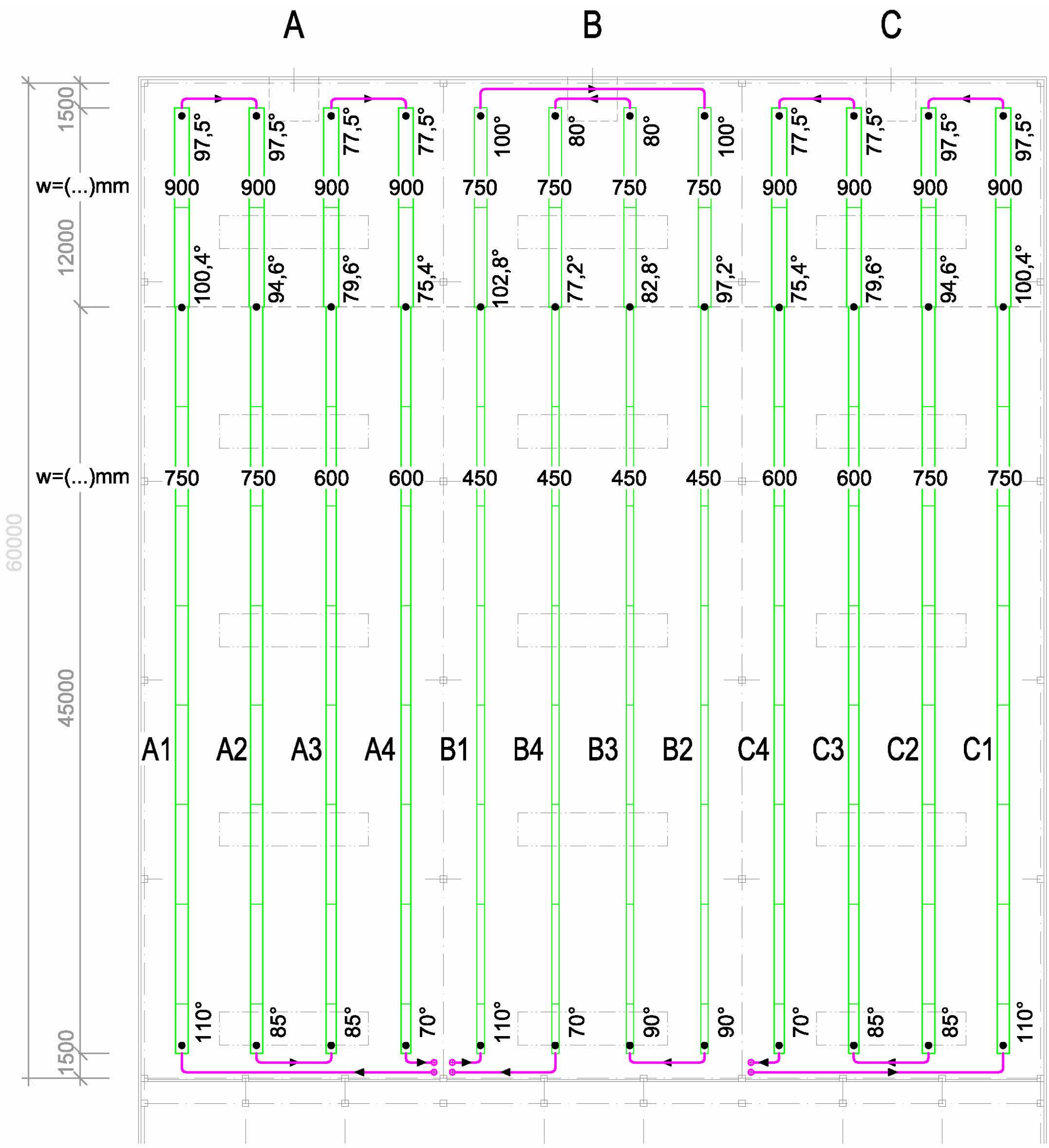



***Figure 4.4*** *Optimal location and connection of radiant strips.*

### Marginal bay – section A, resp. C

In order to achieve uniform conditions, it is essential to suspend the strips maximally in distances $l \leq h$ ($l$ – distance between strips, $h$ – height of suspension). Second condition is even number of strips – significant reduction of distribution pipes. At this specific case 4 strips are designed, 57 meters long. Distance between strips $l = 4.5$ m; at the external wall $l/2 = 2.25$ m. The connection should be done so to follow space requirements. The inlet water with the highest temperature should be connected first to the marginal strip A1. Then the connection is done gradually from the external to the inner part A2, A3 and finaly A4. Design heat load for both halves of the bay are: $Q_{1+2}$ = 73 620 W for marginal half and $Q_{3+4}$ = 56 340 W for inner half. Design water temperature difference 110/70°C. All required information can be found in Table 4.4.

Zone 2 can be determined the same way. The rest zones 3, 4 are connected to the water outgoing from zone 2: $\theta_2 = \theta_3$ = 85°C. Outflow temperature from the zone 4 is: $\theta_4$ = 70°C. The total temperature difference $\Delta\theta$ = 40 K is therefore utilized in full range and heat will be delivered to the required places according to the needs.

Radiant panels placed in zones:

| Zone | | w | Modules |
|---|---|---|---|
| 1 | 2 x 12 m | 900 mm | 6 |
| 2 | 2 x 45 m | 750 mm | 5 |
| 3 | 2 x 12 m | 900 mm | 6 |
| 4 | 2 x 45 m | 600 mm | 4 |

| | | |
|---|---|---|
| $Q_1$ | Design zone's heat load | = 17 280 W |
| $l_1$ | Length of strips in zone 1 | = 2 x 12 m = 24 m |
| $q_{\circ}$ | Required specific power output | = 17 280 : 24 = 720 W/m |
| $n$ | Utilization of heat transfer medium temperature difference | = 17 280 W / 118 220 W x 100 = 14.62% |
| $\Delta\theta_1$ | Utilized part of heat transfer medium temperature difference | = $\Delta\theta_1$ = ((110–70) x 14.62)/100 = 5.85 K |
| R | Heat-transfer medium temperature difference | = $\theta_1$ / $\theta_2$ = 100.4 / 94.6ºC |
| | | |
| Average temperature in the middle of the loop = (110 + 85) / 2 = 97.5°C | | |
| $\theta_1$ | = 97.5 + (5.85 / 2) = 100.4°C resp. | |
| $\theta_2$ | = 97.5 – (5.85 / 2) = 94.6°C | |
| $\Delta\theta_2$ | Temperature difference during operation (difference between average heat-transfer medium temperature and design temperature in heated space) | $\Delta\theta_2$ = ((110 + 85°C)/2) – 18°C = 79.5 K |
| $q_i$ | installed specific power output (manufacturer) | = 751 W/m |
| $w$ | width of the panel | = 900 mm |
| $Q_i$ | installed power output: $q_i = q_i \cdot l_1\ l_1$ = | = 751 x 24 = 18 024 W |

***Table 4.4*** *Marginal bay - heat balance + determination of radiant strips' size.*

| **z** [-] | **$Q_0$** [W] | ***l*** [m] | **$q_0$** [W/m] | **$n_1$** [%] | **$n_2$** [K] | **$\theta_1 / \theta_2$** [°C/°C] | **$\Delta\theta_2$** [K] | **$q_i$** [W/m] | **w** [mm] | **$Q_i$** [W] |
|---|---|---|---|---|---|---|---|---|---|---|
| 1 | 17 280 | 24 | 720 | 14.62 | 5.85 | - | 79.5 | 751 | 900 | 18 024 |
| 2 | 56 340 | 90 | 626 | 47.65 | 19.05 | - | 79.5 | 640 | 750 | 57 600 |
| 1 + 2 | 73 620 | - | - | 62.27 | 24.90 | 110 / 85 | 79.5 | - | - | 75 624 |
| 3 | 12 729 | 24 | 530 | 10.77 | 4.32 | - | 59.5 | 532 | 900 | 12 768 |
| 4 | 31 871 | 90 | 354 | 26.96 | 10.78 | - | 59.5 | 374 | 600 | 33 660 |
| 3 + 4 | 44 600 | - | - | 37.53 | 15.1 | 85 / 70 | 59.5 | - | - | 46 428 |
| 1…4 | 118 220 | - | - | 100 | 40 | 110 / 70 | - | - | - | 122 052 |

*Definition of terms in tables 4.4 and 4.5:*

| | |
|---|---|
| z [-] = zone | $\theta_1$ / $\theta_2$ [°C/°C] = heat transfer medium temperature difference |
| $Q_0$ [W] = design temperature | $\Delta\theta_2$ [K] = temperature difference during operation ($\theta_m$ - $\theta_0$) |
| *l* [m] = length of a strip | $q_i$ [W/m] = installed specific total power output |
| $q_0$ [W/m] = total specific power output | *w* [mm] = panel width |
| $n_1$ [%] = power output rate | $Q_i$ [W] = installed total power output |
| $n_2$ [K] = part of temperature difference | |

## Inner bay – Section B

On the contrary to marginal zones, there is no asymmetrical cooling. Therefore the connection is done so, to achieve the same average temperatures in both parts B1, B2, B3 and B4. Average temperature for both parts is $\theta_{av}$= 90°C. Required values are shown in Table 4.5.

Radiant panels placed in zones:

| **Zone** | | **w** | **Modules** |
|---|---|---|---|
| 5 | 4 x 12 m | 750 mm | 5 |
| 6 | 4 x 45 m | 750 mm | 3 |

***Table 4.5*** *Inner bay – heat balance + determination of radiant strips' size.*

| **z** [-] | **$Q_0$** [W] | ***l*** [m] | **$q_0$** [W/m] | **$n_1$** [%] | **$n_2$** [K] | **$\theta_1 / \theta_2$** [°C/°C] | **$\Delta\theta_2$** [K] | **$q_i$** [W/m] | **w** [mm] | **$Q_i$** [W] |
|---|---|---|---|---|---|---|---|---|---|---|
| 5 | 26 143 | 48 | 545 | 28354 | 11.42 | - | 72 | 571 | 750 | 27 408 |
| 6 | 65 457 | 180 | 364 | 71.46 | 28.58 | - | 72 | 369 | 450 | 66 420 |
| 5 + 6 | 91 600 | - | | | | 110 / 70 | | | | 93 828 |

## 4.3 Influence of surface temperature and width of a panel – additional design aspects

In order to create as economical system as possible every design detail should be carefully considered. Besides standard parts of design procedure (recommended by manufacturers) there are other aspects influencing significantly operation economy. From this point of view significant is relation between surface temperatures and widths of radiant panels. Currently manufactured radiant panels (active heating surfaces respectively) are made either of aluminum or steel. Majority of manufacturers use tube register distances either 150 or 100 mm (Figure 4.5). There on the top panels are equipped with usually 40 mm thick insulation equipped with aluminum foil in order to reduce heat losses. Panels are normally made in standard lengths 2…6 m, widths 300…1200 mm. Finally on site the radiant strips are made of radiant panels 6 m long connected at the end with shorter panels according to the requirements. Panels are connected either by welding or by fitting pressing.

For determination of heating surface size the designer should start with evaluation of design heat load required for each thermal zone. Then radiant panels are preliminary placed into the thermal zones and theoretical total length of the panel within the specific zone is obtained. Dividing the required design heat load by expected length, minimal required specific power output is calculated. Finally, from the manufacturer's nominal specific power output values, the optimal width for given temperature difference and design temperature is chosen, with respect to previous results.

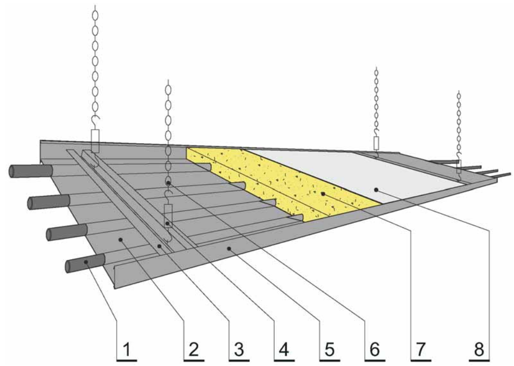



***Figure 4.5*** *Typical construction of suspendable radiant strip. 1 – distribution pipe; 2 – active heating surface (aluminum; steel); 3 – crossbar; 4 – tension bolt; 5 – side cover; 6 – nodal chain; 7 - insulation; 8 – aluminum foil.*

In case of manufacturers values are not at disposal, following relations can be used (power output values among manufacturers vary minimally):

$$q_o = 1.1 \cdot C \cdot \Delta\theta^n \tag{4.5}$$

where:

$$\Delta\theta^n = \frac{\theta_{W1} + \theta_{W2}}{2} - \theta_o \tag{4.6}$$

$\theta_{w1}$ [ºC] = heat transfer medium - inlet temperature;
$\theta_{w2}$ [ºC] = heat transfer medium - return temperature;
$\theta_o$ [ºC] = referent temperature.

Radiant heating however works on completely different principle than are designers familiar with (panel radiators, convective heating units). Location of heating surface in higher parts of heated building and construction of the panels allow differentiation of total power output between radiant and convective part. This must be taken into account when designing such heating system.

***Table 4.6a** Specific total power output $q_0$ of a radiant panels (DIN EN 14037-1-2-3) depending on water temperature difference and dimensions.*

| Δθ [K] | Specific total power output $q_0$ [W/m] $q_0 = K_1 \cdot \Delta\theta^n$ | | | | | | |
|---|---|---|---|---|---|---|---|
| | Panel widths *B* [mm] | | | | | | |
| | 300 | 450 | 600 | 750 | 900 | 1050 | 1200 |
| 30 | 93 | 131 | 166 | 201 | 237 | 272 | 307 |
| 32 | 101 | 141 | 180 | 218 | 256 | 294 | 332 |
| 35 | 112 | 157 | 200 | 242 | 284 | 327 | 369 |
| 38 | 123 | 173 | 220 | 267 | 313 | 360 | 407 |
| 40 | 131 | 184 | 234 | 284 | 333 | 383 | 432 |
| 42 | 139 | 195 | 248 | 301 | 353 | 406 | 458 |
| 43 | 143 | 201 | 255 | 309 | 363 | 417 | 471 |
| 45 | 150 | 212 | 270 | 326 | 383 | 440 | 497 |
| 47 | 158 | 223 | 284 | 344 | 404 | 464 | 523 |
| 48 | 162 | 228 | 291 | 352 | 414 | 475 | 537 |
| 50 | 170 | 240 | 306 | 370 | 434 | 499 | 563 |
| 52 | 178 | 251 | 320 | 388 | 455 | 523 | 590 |
| 53 | 182 | 257 | 328 | 397 | 466 | 535 | 604 |
| 55 | 190 | 268 | 342 | 414 | 487 | 559 | 631 |
| 57 | 199 | 280 | 357 | 432 | 508 | 583 | 658 |
| 58 | 203 | 286 | 365 | 441 | 518 | 595 | 672 |
| 60 | 211 | 297 | 380 | 460 | 540 | 620 | 700 |
| 62 | 219 | 309 | 395 | 478 | 561 | 644 | 727 |
| 63 | 223 | 315 | 402 | 487 | 572 | 657 | 741 |
| 65 | 232 | 327 | 418 | 506 | 594 | 682 | 769 |
| 68 | 244 | 345 | 441 | 533 | 626 | 719 | 812 |
| 70 | 253 | 357 | 456 | 552 | 648 | 744 | 840 |
| 75 | 274 | 387 | 495 | 599 | 704 | 808 | 912 |
| 80 | 296 | 418 | 535 | 647 | 760 | 872 | 985 |
| 85 | 318 | 449 | 575 | 696 | 817 | 937 | 1058 |
| 90 | 340 | 481 | 615 | 745 | 874 | 1003 | 1132 |
| 95 | 362 | 512 | 656 | 794 | 932 | 1070 | 1208 |
| 100 | 385 | 544 | 698 | 844 | 991 | 1137 | 1283 |
| 105 | 407 | 577 | 739 | 895 | 1050 | 1205 | 1360 |
| 110 | 430 | 609 | 781 | 946 | 1110 | 1274 | 1437 |
| 115 | 453 | 642 | 824 | 997 | 1170 | 1343 | 1515 |
| 120 | 477 | 676 | 867 | 1049 | 1231 | 1412 | 1594 |
| 125 | 500 | 709 | 910 | 1101 | 1292 | 1483 | 1673 |
| 130 | 524 | 743 | 953 | 1154 | 1354 | 1553 | 1753 |

***Table 4.6b***
*Panel widths B [mm].*

| *B* [mm] | $K_1$, [–] | n [–] |
|---|---|---|
| 300 | 1.7149 | 1.1754 |
| 400 | 2.3418 | 1.1832 |
| 600 | 2.8947 | 1.1910 |
| 750 | 3.5162 | 1.1902 |
| 900 | 4.1419 | 1.1894 |
| 1050 | 4.7715 | 1.1886 |
| 1200 | 5.4049 | 1.1878 |

Radiant part (Figure 4.6 - R) has significant influence on creation of suitable thermal conditions. On the other hand the convective part (Figure 4.6 - C) increases heat losses of the roof to the external environment.

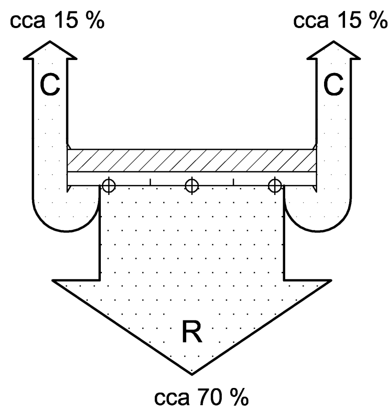



***Figure 4.6** Dividing of total power output of radiant panel in R (radiant) and C (convective) part.*

## 4.4 Analysis of strips' power output

Certified test laboratory provide every manufacturer with power output data for three widths (w = 300, 600, 1200 mm) in relation with referent temperature $\theta_{ref}$ [°C] and average temperature of heat transfer medium $\theta_m$ [°C]. Further data fill required information on heat delivery principle into the building: total specific power output $q_o$ [W/m]; radiant power output $q_r$ [W/m]; convective power output $q_k$ [W/m] and radiant efficiency (radiant rate) $\eta_s$ [%].

The particular values are shown for specific panel's widths in tables 4.7, 4.8, 4.9. Along with other panels' widths summary derived values are shown graphically in Figure 4.7.

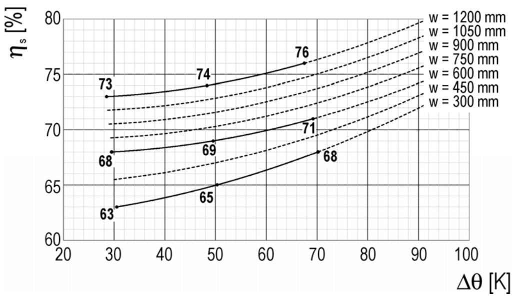



***Figure 4.7** The relation between radiant efficiency $\eta_s$ [%] and temperature difference $\Delta\theta = (\theta_{W1} + \theta_{W2}) / 2 - \theta_i$ [K].*

***Table 4.7** Radiant panels' power output characteristics from the test laboratory – panel width w = 300 mm.*

| Mean water temp. | Referent temp. | Temp. difference | Tot. spec. power output | Radiant spec. power output | Convective spec. power output | Radiant efficiency |
|---|---|---|---|---|---|---|
| $\theta_m$ [°C] | $\theta_g$ [°C] | $\Delta\theta$ [K] | $q_0$ [W/m] | $q_r$ [W/m] | $q_k$ [W/m] | $\eta_s$ [%] |
| 90.21 | 19.75 | 70.46 | 216 | 147 | 69 | 68 |
| 70.40 | 19.92 | 50.48 | 145 | 94 | 51 | 65 |
| 50.47 | 19.92 | 30.55 | 81 | 51 | 30 | 63 |

***Table 4.8** Radiant panels' power output characteristics from the test laboratory – panel width w = 600 mm.*

| Mean water temp. | Referent temp. | Temp. difference | Tot. spec. power output | Radiant spec. power output | Convective spec. power output | Radiant efficiency |
|---|---|---|---|---|---|---|
| $\theta_m$ [°C] | $\theta_g$ [°C] | $\Delta\theta$ [K] | $q_0$ [W/m] | $q_r$ [W/m] | $q_k$ [W/m] | $\eta_s$ [%] |
| 89.18 | 19.92 | 69.26 | 381 | 271 | 110 | 71 |
| 69.40 | 19.91 | 49.49 | 253 | 175 | 78 | 69 |
| 49.96 | 20.13 | 29.83 | 140 | 95 | 45 | 68 |

***Table 4.9** Radiant panels' power output characteristics from the test laboratory – panel width w = 1200 mm.*

| Mean water temp. | Referent temp. | Temp. difference | Tot. spec. power output | Radiant spec. power output | Convective spec. power output | Radiant efficiency |
|---|---|---|---|---|---|---|
| $\theta_m$ [°C] | $\theta_g$ [°C] | $\Delta\theta$ [K] | $q_0$ [W/m] | $q_r$ [W/m] | $q_k$ [W/m] | $\eta_s$ [%] |
| 87.41 | 20.08 | 67.33 | 681 | 518 | 163 | 76 |
| 68.41 | 20.02 | 48.48 | 451 | 334 | 117 | 74 |
| 48.92 | 20.28 | 28.65 | 245 | 179 | 66 | 73 |

By continuous lines there are shown values exemplar obtained from laboratory, dashed lines are derived. From the above mentioned analysis following conclusions can be stated:

a) as the width of the strip is larger, the radiant efficiency is higher;
b) as the temperature of water is higher (resp. $\Delta\theta$), the radiant efficiency is higher.

It is obvious that mentioning the radiant efficiency as single value is insufficient. However obtained values discover completely different problem of radiant heating systems. Values published as radiant strips' power output $q_0$ (given values by certified laboratory to all manufacturers) are used for evaluation of required heating surface. Since for creating of thermal comfort in occupancy zone the radiant part is decisive, the convective part can be considered as heat loss. Then, it is interesting to compare amount of heat delivered to the ambient by convection and by radiation and how this rate varies with panel's width. Radiant part is also mentioned in laboratory report.

In following examples there are compared all three chosen width of radiant strips:

Design heat load: $\phi_{HL}$ = 100 000 W
Mean water temperature: $\theta_m$ = 80°C
Reference temperature: $\theta_g$ = 18°C
$\Delta\theta$ = 62 K

**Radiant strip of width w = 300 mm**

$q_o$ = 218 W/m
$l$ = 458.7 m ⇒ $S$ = 137.6 m²
$\eta_s$ [%]= 0.665
*Radiant part of power output:*
$q_r$ = 100 000 x 0.665 = 66 500 W

**Radiant strip of width w = 600 mm**

$q_o$ = 399 W/m
$l$ = 250.6 m ⇒ $S$ = 150.4 m²
$\eta_s$ [%]= 0.72
*Radiant part of power output:*
$q_r$ = 100 000 x 0.72 = 72 000 W

**Radiant strip of width w = 1200 mm**

$q_o$ = 725 W/m
$l$ = 137.9 m ⇒ $S$ = 165.5 m²
$\eta_s$ [%] = 0.755
*Radiant part of power output:*
$q_r$ = 100 000 x 0.755 = 75 500 W

If a radiant strip of width 300 mm is taken as referent, than radiant strip of width 600 mm would deliver by 8.3% more energy into occupancy zone and a strip of width 1200 mm even by 13.5%.

Since the radiant part of the strips' power output is crucial for the thermal comfort in occupancy zone, it is obvious that by using wider panels it would be possible to reduce strips' length (total surface) while maintaining the same thermal comfort.

**Examples**

$$S_{1200}^{red} = S_{1200} \cdot \frac{\eta_{300}}{\eta_{1200}} = 165.5 \cdot \frac{0.665}{0.775} = 147.7 \text{ m}^2.$$

Radiant strips' surface can be up to 12.0% lower ⇒ $q_{red}$ = 100 000 W · 0.880 = **88 000 W**.

$$S_{600}^{red} = S_{600} \cdot \frac{\eta_{300}}{\eta_{600}} = 150.4 \cdot \frac{0.665}{0.720} = 138.9 \text{ m}^2.$$

Radiant strips' surface can be up to 7.8% smaller ⇒ $q_{red}$ = 100 000 W · 0.922 = **92 200 W**.

***Table 4.10** Evaluation of power output of compact and divided radiant panels 90 / 70 ˚C, $\theta_o$ =18 ˚C, $\Delta\theta$ = 62 K.*

| w [mm] | $q_0$ [W/m] | $\eta_s$ [%] | $q_s$ [W/m] | $q_k$ [W/m] | $q_0^{kor}$ [W] | $q_s^{kor}$ [W] | $q_k^{kor}$ [W] |
|---|---|---|---|---|---|---|---|
| 300 | 219 | 66.4 | 145 | 74 | 1506 | 1000 | 506 |
| 450 | 309 | 68.3 | 211 | 98 | 1464 | 1000 | 464 |
| 600 | 395 | 70.2 | 277 | 118 | 1425 | 1000 | 425 |
| 750 | 480 | 71.5 | 343 | 137 | 1399 | 1000 | 399 |
| 900 | 561 | 72.7 | 408 | 153 | 1376 | 1000 | 376 |
| 1050 | 644 | 74.0 | 477 | 167 | 1351 | 1000 | 351 |
| 1200 | 727 | 75.1 | 546 | 181 | 1332 | 1000 | 332 |
| 2x300 | 438 | 66.4 | 290 | 148 | 1506 | 1000 | 506 |
| 1x300 | 219 | 66.4 | 145 | 74 | 1506 | 1000 | 506 |
| 1x450 | 309 | 68.3 | 211 | 98 | 1464 | 1000 | 464 |
| Σ | 528 | 67.4 | 356 | 172 | 1483 | 1000 | 483 |
| 3x300 | 657 | 66.4 | 435 | 222 | 1506 | 1000 | 506 |
| 2x300 | 438 | 66.4 | 290 | 148 | 1506 | 1000 | 506 |
| 1x450 | 309 | 68.3 | 211 | 98 | 1464 | 1000 | 464 |
| Σ | 747 | 67.1 | 501 | 246 | 1491 | 1000 | 491 |
| 4x300 | 876 | 66.4 | 580 | 296 | 1506 | 1000 | 506 |

*Definition of terms in Table 4.10:*

| | |
|---|---|
| $w$ = panel width | $q_k$ [W/m] = convective part of power output |
| $q_0$ [W/m] = specific power output of radiant strips | $q_0^{cor}$ [W/m] = corrected total power output |
| $H_S$ [-] = radiant efficiency | $q_s^{cor}$ [W/m] = corrected radiant power output |
| $q_s$ [W/m] = radiant part of power output | $q_s^{cor}$ [W/m] = corrected convective power output |

Comparison (to reach $q_s^{kor}$=1000 W/m) with various panel dimensions:

| | | | |
|---|---|---|---|
| $w$ = 600 mm | $w$ = 2x 300 mm | $w$ = 1050 mm | $w$ = 2x 300 mm + 1x 450 mm |
| $q_o^{kor}$ = 1425 W/m | $q_o^{kor}$ = 1506 W/m | $q_o^{kor}$ = 1351 W/m | $q_o^{kor}$=1491 W/m |
| $q_k^{kor}$ = 425 W/m | $q_k^{kor}$ = 506 W/m | $q_k^{kor}$= 351 W/m | $q_k^{kor}$=491 W/m |
| $n$ = 1506 / 1425 = 1.057 => **5.68%** | | $n$ = 1493 / 1351 = 1.104 => **10.36%** | |
| $w$ = 750 mm | $w$ = 1x 300 mm + 1x 450 mm | $w$ = 1200 mm | w = 4x 300 mm |
| $q_o^{kor}$ = 1399 W/m | $q_o^{kor}$ = 1483 W/m | $q_o^{kor}$ = 1332 W/m | $q_o^{kor}$ = 1506 W/m |
| $q_k^{kor}$ = 399 W/m | $q_k^{kor}$ = 483 W/m | $q_k^{kor}$ = 332 W/m | $q_k^{kor}$ = 506 W/m |
| $n$ = 1483 / 1399 = 1.061 => **6.00%** | | $n$ = 1506 / 1332 = 1.131 => **13.06%** | |
| w = 900 mm | w = 3x 300 mm | | |
| $q_o^{kor}$ = 1376 W/m | $q_o^{kor}$ = 1506 W/m | | |
| $q_k^{kor}$ = 376 W/m | $q_k^{kor}$ = 506 W/m | | |
| $n$ = 1506 / 1376 = 1.094 => **9.44%** | | | |

***Table 4.11*** *Relative evaluation of a heating water temperature influence on the utilization of radiant part of radiant strips' power output – compact vs. divided radiant strips*

| w [mm] | 90/70°C Δθ = 62 K | 110/70°C Δθ = 72 K | 130/70°C Δθ = 82 K |
|---|---|---|---|
| 600<br>2x300 | 5.68% | 5.08% | 4.45% |
| 750<br>1x300+1x450 | 6.00% | 5.24% | 4.01% |
| 900<br>3x300 | 9.44% | 8.66% | 8.00% |
| 1050<br>2x300+1x450 | 10.36% | 9.10% | 8.37% |
| 1200<br>4x300 | 13.06% | 12.30% | 11.62% |

The difference between heating surface determined by using values $q_o$ (total power output) given by testing laboratory and reduced values can be used for radiant strips' design in cases when $q_o$ of strip of chosen width doesn't fulfill desired power output.

Adequate operation parameters are provided by suitable control.

One of the most important aspects for the highest radiant efficiency is surface temperature of active heating surface.

Surface temperature between two modules doesn't almost differ from the surface temperature at the tube. On the Figure 4.8 there are shown measured surface temperatures in particular points at surface of a profile. Also this picture proves that width of a panel has influence on the summary radiant output (higher average temperature).

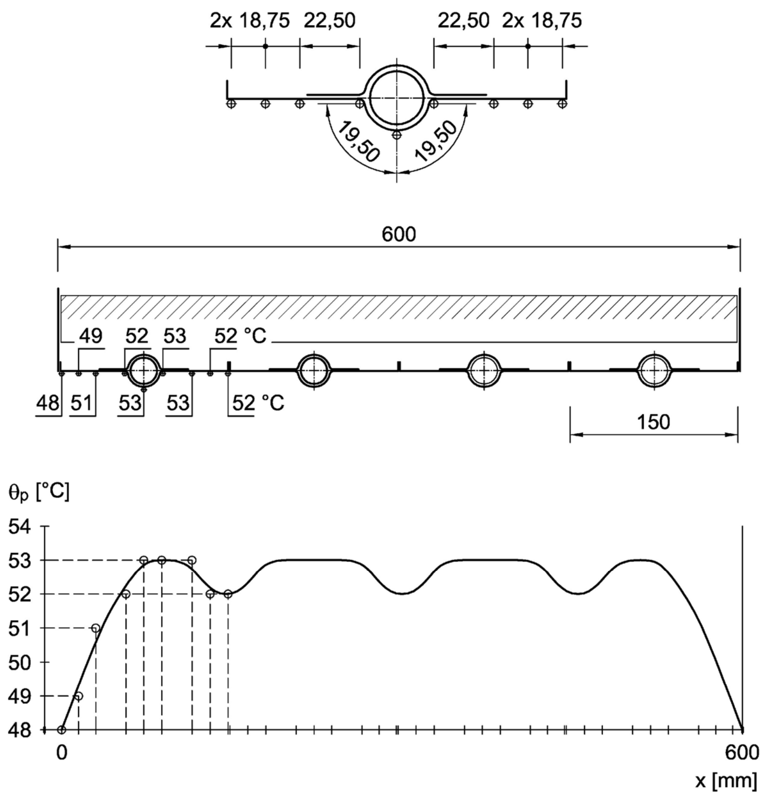



***Figure 4.8*** *Measured surface temperatures on a radiant strip 600 mm wide.*

Average strip temperature when the surface temperature in the middle of a module is $\theta_{max} = 53°C$:

***Marginal module part:***

| | |
|---|---|
| 18.75 mm x 48.5°C = | 909.38 mm°C |
| 18.75 mm x 50.0°C = | 937.50 mm°C |
| 22.50 mm x 51.5°C = | 1158.75 mm°C |
| 19.50 mm x 52.5°C = | 1023.76 mm°C |
| 79.50 mm | 4029.39 mm°C |

Average temperature of marginal part
$\theta_1$ = 4029.39 : 79.50 = **<u>50.7°C</u>**.

***Inner module part:***

| | |
|---|---|
| 18.75 mm x 52.0°C = | 975.00 mm°C |
| 18.75 mm x 52.5°C = | 984.38 mm°C |
| 22.50 mm x 53.0°C = | 1192.50 mm°C |
| 19.50 mm x 53.0°C = | 1033.50 mm°C |
| 79.50 mm | 4185.38 mm°C |

Average temperature of inner part
$\theta_2$ = 4185.38 : 79.50 = **<u>52.6°C</u>**.

Average surface temperatures when the temperature in the middle of a module is $\theta_{max} = 53.0°C$:

w = 300 mm: $$t_p^{300} = \frac{2 \cdot 50.7 + 2 \cdot 52.6}{4} = \frac{101.4 + 105.2}{4} = 51.7°C;$$

w = 450 mm: $$t_p^{450} = \frac{2 \cdot 50.7 + 4 \cdot 52.6}{6} = \frac{101.4 + 210.4}{6} = 52.0°C;$$

w = 600 mm: $$t_p^{600} = \frac{2 \cdot 50.7 + 6 \cdot 52.6}{8} = \frac{101.4 + 315.6}{8} = 52.1°C;$$

w = 750 mm: $$t_p^{750} = \frac{2 \cdot 50.7 + 8 \cdot 52.6}{10} = \frac{101.4 + 420.8}{10} = 52.2°C;$$

w = 900 mm: $$t_p^{750} = \frac{2 \cdot 50.7 + 10 \cdot 52.6}{12} = \frac{101.4 + 526.0}{12} = 52.3°C;$$

w = 1050 mm: $$t_p^{1050} = \frac{2 \cdot 50.7 + 12 \cdot 52.6}{14} = \frac{101.4 + 631.2}{14} = 52.3°C;$$

w = 1200 mm: $$t_p^{1200} = \frac{2 \cdot 50.7 + 14 \cdot 52.6}{16} = \frac{101.4 + 736.4}{16} = 52.4°C.$$

From the results, following outcomes are obvious:

a) as width of a strip increases the average surface temperature increases as well ⇒ influence on radiant efficiency;

b) the strip construction allow minimal temperature difference between average surface temperature and temperature at the top of the tube ($\theta$ = 53°C). ($\Delta\theta_{300}$ = 53.00 – 51.66 = 1.34 K ; $\Delta\theta_{1200}$ = 53.00 – 52.39 = 0.61 K).

## 5.1 Overview

Heating of industrial and other large space buildings with gas-burning suspendable heaters has made lately significant progress. Enormous reduction of energy consumption given by the heat delivery principle has become the only argument for many assembly companies and designers when persuading the investor about advantages of such a system. The problem is that even the best system with theoretically the lowest operation costs, when designed and/or installed wrongly, will give poor results. All advantages can be ruined.

As any other product also infrared heaters have their technical respectively construction pros and cons and different function characteristics. From the design point of view the radiant heat delivery principle itself requires quite a specific approach when desired power output and location in a space is evaluated. When assessing the suitability of such radiant heating system following aspects have to be considered:

- radiant heaters´ construction and the way how the heat is delivered into the considered space
- operation characteristics
- control of the system
- device maintenance

Infrared heaters (1) deliver heat into the occupancy zone by means of radiation (Figure 5.1). Firstly, all irradiated surfaces' (2) temperature is increased and just after, secondly, ambient air is by convection heated up. Vertical air temperature gradient for this type of heating is very low (0.3…0.5 K/m) and so this reduces significantly heat losses through higher parts (ceiling, skylights) of buildings‘ envelopes. Below the ceiling, which belongs with skylights to the largest cooled surfaces, there is relatively low temperature and therefore heat loss is lower than in case of systems that creates a high temperature air pillow below the ceiling (typically convective heating systems with low air circulation).

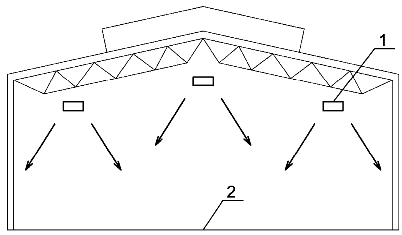



***Figure 5.1*** *Industry hall heating with infrared heaters. 1 – radiant heater 2 – irradiated (warm) floor.*

Infrared heaters can be divided according to the surface temperature (wavelength of emitted radiation) into low- medium- and high- intensity radiant heaters (ASHRAE). In some countries very popular differentiation is also “bright” and “dark” heaters, which is related to the way of how the gas-air mixture is burned (open fire on ceramic plaques – bright, burning within tubes - dark). Somewhere even “plaque” and “tube” heaters specification is used.

## 5.2 Tube radiant heaters

Tube radiant heaters (Figure 5.2) burn the mixture of gas with air by burners placed in control box. From the box the hot burnt gases are driven to the heating tubes covered from the top by reflective metal sheet (reflectors). Burnt gasses are forced by the fan to the chimney and drawn out of heated space. Surface temperature of the tubes varies from the beginning (burner side) 500°C up to 180°C at the end (exhaust fan side). Average surface temperature is about 340 – 350°C. Around these temperatures the efficiency of the burning process is maximal. Some manufacturers declare very high power outputs of their devices even when the heater is very short. However in such heater types the heat exchange surface cannot transfer produced heat into the space and therefore exhausts temperature rises up to 250…280°C. The chimney heat loss is very high and therefore efficiency low. This results in higher running costs. The optimal suspension heights for the tube radiant heaters are about 5 – 8 meters above the floor. Tube radiant heaters are produced in power outputs line from approx. 5 to 50 kW, special "radiant modules" ones even up to 150 kW.

### Construction

To the main parts of tube radiant heater belong control unit with burner and exhaust fan, heating tubes, reflective cover (reflector) and hangers (Figure 5.3). Other components depend on a particular manufacturer and also an investor (insulation).

The majority of tube radiant heaters are manufactured very simply, in almost trivial shape. Side parts of the reflector are relatively shallow (Figure 5.4). Such a solution influences its radiant efficiency (about 50%). Here about 50% of the produced heat is delivered to the ambient by convection (can be considered as heat loss – just small part is re-utilized). The device losses its specification "radiant". When such heater is suspended tilted (30°), the efficiency falls even more, to about 45%. Another technical draw back can be absence of front and rear covers that also reduces the convective part and increases total efficiency of heat delivery.

***Figure 5.2*** *Typical tube radiant heater.*

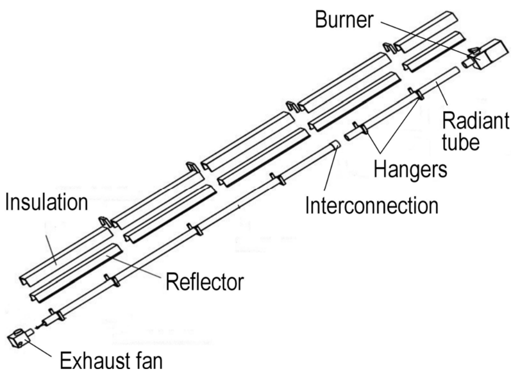



***Figure 5.3*** *Tube radiant heater's construction.*

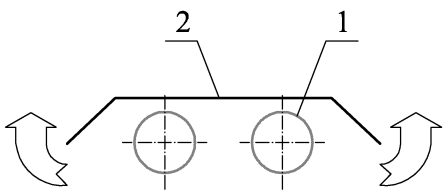



***Figure 5.4*** *Tube heater with low radiant efficiency, shallow reflector.*

On the Figure 5.5 there is a tube radiant with completely different reflector shape. Such construction reduces amount of heat rising up by convection and also reduces energy that would irradiate side walls and ceiling when the heater would be suspended tilted. Radiant efficiency of such a heater is about 63% in standard position (58% when tilted 30°).

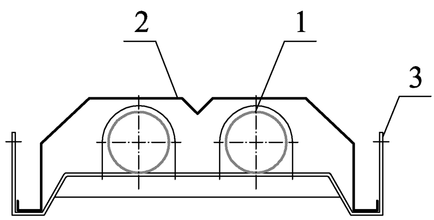



***Figure 5.5*** *Non-insulated tube radiant heater with deep reflector and front and rear covers*

Radiant efficiency can be further more improved with insulation placed on the top of the reflector (Figure 5.6). In such a case radiant efficiency is about 72% and 67% in tilted position.

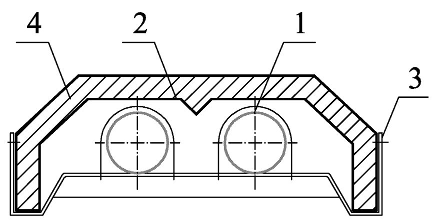



***Figure 5.6*** *Insulated tube radiant heater with deep reflector and front and rear covers*

### *Radiant modules (compact infrared heaters)*

Completely separate group of tube heaters are so called "radiant modules" (Figure 5.7). Although the main principles are the same, there are some differences. At first, the power output of these heaters is typically higher (100 to 150 kW). At second, heating tubes are much longer (up to 50 m). At third, because of longer length the average surface temperature is much lower (about 200°C). And at last, to be able to obtain low surface temperature decrease along the tube length, large tubes, large exhaust fan and so large exhaust velocities (15 m/s) are kept. The surface temperatures can be also maintained high by installing additional burners along the tubes' length. The biggest advantage of the system lies in almost uniform, relatively low temperature at large scale (the power output can be delivered in larger area - low specific heat delivery W/m²). This is becoming more and more important when buildings are being well insulated – low specific heat demand. On the other hand such system cannot be used for heating of lonely workplaces or it is not possible to switch off places where there is temporary no need for heating.

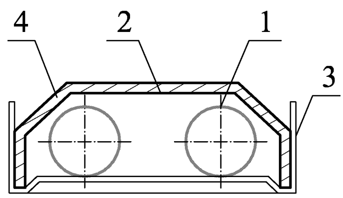



***Figure 5.7*** *Radiant module in cross section.*

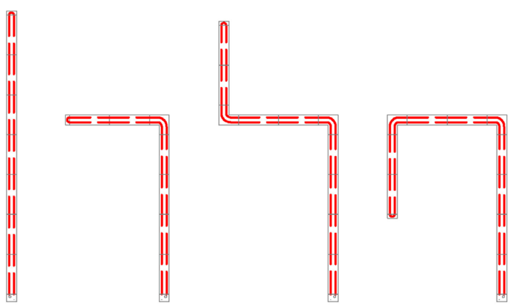


***Figure 5.8*** *Radiant module's connection possibilities.*

### Function

Traditional gas burner is located together with control unit for pressure regulation in front of the heating tube. Flame right after the burner, entering the heating tube heats up its surface up to 500°C. After passing whole length of the tubes the exhausts temperature decreases to about 160…250°C. Heating tubes are either U-shaped (Figure 5.9 top) or linear (I-shaped – Figure 5.9 below). In case of U-tube type both burner and exhaust fan are located on one side in one box. In case of linear type the burner is on one side and the exhaust fan is on another (Figure 5.3). It is very important to consider the surface temperature decrease along the tubes, especially in linear heater type, where differences in heat delivery at the beginning and at the end are significant.

The first critical point on economical operation can be found in exhausts temperatures. The mentioned temperature difference of the exhaust outlet (160…250°C) is given by length of the tubes. When the same length transfers once 20 kW (160°C) and once 30 kW (250°C) the higher value is less economical because the chimney heat loss is much higher. Tolerable are differences about 2…3 kW.

Shape of the reflector and heater's placement in considered space are also very important issues for economical heating system. U-tube heater with shallow reflector and without front and rear sheets (Figure 5.10 on top) can be marked as completely unsuitable construction solution. Its radiant efficiency is about 50% (rate between heat flux delivered to the ambient by radiation and total heat flux delivered to the ambient). This value is on the edge between radiant and convective device. Its placement in higher part of the room and low air velocities ensures that whole half of the heater's power output will fall in losses. Material of which the reflector is made of has also negative influence on convective/radiant rate. Polished aluminum has higher reflectivity than stainless metal sheet but shorter lifecycle (reflectivity changes rapidly in operation time). Thus it is necessary to weight, which aspect is in particular case more important. Completely wrong is to suspend such heater in tilted position (Figure 5.10 below). Radiant efficiency decreases lower and because of shallow reflector even some radiation is useless.

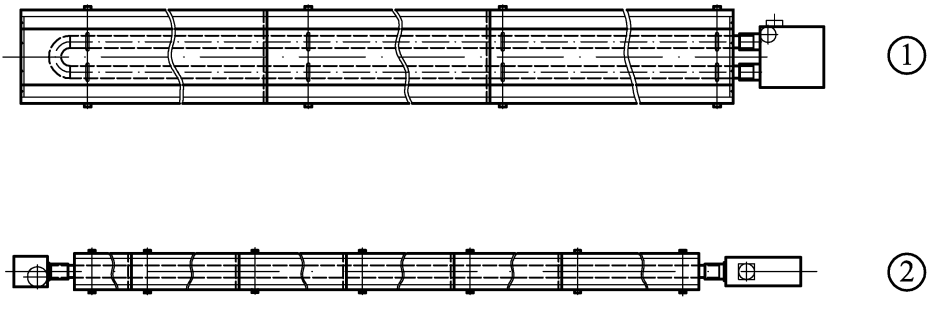



***Figure 5.9*** *Tube radiant heater 1 – U-tube version; 2 – linear version.*

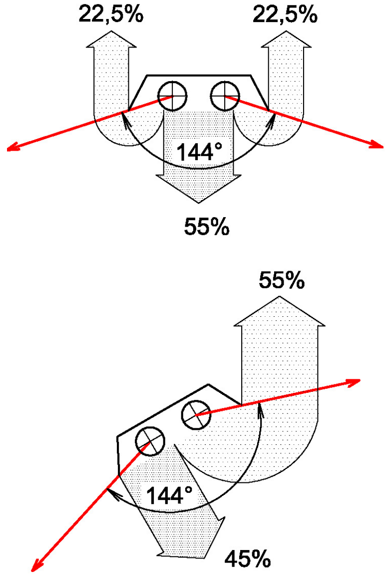



***Figure 5.10** U-tube infrared heater with shallow reflector - horizontal position (top); tilted position (below).*

Let's have a specific heater suspended tilted 30°, 6 m above the floor (Figure 5.11). Then, let's have an external wall in distance of about 18 m. In case of external wall recommended irradiation to eliminate cold convective flows near the wall is about 2.5 m above the floor. It means that all the energy that irradiates wall above this height (points 6 – 4, angle 24°) can be considered as heat loss. When the distance is shorter (12 m), the amount of waste energy is even higher (angle 29°). Hence, infrared heaters with shallow reflectors should never be installed in tilted position.

Beside reflector shape and material, very important is also heater's placement within the space. Heaters with smaller core radiation angle (Figure 5.12 left) are more suitable for narrower halls, or between shelters. On the other hand heaters with larger core radiation angle (Figure 5.12 right) are more suitable for whole-space heating. The difference in this case is given just by different heater's power output (30 kW - left, 10 kW – right) and hence different tubes' dimensions but there are even manufacturers offering various core angles at same power output.

In case of the heaters with deeper reflector, it is possible to suspend it also tilted. It is not ideal solution (radiant efficiency sinks down) but in certain cases (for example crane installation) there is no other way.

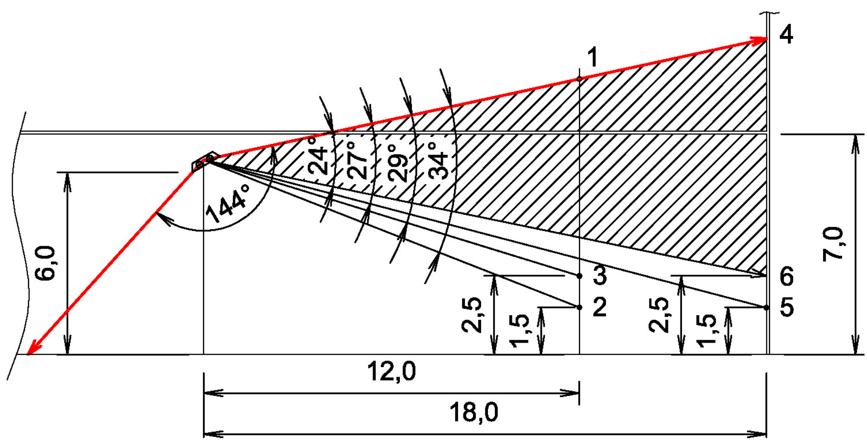



***Figure 5.11** Radiation heat loss from tilted (30°) U-tube infrared heater with shallow reflector.*

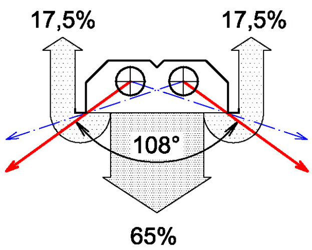



***Figure 5.12*** *U-tube radiant heaters 30 kW (left) and 10 kW (right).*

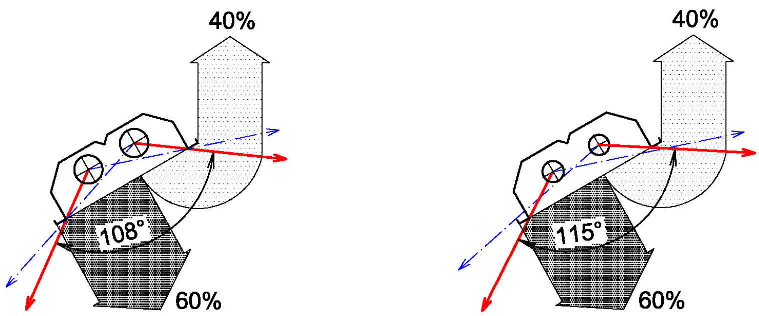



***Figure 5.13*** *U-tube radiant heaters 30 kW and 10 kW in tilted suspension.*

In case of the same example as above the radiation heat loss angle is just 6°. It means that almost all the energy is already utilized and when there is a desire to increase system's efficiency even more it is possible to reduce suspension angle to $\gamma = 30° - 6° = 24°$. If the external wall is closer -12 m (Figure 5.14 point 3) suspension angle can be just $\gamma = 18°$. Because the radiation is absorbed on dust particles it is recommended not to extend reach from 12 up to 15 m.

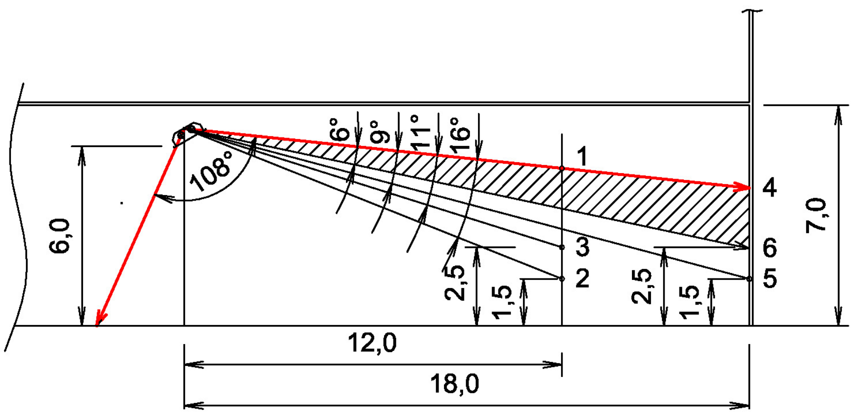



***Figure 5.14*** *Radiation heat loss from tilted (30°) U-tube infrared heater with deep reflector.*

Further improvement of radiant efficiency can be achieved by installing the insulation on the top of the reflector (Figure 5.15), which influences the utilization of energy by approx. 3…5% (20 mm thick). Recommended type of insulation is with aluminum foil on the top reducing emissivity of the surface.

***Figure 5.15** Thermal insulation of U-tube heaters.*

Standard tube radiant heaters are manufactured with relatively dense power output line from 5 up to 50 kW, and lengths 3…12 m in U-tube version and 5…20 for linear version (I-type).

## 5.3 Plaque radiant heaters

***Figure 5.16** Tilted version of a plaque radiant heater suspended near ambient wall.*

Plaque radiant heaters have surface temperature about 850…950°C. The heating-active surface is created from porous ceramic tiles. These heaters are sometimes, because of their high surface temperatures, called "bright". There is performed on the active surface "flameless" diffuse burning of gas – air mixture while large amount of heat is created. Emissivity of such surface is about 0.93…0.95. The most significant parameters for power output are surface temperature and active surface area. Some manufacturers install in front of the ceramic plaques wired grid from austenitic steel to improve radiant properties of the heater. The grid has also a safety function in case the ceramic tile would break. On the other hand the breakage happens quite seldom and even if some part would fall down the ceramic material has very low thermal capacity so the part cools down before it reaches floor level. Radiant efficiency can further be improved by thermal insulation or by utilizing heat from exhaust gasses to preheat the air-gas mixture. Power output variants are similarly to tube heaters from about 4 to 50 kW. Optimal suspension heights are from 4 up to about 20 m. For heating of high spaces and separate workplaces they are irreplaceable.

### Construction

Although heater's construction varies according to the manufacturer, main parts are the same. Major construction parts of typical plaque radiant heater are mixing chamber (1), ceramic plaques (2), reflector (3), ignition and ionization electrodes (4), inlet nozzle (5), control unit (6) and hangers (7).

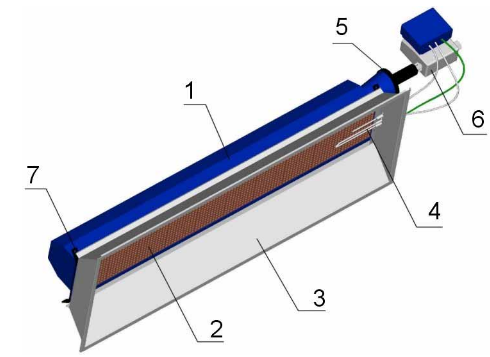



***Figure 5.17** Typical plaque radiant heater 1 – mixing chamber, 2 – ceramic plaques, 3 – reflector, 4 – ionization and ignition electrodes, 5 – inlet nozzle, 6 – control unit, 7 – hanger.*

Particular construction types differ in sizes and shape of active surface, shape of mixing chamber and reflector, application of insulation, way of suspension or other construction details. Assessment of the influence of various construction details was done by SCHWANK GmbH Köln company in Gaswärmeinstitut Essen (Germany) (Mr. Schwank is the inventor of burning principle on plaque radiant heaters in 1950s).

**Construction types available:**

- Plaque heater with open combustion chamber (Figure 5.18)

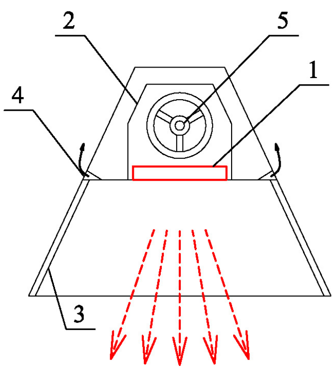



***Figure 5.18*** *Plaque radiant heater with open combustion chamber.*

Ceramic plaques (1) are placed in lower part of mixing chamber (2). The reflector is separated from the mixing chamber by a crevasse hole (4) that is used for exhausts. Radiant efficiency is about 58%.

- Plaque heater with open combustion chamber and austenitic grid (Figure 5.19)

The austenitic grid (6) is located right below the ceramic plaques. Exhausts are rising out of the combustion chamber by the crevasse (4) between mixing chamber and reflector. Radiant efficiency is about 63%.

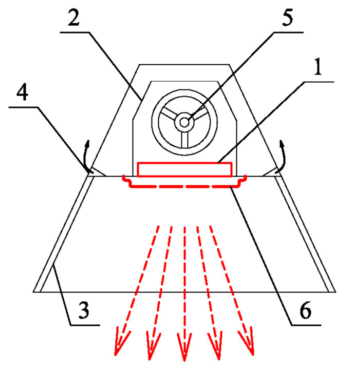



***Figure 5.19*** *Plaque radiant heater with open combustion chamber and austenitic grid.*

- Plaque radiant heater with austenitic grid and pre-heating of burning mixture, the reflector is insulated (Figure 5.20)

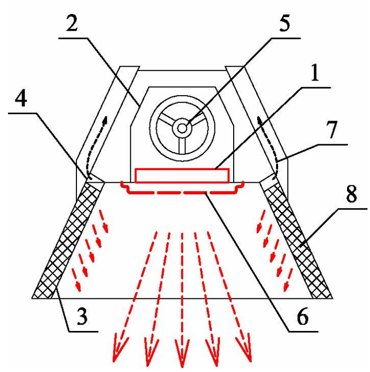



***Figure 5.20*** *Plaque radiant heater with austenitic grid and pre-heating of burning mixture, the reflector is insulated.*

Apart from parts described in previous version, the heater is equipped with exhaust channels (7) improving pre-heating of burning mixture. Radiant efficiency at this heater's type rises up to 73%.

- Combined heater with closed "delta" combustion chamber (Figure 5.21).

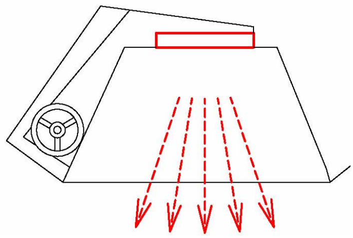


*Figure 5.21 Combined plague heater with "delta" combustion chamber.*

This radiant heater's type is designed to transfer significant portion of energy by secondary radiation of side covers. The gas-air mixture is led to the burner from the side all around the side cover to utilize maximally the pre-heating principle. Radiant efficiency of this type starts at about 67%.

- Combined heater with "delta" chamber, fully insulated with austenitic grid (Figure 5.22).

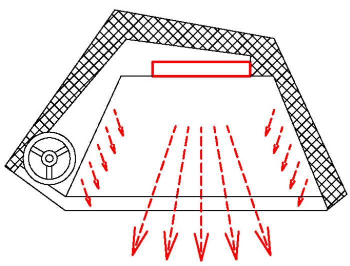


*Figure 5.22 Plaque radiant heater with closed "delta" chamber, fully insulated and equipped with austenitic grid.*

This type combines all the advantages of previous models (austenitic grid, pre-heating and insulation)). Radiant efficiency reaches the highest value of all plaque radiant heaters so far introduced to the market 82%.

For following technical specifications an average quality plaque radiant heater was chosen to be able to show particular values for designers. Most of the manufacturers have the values similar and so there are just minor corrections.

**Standard horizontal heater – specification A**: power output line - 7, 11, 15, 18 kW – ceramic plaques in one row; 25, 36, 43 kW – ceramic plaques in two rows; angle between the reflector and horizontal plane 60°. Core radiation angle is 88° *(*Figure 5.23*)*. This heater is more suitable for places where higher concentration of radiation is needed (near to fully glassed walls, gates, among shelters etc.). The reflector is made of stainless steel with high reflectivity. Combustion chamber is open; exhausts slightly pre-heat the mixing chamber. Radiant efficiency is approx. 70%.

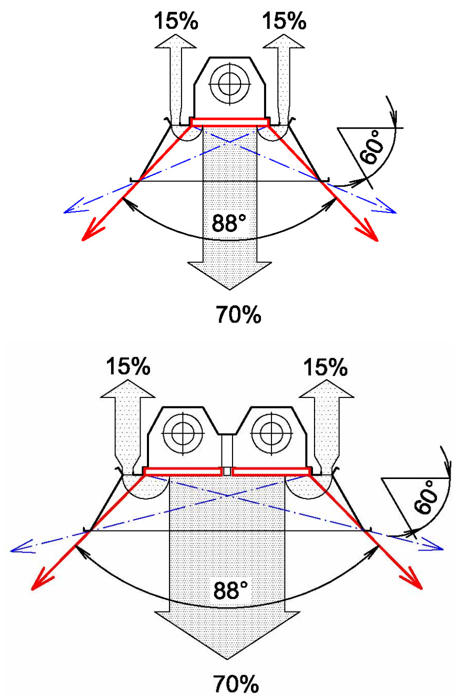



*Figure 5.23 Typical plaque radiant heaters - A; one respectively two rows of ceramic plaques.*

**Standard horizontal heater – specification B**: power output line - 7, 11, 15, 18 kW – ceramic plaques in one row; 25, 36, 43 kW – ceramic plaques in two rows; angle between the reflector and horizontal plane 45°. Core radiation angle 112° (Figure 5.24). Wide coverage is optimal for places with lower specific heat demand [W/m²] – typically inner parts of heated spaces. Radiant efficiency is approx. 70%.

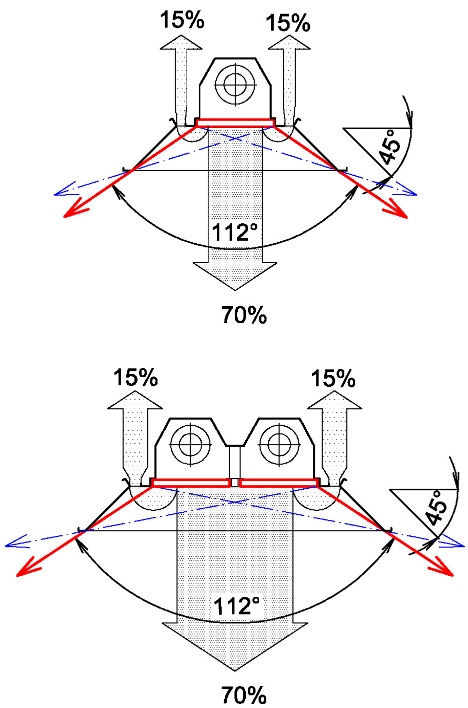



***Figure 5.24** Typical plaque radiant heaters - B; one and two rows of ceramic plaques respectively.*

**Standard tilted heater – longer** (Figure 5.26) and shorter (Figure 5.25) reach: power output line 7, 11, 15, 18 kW – one row; 25, 36, 43 kW – two rows; they differ in construction of reflector. Tilted version is optimal for cases where it is not possible or suitable to suspend heaters from the ceiling. They are designed for wall suspension (standard wall suspension angle 30°). In case a standard construction type (A or B) would be suspended tilted, significant amount of heat would be delivered to space where it is not needed – heat loss. Shape of the reflector in both tilted versions is designed to satisfy both local and overall heating.

**Longer reach variant** (Figure 5.25) is designed to cover space of 2 x h from the heater's placement on the wall, where h is height of suspension. This variant is optimal for overall heating.

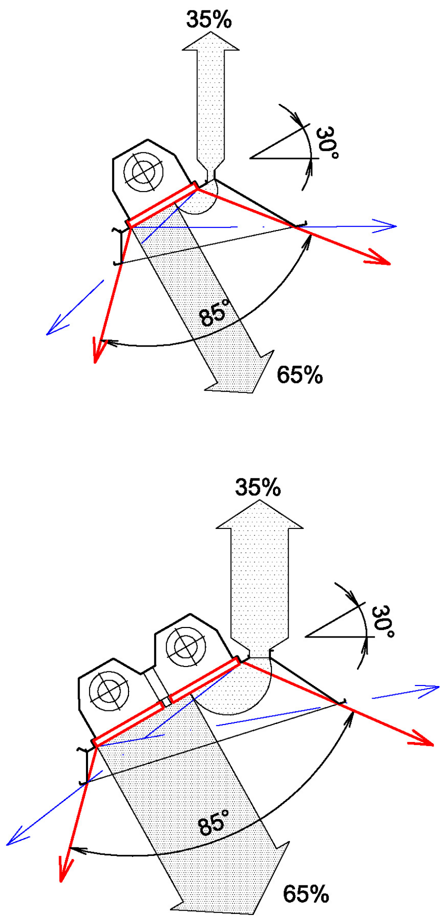



***Figure 5.25** Tilted plaque heaters with longer reach; one respectively two rows of ceramic plaques.*

The variant with shorter reach (Figure 5.26) is designed for coverage of 1.5 x *h*, where *h* is height of suspension. Heater is ideal for local heating of separate workplaces and areas with higher heating demand (external walls, gates, windows).

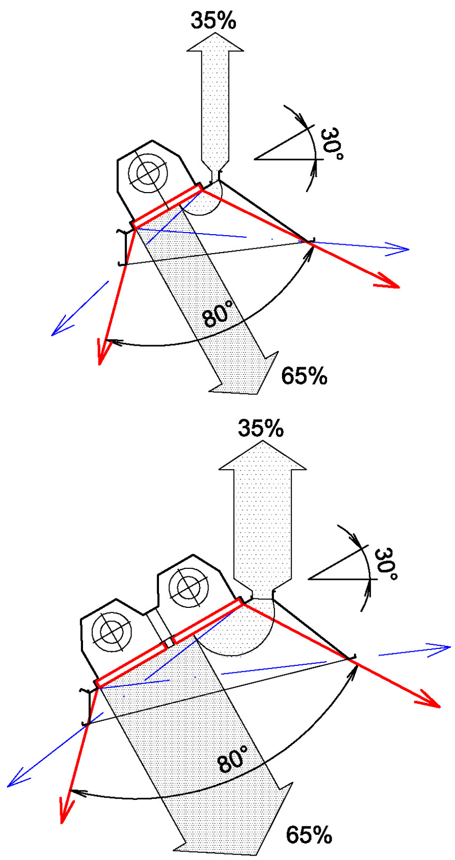



***Figure 5.26*** *Tilted plaque heaters with shorter reach; one respectively two rows of ceramic plaques.*

### Function

Plaque radiant heater burns mixture of natural gas (propan - butan) with ambient air on the surface of porous ceramic plaques. Produced heat is partly delivered to the ambient by convection (to the ceiling level – heat losses) but mainly is transferred by means of radiation to the occupancy zone. Active surface temperature varies from about 850 to 900°C. The flame layer can be directly seen during operation - so called open-burning device. Because the active surface is relatively small but its temperature on the other hand high, the heat delivery is very concentrated. This is crucial characteristic of plaque radiant heaters. Since it is nearly a point heat source it is easily possible to aim produced heat even from longer distances to local workplaces.

## 5.4 Heating load evaluation and design procedure

It is proved statement that when using radiant instead of convective heating even about 30 up to 40% theretically consumed energy can be spared. However such claim is not universal and it is not applicable to all cases. For example at lower buildings (lower than 4 m), radiant systems can cause local overheating. There, convective heating is reasonable alternative. There is always necessity to obey design procedures and recommendations. The statement is based on the assessment of thermal comfort of people and vertical air temperature gradient. When radiant heating is used, mean radiant temperature is higher and therefore air temperature can be lower. It is not necessary to heat up the whole air volume of the room. This is significant especially in high buildings where vertical air temperature gradient causes higher heat losses. Hence, design should be also considered differently. While in case of convective heating systems the design heat load should be higher than summary heat losses (evaluated for example according to EN 12831), in case of radiant heating systems, the design heat load can be even lower.

Recommended methodology is for such cases based on German technical rules G 638-1 and G 638-2. Heat losses are evaluated normally according to valid standards (for ex. EN 12831). Then, the resulting value of heat loss is “corrected” with various coefficients (type of heater, height of suspension, radiant efficiency etc). Detail procedure will be described in following chapter.

### Definitions

- ***Design heat load*** $\Phi_{HL}$ [W] is required power output needed for maintaining design conditions EN 12831.
- ***Design heat loss*** $\Phi_i$ [W] is amount of heat per unit of time leaving the building to the external environment under defined design conditions EN 12831.
- ***Indirect exhaust discharge*** is a way of combustion products removal. Exhausts are not directly led outside by chimney, but instead, first they are mixed with ambient air, afterwards the mixture rises to the highest point of the room (supposing exhausts are warmer than the ambient air) and finaly the mixture is driven away from the highest point of the room. Nevertheless such method is possible only for plaque radiant heaters where resulting emissions are very low.
- ***View factor*** $\varphi$ [-] is ratio between heat transferred by radiation to the particular surface and total amount of heat transferred by the radiation from the source surface to the ambient. It expresses the percentage of transferred heat to the particular surface.
- ***Direct exhaust discharge*** is standard way of combustion products removal by chimney.
- ***Reflector*** is metal cover mounted to the heater’s body in order to increase the amount of heat transferred to the core radiation area.
- ***Radiant efficiency*** $\eta_S$ [-] is ratio between heat flow transferred to the ambient by means of radiation and total heat flow transferred to the room.
- ***Radiant heaters’ power input*** $\Phi_{ZP}$ [W] is defined as an amount of energy consumed per unit of time corresponded to the volume or mass flow of gas; used heat value is either gross or net calorific value.
- ***Radiant device’s efficiency*** $\eta_c$ [-] is ratio between total heat flow transferred to the room and total heat power input of the radiant heater.
- ***Core radiation angle*** $\alpha$ [°] is an angle between “marginal” rays connecting characteristic points of the active heating surface (plaque heaters – edge of ceramic plaques, tube heaters – centre of the tube) and adjacent lower edge of the reflector.
- ***Total radiation angle*** $\beta$ [°] is an angle between “marginal” rays connecting characteristic points of the active heating surface (plaque heaters – edge of ceramic plaques, tube heaters – centre of the tube) and opposite lower edge of the reflector.
- ***Setting angle of the reflector*** $\gamma$ **resp.** $\delta$ [°] is acute angle between the reflector and active surface’s plane in xz, resp. yz cross section.
- ***Ventilation by free-convection of air*** represents convective discharge of exhausts-air mixture through defined holes in a roof or walls of a building.
- ***Ventilation by forced convection of air can be described as*** discharge of exhausts-air mixture by one or more fans located in a roof or walls of a building.

- ***Ventilation by air infiltration*** represents convective discharge of exhausts-air mixture through not-defined holes in a roof or walls of a building based on temperature and pressure differences between indoor and outdoor of a building.
- ***Overall heating*** is aimed on creation of thermal comfort for occupants uniformly over whole room surface area.
- ***Local heating*** is aimed on creation of thermal comfort for occupants just in specific parts of room surface area.

### Influencing factors

The biggest part of responsibility for optimally operateable, economical heating system is lying on designer. He has to consider all possibilities and limitations and recommend the most suitable solution. Different requirements in different zones of treated space directly influence choice of type and construction variant of the heater. This might result in heating systems consisted from both tube and plaque radiant heaters. Such solution reduces gas consumption and so operation costs.

When such heating system is designed, it is necessary to consider always whole energetic system. It means do not desing particular parts separately but always care them as parts of system. We have to take into account a lot of parameters, parameters of a building we heat, parameters of system we design, including outside environment that is surrounding the building.

**Parameters related to a building**
- Construction (dimensions, materials, windows, gates, skylights, crane)
- Building orientation
- Fire risks (environment, dustness)
- Location of technology, lights
- Indoor environment (heat gains, air distribution pattern)
- Supposed location of workplaces
- Workplaces operation specifics (number of shafts)
- Hygienic parameters regarding type of work and technology

**Parameters related to a heating system**
- Core and total radiation angle of radiant heaters
- Radiant efficiency
- Power output variants at disposal
- Operation characteristics (heat-up period)
- Infrared heaters' construction (lenght, surface temperatures)
- Suspension possibilities
- Minimal safety clearences
- Ventilation (air inlet and outlet)

**Surrounding environment parameters**
- Building location and placement within surroundings
- Climatic conditions
- Height above the sea

### Classes and types of heaters

First step of the design procedure should be environment assessment according to valid standards of fire safety (whether it is possible to install such devices in the building – the environment shouldn't be flameable or even explosible). Following step is to choose suitable heater's class. There are three possibilities: plaque (900°C), tube (350°C) and compact (200°C) radiant heaters. The installation principles are relatively similar.

From the assessment done by Gaswärmeinstitut in Essen came out that comparable economical application of plaque and tube radiant heaters lays in height of 6 m above the floor. In height of 10 m radiant efficiency of tube heater is about 15% and in height of 15 m even about 27% lower than in case of plaque radiant heaters. On the other hand in case of lower suspension heights it is recommended to use rather heaters with lower surface temperature or use fewer heaters with lower power output both to avoid local overheating. Furhter minimization of suspension heights can be achieved by tilted suspension of suitable type of radiant heaters on side walls. Such suspension is however recommended just for plaque radiant heaters where radiant efficiency decreases just minimally. To conclude, for buildings and spaces with height of suspension $h \leq 6$ m it is more economical to choose tube radiant heater that can be in places with higher heating demands (frequently opened windows, expedition spaces etc.) fitted with plaque heaters with faster reaction time. For spaces with suspension height $h > 6$ m the most suitable on the ther hand is plaque radiant heater.

Also within one heater's class there are different construction types with different core $\alpha$ [°] and total $\beta$ [°] radiation angles (Figure 5.27). Construction influences the way how the heat is transferred to the ambient.

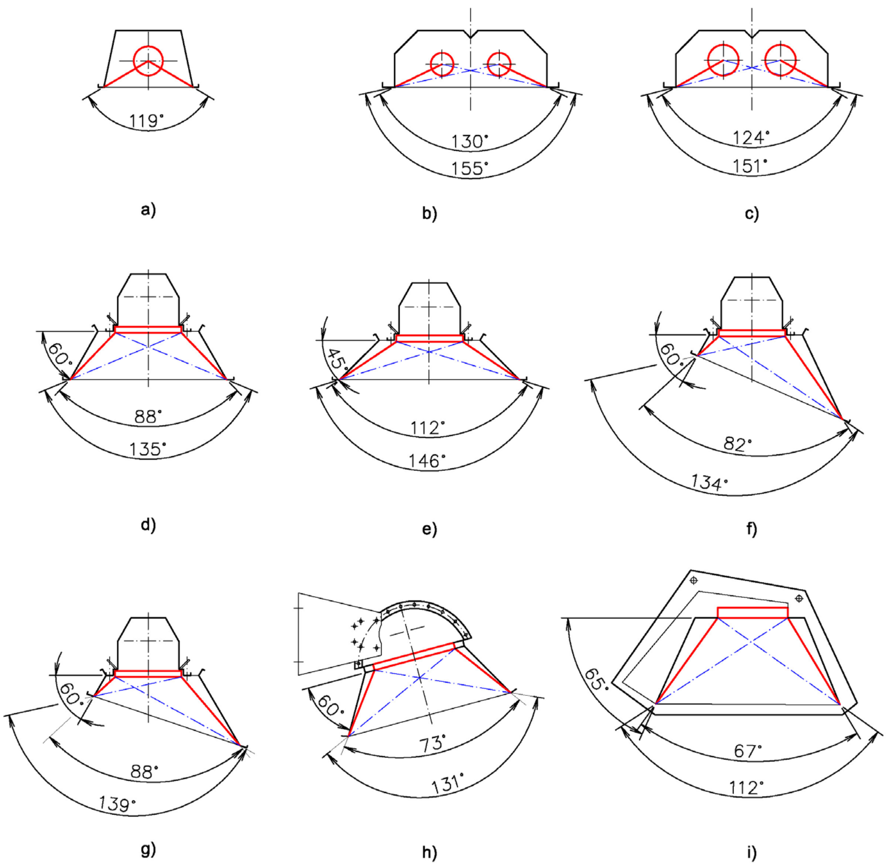



***Figure 5.27*** *Core (red rays) and total (blue rays) radiation angles of various heaters classes (types). a to c – tube heater, linear- and U- tube; d to g – plaque heater, open combustion chamber – horizontal and tilted construction; h to i – plaque heaters, closed combustion chamber.*

## Infrared heater's placement

### Core and total radiation angles

The most important parameter for placement design is so called core radiation angle (Figure 5.27 and 5.28 – red color). It defines area to which incidents about 90% of heat delivered to the space by means of radiation. The area is called core radiation area. The core radiation angle is limited by margin rays connecting key points of active surface (plaque – corners, tube – tubes' centres) with lower edges of anjacent reflector (Figure 5.28). These values or equation for evaluation of optimal heaters' distances should offer every manufacturer.

### Tube radiant heaters

Praxis shows that for uniform heat delivery into the occupancy zone, marginal rays should cross between two radiant heaters at height minimally 1.5 m above the floor (Figure 5.29). In case of outside walls the cross between marginal ray and the wall shouldn't be lower than 2.5 m. The higher height at external wall is recommended to reduce cold convective flows that might appear in this area. In case of workplaces located right next to the cooled window, it is possible to place the crossing even higher. In case there is a corridor between two radiant heaters it is possible to use total radiation angle crossing and the height >2.5 m. Typical core radiation angles for tube radiant heaters when there are no informations from manufacturers can be:120° in cross section and 70° along the heater (in this direction length of the heater has to be taken to account – Figure 5.29).

### Horizontal suspension of tube radiant heaters

Some manufacturers offer two construction types of their radiant devices in order to fulfill two different requirements. At external wall there is a need to concentrate the radiated energy as much as possible, therefore radiant heater with narrow core radiation angle – let's mark it α. On the other hand for overall heating (inner spaces) larger radiation spread to cover larger area is needed (Figure 5.30).

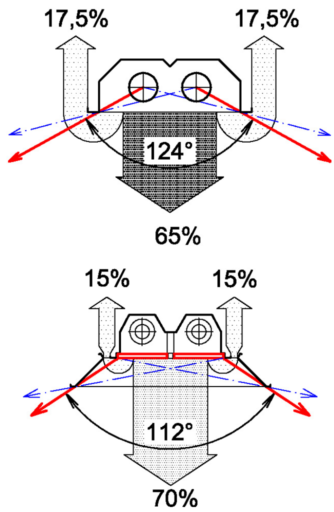



*Figure 5.28* *Core (red) and total (blue) radiation angles at typical plaque (top). Resp. tube radiant heater (below).*

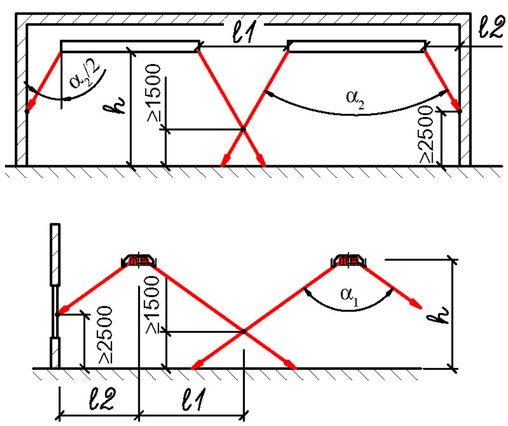



*Figure 5.29* *Optimal distances between tube radiant heaters and between heaters and walls in cross section (top) and along the heaters (below).*

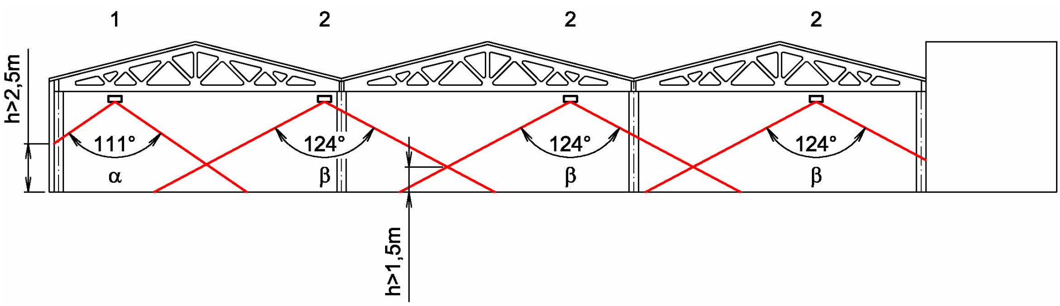



***Figure 5.30*** *Elimination of negative influence of external wall with tube radiant heaters (1) core radiation angle α = 114°, (2) core radiation angle β = 124°*

### *Tilted suspension of tube radiant heaters*

The same principles must be kept also for tilted suspension. (Figure 5.31). Even for the heaters' types with deep reflector the maximum angle of suspension is $\delta$ = 30°. Heaters with shallow reflector should be suspended tilted at all. It is recommended to suspend the heater so to keep the warmer "burner tube" upper. This is the way how the amount of wasted radiation energy can be reduced. To conclude, it is not optimal anyway to suspend tube radiant heaters tilted, but in cases when it is necessary to do so, it is recommended to use heaters with deeper reflector and with warmer tube in higher position.

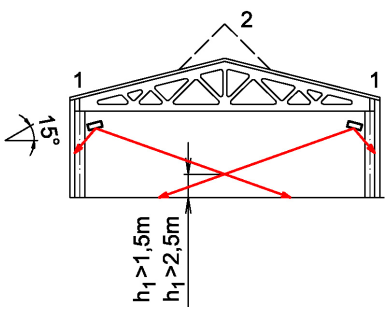



***Figure 5.31*** *Location of tilted heaters (1) case without skylight α = 114°, suspension angle γ = 30°, $h_1$ = 1.5 m. (2) case with skylight $h_1$ = 2.5 m.*

### *Safety conditions*

Tube radiant heaters are heating devices with high surface temperature (up to 500°C). Therefore it is necessary to avoid high irradiation of flammable materials in their neighborhood.

Recommended minimal distances between tube radiant heaters and their surrounding are presented in Figure 5.32.

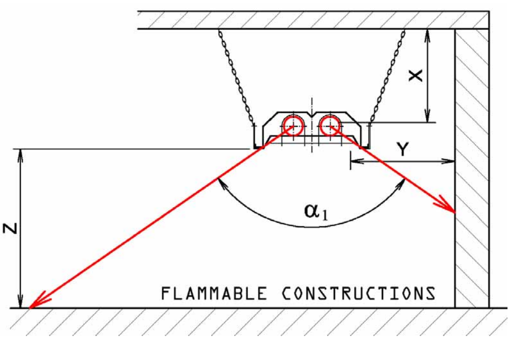



***Figure 5.32*** *Minimal distances between tube radiant heaters and their surroundings.*

| | |
|---|---|
| X = 800 mm | horizontal suspension |
| X = 1200 mm | tilted suspension |
| Y = 1000 mm | distance from vertical surfaces |
| Z = 2000 mm | flammable materials in core radiation area |

Safe clearance of flammable constructions from non-insulated parts of chimney is 1000 mm.

Safe clearance of electro-wires (cables), where surface temperature shouldn't exceed 35°C, is 1500 mm.

- inside core radiation area 1500 mm
- outside core radiation area 900 mm

For smaller distances it necessary to cover cables with metal (optimally reflective) sheet Figure 5.33.

Often it is needed to place the infrared heater above the crain. There cables can be also very close to the heater's active surface. It is recommended in such cases place the reflective sheet right on the crane to the most critical place and avoid overheating this way. The cover should be attached with air gap (at least 20 mm) to minimize conduction. Moreover the sheet should exceed core radiation angle area at least of 100 mm (Figure 5.34).

All other constructions that could be harmed by overheating can be covered by reflective sheet likely previous cases.

### *Recommended minimal distances of tube heaters – hygienic view*

From the hygienic point of view (thermal comfort of occupants), it is recommended to keep minimal suspension heights that are given by manufacturers. Table 5.1 can serve for basic orientation.

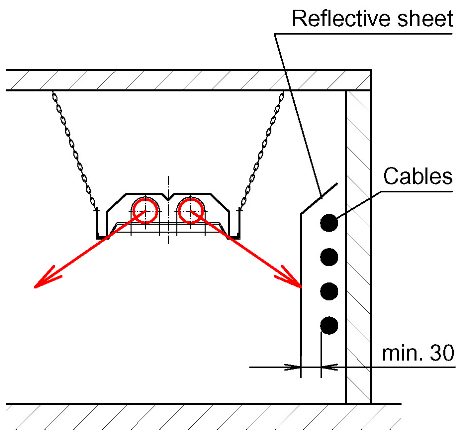



***Figure 5.33*** *Cable guard against overheating.*

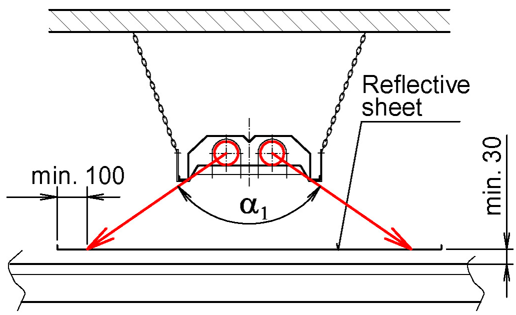



***Figure 5.34*** *Crain construction guard against overheating.*

***Table 5.1*** *Recommended minimal suspension heights of tube radiant heaters – overall heating.*

| Power output [kW] | Suspension height *h* [m] | | |
|---|---|---|---|
| | Suspension angle [°] | | |
| | 0° | 15° | 30° |
| 12 | 4.2 | 4.0 | 4.0 |
| 17 | 4.3 | 4.0 | 4.0 |
| 24 | 4.5 | 4.3 | 4.0 |
| 30 | 4.7 | 4.5 | 4.2 |
| 36 | 4.7 | 4.5 | 4.2 |
| 45 | 4.9 | 4.7 | 4.5 |

### *Plaque radiant heaters*

The same rules valid for tube radiant heaters can be used for plaque ones. The only differences are in core radiation angles (Figure 5.35, 5.36). At plaque radiant heaters tilted suspension is utilized much more because convection losses doesn´t vary that much when suitable heater construction is chosen. Some manufacturers offer special reflector construction for tilted suspension.

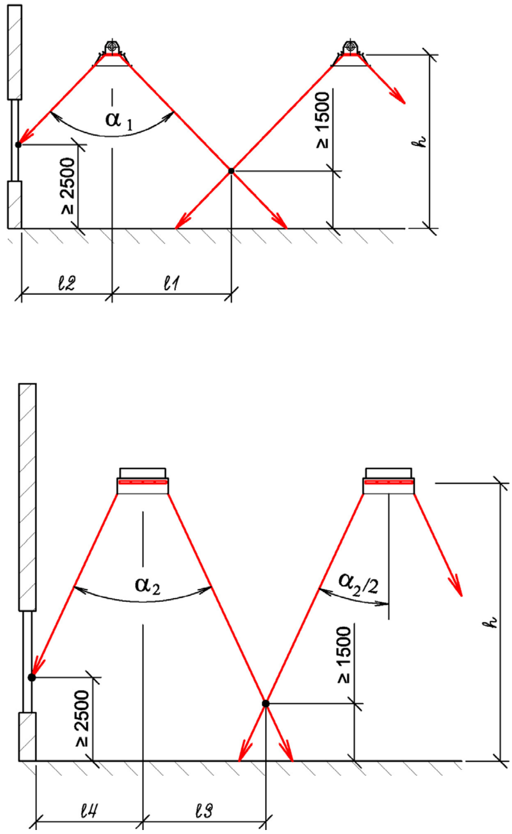



**Figure 5.35** *Optimal distances between plaque radiant heaters and between a plaque radiant heater and a wall in cross direction (top) and along the heater (below) - horizontal suspension*

### *Safety conditions*

Plaque radiant heaters are heating devices with very high surface temperature (up to 900°C). Therefore it is necessary to avoid high irradiation of flammable materials in their neighborhood.

Safe distances between heater's active surface and flammable construction depending on power output are shown in on Figure 5.37 and in Table 5.2. The distance ensures that surface temperature does not exceed 85°C. In case of there is a crain construction below the heater it can be guarded similarly as in case of tube radiant heaters (Figure 5.38).

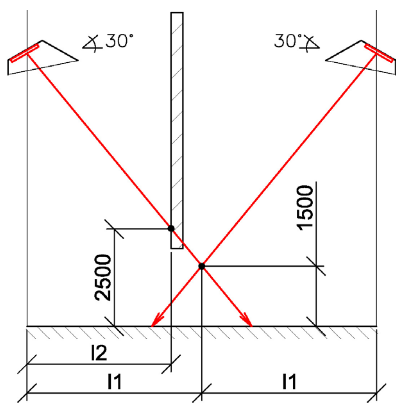



**Figure 5.36** *Optimal distances between plaque radiant heaters and between plaque radiant heater and a wall - tilted suspension*

**Table 5.2** *Safe distances from flammable surfaces.*

| Power output [kW] | Safe distances [m] | | |
|---|---|---|---|
| | X | Y | Z |
| 7 | 1.0 | 1.0 | 2.3 |
| 11 | 1.0 | 1.0 | 2.4 |
| 15 | 1.2 | 1.0 | 2.6 |
| 18 | 1.3 | 1.2 | 2.6 |
| 25 | 1.4 | 1.3 | 2.7 |
| 36 | 1.6 | 1.3 | 2.8 |
| 43 | 1.8 | 1.4 | 3.0 |

All other constructions that could be harmed by overheating can be covered by reflective sheet likely previous cases and tube radiant heaters. The reflective cover should always be connected with air gap to allow air convection Figure 5.39.

**Recommended minimal distances of plaque heaters – hygienic view**

From the hygienic point of view (thermal comfort of occupants), it is recommended to keep minimal suspension heights that are given by manufacturers. Table 5.3 can serve for basic orientation.

*Evaluation of design heat output*

The first step is determination of approximate location of radiant heaters in space. Input parameters are core radiation angle (in first step average value is chosen), operation requirements (reaction time) and real suspension possibilities (ceiling construction, crane, technology etc.). When location draft is done, heaters are divided in zones with similar thermal conditions and/or with the same heater's classes. It is obvious that a zone with external wall, cooled windows and gates will not have the same heat demand as an inner zone (Figure 5.40). Besides heaters belonging to one zone should be suspended and operated the same way (horizontal x tilted suspension, number of shafts, control loop) To conclude, first locate heaters within the space, divide the space into zones with similar thermal behaviour and these further more into zones with the same heater's classes (Figure 5.40 and 5.41).

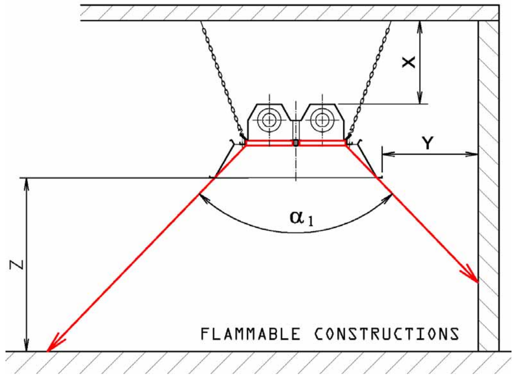



***Figure 5.37*** *Minimal distances of plaque radiant heaters from flammable constructions.*

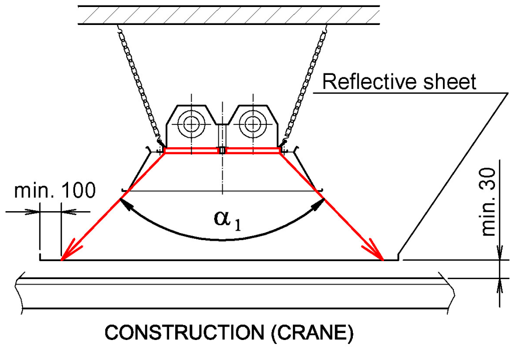



***Figure 5.38*** *Construction overheating guard.*

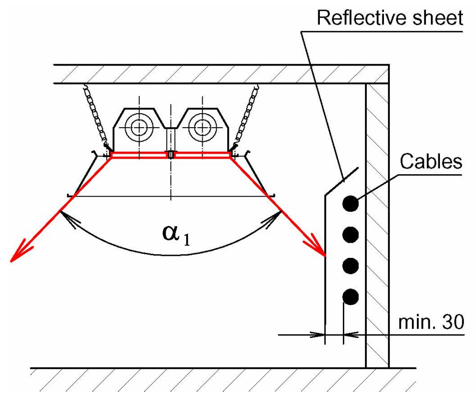



***Figure 5.39*** *Cable guard against overheating.*

***Table 5.3*** *Recommended minimal suspension heights of plaque radiant heaters – overall heating.*

| Suspension height | Type of heater | Suspension angle | Heat output of nominal heater [kW] | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| | | | 7 | 11 | 15 | 18 | 25 | 36 | 43 |
| *h* [m] | Horizontal | 0° | 4.6 | 5.3 | 5.8 | 6.1 | 7.3 | 7.7 | 8.8 |
| | Tilted | 30° | 4.2 | 4.8 | 5.3 | 5.5 | 6.6 | 6.9 | 7.9 |

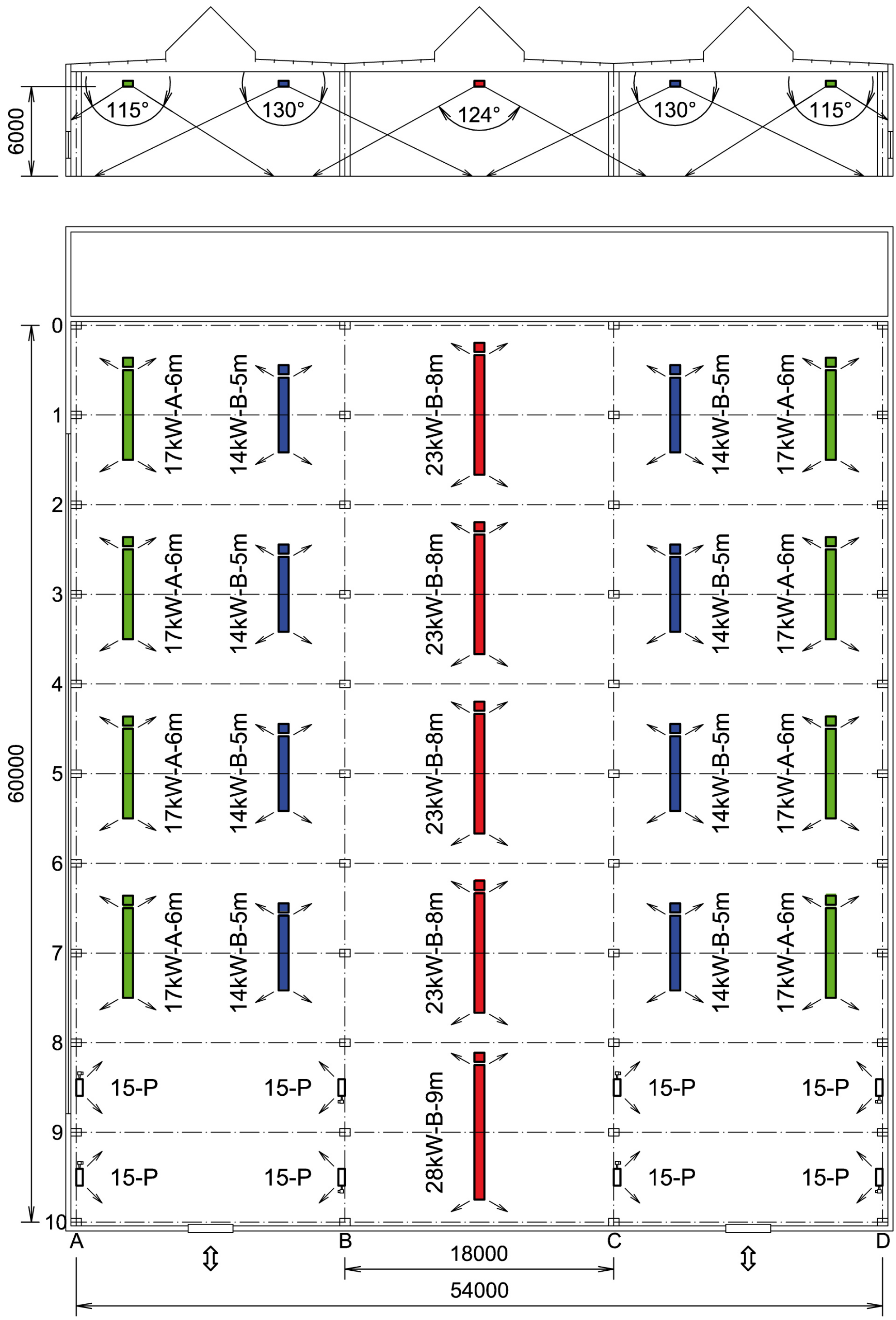



***Figure 5.40*** *An example of tube radiant heaters' location in three-bay industrial hall, height of 6 m. red – 28, 23 kW, green – 17 kW, blue – 14 kW.*

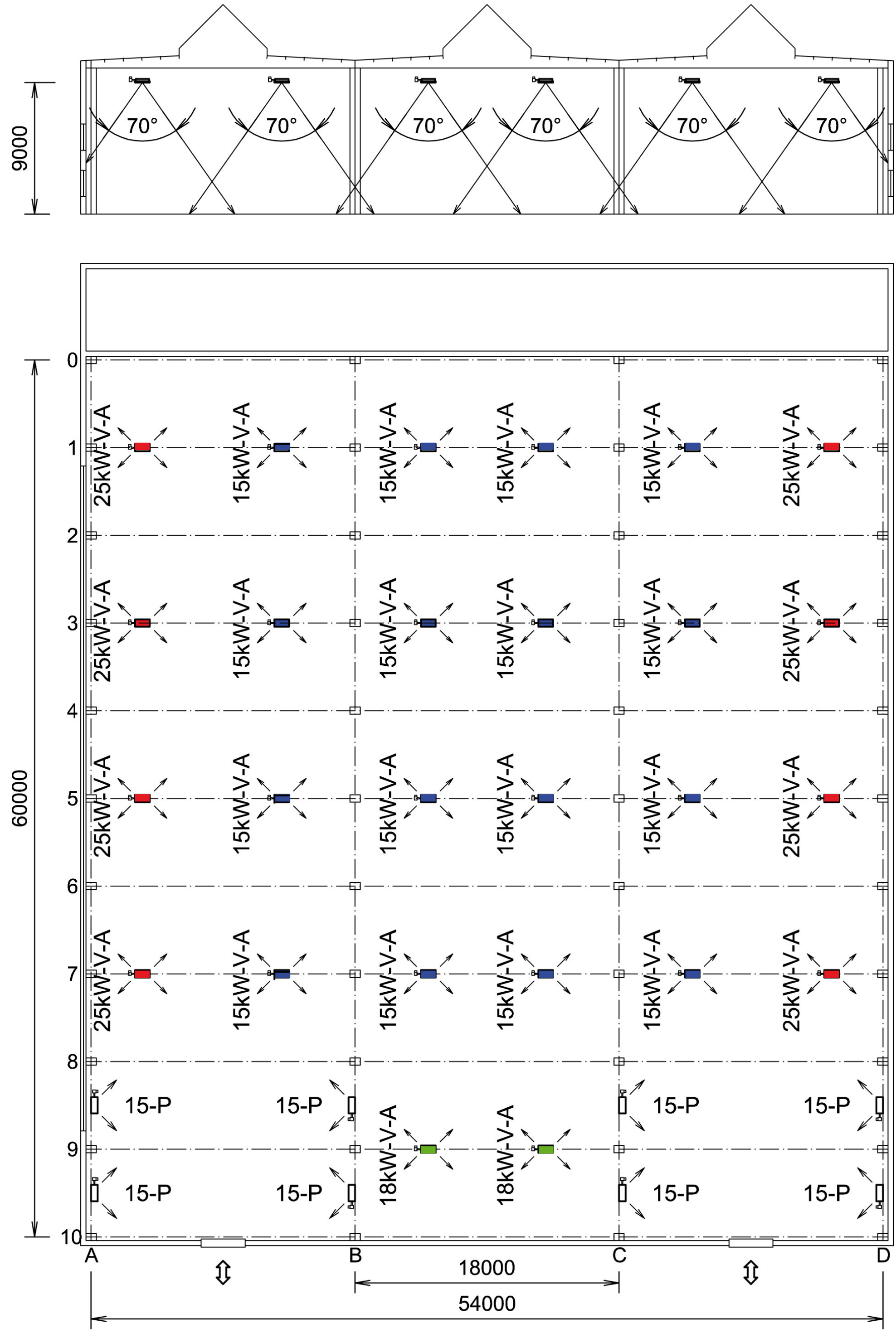



***Figure 5.41** An example of plaque radiant heaters' location in three-bay industrial hall, height of 9 m (red – 25 kW, green – 18 kW, blue – 15 kW).*

When radiant heaters are divided into zones, design heat load (heat losses) should be for each zone evaluated (for example according to EN 12 831). Obtained values $\Phi_{HL}$ are then corrected by radiant efficiency, floor area, way of suspension and a way how the exhausts are driven out.

First the dimensionless correction factor $K_S$ should be evaluated:

$$K_S = \frac{\eta_L \cdot c_S \cdot \alpha \cdot \varphi}{H_m \cdot c_L \eta_F \varphi_m (\varphi - 1)} \qquad (5.1)$$

where:

$\eta_L$ = radiant device's efficiency (Table 5.4) [-];

$c_S$ = specific heat for heating device (Figure 5.42) [W/(m²K)];

$\alpha$ = absorption coefficient (dust, $H_2O$, $CO_2$) of environment depending on suspension height (Figure 5.43) [-];

$H_m$= mean height of the space (total volume of the hall / total floor surface) [m];

$c_L$ = specific heat depending for heated space [W/(m²K)];

$$c_L = \frac{\Phi_{HL}}{V_R \cdot (\theta_i - \theta_{ev})}$$

where:

$\Phi_{HL}$ = design heat load (for ex. EN 12 831) [W];

$V_R$ = room volume [m³];

$\theta I$ = internal air design temperature (EN 12831 – national annex) [°C];

$\theta_{ev}$ external design temperature (EN 12831 – national annex) [°C];

$\eta_F$ = rate of floor coverage (Figure 5.44 a 5.45) [-];

$\varphi$ = characteristic radiation coefficient according to EN 419-1,2; EN 416-1 can be obtained from radiant efficiency as $\varphi = 1/(1-\eta_S)$ [-];

$\varphi_m$ = mean view factor between a heater and a person in occupancy zone (infrared heater suspended horizontaly 0.40; infrared heater suspended tilted 30…45° - 0.70) [-].

Next step: evaluation of actual air temperature.

$$\theta_L = \frac{\theta_i \cdot K_S + \theta_{ev}}{1 + K_S} \text{ [°C]} \qquad (5.2)$$

Total required nominal heat output is

$$\Phi_{HL,O} = V_R \cdot c_L \cdot (\theta_L - \theta_{ev}) \text{ [W]}$$

and hence nominal heat input:

$$\Phi_{HL,1} = \frac{\Phi_{HL,0}}{\eta_L} \qquad \text{[W].}$$

***Table 5.4** Radiant device's efficiency related to considered space.*

| | $\eta_L$ [-] |
|---|---|
| Tube radiant heater, direct exhausts removal by a chimney (exhausts' temperature 180°C) | 0.85 |
| Tube radiant heater, direct exhausts removal by a chimney (exhausts' temperature >180°C) | 0.70 |
| Plaque radiant heaters, free installation, indirect exhausts removal | 0.95 |
| Plaque radiant heaters equipped with exhausts' hood, direct exhausts removal, open combustion chamber | 0.60…0.73 |
| Plaque radiant heaters equipped with exhausts' hood, direct exhausts removal, delta combustion chamber, insulated | 0.70…0.82 |

***Table 5.5** Radiant efficiency $\eta_S$.*

| | horizontal | tilted 30° |
|---|---|---|
| Tube radiant heater with deep reflector, non-insulated (Figure 7.5) | 0.63 | 0.58 |
| Tube radiant heater with deep reflector, insulated (Figure 7.6) | 0.72 | 0.67 |
| Plaque radiant heater with open chamber (Figure 7.18) | 0.58 | 0.55 |
| Plaque radiant heater with delta chamber, non-insulated (Figure 7.21) | 0.67 | 0.62 |
| Plaque radiant heater with delta chamber, insulated, grid (Figure 7.22) | 0.82 | 0.78 |

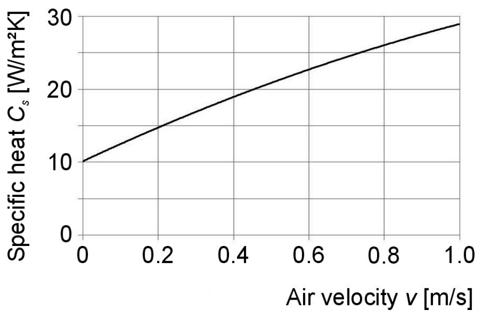



***Figure 5.42** Relation between specific heat demand $C_s$ and air velocity v in considered space.*

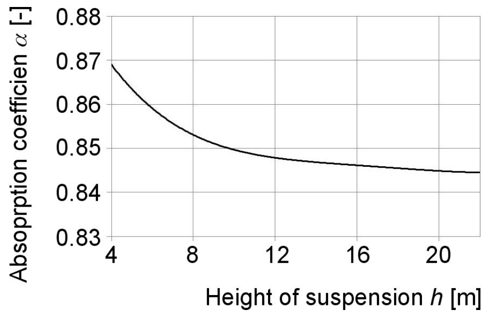



***Figure 5.43** Relation between environment absorption $\alpha$ and suspension height h.*

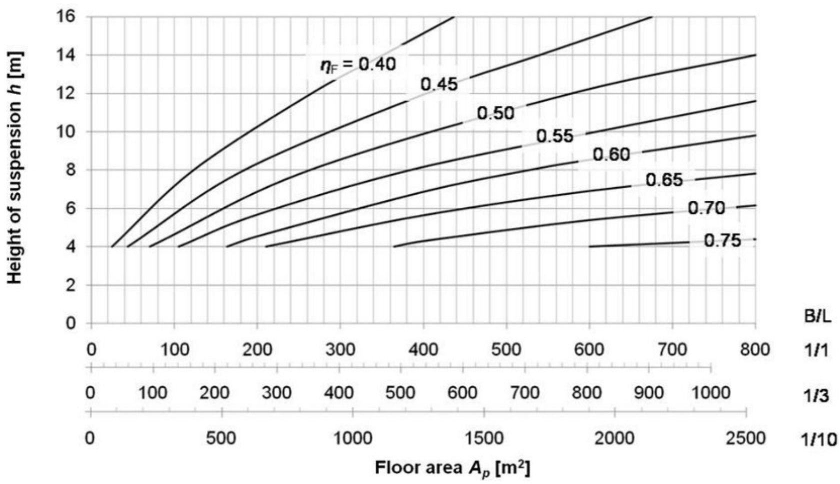



***Figure 5.44** Relation between rate of floor coverage $\eta_F$, floor surface area $A_p$ and height of suspension h for smaller buildings.*

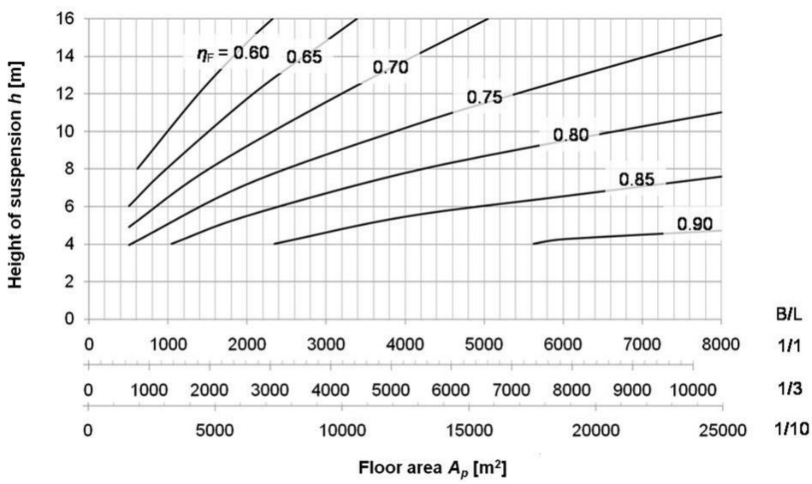



***Figure 5.45** Relation between rate of floor coverage $\eta_F$, floor surface area $A_p$ and height of suspension h for larger buildings.*

Finally, after evaluation of required summary power output of heaters in each zone, required power output for one heater is obtained by simple dividing of summary value by considered number of heaters in particular zone. Now, from the manufacturers' brochures the heater with nearest higher power output is chosen. In the last step it is necessary to check again core radiation angles (this time exact values according to the brochures), suspension possibilities, safety conditions, minimal hygienic suspension heights, possible chimney placement, etc. and if anything would change, correct particular part in design procedure. (Table 5.4 and 5.5)

#### Control and operation

During the heating season outside temperatures are the most of the year higher that design outside temperature the heating system is designed for. A goal for good quality control is to limit temporary not required power output. Control system is another aspect that significantly influences annual energy consumption. Corect placement of sensors and allocation of heaters into control loops plays therefore a keyrole.

A control loop Figure 5.46 consists of a controller, a sensor (3), and radiant heaters (1,2) assigned to the loop from which one or two (1) must be chosen as "reference" for operation. Control sensor should be placed on a place where it cannot be influenced both in winter by cold draft and in spring/autum by direct sun radiation. Also the sensor should be placed in core radiation area of the chosen reference radiant heater. Operation of the zone is fully connected with operation of the reference infrared heater(s). While the other heaters from the zone can be manually swiched of (disconnected from the zone's automatic control), the reference heater must remain in operation. If reference heater is not working whole zone is schwitched of. For radiant heating systems recommended sensor is a sort of globe thermometer Figure 5.47 (measuring globe temperature). On the other hand for zones with external gates there are recommended standard air temperature sensors (termostats) to be able to faster react on changes caused by for example opening of a gate. On the controller it is possible to set all the standard features, time program, night temperature reduction, antifreeze control, etc.

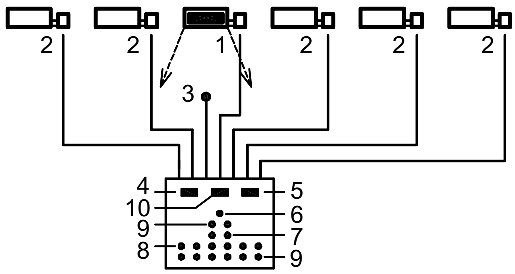



***Figure 5.46*** *Control loop scheme 1 – reference heater, 2 – switchable heaters, 3 – sensor in core radiation angle, 4 – switches AUTO – MAN - OFF, 5 – deblocation buttons.*

***Figure 5.47*** *Globe sensor for radiant heaters.*

The heaters' allocation into control loops will come out from heat demands of particular parts of a heated space (number of workplaces, operation time of workplaces, heat load). Therefore it is necessary to get detail information about operation, resp. time schedule of workplaces (number of shafts). During operation time it is recommended to switch of heaters that are at present not needed. This way can be achieved significant savings. This is the main advantage of decentralized system.

The control itself can be either standard - wired (Figure 5.48) or wireless (Figure 5.49), where there is on every heater a receiver and on the controller a transmitter.

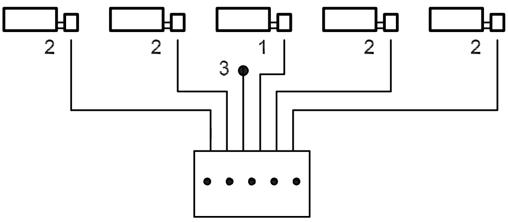



***Figure 5.48*** *Standard control loop.*

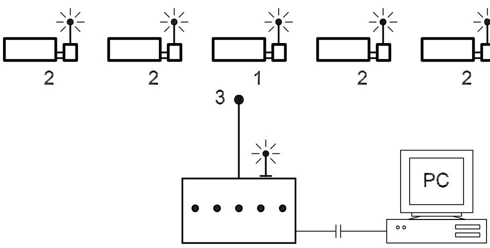



***Figure 5.49*** *Wireless control loop.*

## Electrical connection

The connection is influenced by type of the heater (controller) and also availability of beblocation function. The deblocation function is recommended for tube radiant heaters for safety reasons. In case of any malfunction the staff is needed to set the heater after check again manually into operation. Therefore these heaters are connected by cable type CYKY 7C x 1.5. Regarding a need to control in case of plaque radiant heaters a venting fan it is necessary to equip the control box with suitable equipment. It is recommended to set the fan into operation with operation of the zone. According to EN 13410 there are other possibilities how the exhausts can be driven away and fresh air supplied but the mechanical ventilation is the most reliable. Plaque radiant heaters are connected by cable type CYKY 3C x 1.5. Supplied voltage is mostly 230V, 50 Hz. Electrical power input varies up to 100 W/pcs.

### Exhausts venting and combustion air intake

#### *Tube radiant heaters*

In tube radiant heaters systems exhausts venting is done by standard flue gas ducting. The ducts can be led either through a wall or through a roof. Detail solution is presented on Figure 5.50.

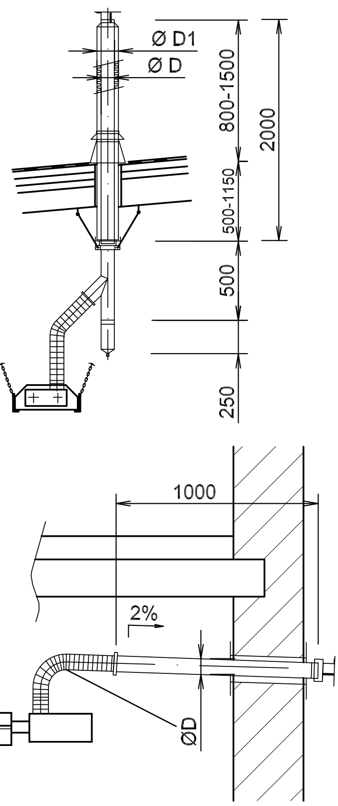



***Figure 5.50*** *Exhausts venting via roof (top) and via wall (below).*

Combustion air intake is done either directly from the heated space without any ducts or from the outside. The whole system exhaust outflow/combustion air intake can be also done by “turbo” system where two concentric tubes are used, inside for flue gas exhaust and outside for combustion air intake.

In cases where it is necessary to led combustion air from the outside (dusty environment, pressure problems), intake ducting is installed according to Figure 5.51.

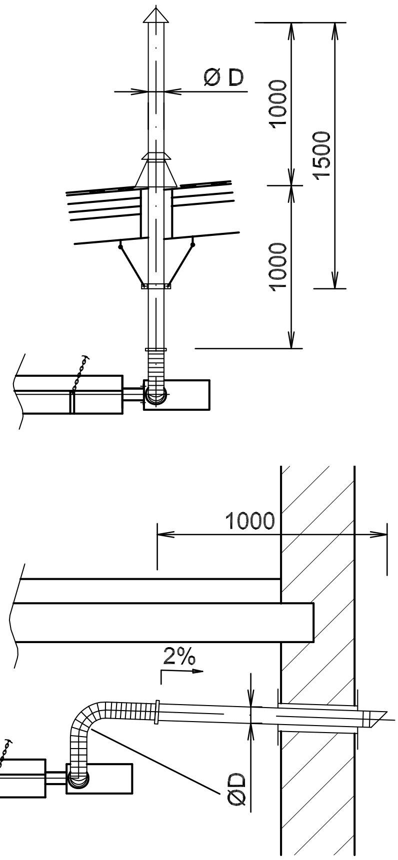



***Figure 5.51*** *Combustion air intake for tube radiant heaters.*

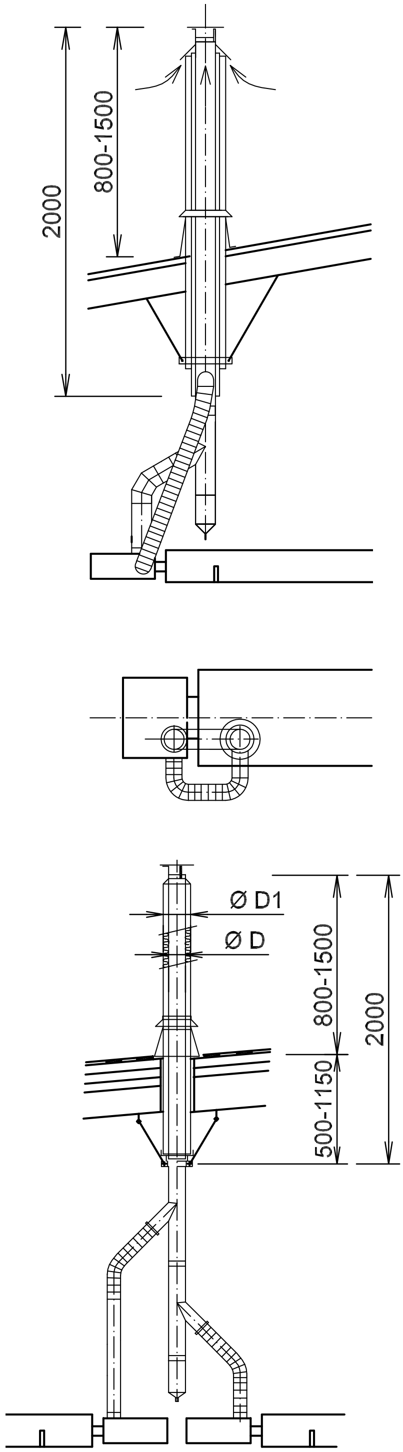



***Figure 5.52*** *Common exhaust venting for two heaters (top), “turbo” venting system (below).*

***Plaque radiant heaters***

Design of combustion air intake and exhausts venting is standardized according to EN 13410. Basic condition that must be kept is requirement on minimal space volume where plaque radiant heaters are installed 10 m³/1 kW of installed heat power input. Exhausts must be driven away from the space where radiant heaters are installed to the outside of the building. At plaque radiant heaters (A type device), exhaust can be driven away indirect. First exhausts are mixed with ambient air and driven away just after.

Venting can be achieved by one of the following principles: (1) venting of mixture (exhausts with ambient air) by convection of heated air; (2) venting of mixture (exhausts with ambient air) by forced air-exchange; (3) venting of mixture (exhausts with ambient air) by natural air-exchange.

**(1) Venting by convection of heated air**

Venting by convection is sufficient when 10 m³/h of flue gas / ambient air mixture per 1 kW of in-operation heat power input is driven away. The mixture must be withdrawn from above the heaters as close as possible to their top edge, by means of openings designed for air / mixture venting. The openings must be constructed and located to avoid misoperation by wind influence. Application of closing and flow limitation equipment is possible when self working device ensures opening whenever heaters will be in operation. Number and location of openings is related to location of heaters and construction of considered space itself. Horizontal distance between a heater and an opening musn't be, in case of wall openings, higher than six times height of the opening (measured to the middle of the opening). In case of roof openings the distance mustn't exceed three times height of the opening (measured to the middle of the opening).

**(2) Venting by forced air-exchange**

Venting by forced ventilation is sufficient when 10 m³/h of flue gas / ambient air mixture per 1 kW of in-operation heat power input is driven away. The mixture must be withdrawn from above the heaters by mechanically by fans. Only fans with steep operation characteristic must be used. Operation of heaters must be conditioned by operation of fans. Horizontal distance between a heater and a fan musn't be, in case of wall installation, higher than six times height of the fan installation (measured to the middle of the fan). In case of roof fans the distance mustn't exceed three times height of the installation (measured to the middle of the fan).

**(3) Venting by natural air-exchange**

Plaque radiant heaters can be operated without any special venting system only if the flue gas/air mixture is driven away to the outside of the building by sufficient existing air-exchange in the building.

Venting by heated air convection and mechanical air-exchange is not necessary:

a) at buildings with higher air-exchange rate than 1.5 times air volume of considered space per hour;
b) at buildings with maximal heat power in operation 5 W/m³ (related to total volume of space).

## Air intake

*Evaluation of free opening area inside spaceous building*

Required amount of air intake will be determined according to installed heat power output of heaters above a surface where the air intake from windows, gates and doors cannot be taken into account. Area of influence of these parts is given by their dimensions. The vented area at gates varies from 18 up to 24 m, at external walls with window varies from 12 up to 18 m. From their installed heat power output then:

$$M_v = Q_i^v \cdot M_1 \quad [\mathrm{m^3/h}] \qquad (5.3)$$

where:

$M_v$ = the air flow rate from external environment [m³/h];

$Q_i^v$ = installed heat power output above “inner” building’s surface [kW$_i$];

$M_1$ = the air flow rate per installed kW [m³/h/kW$_i$]

$M_1$ = 10 m³/h/kW$_i$

Free surface of inlet openings:

$$S = \frac{M_v}{v} \quad [\mathrm{m^2}] \qquad (5.4)$$

where $v$ = 1.2 m/s (velocity in inlet ducts $v_1$ = 1.3…1.5 m/s).

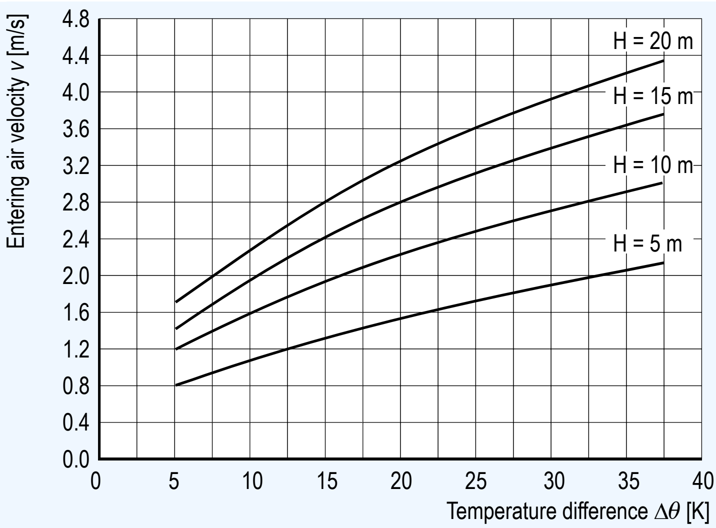
Inlet openings are located in horizontal way if possible as far from heaters as possible – the best location is between heaters. Verticaly the opening should be minimally 150 mm below heater’s installation plane. In case of tilted suspension inlet openings are designed to the plane tie’s lower edge, ideally above pathways.

*Evaluation of ventilation openings for natural convection of heated air*

The amount of exhaust air is given by:

$$M_{celk} = Q_{is} \cdot M_1 \quad [\mathrm{m^3/h}] \qquad (5.5)$$

Air velocity [m/s] through a opening can be read from Figure 5.53. The figure is valid for free opening without any ducts or any embedded return channels.




$H_1$ = installation height of outlet opening from the floor or height between inlet and outlet opening [m];

$v$ = air velocity in the opening [m/s];

$\Delta\theta_1$ = temperature difference $\theta_i$ - $\theta_{ev}$ [K];

$\theta_i$ = internal design air temperature [°C];

$\theta_{ev}$ = external design temperature for ventilation $\theta_{ev} = \theta_e - 8$ K (when radiant system is used, draft influences thermal comfort much more than in convective systems – safety addon) [°C];

$\theta_e$ = local design temperature [°C].

***Figure 5.53*** *Determination of air velocity v within a free opening in relation with height of the opening.*

Free surface area of outlet opening can be obtained from following equation:

$$S_1 = \frac{M_{sum}}{v \cdot 3600 \cdot n_1} \quad [\mathrm{m}^2] \qquad (5.6)$$

Where $S$ is the free surface area of one outlet opening [m²]; $M_{sum}$ is the summary amount of outlet air [m³/h]; $v$ is the outgoing velocity [m/s] and $n_1$ is the number of openings for outlet air withdrawal [-].

### Design of gas piping

As an input to this part of design process is information from a manufacturer Table 5.6 and 5.7 (desired overpressure, gas consumption, connection dimension etc.)

Dimensioning and designing of gas piping must be in agreement with operation regime of the system. In case the devices are connected individualy, it is possible to design whole piping traditionally with various piping branches. On the other hand when there occurs a case with larger amount of devices assigned to one control loop there can be a problem with insufficient pressure at the end of piping when all the heaters would switch on at once. It is recommended for such cases to design so called "gas piping frame". The backbone of the frame is designed for higher gas flow rate which solves the problem with insufficient gas Figure 5.54 and 7.55. Figure 7.54 shows frame construction for a three-bay industrial hall where operation costs can be billed in a sum. Hence there frame piping is considered. On the other hand on the Figure 7.55 there was a requirement to measure gas consumption of every bay separately and therefore branch piping must be used instead. Larger dimensions must be considered in this case.

***Table 5.6** Example of requirements on gas supply piping for tube radiant heaters.*

| | | | |
|---|---|---|---|
| Nominal overpressure at device's inlet | NG | kPa | 2…5 |
| | PB | | 3…5 |
| Connection dimension | G ¾“ | | |
| Electric protection | IP 40 | | |
| Nominal voltage [V/Hz] | 230 / 50 | | |
| Electric power input [W] | 10…30 kW | | 50 W |
| | 36, 45 kW | | 75 W |
| Maximal noise level 1m from a heater [dB(A)] | 55 | | |

***Table 5.7** Example of requirements on gas supply piping for plaque radiant heaters.*

| Characteristics | | Units | Type | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| Heat power output | | kW | 7 | 11 | 15 | 18 | 25 | 36 | 43 |
| Minimal connection overpressure | Natural gas | kPa | 2.0 | | | | | | |
| Maximal connection overpressure | | kPa | 5.0 | | | | | | |
| Gas consumption | | m³/h | 0.72 | 1.08 | 1.44 | 1.80 | 2.52 | 3.60 | 4.32 |
| Connection of gas hose $l$ = 750 mm | | DN | ½“ | | | | | | |
| Gas piping connection | | DN | ¾“ | | | | | | |

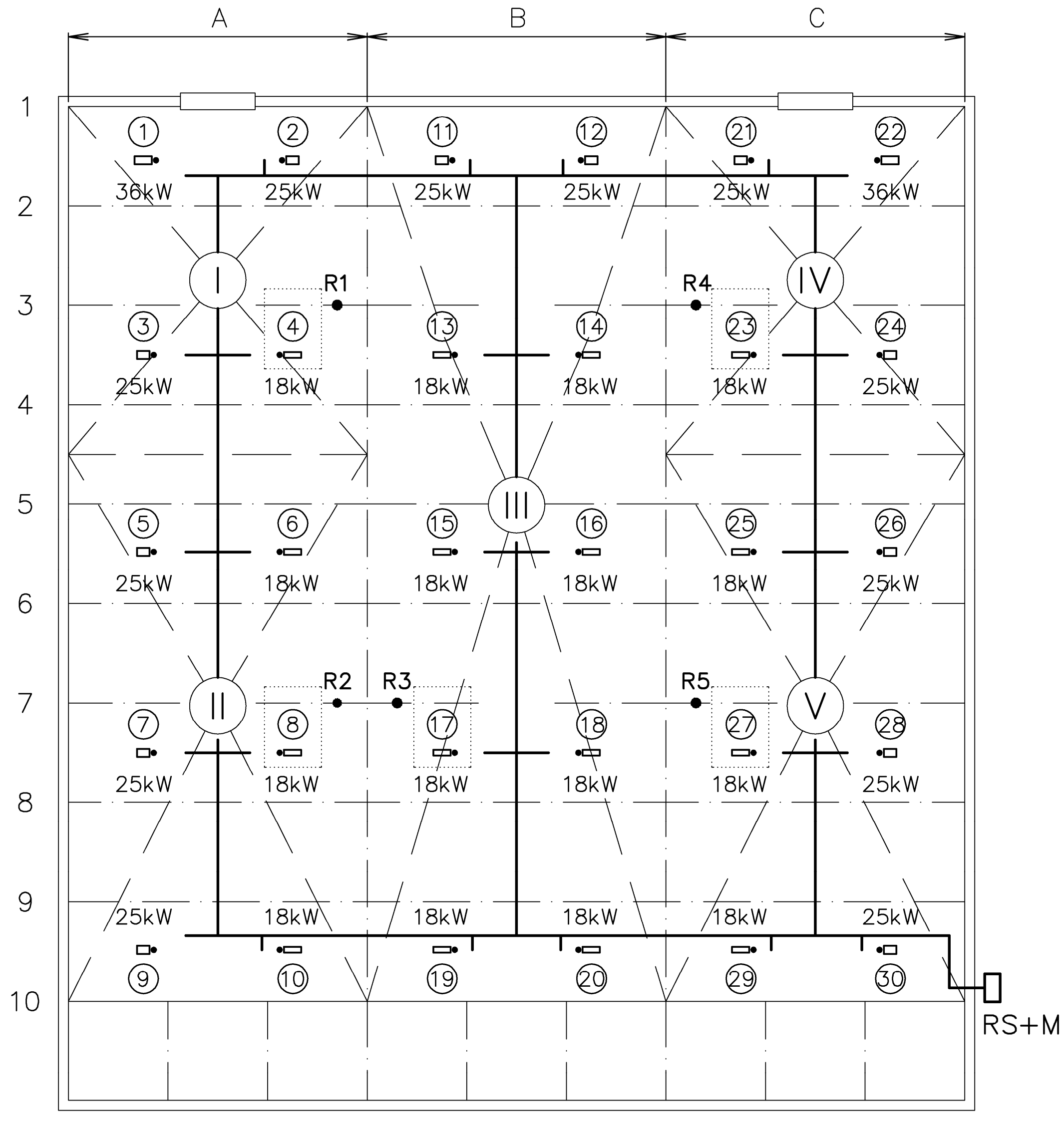



*Figure 5.54* Gas piping in building where operation costs are billed as sum for whole building RS – regulation (pressure) station, M – measurement place. Control loops: R1 – heaters 1…4 (R1); R2 – heaters 5…10 (R2); R3 – heaters 11…20 (R3); R4 – heaters 21…24 (R4); R5 – heaters 25…30 (R5).

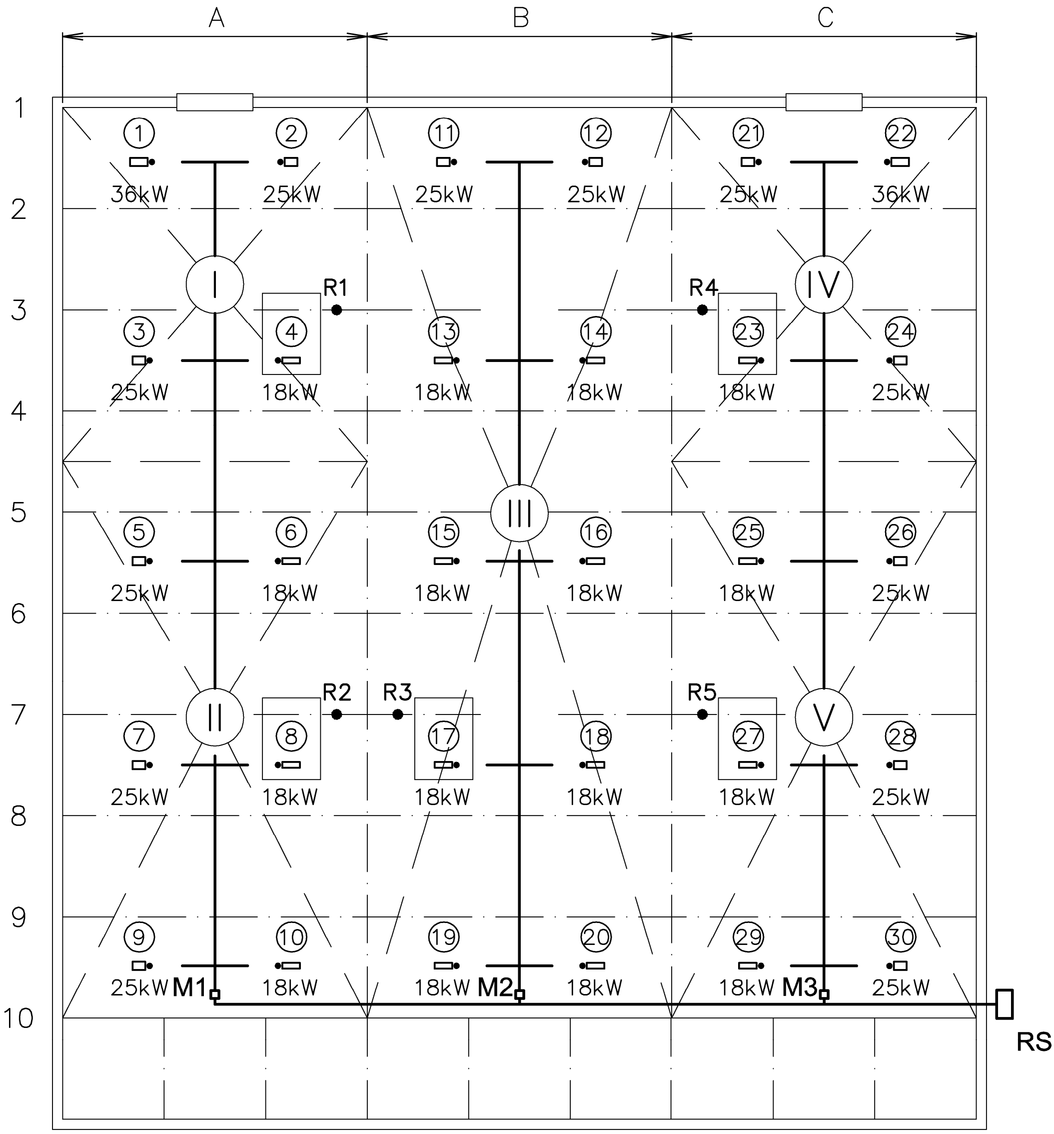



***Figure 5.55*** *Gas piping in building where operation costs are billed partially for every bay RS – regulation (pressure) station, M – measurement place. Control loops: R1 – heaters 1…4 (R1); R2 – heaters 5…10 (R2); R3 – heaters 11…20 (R3); R4 – heaters 21…24 (R4); R5 – heaters 25…30 (R5).*

Here, plaque radiant heaters were chosen but the same principles should be kept when tube radiant heaters are considered.

The recommended design procedure is following:

1. According to location of heaters, initial pipeline path is designed;
2. Gas pipeline is divided into sections with constant design volume flow rate (nod points are T-points or vertical parts);
3. Volume flow rates in all sections must be evaluated;
4. Number of the same class devices in section is determined, coefficient of concurrence is evaluated and reduced volume flow rate Vr for all the sections is calculated; all the pipeline sections are designed for particular reduced flow rate.

$$\dot{V}_r = k_1 \cdot \dot{V}_1 + k_2 \cdot \dot{V}_2 + k_3 \cdot \dot{V}_3 \qquad (5.7)$$

Where $V_1$ [m³/h] is the summary gas consumption of devices connected for cooking and heating of water in particular pipeline section;

$$k_1 = \frac{1}{\log(10 \cdot n)} \qquad (5.8)$$

Where $n$ is the number of devices; $V_2$ is the summary consumption of local heaters of particular pipeline section [m³/h];

$n = 1$ or $2$ $\quad k_2 = 1.0$
$n \geq 3$ $\quad k_2 = 0.8$

$V_3$ is the summary consumption of boilers of particular pipeline section [m³/h];

$n = 1$ or $2$ $\quad k_3 = 1.0$
$n \geq 3$ $\quad k_3 = 0.8$

5. Length of the section is measured $l$
6. Ekvivalent length of local pressure losses is evaluated
   $$l_x = 0.5 \cdot l \qquad (5.9)$$
7. Total length is calculated
   $$L = l + l_x = 1.5 \cdot l \qquad (5.10)$$
8. Preliminary pressure loss is obtained
   $$R = \frac{\Delta p}{L} \; [Pa/m] \qquad (5.11)$$

According to Table 5.8 for specific pressure losses and volume flow rates, an adequate pipeline section dimension is chosen and $\Delta p$ is controlled. At vertical pipeline $\Delta p$ must not be determined, for reduced consumption $V_r$ exceeds buoyancy pressure. In some cases, when some parts of pipeline go through boundaries of fire sections, it is mandatory (instead of fire resistant sealing) to lead the pipeline to an accessible height (1.5 m) and there equip the pipeline with a closing valve. Moreover, one fire section has to be closable from one place and at accessible height.

### Heating of separate workplaces

Heating of separate workplaces in unheated buildings belongs to the most difficult solutions in HVAC. It is important to eliminate the influence of surrounding cold environment with increased radiant intensity. For such cases the most suitable heating system is plaque infrared heaters because of their high intensity and directional properties. Ideal suspension for such cases is tilted with special reflector assembled at workplace perimeter to irradiate mainly backs of occupants. For such case tube radiant heaters are totally unsuitable (small radiation intensity, large irradiation spread).

***Table 5.8** Specific pressure losses and natural gas volume flow rates for specific pipelines' dimensions.*

| DN | Specific pressure loss $\Delta p$ [Pa] for 1 m long pipeline | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Natural gas volume flow rate [m³/h] | | | | | | | | | | | | | |
| | 20 | 10 | 8 | 5 | 4 | 3 | 2 | 1 | 0.667 | 0.5 | 0.4 | 0.33 | 0.25 | 0.2 |
| **10** | 1.3 | 0.9 | 0.8 | 0.7 | 0.6 | | | | | | | | | |
| **12** | 2.1 | 1.5 | 1.3 | 1.0 | 0.9 | 0.8 | | | | | | | | |
| **15** | 3.6 | 2.6 | 2.3 | 1.8 | 1.6 | 1.4 | 1.1 | | | | | | | |
| **20** | 7.4 | 5.2 | 4.7 | 3.7 | 3.3 | 2.9 | 2.3 | 1.7 | 1.3 | 1.2 | 1.1 | 1.0 | 0.8 | 0.7 |
| **25** | 13.0 | 9.2 | 8.2 | 6.5 | 5.8 | 5.0 | 4.1 | 2.9 | 2.4 | 2.1 | 1.8 | 1.7 | 1.5 | 1.3 |
| **32** | 24.0 | 17.0 | 15.2 | 12.0 | 10.7 | 9.3 | 7.6 | 5.4 | 4.4 | 3.8 | 3.4 | 3.0 | 2.7 | 2.4 |
| **40** | 42.0 | 29.7 | 26.5 | 21.0 | 18.8 | 16.2 | 13.3 | 9.4 | 7.7 | 6.6 | 5.9 | 5.4 | 4.7 | 4.2 |
| **50** | 73.3 | 51.8 | 46.3 | 36.6 | 32.8 | 28.4 | 23.2 | 16.4 | 13.4 | 11.6 | 10.4 | 9.4 | 8.2 | 7.3 |
| **60** | 116.0 | 81.7 | 73.1 | 57.8 | 51.7 | 44.8 | 36.6 | 25.8 | 21.1 | 18.3 | 16.3 | 14.8 | 12.9 | 11.6 |
| **70** | 170.0 | 120.0 | 107.0 | 85.5 | 76.0 | 65.8 | 53.7 | 38.0 | 31.0 | 26.9 | 24.0 | 21.8 | 19.0 | 17.0 |
| **80** | 237.0 | 168.0 | 150.0 | 119.0 | 106.0 | 91.9 | 75.0 | 53.1 | 43.3 | 37.5 | 33.6 | 30.5 | 26.5 | 23.7 |
| **100** | 415.0 | 293.0 | 262.0 | 207.0 | 185.0 | 161.0 | 131.0 | 92.7 | 75.6 | 65.5 | 58.6 | 53.2 | 46.3 | 41.5 |

*Evaluation of power output of the radiant heaters*

Main goal is the same as in overall heating to achieve desired globe (operative) temperature $\theta_g$ ($\theta_o$) at occupancy zone regarding influence of air temperature $\theta_i$ of surrounding environment

General equation for evaluation of radiant heaters' power output is following:

$$\phi_{HL,I} = \frac{I_s \cdot A_p \cdot f_1}{f_4 \cdot 1000} \text{ [kW]}, \qquad (5.12)$$

Where

$$I_s = \frac{\theta_o - \theta_i}{0{,}0716} \text{ [W/m}^2\text{]} \qquad (5.13)$$

$I_s$ is the irradiation intensity; $\theta_o$ [°C] is the operative (globe) temperature; $\theta_i$ [°C] is the air temperature in surrounding unheated space; $A_p$ [m²] is the irradiated area; $f_1$ [-] is the absorption coefficient for dusty environments as a function of distance between heater and supposed occupants:

$h$ = 7.5 m  $f_1$ = 1.10;
$h$ = 9.0 m  $f_1$ = 1.20;
$h$ = 12 m  $f_1$ = 1.35;

$$f_4 = \eta_S \cdot \varphi_m \cdot a_S \text{ [-]} \qquad (5.14)$$

Where $f_4$ is the placement coefficient; $\eta_S$ [-] is the radiant efficiency of a heater (Table 5.5); $\varphi_m$ [-] is the mean view factor as a function of suspension ($\varphi$ = 0.4 (horizontal), $\varphi$ = 0.7 (tilted)) and $a_S$ [-] is the mean absorption coefficient of irradiated surfaces: $a_S$ = 0.85.

## Annual energy and gas need

Annual energy and gas need is influenced by climatic conditions, building type and contruction, operation, control strategy, etc. Energy need is evaluated from following equation:

$$Q_r = \frac{b_v \cdot \phi_{HL,O} \cdot D}{3{,}6 \cdot \Delta\theta_{max} \cdot \eta_L} \text{ [MJ]}, \qquad (5.15)$$

Where $Q_r$ [MJ] is the annual energy need for heating; $b_v$ [h/day] is the number of daily operation hours - maximall power output required to cover energy demand according to external design temperature for specific climatic zone (5 h for one shaft, 9 h for two shafts, 12 h for three shafts); $\phi_{HL,O}$ [kW] is the nominal heat output; $D$ [day·K] is the number of day-degrees during heating season ($D = d \cdot (\theta_{im} - \theta_{em})$); $\Delta\theta_{max}$ [K] is the maximal difference between indoor operative temperature $\theta_o$ (ISO EN 7730) and external design temperature $\theta_{ev.}$

$\theta_{im}$ is the mean indoor design temperature; $\theta_{em}$ is the mean outdoor design temperature during heating season– national standards; $d$ is the number of heating days during heating season.

Approximate annual gas need can be calculated from the following equation:

$$V_r = \frac{b_v \cdot \phi_{HL,O} \cdot D}{3{,}6 \cdot \Delta\theta_{max} \cdot H_u \cdot \eta_L} \tag{5.16}$$

Where $V_r$ [m³] is the annual gas need; $H_u$ [MJ/m³] is the operative specific heating value of the gas (heating value of natural gas is approx. 33.48 mJ/m³).

## Design examples: Overall heating

**Example 1:**
Design overall heating with gas burning radiant heaters of three-bay industrial hall accordning to Figure 5.56. Consider indoor design temperature $\theta_i$ = 16°C. External design temperature consider $\theta_{ev}$ = –17°C. There are no persistant heat gains. There is no crane. Gates are (4 x 3 m) just in western fasade. Windows are both in southern (30 x 3 m and 30 x 3 m), and northern fasade (66 x 3 m). On the roof of each bay there is a skylight (2 x 60 m and 1 x 24 m). Height of the hall is 10 m to the tie level. The construction is older, consider air velocity within occupancy area $v$ = 0.3 m/s. The operation should be for every bay independent. Gates are opening just once a while for material supply.

Building envelope surfaces have heat transfer coefficients according to Table 5.9.

First, optimal heater class must be chosen. Considering relatively high building height ($H$ = 10 m) and age of the building with significant air draft plaque radiant heaters are the most suitable solution.

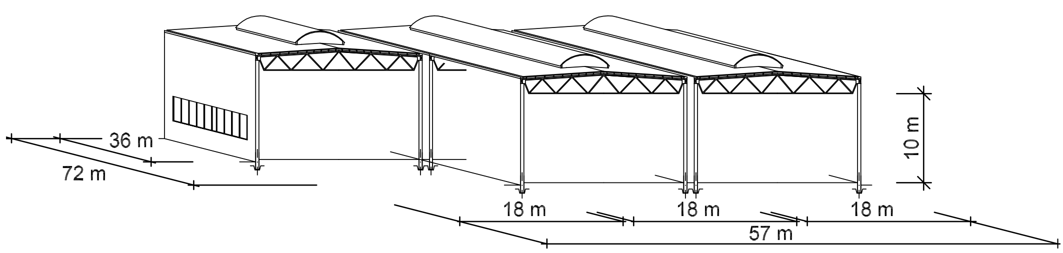



***Figure 5.56** Three bay industrial hall.*

***Table 5.9** Heat transfer coefficients.*

| Constructions | *U* [W/m²·K] |
|---|---|
| External wall | 1.10 |
| Window | 3.00 |
| Gates | 3.50 |
| Floor | 1.00 |
| Roof | 0.45 |
| Skylight | 3.50 |

*Location of radiant heaters*

Furhter on, core radiation angles or optimal heaters distances from a manufacturer should be obtained. Based on these and suspension possibilities location of heaters is prepared. For this case horizontal heaters with open combustion chamber and medium efficiency were chosen. During design process the designer should have in mind that more heaters with lower power output means higher investment and operation costs. It means that if it is possible install rather smaller amount of devices with higher power output. From the suspension point of view it is recommended to keep column distance and avoid additional suspension construction if possible. At this case, suspension height about +/- 10 m the best solution would be a combination of radiant heaters with core angles A = 88°, resp. B = 112° Figure 5.57.

Although in rear parts of building (zones 3 and 4) the coverage is not ideal, since there are no windows or gates it is possible to tolerate it.

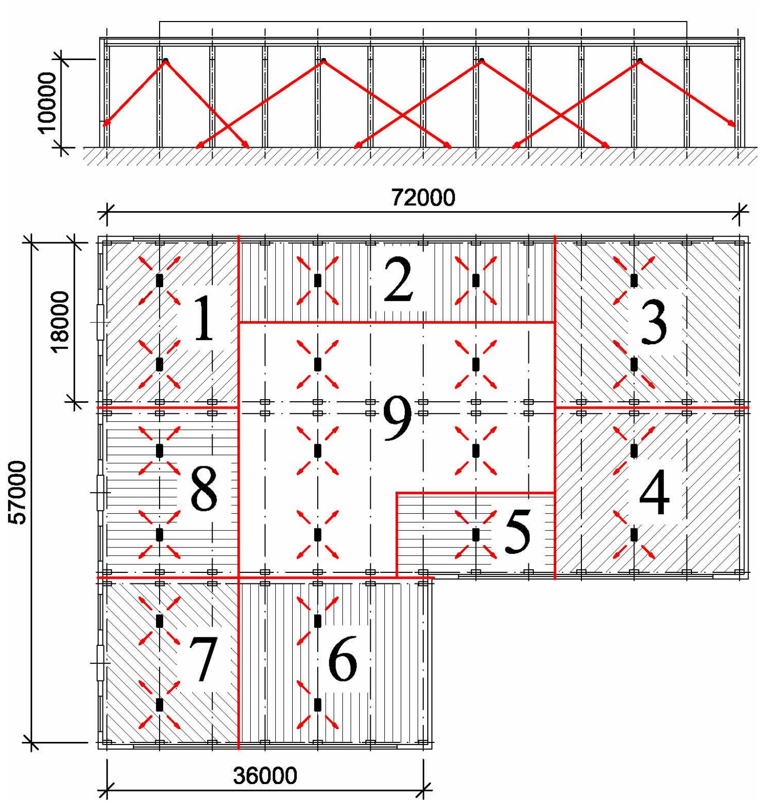



***Figure 5.57** Preliminary location of radiant heaters in the hall.*

*Evaluation of installed power output*

When first draft of the placement of heaters is done, it is necessary to divide the zones according to the same thermal behaviour. Required installed power output is calculated for each zone calculated separately. Only this way can be achieved adequate heat delivery to places according to needs. Here, nine zones were created (Figure 5.57). Resulting design power outputs are shown in Table 5.10.

**Example 2:**

*Design heat load correction for section nr. 9*

From tables 5.4 and 5.5 and figures 5.49, 5.50. 5.51, 5.52 we get the following data:

$c_s$ = 17.0 W/(m²·K)
(Figure 5.42), v = 0.3 m/s;

$a$ = 0.85 (Figure 5.43), $h$ = 10 m;

$\eta_F$ = 0.45 (Figure 5.44), $A_p$= 270 m²,
$h$ = 10 m, B/L~1;

$\eta_L$ = 0.95 (Table 5.4), plaque radiant heater indirect exhaust removal;

$\eta_S$ = 0.58 (Table 5.5), the simplest plaque radiant heater, open combustion chamber.

Hence $\varphi$ = 1/(1-0.58) = 2.38;
$\varphi_m$ = 0.40 (horizontal suspension).

Now, after substitution to (5.1)

$$K_s = \frac{\eta_L \cdot c_S \cdot \alpha \cdot \varphi}{H_m \cdot c_L \cdot \eta_F \cdot \varphi_m(\varphi - 1)} = \frac{0.95 \cdot 17 \cdot 0.85 \cdot 2.38}{11 \cdot 0.1595 \cdot 0.45 \cdot 0.40 \cdot (2.38 - 1)} = 74.97$$

where:

$$H_m = \frac{V}{A_p} = \frac{2970}{270} = 11\ \mathrm{m}$$

$$c_L = \frac{\Phi_{\mathrm{HL}} \cdot 1000}{V_R \cdot (\theta_i - \theta_{ev})} = \frac{46.9}{(3 \cdot 18 \cdot 11 + 18 \cdot 9 \cdot 11) \cdot (16 - (-17))} = 0.1595.$$

For actual air temperature, equation (5.2) is used:

$$\theta_L = \frac{\theta_i \cdot K_S + \theta_{ev}}{1 + K_S} = \frac{16 \cdot 74.97 + (-17)}{1 + 74.97} = 15.6\,°\mathrm{C}.$$

Following step is to evaluate total required nominal heat output:

$$\Phi_{\mathrm{HL,O}} = \frac{V_R \cdot c_L(\theta_L - \theta_{ev})}{1000} = \frac{8910 \cdot 0.1595 \cdot (15.6 - (-17))}{1000} = 46.3\ \mathrm{kW}.$$

and hence nominal heat input:

$$\Phi_{\mathrm{HL,I}} = \frac{\Phi_{\mathrm{HL,O}}}{\eta_L} = \frac{46.3}{0.95} = 48.8\ \mathrm{kW}.$$

Since for section 9 there are considered five heaters, installed power output per heater should be at least 9.7 kW. From the manufacturers product line heater with power output 11 kW was chosen. Into sections 1, 7 and 8 radiant heaters A with core radiation angle 88° because they are able to concentrate radiation on small area. Resulting power outputs are obvious from layout presented on Figure 5.58.

***Table 5.10** Design and required installed power output of particular heaters of each zone.*

| Zone | $\phi_{HL}$ [kW] | $\phi_{HL,I}$ [kW] | Number of heaters | Min. per heater [kW] | Chosen heater | |
|---|---|---|---|---|---|---|
| 1 | 43.1 | 43.8 | 2 | 21.9 | A | 25 kW |
| 2 | 48.8 | 49.6 | 2 | 24.8 | B | 25 kW |
| 3 | 45.4 | 46.5 | 2 | 23.3 | B | 25 kW |
| 4 | 43.8 | 44.9 | 2 | 22.5 | B | 25 kW |
| 5 | 18.8 | 19.3 | 1 | 19.3 | B | 25 kW |
| 6 | 43.8 | 44.9 | 2 | 22.5 | B | 25 kW |
| 7 | 40.4 | 41.1 | 2 | 20.6 | A | 25 kW |
| 8 | 26.3 | 27.1 | 2 | 13.6 | A | 15 kW |
| 9 | 46.9 | 48.8 | 5 | 9.8 | B | 11 kW |
| **Totally** | **357.3** | | **20** | | | |

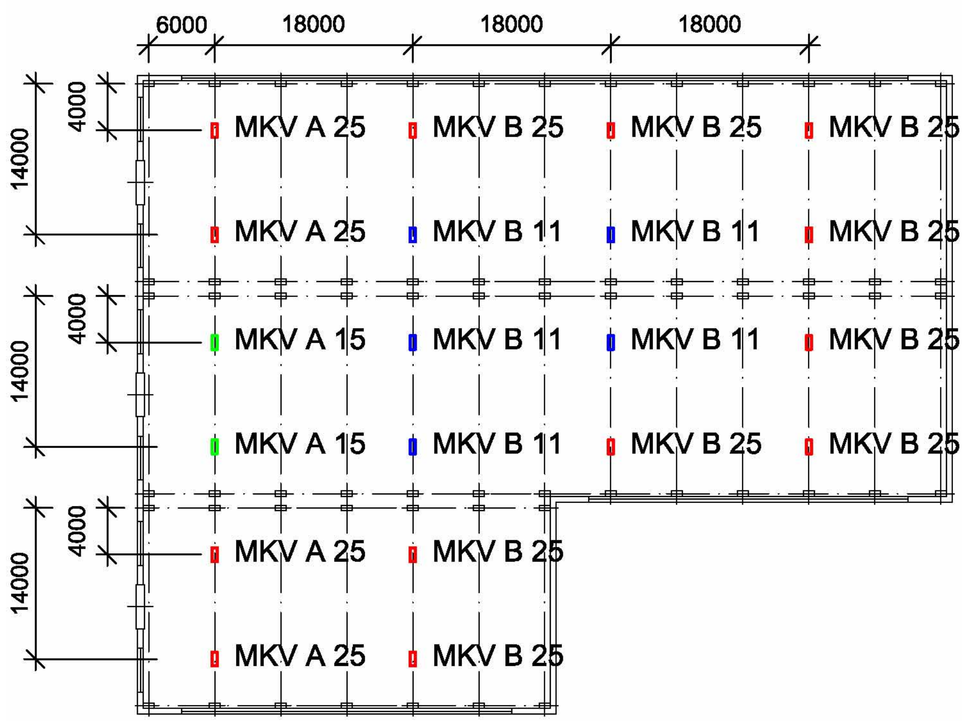



***Figure 5.58** Power outputs and location of plaque radiant heaters.*
*red – 25 kW, green – 15 kW, blue – 11 kW.*

Because floor coverage (core radiation angles) can slightly vary with heat output, it is recommended before any further step to check once more area coverage of all installed heaters, then check minimal distances from flammable materials or electrical wires.

*Control and operation*

Next step is to assign installed heaters according to required operation into control loops. Here, three control loops will be made (1, 2, 3), operating according to signal from sensors (R1, R2, R3). Considering seldom opening of the gates globe temperature sensors are chosen for their more precise operation. The closest radiant heater next to the sensor will be reference one (Figure 5.59).

*Gas pipeline design*

Standard design and evaluation of pipeline dimensions will not be shown here in detail. There is just the requirement to connect the gas pipeline of large control loops into a frame. During switch on period the risk of insufficient pressure is reduced. On Figure 5.59 there is shown possible supply pipeline and location of regulation and measurement station (RS+M).

*Exhausts removal and combustion air intake*

In this case indirect exhausts removal is used by installation of an axial fan to the highest point of the front fasade. Fans are designed according to the standard EN 13410 for 10 m³/h per 1 installed kW. In this case:

**V1:** 6 x 25 kW + 2 x 11 kW = 172 kW
$\dot{V}_{min}$ = 1720 m³/h;

**V2:** 3 x 25 kW + 2 x 15 kW + 3 x 11 kW = 138 kW; $\dot{V}_{min}$ = 1380 m³/h;

**V3:** 4 x 25 kW = 100 kW;
$\dot{V}_{min}$ = 1000 m³/h.

Fans should be placed either in highest point of front façade or in the vertical surface of a skylight. Maximal distance between the fan and the heater is six times height of installation which is here 6 x 11 m = 66 m. All the installed heaters are within this range. Location in layout is shown on Figure 5.59.

Combustion air intake is sufficient here by infiltration because of low tightness of the building envelope. However for newly build buildings this point musn't be forgotten!

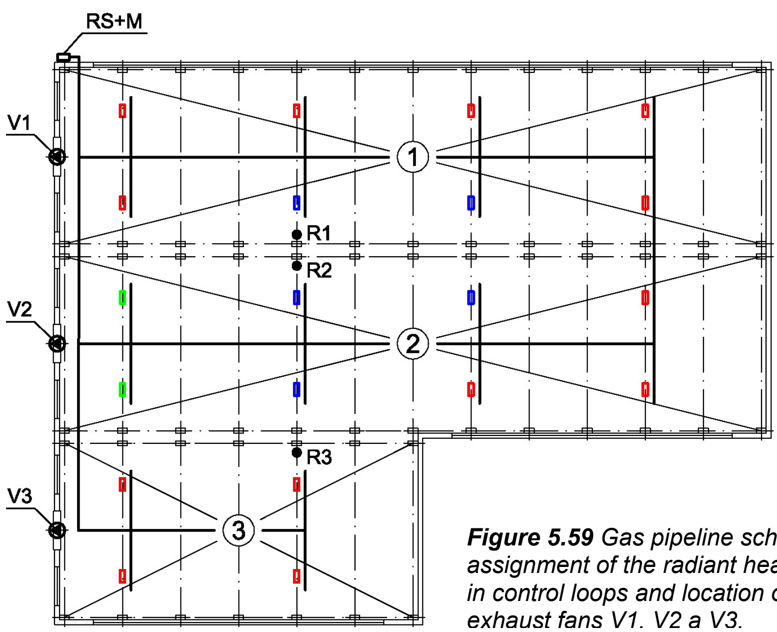



***Figure 5.59*** *Gas pipeline scheme, assignment of the radiant heaters in control loops and location of exhaust fans V1. V2 a V3.*

## *Design examples: Heating of a separate workplace*

**Example 3:**
In an unheated space ($\theta_i$ = + 5°C), heat up a workplace with floor area of 12 x 18 m on design temperature $\theta_g$ = 18°C (Figure 5.60. area 1). Since it is necessary to ensure design temperature all over the workplace for design purposes extended area by 1.5 m is considered. Then $A$ = 15 x 19.5 = 292.5 m².

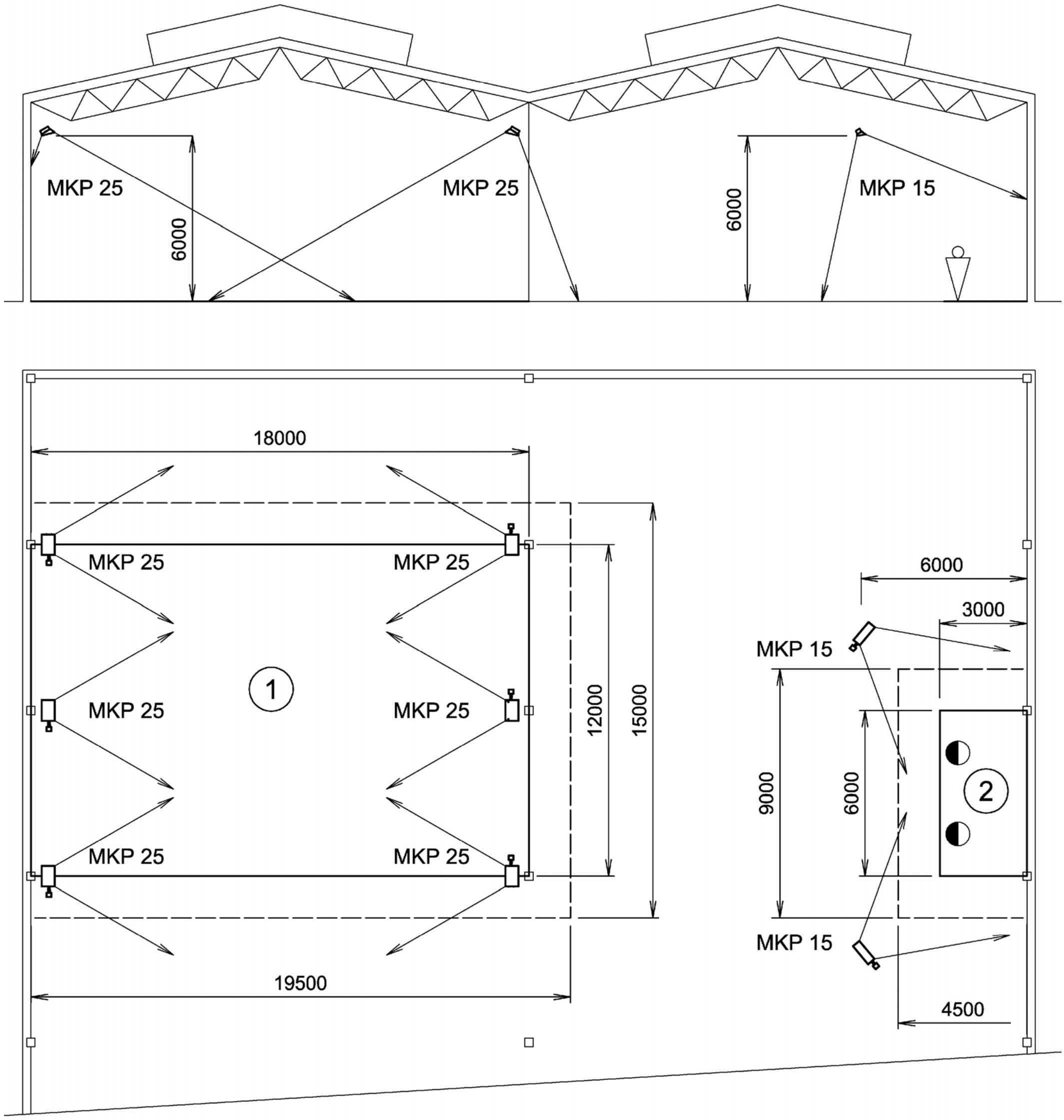



***Figure 5.60*** *Heating of separate workplaces in unheated hall.*
*1 – floor surface 12 x 18 m, 2 – workplace near the window 6 x 3 m.*

Required installed power output (5.12, 5.13, 5.14):

$$\varphi_{HL,I} = \frac{A_p \cdot f_1 \cdot \Delta\theta_{max}}{0.0716 \cdot \eta_S \cdot \varphi_m \cdot A_S \cdot 1000} = \frac{292.5 \cdot 1.2 \cdot (18-5)}{0.0716 \cdot 0.65 \cdot 0.70 \cdot 0.85 \cdot 1000} = 137\,\text{kW}.$$

Required power output of one heater:

$$\frac{\varphi_{HL,i}}{n} = \frac{137}{6} = 22.84\,\text{kW}.$$

With regard to the height of suspension $h$ = 6 m – tilted suspension, it will be necessary to install 6 plaque radiant heaters, each with nominal power output 25 kW. Sometimes to improve irradiation marginal heaters are suspended against boundary corner (Figure 5.60 - case 2).

Installed power output:

$$\varphi_{HL,I} = 6 \cdot 25\,\text{kW} = 150\,\text{kW}$$

When considering heating of separate workplaces finding of suitable control strategy can be problematic. It emerges because it is very often difficult to place control sensor. If it is possible the most suitable location would be within the heated area. When it is not possible, time loop operation is recommended. On the controller running time 1/2 h is set for spring and autum (till $\theta_e$ = + 3°C); 3/4 h – mid-winter season ($\theta_e$ from - 5°C to 3°C) and 1 h – winter extreme ($\theta_e$ lower than - 5°C). The most suitable intervals will be shown by praxis. After the time interval heaters will remain in stand-by mode until staff will not set them back into operation. Design air temperature of unheated space is valid until temperature limit $v_0$ = 0.2 m/s; (designation $\theta_i^{0.2}$). When considering higher air velocities, it is necessary to lower design air temperature $\theta_i$ according to following relation:

$$\theta_i = (\theta_i^{0,2} - \Delta\theta_1) \qquad (5.17)$$

As the air velocity increases, the temperature difference increases as well:

- $v_0$ = 0.4 m/s  $\Delta\theta_1$ = 2 K;
- $v_0$ = 0.6 m/s  $\Delta\theta_1$ = 4 K;
- $v_0$ = 0.8 m/s  $\Delta\theta_1$ = 6 K;
- $v_0$ = 1.0 m/s  $\Delta\theta_1$ = 8 K.

In case we would consider the same input values but we would increase air velocity to $v_0$ = 0.8 m/s the required power output would change significantly (5.12, 5.13, 5.14):

$$\varphi_{HL,I} = \frac{(\theta_g - (\theta_i^{0,2} - \Delta\theta_1)) \cdot A_P \cdot f_1}{0.0716 \cdot \eta_s \cdot \varphi_m \cdot a_S \cdot 1000} = \frac{(18-(5-6)) \cdot 292.5 \cdot 1}{0.0716 \cdot 0.65 \cdot 0.70 \cdot 0.85 \cdot 1000} = 200.7\,\text{kW}$$

Required power output of one heater:

$$\frac{\varphi_{HL,I}}{n} = \frac{200.7\,\text{kW}}{6} = 33.5\,\text{kW}$$

Finally, six radiant heaters are chosen with nominal power output 36 kW.

Results show that air velocity has very high impact on required power output of radiant heaters. Therefore it is important to care a lot about protection of separate workplaces against draft. It is recommended to install curtains of height at least 2 or 3 m (Figure 5.60 bold line).

Different approach show following equation:

$$\phi_{HL,I} = \frac{q_0 \cdot \Delta\theta_{\max} \cdot A_p \cdot f_1}{\eta_s \cdot 1000} \text{ [kW]}; \qquad (5.18)$$

Where $q_o$ = 25 W / K is the unit power output per 1 K of temperature difference. Check will be done by substitution into (5.18):

$$\phi_{HL,I} = \frac{q_o \cdot A_p \cdot f_1 \cdot \Delta\theta_{\max}}{\eta_s \cdot 1000} =$$

$$\frac{25 \cdot 292.5 \cdot 1 \cdot (18-5)}{0.65 \cdot 1000} = 146.2 \text{ kW}.$$

It is apparent that both procedures offer similar results (137 kW x 146 kW).

**Example 4:**

The workplace at external wall (Figure 5.60. area 2). Occupancy area $A_0$ = 6 x 3 m; extension 1.5 m, then $A_P$ = 9 x 4.5 m = 40.5 m²; $\theta_o$ = 18°C; $\theta_i$ = + 3°C.

By substitution into equations 5.12, 5.13, 5.14 we obtain:

$$\varphi_{HL,I} = \frac{A_p \cdot f_1 \cdot \Delta\theta_{\max}}{0.0716 \cdot \eta_S \cdot \varphi_m \cdot a_S \cdot 1000} =$$

$$\frac{40.5 \cdot 1.2 \cdot (18-3)}{0.0716 \cdot 0.65 \cdot 0.70 \cdot 0.85 \cdot 1000} = 26.3 \text{ kW}.$$

Two 15 kW plaque radiant heaters are chosen, which means 2 x 15 kW = 30 kW. Suspension in ceiling area tilted at height of 7.5 m. Approx. distance from occupants ~9.0 m, hence $f_1$ = 1.2.

Check evaluation by second procedure (5.18):

$$\varphi_{HL,I} = \frac{q_o \cdot \Delta\theta_{\max} \cdot A_p \cdot f_1}{\eta_S \cdot 1000} =$$

$$\frac{25 \cdot 15 \cdot 40.5 \cdot 1.2}{0.65 \cdot 1000} = 28.0 \text{ kW}.$$

Again result is almost the same.

In this case it is possible to control power output by standard globe sensor and controller with week program.

## Discussion

From both examples of separate workplaces heating it is obvious that requirements on installed power output are much higher that for overall heating (from 2 to 4 times higher power output). It is also necessary to consider reaction time and mainly concentration of heat flow into the workplace area. For such cases plaque radiant heaters are irreplaceable. Both airhandling units and tube radiant heaters are for such a case completely unsuitable. The same methodology can be also used for heating design of entering area to the heated building where gates are frequently opening – area about (12 x 18 m). According to the opening frequency design air temperature needs to be chosen $\theta_i$ = (8 to 12°C). Control sensor for air temperature is recommended ($\theta_i$).

# 6 WARM AIR HEATING

## 6.1 Introduction

The main difference between radiant and convective heating is the way how the heat is being transferred to the considered space. Radiant heating first heats up irradiated surfaces and only after secondary air is heated up by convection from the surfaces. This principle causes lower air temperatures but on the other hand higher mean radiant temperature (MRT) of treated space. Convective heating works differently. Air is heated in heat exchangers and it is driven to the considered space. The air temperature is higher but MRT lower. Because of the fact that warmer air is due to its lower density rising to the higher parts of a space, convective heating is not suitable for higher spaces (higher than approx. 6 up to 7 m). In case of lower spaces the situation is different. Radiant systems can face a different problem - local overheating due to high surface temperature. Surfaces with higher temperatures close to the occupants can cause local discomfort.

## 6.2 Wall mounted gas-burning air-handling units

Heating of large spaces with wall-mounted air-handling units is quite popular solution, however majority of installed systems (units heated either by water or steam) work inefficiently. Besides large heat consumption, desired comfort for occupants (uniform air temperature in occupation zone) is very often not reached. This is caused either by insufficient air distribution or inappropriate application in high spaces. It results in high air temperatures below the roof (when the roof is well insulated up to $\theta_{a,roof} = 30°C$) and insufficient heating in occupation zone.

On the other hand also warm air heating can be economical, however certain principles have to be kept. The first principle is heat delivery into the heated space and the second is physical background of warm air behaviour. Moreover there is another advantage of gas-burning air-handling units, the flow pattern is during whole heating season almost constant and therefore the heat delivery control is done only by switching off particular units.

### Product line

Wall-mounted gas-burning air-handling units are offered in two variants. The first variant is equipped with axial fan and therefore there cannot be any distribution system attached (low disposition pressure). On the other hand units with radial fan are designed for cases where distribution system is intended (either within the heated space or as an fresh air intake). Distribution system design is the same as in case of standard air-handling units. First volume air flows are evaluated according to local needs and then pressure losses are calculated. Finally, total power output and the type of radial fan are specified. The power output of such units varies from 10 up to 100 kW. There is also possibility to choose condensing operation and hence obtain higher efficiency. Such solution reduces significantly running costs of the system.

Very important thing for the flow distribution pattern in space is type of diffuser. The designer can choose among various types and is able to influence the air temperature distribution in heated space. Examples of various distribution elements for an axial fan wall-mounted unit can be found on Figure 6.1. When ceiling diffuser (Figure 6.2) is used the radial fan should be used instead. Axial fan would have problem to handle the pressure loss of the diffuser.

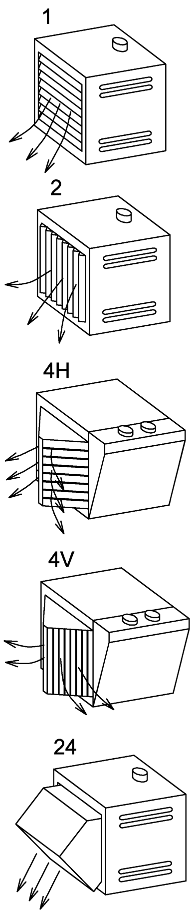



***Figure 6.1*** *Examples of various diffusers for axial-type wall-mounted air-handling units.*
*1 – diffuser with horizontal lamellas, 2 - diffuser with vertical lamellas, 4H - diffuser 2 x 45° with horizontal lamellas, 4V - diffuser 2 x 45° with vertical lamellas, 24 – ceiling diffuser 45° without lamellas.*

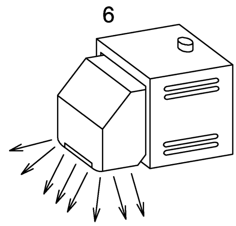



***Figure 6.2*** *Ceiling diffuser 90° for units with radial fan.*

## Evaluation part

For evaluation of design heat load EN 12 831 (or equivalent for ex. older DIN 4701) can be used with following corrections:

### *Temperature gradient*

For evaluation of heat losses over the roof and skylights, vertical temperature gradient have to be considered 0.8…1 K/m. If circulation rates are fulfilled according to Table 6.1 it is possible to use gradient 0.4…0.5 K/m. The circulation rate includes both volume air flows of installed air-handling units and volume air flows of recirculation fans placed at roof level.

### *Heat-up coefficient*

Air-handling units reach full power output within very short time period (10…20 s). After operation breaks the inside air is cooled down very fast because both building constructions and mechanical equipment doesn't accumulate from the air much energy. Therefore heat-up coefficient is necessity (20% for 1 shaft and 15% for 2 shafts). Previous values suppose that globe temperature for reduced operation would be:

$$\theta_{g,red} = \theta_g - 5\ \text{K}, \quad [°\text{C}] \qquad (6.1)$$

where $\theta_g$ [°C] is the globe temperature.

### *Infiltration*

For the heat losses due to infiltration following design external temperature should be used:

$$\theta_{g,red} = \theta_g - 8\,\mathrm{K}, \quad [°C] \qquad (6.2)$$

where $\theta_e$ [°C] is the external design temperature in winter period.

To be able to supply heat uniformly to the heated space following circulation rates should be kept Table 6.1.

***Table 6.1** Required air circulation rates.*

| Inner space volume [m³] | Circulation rate Σ (units + circ. fans) [x/h] |
|---|---|
| 1 000 | 5.0 |
| 2 000 | 4.5 |
| 3 000 | 4.0 |
| 4 000 | 3.5 |
| 6 000 | 3.0 |
| 8 000 | 2.5 |
| 10 000 | 2.0 |
| >10 000 | 1.5 |

### *Design methodology of wall mounted air-handling units*

Convective heating by wall mounted air-handling units can be economical up to the heights of about $H$ = 8…10 m. Essential influence on the operation costs has the circulation rate in heated space. It is therefore not sufficient to evaluate just design heat losses, volume air flow rate should be considered as well. In case of insufficient air flow, a pillow of warm air is being created below the ceiling and the occupancy area is heated insufficiently. From this point of view, very important device are recirculation fans. These are mainly axial fans suspended at the roof level, pushing large amount of warm air back to the occupancy zone.

Figure 6.3 shows throw of the units ($d_1$) – units with axial fan and standard diffusers. Figure 6.4 shows throw of units equipped with ceiling diffuser 45° ($d_2$).

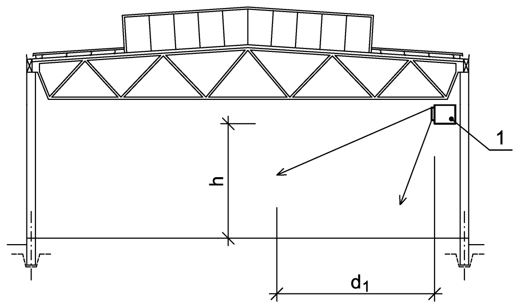



***Figure 6.3** Throw of the units with diffusers 1, 2, 4H, 4V. 1 – gas-burning wall-mounted unit, h – height of suspension, $d_1$ – throw of the flow.*

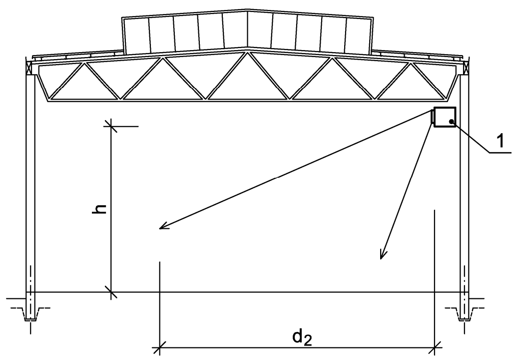



***Figure 6.4** Reach of the flow with diffuser 24. 1 - gas-burning wall-mounted unit, h – height of suspension, $d_2$ – throw of the unit.*

Higher height of suspension than mentioned in the Table 6.2 means in case without destratification unit creation of warm air pillow and insufficient heating in occupancy zone. For higher buildings and higher suspension heights it is necessary to use rather units with radial fan and its location below the ceiling between the girders.

***Table 6.2** Recommended height of suspension and reach of the flow (for orientation).*

| Power output [kW] | Volume air flow [m³/h] | Diffuser 1, 2, 4H a 4V | | Diffuser 24 | |
|---|---|---|---|---|---|
| | | $h_{max}$ [m] | $d_1$ [m] | $h_{max}$ [m] | $d_2$ [m] |
| 20 | 2 200 | 2.5 | 10 | 3.5 | 10 |
| 25 | 2 750 | 2.8 | 12 | 4.0 | 12 |
| 30 | 3 350 | 3.0 | 14 | 4.5 | 14 |
| 40 | 4 450 | 3.2 | 22 | 4.8 | 22 |
| 50 | 5 550 | 3.4 | 23 | 4.9 | 23 |
| 60 | 6 650 | 3.6 | 24 | 5.0 | 24 |
| 80 | 8 850 | 4.0 | 27 | 5.5 | 27 |

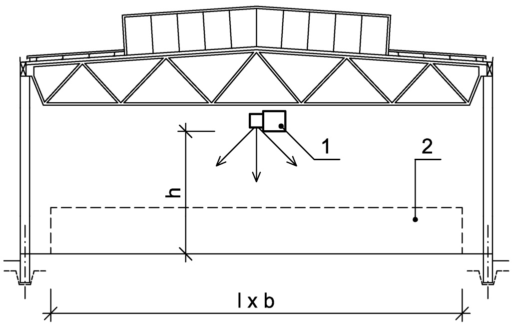



***Figure 6.5** Area of coverage – unit with radial fan and ceiling diffuser 90°. 1 – gas-burning wall-mounted unit, 2 – Area of coverage, h – height of suspension, (l x b) – layout dimensions of covered area.*

When location of units is designed, the most important are volume air flow and power output per one unit. It is not always possible to install such amount of units to ensure sufficient circulation in the building. When considering warm air heating the velocity of air at the end of the units' throw might be about 0.25 m/s. Such velocity causes immediate rising of large, still warm, amount of air towards the roof. Limiting reach is about 2/3 of building width (*B*), which is sufficient enough to deliver heat along the whole width (Figure 6.6, where $d_1$ resp. $d_2$=2/3*B*). However in most of the cases required circulation rate is not achieved and therefore recirculation fans are the only solution. (Figure 6.7).

In case of the fans are used wisely, significant part of both acquisition costs and running costs can be reduced. From the economical point of view, it is less expensive to install rather smaller number of units with higher power output that the other way around.

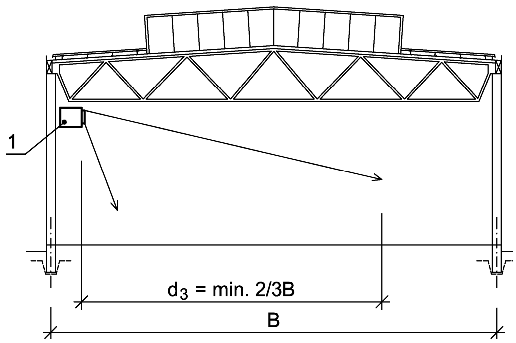



***Figure 6.6** Minimal recommended throw of the air. 1 – gas-burning wall-mounted unit, $d_3$ – throw of the unit, B – width of the building.*

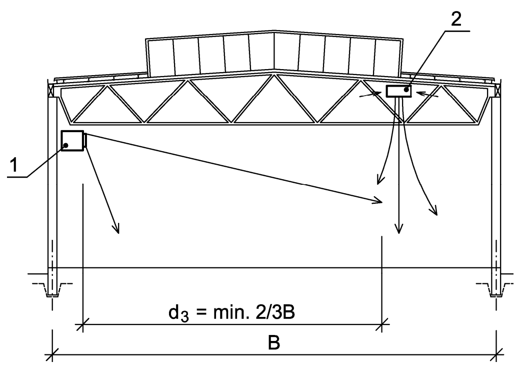



***Figure 6.7** An increase of unit reach by using the recirculation fan. Suitable for building with height h > 4 m and without significant pollution sources. 1 – gas-burning wall-mounted unit, 2 – recirculation fan, $d_3$ – unit throw, B – width of the building.*

***Table 6.3** Area of coverage for units with ceiling diffusers.*

| Power output [kW] | Volume air flow [m³/h] | Ceiling diffuser 90° (5 open diffusers) | | | | | Ceiling diffuser 90° (3 open diffusers) | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | *h*=3 m | *h*=4 m | *h*=5 m | *h*=6 m | *h*=7 m | *h*=3 m | *h*=4 m | *h*=5 m | *h*=6 m | *h*=7 m |
| | | $l_1$ x $b_1$ [m] | | | | | $l_2$ x $b_2$ [m] | | | | |
| 20 | 2200 | 10x10 | | | | | 10x6 | | | | |
| 25 | 2750 | 12x12 | 10x10 | | | | 14x7 | 12x7 | 11x6 | 10x6 | |
| 30 | 3350 | 15x15 | 13x13 | 11x11 | | | 19x7 | 16x7 | 14x7 | 12x7 | 9x7 |
| 40 | 4450 | 15x15 | 13x13 | 11x11 | | | 20x9 | 17x9 | 16x9 | 14x9 | 12x9 |
| 50 | 5550 | 17x17 | 16x16 | 13x13 | | | 25x11 | 22x11 | 20x11 | 18x11 | 16x11 |
| 60 | 6650 | 18x18 | 16x16 | 14x14 | 12x12 | | 27x12 | 23x12 | 22x12 | 20x12 | 18x12 |
| 80 | 8850 | 19x19 | 17x17 | 15x15 | 13x13 | 12x12 | 38x15 | 36x15 | 34x15 | 32x15 | 31x15 |

Figure 6.8 shows ideal location of units from the throw point of view without recirculation fans. From the economical point of view, rather less number of units of larger power output in combination with recirculation fans would be more suitable.

## Placement limits

For the correct function of the units, good accessibility for operation and maintenance is essential. Therefore it is necessary to maintain minimal distances from building constructions recommended from every manufacturer. There are on the Figure 6.9 minimal distances from other constructions for one diffuser type.

## Gas connection

Gas connection is the most often 3/4". There is a closing valve 3/4" after the piping outlet. Most often the connection is done by flex hose. The fuel can be either natural gas or propane (event. + butane). Minimal connection over-pressure is 2 kPa; maximal 6 kPa. The units are supposed to be installed just in basic environment (not flammable, not explosive).

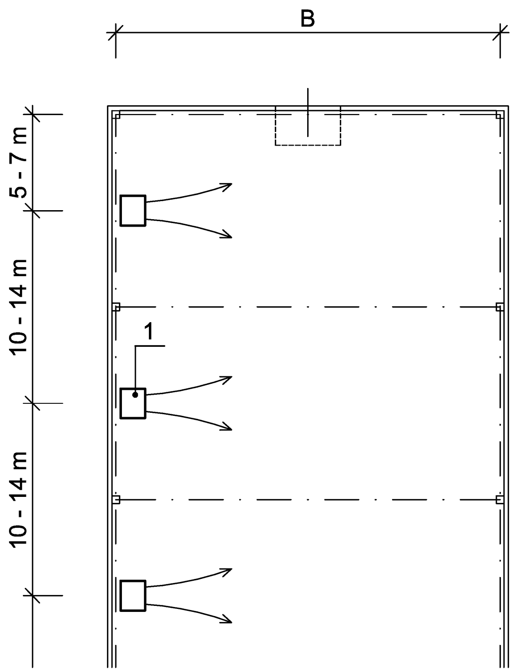



***Figure 6.8** An example of air handling units' placement in case with no recirculation fans. 1 – gas-burning wall-mounted unit, B – width of the building.*

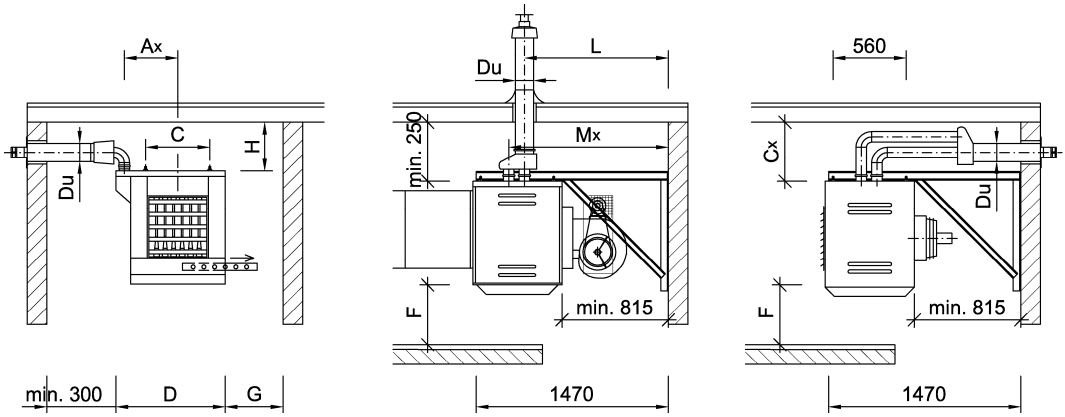



***Figure 6.9*** *Minimal distances from building constructions.*

***Table 6.4*** *An example of minimal distances from the building constructions [mm].*

| Type [kW] | Ax | Bx min | Cx min | C | D | F min | G min | H min | L | Mx | ø Du |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 20 | 345 | 170 | 345 | 365 | 714 | 75 | 490 | 220 | 1115 | 1235 | 126 |
| 25 | 377 | 170 | 345 | 430 | 779 | 75 | 555 | 220 | 1115 | 1235 | 126 |
| 30 | 409 | 170 | 345 | 495 | 844 | 75 | 620 | 220 | 1115 | 1235 | 126 |
| 40 | 475 | 170 | 345 | 625 | 974 | 75 | 750 | 220 | 1115 | 1235 | 126 |
| 50 | 540 | 175 | 365 | 775 | 1104 | 75 | 850 | 250 | 1115 | 1235 | 151 |
| 60 | 640 | 175 | 365 | 885 | 1289 | 75 | 1010 | 250 | 1137 | 1277 | 151 |
| 80 | 770 | 175 | 365 | 1145 | 1549 | 75 | 1270 | 250 | 1137 | 1277 | 151 |

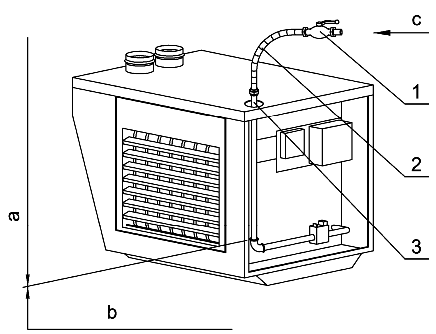



***Figure 6.10*** *Gas connection. a – gas piping, b – supplier delivery (ended with knee 90° - G3/4"),*
*c – gas connection G20 (overpressure min. 2 kPa, max. 6 kPa), 1 – closing valve DN 20. 2 – flex hose, 3 – hose connection place.*

## Flue gas venting and combustion air intake

Flue gases must be vented outside the heated space. The tubes for venting of flue gases and combustion air intake are recommended to keep of the same length and shape. Standard chimney construction is through the roof (vertically). The second solution is via facade which is cheaper but in some EU countries this way is limited just for units with lower power output than 40 kW. The maximum length of the flue gas tubes depends on every unit, but for example for the type shown on the Figure 6.11 the maximum length of flue gas venting tubes and air intake tubes is 6 m, including two elbows 90°. Each additional elbow further reduces the length by 1 m. It is recommended to use rather partial elbows 45°. The tubes should be the whole length of the same diameter and it is not allowed to descend it. In case of shorter tubes than 3 m and without elbows, it is necessary to install some element with pressure drop (e.g. shutter). Besides the unit, also the tubes must be suspended. Recommended suspension might be each 1…2 m.

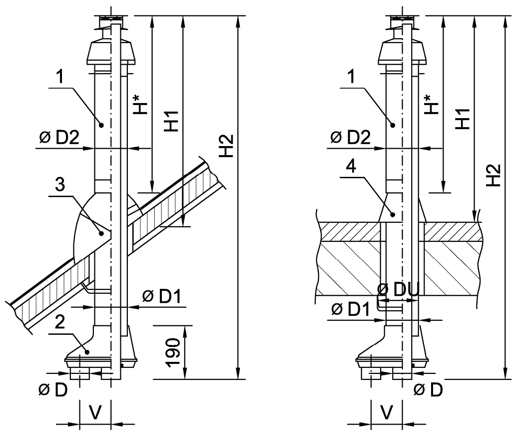



***Figure 6.12*** *Combustion air intake and flue gas venting by coaxial tubes through the roof 1 – flue gas venting / combustion air intake coaxial tubes; 2 - T piece, 3 – tilted roof cover, 4 – horizontal roof cover.*

**<u>Examples of installation</u>**

On the Figure 6.11 there is an example of exhausts and air intake tube connection both for ceiling and facade leading. At the end both tubes are led into one allowing passing the construction with just one hole.

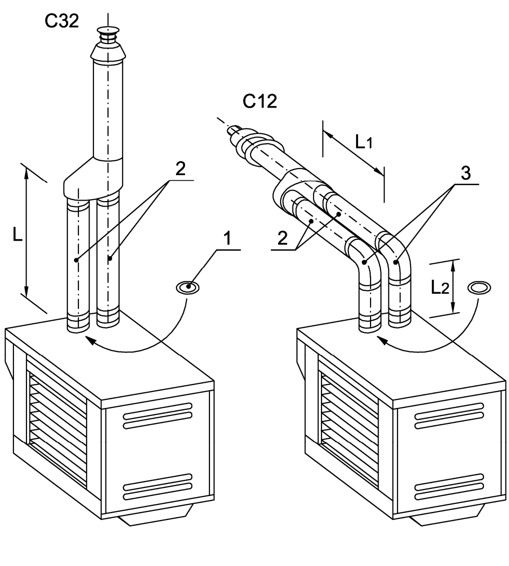



***Figure 6.11*** *Forced flue gases venting and combustion air intake from the exterior (ceiling – left, wall - right) 1 - shutter, 2 – extension package, 3 - elbows 90°.*

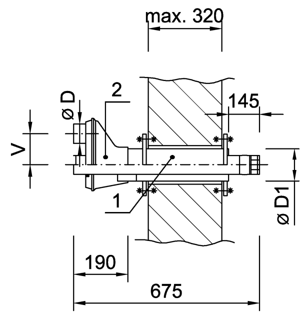



***Figure 6.13*** *Combustion air intake and flue gas venting by coaxial tubes through the wall 1 – flue gas venting / combustion air intake coaxial tubes; 2 - T piece.*

Technical solution for roof assembly is shown on the Figure 6.12. The wall assembly is shown on the Figure 6.13.

## 6.3 Wall-mounted air-handling units in combination with recirculation fans

For the improvement of installation efficiency, recirculation fans are recommended placed to the highest points of heated space. The fans are supposed to return warm air from the roof level back to the occupancy zone (winter operation) or to remove warm air from the roof to the outside environment in summer. It is necessary to note that recirculation fans can be just in cases with low pollution sources and in cases where higher air movement cannot cause any technological or other problem.

**Winter operation**

In case of just winter operation is considered, basic version of recirculation fan is sufficient. The type is chosen according to maximum height of suspension (manufacturer recommendation), air flow rate and volume air flow.

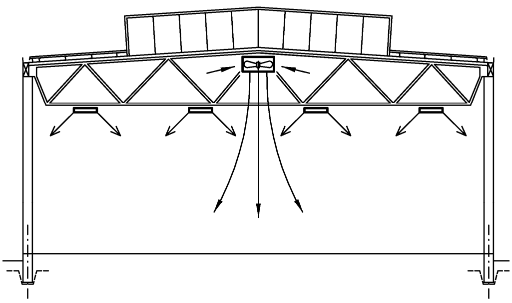


*Figure 6.14 Recirculation fan placement in combination with radiant panels.*

The fan is set to operation automatically when air temperature at thermostat reaches set level.

**Winter and summer operation**

One of the most actual problem of industrial halls these days (new buildings are already well insulated and tight) is creation of optimal working conditions in summer period. Recirculation fan in "whole year" version can help to improve conditions by forcing the warm air from the roof in summer to the outside environment Figure 6.15. Fresh and colder air can be supplied by under pressure with holes made at floor level. These can be closable for winter conditions. In winter conditions recirculation fan works in normal operation (pushing the warmer air to the floor).

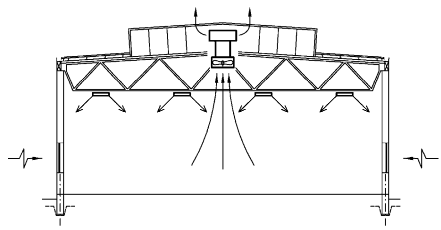


*Figure 6.15 Recirculation fan placement.*

**Recirculation fan throw and its reach**

Very important aspect when using recirculation fans is to maintain maximal velocity at the height of 1.5 m above the floor ($v_{max}$ = 0.2 m/s). For this purpose there are adjustable lamellas at the outflow direction from the fan's case. The lowest suspension height is about 4 m (Figure 6.16, variant C). Construction of recirculation fans differs according to the manufacturer. The type shown here has the diffuser surface divided into four squares to be able to set the outflow direction separately. The highest height of suspension can be affordable with lamellas set to position A. Maximal heights of suspension can be obtained from the manufacturer (Figure 6.16).

The fan is set to the operation automatically when air temperature at recirculation fan thermostat exceeds set temperature. The setting range is 10…30°C. Normally the temperature set at the thermostat is recommended to be about 2…5 K higher than design air temperature in occupancy zone.

## 6.4 Examples of units' placement

Examples of gas-burning, wall-mounted units' and recirculation fans' placement can be found on figures 6.17 and 6.18.

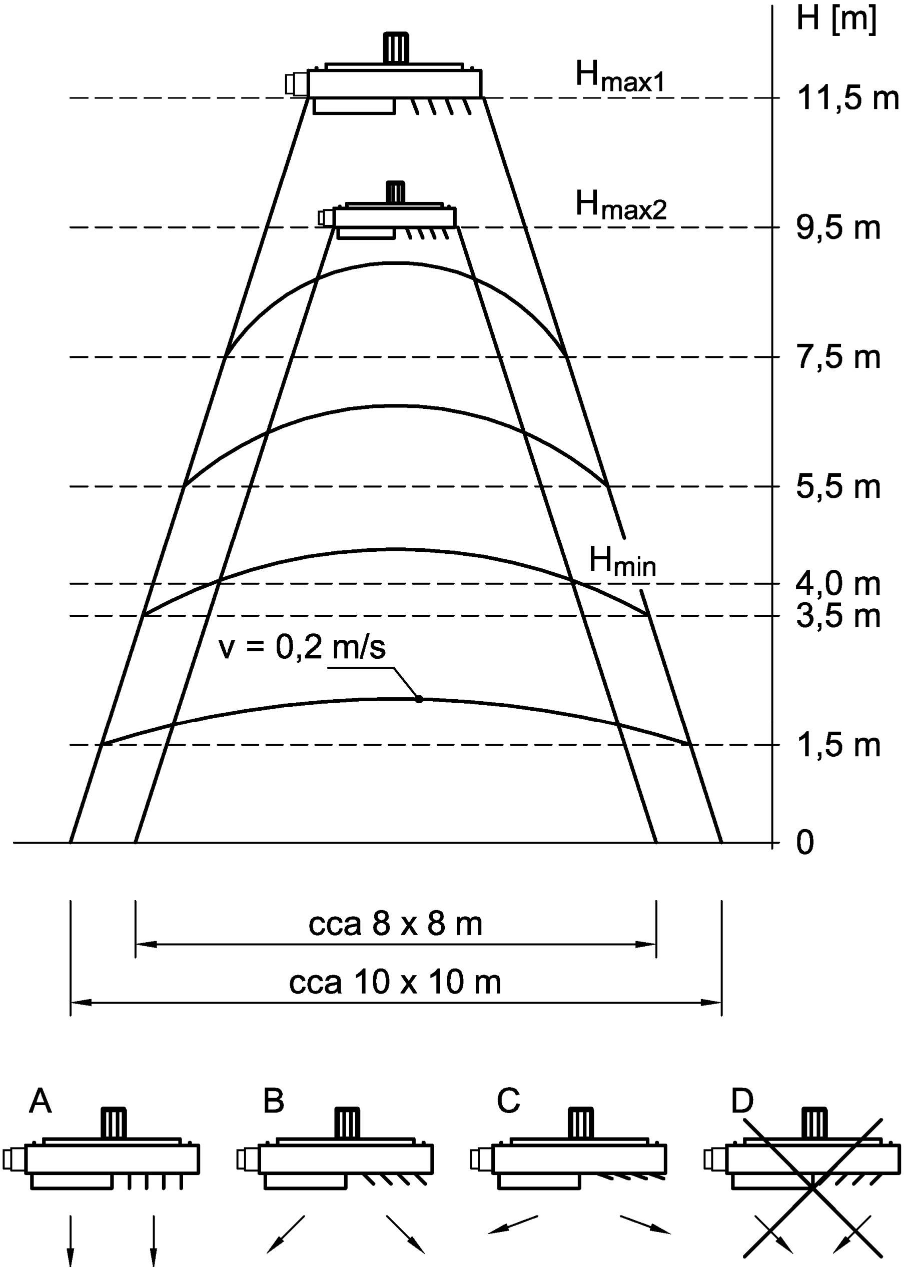



*Figure 6.16 Recirculation fans' throw.*

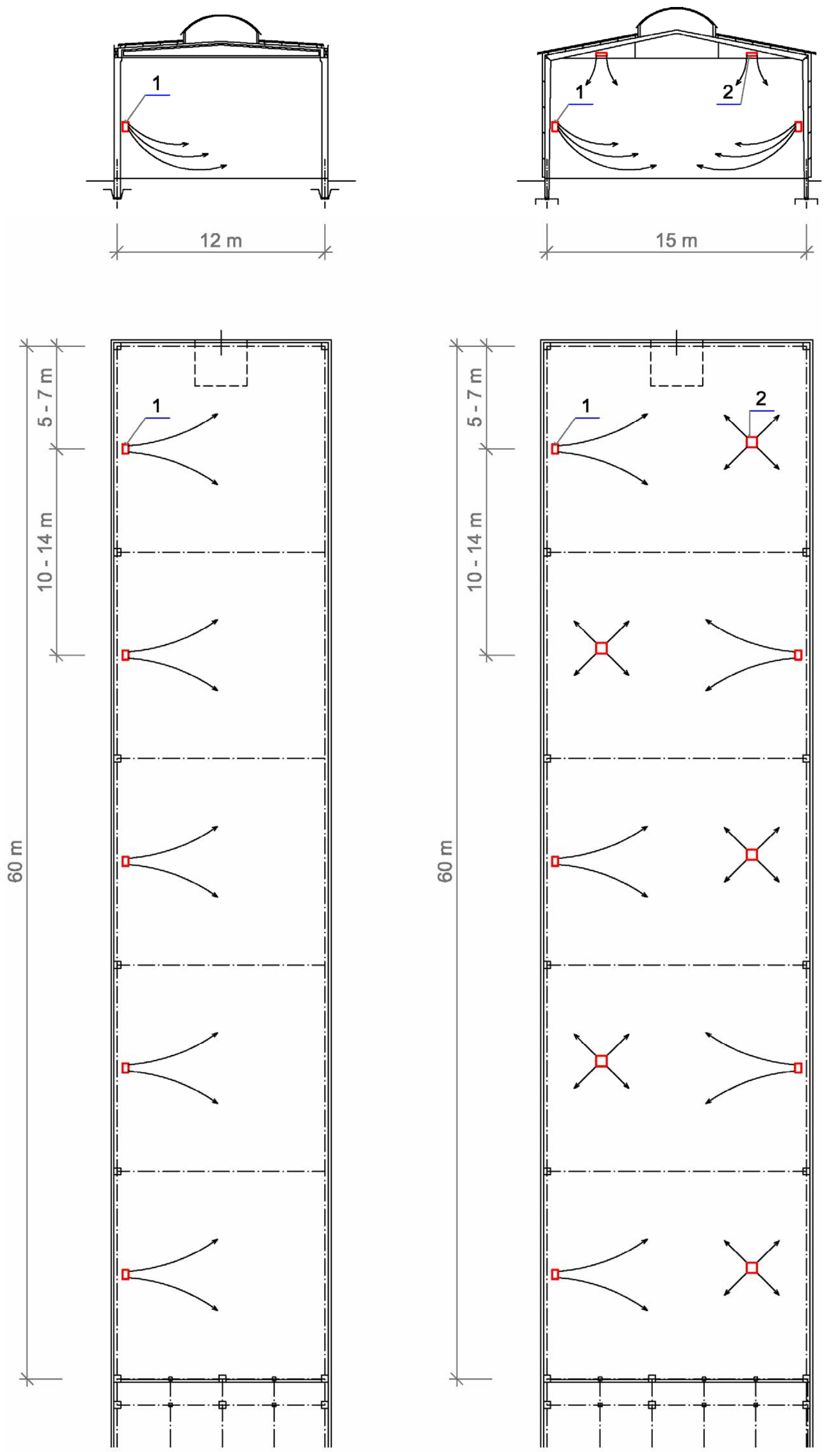



***Figure 6.17** Examples of placement of gas-burning wall-mounted units in combination with recirculation fans – building sizes 12, 15 m. 1 – gas-burning wall-mounted unit, 2 – recirculation fan.*

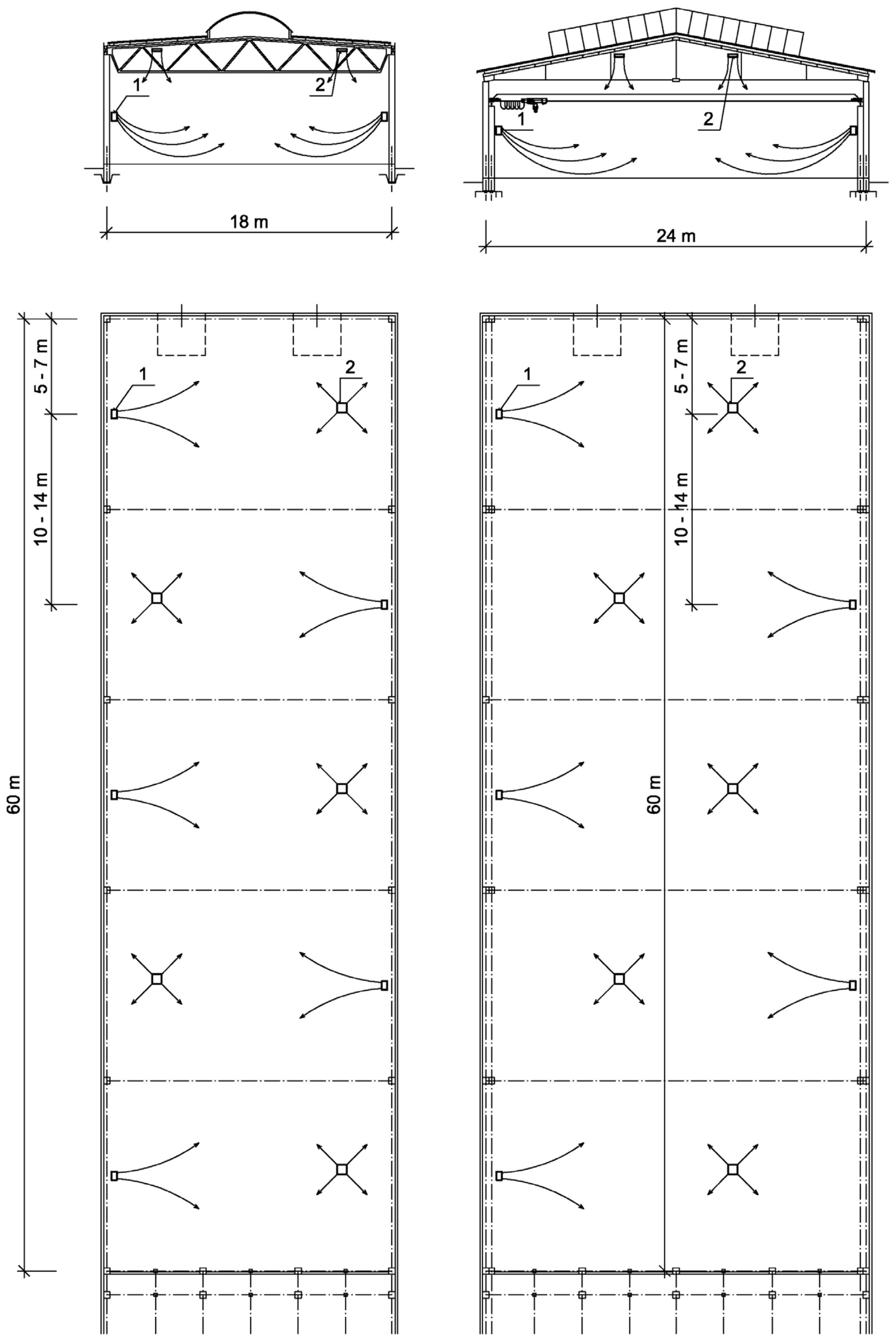



***Figure 6.18*** *Examples of combination of wall mounted wall-mounted units with recirculation fans – building sizes 18, 24 m. 1 – gas-burning wall-mounted unit, 2 – recirculation fan.*

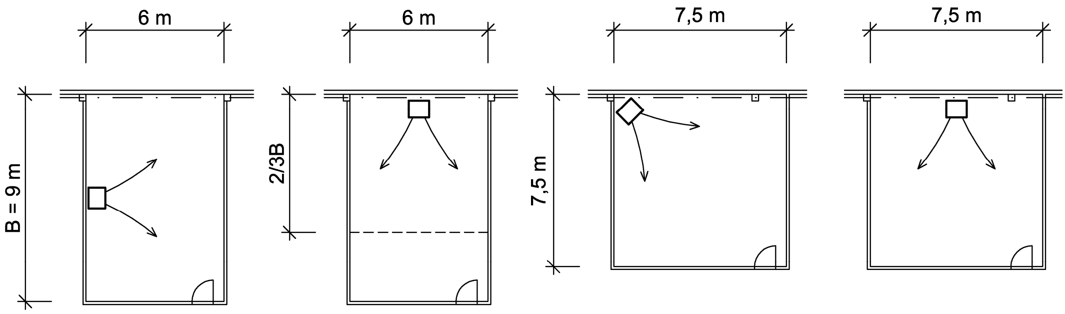



***Figure 6.19*** *Heating of small workshops with wall-mounted units.*

From the previous figures it is worthy of mentioning the location of the fans within the space. In case of wider buildings, where throw of the units ends (with $v_{end}$ = 0.25 m/s) at about 2/3$^{rd}$ of hall width B it is recommended to place the fan against the unit above the area with slow flow velocity. Suitable placement is also in the middle of the hall between two or more units. The total amount and location within the space should be evaluated to fulfill requirements for sufficient circulation rate and the ceiling coverage. Every fan is able to influence the flow pattern in the square of roughly about 20 x 20 m. Optimally the fans should be designed so to return maximum of the warm air at the roof level back to the occupancy zone. The operation economy increases.

At narrow and relatively low buildings and small workshops with higher heat losses there is normally no need of recirculation fans because volume air flow corresponds with demanded circulation rate (Figure 6.19).

# 7 RADIANT FLOOR HEATING FOR INDUSTRIAL BUILDINGS AND MAINTENANCE FACILITIES

## 7.1 Introduction

This chapter discusses only radiant floor heating systems that use hydronic pipes embedded in the building structure in order to heat large spaces.

Hydronic radiant floor heating makes it possible to heat large industrial spaces with high ceilings in a manner that is not only energy- and cost-efficient, but also allows space to be used flexibly. By using hydronic floor heating, room temperatures can be maintained at required temperatures, for example as defined by the *Arbeitsstättenrichtlinie* (German Workplaces Guideline), without taking additional measures.

Radiant floor heating for industrial buildings also offers good technical conditions for the use of geothermal and solar energy in heating large halls.

Radiant floor heating for industrial purposes can be found in:
- Production plants
- Assembly and maintenance facilities
- Warehouses / distribution centres
- High rack warehouses / logistics centre
- Aircraft hangars, see Figure 7.1
- Market and exhibition halls.

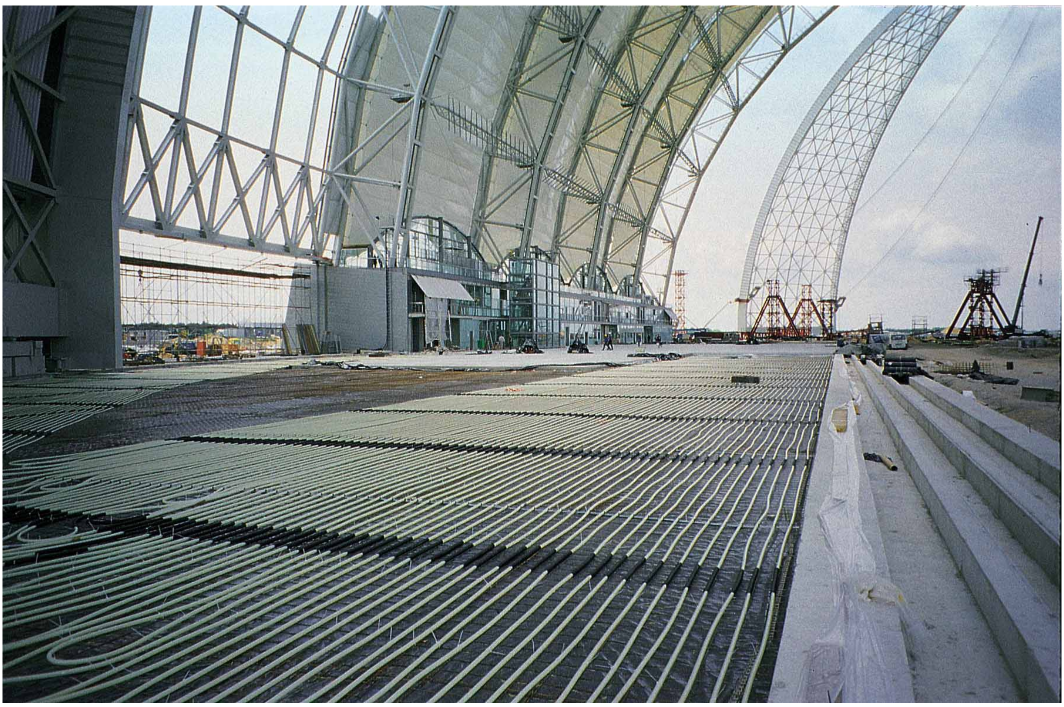


***Figure 7.1*** *Aircraft hangar site (Photo: BVF).*

***Figure 7.2*** *Industrial floor heating pipe circuits on reinforcement (Photo: BVF).*

The basis for the utilization of radiant floor heating in industrial applications was derived from experience with the thermal behaviour of radiant floor heating in residential and office buildings, for which it was mainly used up until the 1980s.

## 7.2 Decision criteria

In halls with high ceilings different vertical temperature profiles result from the chosen heating system, see Figure 7.3.

As a minimum floor surface temperature must be maintained in the occupied zone, according to the cited example of the *Arbeitsstättenrichtlinie*, this is 18°C. In order to ensure the economic operation of the heating system, it must be ensured that this temperature does not rise with increasing ceiling height. Figure 7.3 depicts the vertical temperature profile for radiant floor heating in an industrial application as well as for a forced warm air heating system.

Industrial radiant floor heating fulfils in a nearly ideal manner the criteria for cost-efficient heating and the maintenance of minimum temperatures in occupied zones.

In Figure 7.3 the air temperature profile generated by the forced warm air heating system falls below the required minimum temperature in the occupied zone (floor surface temperature), so that additional technical measures for an increase in the temperatures for some parts of the occupied zone must be adopted. In addition, the forced warm air heating system leads to high temperatures in the upper regions of the hall; this causes increased heat losses and cost-inefficient heating operation.

Figure 7.4 confirms the nearly ideal vertical temperature distribution in the case of an existing industrial hall with a building insulation standard dating from early 2000. Throughout the entire space with a height of 14 metres an indoor temperature of approx. 20°C is achieved.

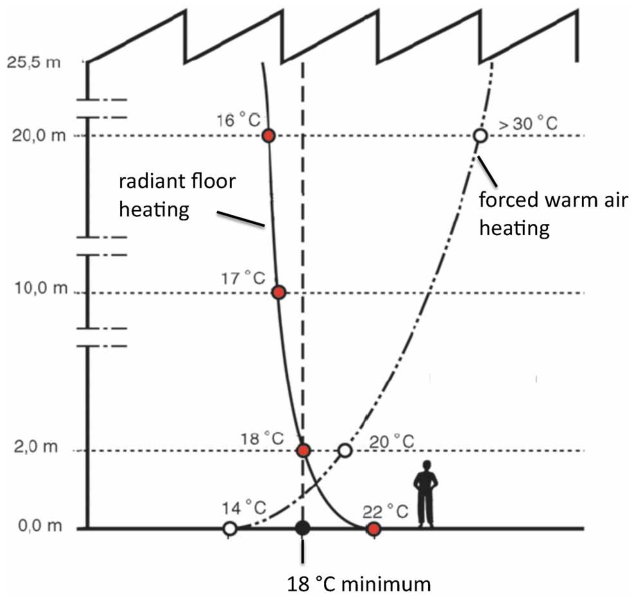



***Figure 7.3*** *Typical vertical air temperature profile for different types of heating in an industrial hall under the same conditions (adapted from BVF).*

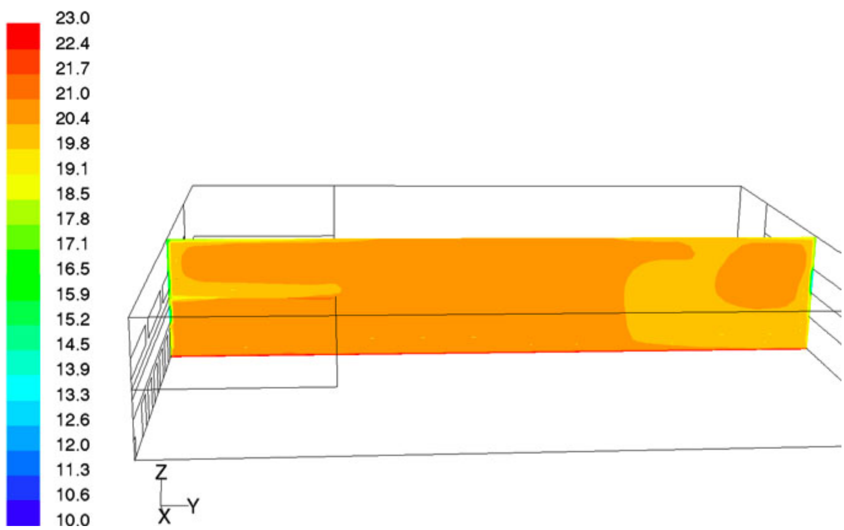



***Figure 7.4*** *Vertical distribution of the air temperature in the middle of the hall (CFD simulation). Cousin, Rene; Sommer, Klaus: "Verification of the air temperatures for an industrial hall with a gallery (y=0 to 19 m, x=0 to 126 m, z=7 m)", (2005, unpublished).*
*Hall geometry: 14 m high (z coordinate), 126 m deep (x coordinate), 76 m wide (y coordinate).*

Further selection criteria for radiant floor heating in industrial buildings are:

- Free and flexible use of space, because heating aggregates do not impose any limitations
- Optimal use of space in the hall
- Low velocity rates
- No dust is whirled up
- Cleaning the floors is tentamount to cleaning the system of distributing heat
- Utilization of renewable energy, such as solar and geothermal, and multiple use of energy for example through the use of waste heat from production processes
- Radiant floor heating can be used for cooling by making relatively modest adaptations in the heating system
- No maintenance costs
- Low noise level

## 7.3 System overview

Since especially high demands are placed on the engineering design of concrete floors in industrial buildings and the heating pipes must be integrated into the concrete slab, very good coordination is needed between the architects, structural engineers and engineers for building services involved in the planning.

Radiant floor heating can be installed in nearly every concrete slab structure built for industrial purposes. The sizing design of the concrete slab must always been determined by the structural engineer. The heating pipes are, as a rule, directly integrated into the load-bearing concrete slab. In the case of concrete slabs with the correct dimensions, no load will be borne by the heating pipes. The position of the heating pipes will depend upon the use, the mounting of equipment in the hall and the structural conditions of the concrete slab, see Figure 7.2 and 7.5.

***Figure 7.5*** *Example of the fixation of heating pipes integrated into the reinforcement and an example for the installation of equipment in the hall (Photo: BVF).*

The pipes for the radiant floor heating must be protected when they cross expansion gaps because of the mechanical stress that is to be expected in the area of the joints by placing them in protective sleeves, see Figure 7.6.

The need for thermal insulation below the heating pipes is regulated by the European Energy Performance of Buildings Directive (EPBD) or by corresponding national directives and must be considered by the building planner within the context of a holistic approach. The goal must be to plan an optimally cost-efficient solution, which can, for example, lead to the planning of peripheral strip insulation measuring maximally 5 metres instead of full-surface insulation of the floor slab. The economical width of the strips can be determined by consulting ISO 13370.

Various materials can be used for the heating pipes. The entire piping system for radiant floor heating for industrial

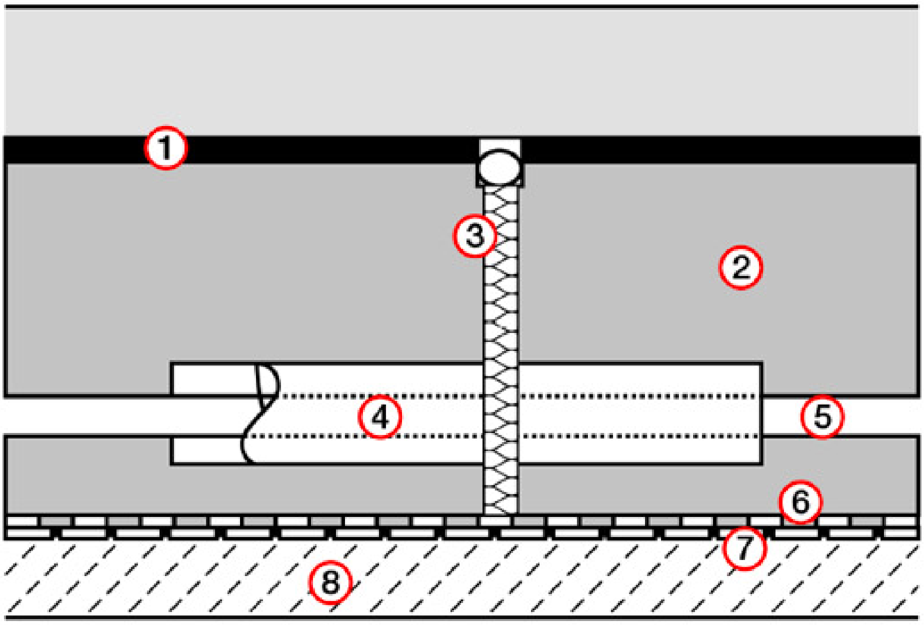



***Figure 7.6*** *Heating pipe with protective sleeve traversing an expansion gap (Photo: BVF)*
*1- wearing layer, 2 – concrete, 3- expansion gap, 4 - pipe protection sleeve, 5 -heating pipe, 6 -separation/gliding layer, 7 - waterproofing, 8 - granular subbase.*

buildings must be extremely robust in order to ensure that it can withstand the stress placed upon it during the construction process. If plastic heating pipes are used, then cross-linked polyethylene with an oxygen diffusion barrier are suitable.

The equipment for controlling and regulating the radiant floor heating for industrial buildings are subject to the regulations of the Energy Performance of Buildings Directive (EPBD) or corresponding national directives. Zone controls for regulating room temperature is possible for "zones of rooms of the same type and purpose".

## 7.4 Design aspects

For the sizing design of the heating system, the design heat load must initially be calculated according to the European standard EN 12831:2003 and, if necessary, according to the corresponding national appendices as well.

In the case of rooms with heights measuring in excess of 5 m, this can be calculated as a "special case B.1" according to the Appendix of EN 12831:2003 which covers compared to the standard cases the increase of the heat losses through the roof area due to the vertical air temperature gradient in halls with high ceilings. This allows for a correction for buildings with a specific design heat loss below 60 W/m², by calculating the overall design heat loss $\Phi_i$ using a "Room Height Correction Factor $f_{hi}$" as follows:

$$\Phi_i = (\Phi_{T,i} + \Phi_{V,i}) \cdot f_{hi} \text{ [W]}. \qquad (7.1)$$

In this equation $\Phi_{T,i}$ is the design transmission heat loss for standard cases in

EN 12831:2003 [W] and $\Phi_{V,i}$ is the design ventilation heat loss for standard cases in EN 12831:2003 [W].

The Room Height Correction Factor $f_{hi}$ is listed in Table B.1 of EN 12831:2003. The numerical values lie between 1.0 and a maximum of 1.6 and are always 1.0 for radiant floor heating used for industrial buildings. This is a considerable advantage over forced warm air heating, which requires a Room Height Correction Factor $f_{hi}$ = 1.60 at most for cross ventilation at a low height in rooms with 10 to 15 metre ceilings.

Together with the calculated overall design heat losses $\Phi_i$ the subsequent sizing design of radiant floor heating for industrial buildings is then undertaken with the aid of product-related performance diagrams according to EN 1264. The performance diagrams enable the spacing of the pipes and the desired design temperature of the heating water to be determined easily. When renewable energy is to be used (solar energy, geothermal energy) the lowest possible design temperature for the heating water should be chosen.

## 7.5 Calculation of warm-water and electrical floor heating

Large area heating is a heating system, which in the present goes through certain renaissance and it is becoming to be increasingly used in residential buildings, in civic amenities, but also in large area halls. Selection of this type of heating is primarily determined by the object itself, which have to fulfill thermal protection conditions so that the average heat loss is lower than 20 W/m³, eventually the average year consumption is under 70 to 80 kWh/m².

### Thermal calculation of warm-water floor heating

Calculation of the floor heating area comes out from the assumption that the mean surface temperature of the floor does not exceed hygienic acceptable values during which time the thermal output of the floor heating area is covering thermal losses of the heated space.

***Calculation of the mean surface temperature of the floor***

Assuming that the temperature on both sides of the heating floor area is the same $\theta_i = \theta_i'$, the mean temperature of the floor in the axis of the pipes $\theta_d$ can be calculated using the equation:

$$\theta_d - \theta_i = (\theta_m - \theta_i) \cdot \frac{tgh(\mathrm{m} \cdot \frac{L}{2})}{\mathrm{m} \cdot \frac{L}{2}} \quad [\mathrm{K}] \qquad (7.2)$$

Where $\theta_i$ is the calculated indoor temperature of the room [°C], $\theta_m$ is the mean temperature of the heating water [°C] and L is distance of axes of the pipes [m], $m$ is coefficient characterizing the heating slab in term of heat transmission [1/m].

Coefficient "$m$" characterizing the heating slab in term of heat transmission is calculated using the equation:

$$m = \frac{2 \cdot (\Lambda_a + \Lambda_b)}{\pi^2 . \lambda_d . d} \quad [1/\mathrm{m}] \qquad (7.3)$$

Where: $\Lambda_a$ is the heat permeability of the layer above the pipes; $\Lambda_b$ is the heat permeability of the layer below the pipes; $\lambda_d$ is the heat conductivity of the material of the layer, in which the pipes are embedded and $d$ is diameter of the pipes.

Heat permeability of the layer above the pipes $\Lambda_a$ can be calculated using the equation:

$$\Lambda_a = \frac{1}{\sum \frac{a}{\lambda_a} + \frac{1}{h_p}} \quad \text{[W/m}^2\text{·K]} \qquad (7.4)$$

Where $a$ is the thickness of the layer above the pipes [m]; $\lambda_a$ is thermal conductivity of the material of the layer [W/m·K]; $h_p$ is coefficient of the heat transfer upwards [W/m²·K].

The heat permeability of the layer below the pipes $\Lambda_b$ can be calculated analogue using the equation:

$$\Lambda_b = \frac{1}{\sum \frac{b}{\lambda_b} + \frac{1}{h_p}} \quad \text{[W/m}^2\text{·K]} \qquad (7.5)$$

Where $b$ is the thickness of the layer below the pipes[m]; $\lambda_b$ is thermal conductivity of the material of the layer [W/m·K]; $h_p$ is coefficient of the heat transfer downwards [W/m²·K].

Values of heat permeability $\Lambda_a$ and $\Lambda_b$ for some of the floor layers used are listed in the following tables (Table 7.1 and Table 7.2).

***Table7.1*** *Heat permeability $\Lambda_a$ of some floor layers.*

| No. | Composition of the layer | $\Lambda_a$ (W/m²·K] |
|---|---|---|
| 1 | Concrete + ceramic floor stuck | 8 |
| 2 | Concrete + ceramic floor mortar | 7 |
| 3 | Concrete + PVC | 8 |
| 4 | Concrete + hard carpet | 5.5 |
| 5 | Concrete + compressed woodchips | 4.5 |
| 6 | Concrete + soft carpet | 3.8 |
| 7 | Concrete without surface layer | 8.5 |
| 8 | Concrete reinforced with steel netting | 9.3 |

***Table 7.2*** *Heat permeability $\Lambda_b$ of some examples of bottom layers and constructions.*

| No. | Composition of the layer | $\Lambda_a$ [W/m²·K] |
|---|---|---|
| 1 | Reinforced concrete slab, levelling concrete layer, KARI netting | 3.2 |
| 2 | Reinforced concrete cavity panel, levelling concrete layer, KARI netting | 2.2 |
| 3 | Ceiling MIAKO (HURDIS), KARI netting | 1.8 |
| 4 | Reinforced concrete slab, levelling concrete layer, polystyrene 4 cm, PE, KARI netting | 0.8 |
| 5 | Reinforced concrete cavity panel + detto | 0.7 |
| 6 | Ceiling MIAKO + detto | 0.67 |
| 7 | Reinforced concrete slab, levelling concrete layer, polystyrene 6 cm, PE, KARI netting, Concrete + soft carpet | 0.6 |
| 8 | Reinforced concrete cavity panel + ditto | 0.55 |
| 9 | Ceiling MIAKO + detto | 0.5 |
| 10 | Concrete, cardboard, levelling concrete layer, polystyrene 6 cm, PE, KARI netting on the ground | 0.35 |

Mean surface temperature of the floor $\theta_p$ is calculated using the equation:

$$\theta_p - \theta_i = \frac{\Lambda_a}{h_p} \cdot (\theta_d - \theta_i) =$$

$$\frac{\Lambda_a}{h'_p} \cdot (\theta_m - \theta_i) \cdot \frac{tgh(m \cdot \frac{L}{2})}{m \cdot \frac{L}{2}} \text{ [K]} \qquad (7.6)$$

We can see that at given input values $\theta_i$ and $\theta_{p,max}$ (24 to 26°C – rooms where people are mostly standing, 28 to 29°C – residential and office, 32 to 35°C – bathrooms, floors, pools) in respect of the material of the walkway layer (Table 7.3), the mean surface temperature of the floor depends foremost on the axis distance of the pipes $L$, other parameters are either more or less constant or have only small influence on the result (e.g. diameter of the pipes $d$).

Floors used by footwear people do not influence thermal state of a man from the point of view of the construction material used. For this case is recommended optimal temperature of the floor 25°C for sitting persons and 23°C for standing and walking persons.

In the same time the mean surface temperature of the floor $\theta_p$ influences the mean temperature of the heating water. This may not exceed the value of 50°C. The temperature difference in inlet and reverse piping is can be maximally 10 K, whereas 5 to 6 K is recommended. In the following two tables (7.3 and 7.4) is listed the mean surface temperature of the floor heating area by the mean temperature of the heating water and values ($\theta_p - \theta_i$) for the temperature difference ($\theta_m - \theta_i$) = 25 K depending on the distance of the heating pipes.

***Table 7.3*** *Optimal surface temperature of the floor used by footwear people.*

| Floor material | Optimal temperature of the floor | | Recommended interval of the surface temperature of the floor $\theta_p$ [°C] |
|---|---|---|---|
| | 1. min | 10. min | |
| Textile | 21 | 24.5 | 21.0 - 28.0 |
| Cork | 24 | 26 | 23.0 - 28.0 |
| Pinewood | 25 | 26 | 22.5 - 28.0 |
| Oak wood | 26 | 26 | 24.5 - 28.0 |
| PVC on concrete | 28 | 27 | 25.5 - 28.0 |
| Hard linoleum on wood | 28 | 26 | 24.0 - 28.0 |
| Gas concrete | 29 | 27 | 26.0 - 28.5 |
| Concrete smudge | 28.5 | 27 | 26.0 - 28.5 |
| Marble | 30 | 26 | 28.0 - 29.5 |

***Table 7.4** Mean surface temperature of the floor heating area $\theta_p$ [°C] at mean temperature of the heating water $\theta_m$ = 35°C.*

| Room temperature $\theta_i$ [°C] | Mean surface temperature $\theta_p$ [°C] at $\theta_i$ [°C] and pipe distance l [m] | | |
|---|---|---|---|
| | 0.1 | 0.15 | 0.2 |
| 10 | 27.3 | 25.9 | 24.3 |
| 12 | 27.9 | 26.6 | 25.2 |
| 14 | 28.6 | 27.3 | 26 |
| 16 | 29.2 | 28.1 | 26.9 |
| 18 | 29.8 | 28.8 | 27.7 |
| 20 | 30.4 | 29.5 | 28.6 |
| 22 | 31 | 30.3 | 29.4 |
| 24 | 31.6 | 31 | 30.3 |

***Calculation of the specific heat output of the floor heating area***

The specific heat output $q$ [W/m²] is a parameter crucial for dimensioning the floor heating. Two cases can occur that the floor heating area divides spaces with equal temperature above and below it or with different temperatures above and below it.

The specific heat output of the heating area at the same temperature above and below the heating floor area (flow) upwards $q$ is calculated using the equation:

$$q = \Lambda_a.(\theta_d - \theta_i) = h_p.(\theta_p - \theta_i) \qquad \text{[W/m²]} \qquad (7.7)$$

and specific heat flow from the floor heating area downwards $q'$ from the equation

$$q' = \Lambda_b.(\theta_d - \theta_i) = \Lambda_b \frac{h_p}{\Lambda_a}.(\theta_p - \theta_i) = \frac{\Lambda_b}{\Lambda_a}.q \qquad \text{[W/m²]} \qquad (7.8)$$

At different temperatures in rooms above and below the floor heating area exact calculation of $\theta_p$ and the output of the floor heating area $q$ is more difficult. For standard practice it is possible to proceed in the same way the temperatures above and below the heating floor are equal and neglected the inaccuracy that arises. The specific heat flow downwards has to be calculated differently according to the equation:

$$q'' = \Lambda_b.(\theta_d - \theta_i) =$$

$$\Lambda_b \frac{h_p}{\Lambda_a}.(\theta_p - \theta_i) + \Lambda_b.(\theta_i - \theta_i') \qquad \text{[W/m²]} \qquad (7.9)$$

Where $\theta_i'$ is temperature of the room below the heating area [°C].

Heat flow $q''$ downwards to a space without heating should not exceed the value of 10% of the heat input to the heated room.

It is necessary to adapt the value of the thermal resistance to this requirement according to the equation:

$$\frac{1}{\Lambda_b} \geq 10 \cdot \left( \frac{1}{\Lambda_a} + \frac{\theta_i - \theta_i'}{q} \right) \qquad \text{[W/m²]} \qquad (7.10)$$

In the case that the heat flow downwards "$q'$" is to a heated space it is added to the output of the floor heating area in the heated room, if its contribution is higher than 5%.

In Table 7.5 are listed values of specific heat output of the floor heating area at the mean heating water temperature of 35°C.

***Table7.5** Specific heat output of the floor heating area q [W/m²] at the heating water temperature $\theta_m$ = 35°C.*

| Room temperature $\theta_i$ [°C] | Specific heat output $q$ [W/m²] at $\theta_i$ [°C] and pipe distance $l$ [m] | | |
|---|---|---|---|
| | 0.1 | 0.15 | 0.2 |
| 10 | 209 | 190 | 173 |
| 12 | 192 | 190 | 159 |
| 14 | 175 | 160 | 145 |
| 16 | 158 | 145 | 131 |
| 18 | 141 | 130 | 117 |
| 20 | 124 | 115 | 103 |
| 22 | 107 | 100 | 89 |
| 24 | 90 | 85 | 75 |

Regarding calculation with values of specific heat output "$q$" and heat loss "$q'$", for rooms situated below other heated rooms the total heating area is calculated using the equation:

$$S_p = \frac{\Phi_{\max}}{q + q'} \quad [\text{m}^2] \qquad (7.11)$$

Where $\Phi_{max}$ is maximal heat loss of the room calculated according to the corresponding standard [W].

For rooms in ground-floor objects or on top floor of tier buildings the heating area is calculated as follows:

$$S_p = \frac{\Phi_{\max}}{q} \quad [\text{m}^2] \quad (7.12)$$

The total heat input of the heating area $\Phi_p$ is in both cases given by the equation:

$$\Phi_p = (q + q')S_p \quad [\text{W}] \qquad (7.13)$$

and the weight flow through the pipes

$$M = \frac{\Phi_p}{c.(\theta_{m1} - \theta_{m2})} \quad [\text{kg/s}] \qquad (7.14)$$

Where $c$ is specific heat capacity (at $t$ = 35 to 40°C, c = 420 J/kg.K)

$\theta_{m1}$ is the input temperature of the heating water [°C]
$\theta_{m2}$ is the output temperature of the heating water [°C]

According to the resultant water flow through the system the pipe distribution of the floor heating in the whole building is consequently dimensioned.

### Thermal calculation of electrical floor heating

When designing a heating system, it is necessary to respect the total heat loss of the room, the character of the heated space, operation period of the room, operation period of the heating system, the size of the floor area that is available for bedding of the heating plane and demands on the floor covering. The calculation itself involves design of the composition of the floor construction and materials of particular layers, determination of thermal conditions in the floor during the operation of the system and determination of the input of the heating source.

***Design of the floor construction composition***

For optimal function of the electrical floor system proper design of the composition of layers above and below the heating plane, their thickness and heat conductivity is crucial. Thermal resistances of these layers influence the surface temperature of the heating component; therefore it is important to ensure meeting criteria for the maximal temperature value stated by the manufacturer, failing which causes overheating of insulation covers and consequently their degradation, which lowers the lifetime of the whole heating component. Therefore it is important to ensure fluent transmission of the heat from the heating plane.

The floor construction compound, especially the thickness of the concrete layer and the type of floor covering used depends on the type of the proposed floor heating system, whether talking about accumulation, half-accumulation or immediate operation.

Heat flows from the heating plane in both directions are proportional to coefficients of heat transmission from the heating plane upwards $U_1$ and downwards $U_3$ and for efficient operation of the system observing of their mutual proportions depending on the character of the space situated below the heated room is necessary.

$U_1 \geq 4 \cdot U_3$ if the space below the heated room is heated,

$U_1 \geq 6 \cdot U_3$ if below the heated room there is a not heated space or ground,

$U_1 \geq 6.5 \cdot U_3$ if the heated room is above the outdoor environment.

The coefficient of heat transmission upwards $U_1$ is given by the equation:

$$U_1 = \frac{1}{\frac{1}{h_p} + \frac{d_{a1}}{\lambda_a} + \frac{d_n}{\lambda_n}} \text{ [W/m}^2\text{.K]} \qquad (7.15)$$

and the coefficient of heat transmission downwards $U_3$ is given by the equation:

$$U_3 = \frac{1}{\frac{1}{h_p} + \frac{d_{a2}}{\lambda_a} + \frac{d_i}{\lambda_i} + \frac{d_{zi}}{\lambda_z}} \text{ [W/m}^2\text{.K]} \qquad (7.16)$$

The thickness of the concrete layer depends on the need for accumulation. It is designed for:

- operation with accumulation 90 – 150 mm,
- operation with half-accumulation 60 – 90 mm,
- immediate operation 50 mm (without need for accumulation).

When using electrical floor heating, power take-off from the power network is not continuous as it is by other types of heating, but the system is operating with heating intermissions. To have thermal comfort during the whole operation the ability of the system to gradually emit the heat stored in the floor in such way that at the end of charging temperature in the room does not fall under 32°C or temperature of the surface of the floor under 23°C is important. Accumulation ability of the floor is possible to express using a time constant $T_e$, which, in certain way, represents time necessary for cooling of the heated up floor construction. The time constant of the heating floor $T_e$ can be determined using the equation:

$$T_e = \frac{d_a.\rho_a.c_a}{U_1 + U_3} \qquad \text{[s] (7.17)}$$

where $d_a$ is total thickness of the accumulation layer [m], $\rho_a$ is specific weight of the accumulation layer [kg/m³], $c_a$ is specific thermal capacity of the accumulation layer [J/kg.K].

Proper function of the accumulation system is ensured by the time constant at least for 12 hours, for objects with operation during the whole day the optimum is 12 to 16 hours. Its value can be lower for half-accumulation systems with possibility of daily charging. On the contrary, at immediate operation of heating, when there is a requirement of flexible operation, the value guarantees dynamics of the heating for maximum 3 hours. As can be seen from the equation listed above, the size of the time constant directly depends on the floor construction compound, thickness and thermal conductivity of particular layers.

***Thermal conditions in the heating plane and on the surface on the floor***

When heating by means of electric cables the heating plane does not constitutes a continual large area source of heat and also the take-off of electrical energy is discontinuous, so the thermal field at the level of heating components is not homogeneous. In the calculation we will assume mean temperature on the heating plane that will settle after an infinitely long period of charging in ¼ of the distance between two adjacent wires $\theta_{t\infty}$ and can be obtain as follows:

$$\theta_{t\infty} = \frac{P_1 + U_1.\theta_i + U_3.\theta_i^{'}}{U_1 + U_3} \qquad \text{[°C] (7.18)}$$

Where $P_1$ is specific installed input of the heating source [W].

It is necessary to ensure the temperature at the level of heating compounds during the operation cycle not to exceed limit values stated by the manufacturer.

Temperature on the surface of the floor during charging of the system $\theta_p$, which should not exceed allowable values from hygienic point of view, can be derived from the temperature on the heating plane $\theta_t$ and temperature of the air above the floor $\theta_i$:

$$\theta_p = (1 - \frac{U_1}{h_p}).\theta_i + \frac{U_1}{h_p}\left[\theta_{t\infty} + (\theta_{t0} - \theta_{t\infty}).e^{-\frac{t}{T_e}}\right] \qquad \text{[°C] (7.19)}$$

After ending the period of basic charging the power input is interrupted, that means $P_1 = 0$. and following is the discharge period of the system. After an infinite period of cooling the temperature in the heating system settle and can be determined by the equation:

$$\theta_{t\infty}^{*} = \frac{U_1.\theta_i + U_3.\theta_i^{'}}{U_1 + U_3} \qquad \text{[°C] (7.20)}$$

The temperature on the surface of the floor during discharging of the system can be than determined by the equation:

$$\theta_p = (1 - \frac{U_1}{h_p}).\theta_i + \frac{U_1}{h_p}\left[\theta_{t\infty}^{*} + (\theta_{t1} - \theta_{t\infty}^{*}).e^{-\frac{t}{T_e}}\right] \qquad \text{[°C] (7.21)}$$

Where $\theta_{t\infty}^{*}$ is temperature on the heating plane at the beginning of charging in time $t = 0$ [°C]; $\theta_{t1}$ is temperature on the heating plane at the end of charging [°C].

In case of the half-accumulation regime after ending the time of discharging the time of supplementary charging follows.

***Thermal input of the floor construction***

Designing the power input of the heating source we come out from the total heat loss of the heated interior $\Phi_c$ and from the area of the floor construction $S_p$, which is needed for storage of the heating plane. Heating components are laid only to a clear, un-built area of the floor. Therefore it is necessary to know, approximately, the placement of the furniture or sanitary equipment or consider its placement by walls.

Required heating area as well as heat flows upwards $q$ and downwards $q'$ can be determined in the same way as in case of the warm-water type. Total heat input of the heating area $\Phi_p$ is determined using the equation:

$$\Phi_p = (q + q')S_p \quad \text{[W]} \quad (7.22)$$

In practice a simplified way of determination of the required power input for the heating system is often used. It comes out from the total heat demand determined from a calculation according to the corresponding standard and from additional power needed for ensuring the dynamics of the heating system and for covering the loss heat flow downwards, assuming that its value represents maximally 10% of the total output.

$$\Phi p = (1.10 \text{ to } 1.30)\cdot\Phi c \quad \text{[W]} \quad (7.23)$$

Where $\Phi_c$ is total heat loss of the heated room without additional power [W].

**a) Accumulation and half-accumulation operation of heating**

The installed input of the heat source P needed is determined using the equation:

$$P = \frac{T}{t_{NN} + t_{DD}}.\Phi_p \quad \text{[W]} \quad (7.24)$$

and specific installed input of the heating source $P_1$ on a unit area of the room using the equation:

$$P_1 = \frac{T}{t_{NN} + t_{DD}}.\frac{\Phi_p}{S_p} \quad \text{[W/m}^2\text{]} \quad (7.25)$$

Where $T$ is total operation period [h]; $t_{NN}$ is time of primer charging [h]; $t_{DD}$ is time of supplementary charging [h].

**b) Immediate operation of heating**

The installed input of the heat source P needed is determined using the equation:

$$P = \Phi_p \quad \text{[W]} \quad (7.26)$$

And specific installed input of the heat source $P_1$ on a unit area of the room using the equation:

$$P_1 = \frac{\Phi_p}{S_p} \quad \text{[W/m}^2\text{]} \quad (7.27)$$

In following tables there are listed recommended values of specific installed inputs depending on the purpose of the heated room, assembled on the basis of practical experiences.

When choosing a specific installed input of an immediate heating it is necessary to respect also recommendations of manufacturers of floor coverings regarding the allowable floor temperature.

***Table 7.6*** *Recommended specific installed inputs for floor heating.*

| Room/object purpose | Recommended specific installed input of the heat source [W/m²] |
|---|---|
| Generally | 160 - 200 |
| Bedroom | 160 |
| Kitchen | 160 |
| Living room, workroom | 180 - 200 |
| Sanitary rooms | 200 - 220 |
| Churches, temples | 250 - 300 |

***Table 7.7*** *Recommended specific inputs for floor heating with immediate operation.*

| Room/object purpose | Recommended specific installed input of the heat source [W/m²] |
|---|---|
| Generally | 80 - 140 |
| Bedroom | 80 - 100 |
| Kitchen | 80 - 90 |
| Living room, workroom | 110 - 120 |
| Bathroom | 100 - 150 |
| WC | 80 - 100 |
| Entrance hall | 80 - 120 |
| Wooden floor | 80 - 100 |
| Renovated floor | 100 - 120 |
| Garages, churches | 100 - 150 |
| Workshops | 80 - 150 |

***Table 7.8*** *Distance of heating wires depending on the specific installed input.*

| Specific load p = 10 W/m | | Specific load p = 18 W/m | |
|---|---|---|---|
| Specific installed input | Distance of heating wires | Specific installed input | Distance of heating wires |
| P1 [W/m²] | m [mm] | P1 [W/m²] | m [mm] |
| 50 | 200 | 80 | 225 |
| 57 | 175 | 90 | 200 |
| 67 | 150 | 103 | 175 |
| 80 | 125 | 120 | 150 |
| 100 | 100 | 144 | 125 |
| 133 | 75 | 180 | 100 |
| 200 | 50 | 240 | 75 |

Heating wires are laid in a shape of meander during which time the distance between particular wires m depends on specific installed input and chosen type of heating cable from the point of view of its specific loading "p" [W/m]:

$$m = \frac{p}{P_1} \qquad \text{[m]} \quad (7.28)$$

Total length of the heating wire "*l*" can be calculated according to the equation:

$$l = \frac{P}{p} \qquad \text{[m]} \quad (7.29)$$

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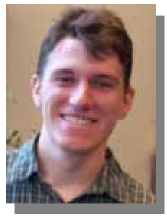
## About authors



**Ondřej Hojer (1980. Praha, Czech Republic)** had finished his masters at CTU (Czech Technical University in Prague), Faculty of Mechanical Engineering, Department of Environmental Engineering in 2005. Diploma thesis "Radiant geometry of plaque radiant heaters" had obtained dean's appreciation for excellent diploma thesis. In 2008 he successfully defended Ph.D. thesis entitled "Optimization of radiant geometry of plaque radiant heaters". During his PhD studies he had spent half year at Technical Universitaet Eindhoven in Netherlands where he broaden his knowledge in the field of computer simulations. Sofar he is partly employed as research scientist at CTU and partly as HVAC designer in company KOTRBATÝ. In 2005 he has become a member of Society of Environmental Engineering in Czech Republic (member of REHVA and ASHRAE) and in 2009 he was admitted to the board of the heating section.

**Karel Kabele (1960. Praha, Czech Republic)** is full-time professor and head of the Department of Microenvironment and Building Services Engineering at the Faculty of Civil engineering CTU Praha. Until 2010 he supervised 8 successful PhD students and more than 101 MSc thesis. He has international experience as visiting professor at National University of Singapore, external examiner at TU Vienna, The Hong Kong Polytechnic University and others. He is professionally focused on intelligent building energy systems, interaction of heating systems with the building, modelling and simulation of energy and environmental building performance and energy auditing, author and co-author of more than 120 international and national conference papers, journal articles and book chapters. In 2008 he was elected as REHVA (European federation of HVAC engineers associations) vice-president. He is also president of the Czech Society for Environmental Technology, member of Czech Chamber of authorised engineers and technicians board founding member of IBPSA-CZ (International building performance simulation association).

**Miroslav Kotrbatý (1953, Praha, Czech Republic)** is graduate of the La Guardia gymnasium and post-graduation course of heat engineering at .technical college (prof.Ing.Václav Pokorný) in Prague in Czech Republic. He enters praxis in 1951 as a ventilation and air-conditioning designer in company Kovoprojekta Praha and shortly after that he started in company Projekta Praha in heating unit. He finished the Czech Technical University in Prague, department of heating and ventilation in 1963 in evening courses while still working as designer. Already in 1954 he met Dr.Cihelka from Czech science academy and with him he started to work on the invention of radiant strips heating systems. He was behind realization of the first installations of radiant strips in Czech republic in 1955 and infrared heaters in 1957. Another occupation was in engineering company Pikaz with aim on automotive industry. He obtained enormous experience from housing development and building-up in Krajském projektovém ústavu Praha (Regional design department Prague). As main specialist he created conceptions and strategies for heat supply of many cities in Czech Republic. After retirement (1990) he founded own design company, that was gradually expanding with help of family (son Martin) into assembly and engineering field and also started importing foreign products (daughter Jitka). In the 1994 manufacture was founded in Pelhřimov, small town between Prague and Brno. The author is well known in Czech Republic for his publishing (more than hundred technical papers, guidebooks, books) and for his contributing on many conferences and seminars in the field of HVAC since 1950s in Czech Republic and Slovak Republic. He was one of the founders of Czech chamber of associated engineers and technicians and still active member of examination committee at Czech technical university in Prague, Faculty of mechanical engineering, Department of environmental engineering.

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**Klaus Sommer** has been a full-time professor and director of the Laboratory for Heating Technology at the Department of Process Engineering, Energy and Mechanical Systems of the Cologne University of Applied Sciences/Germany. He teaches courses in the fields of Heating Technology and Building Energy Simulation. He is a member of the Senate of this University. He has been serving as an expert witness for heating technology in the German judicial system since 2000. On an international level, he is active in the American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE), International Building Performance Simulation Association, USA (IBPSA-USA) and the Federation of European Heating and Air-Conditioning Associations (REHVA).

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**Dušan Petráš (1956, Zvolen, Slovakia)** is university Professor for Building Services at the Faculty of Construction of the Slovak Technical University in Bratislava since 1999.In the years 2000 – 2006 he was Dean of the Faculty of Construction of the Slovak Technical University, 2007 - 2011 Vice Chancellor of the Slovak Technical Unversity. In 2003 was invited as visiting Professor at the Danish Technical University in Lyngby. Since 1998 is ASHRAE fellow. In 2002 - 2005 was elected as President of REHVA (Federation of European Heating, Ventilating and Air-Conditioning Associations). As a President of the Slovak Society for Enviromental Technolog and president of the Editorial Board of the professional journal TZB-Hautechnik (Building Services) organised many seminars and conferences. Professor Petráš is author of 15 books and monographies published in 7 countries and translated into 7 languages, hundreds of scientific and professional articles and lectures published/presented in 25 countries mainly focused on heating and energy auditing issues lead of 130 expert studies, energy audits and concepts. Professor Petráš is the pioneer of energy conservation, energy auditing and energy certification in Slovakia.

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Federation of European Heating, Ventilation and Air-conditioning Associations

**REHVA Office**

Washington Street 40, 1050 Brussels – Belgium
Tel: +32-2-5141171 • Fax: +32-2-5129062

**Orders:** www.rehva.eu • info@rehva.eu